A PROJECT REPORT

ON

FAILURE ANALYSIS OF MECHANICAL SEALS

Submitted by

16ME19, 16ME26, 17DME135

In partial fulfillment for the award of the Degree

Of

BACHELOR OF ENGINEERING

IN

MECHANICAL ENGINEERING

UNDER THE GUIDANCE

OF

PROF.ZIA MOMIN

DEPARTMENT OF MECHANICAL ENGINEERING

ANJUMAN-I-ISLAM

KALSEKAR TECHNICAL CAMPUS NEW PANVEL,

NAVI MUMBAI-410206

UNIVERSITY OF MUMBAI

ACADEMIC YEAR 2019-2020

ANJUMAN-I-ISLAM

KALSEKAR TECHNICAL CAMPUS NEW PANVEL

(Approved by AICTE, rec. By Maharashtra Govt. DTE, Affiliated to Mumbai University)

PLOT #2&3, SECTOR 16, NEAR THANA NAKA, KHANDAGAON, NEW PANVEL,NAVI MUMBAI-410206, Tel.: +91 22 27481247/48 * Website: www.aiktc.org

CERTIFICATE

This is to certify that the project entitled

"FAILURE ANALYSIS OF MECHANICAL SEALS"

Submitted by

DESHMUKH SOBAN ASLAMKHAN

KHAMKAR BILAL ATHAR

PATHAN TAVIT AYUB

To the Kalsekar Technical Campus, New Panvel is a record of Bonafide work carried out by him under our supervision and guidance, for partial fulfillment of the requirements for the award of the Degree of Bachelor of Engineering in Mechanical Engineering as prescribed by **University Of Mumbai**, is approved.

Internal Examiner **External Examiner**

Head of Department **Principal**

ANJUMAN-I-ISLAM

KALSEKAR TECHNICAL CAMPUS NEW PANVEL

(Approved by AICTE, rec. By Maharashtra Govt. DTE,

Affiliated to Mumbai University)

PLOT #2&3, SECTOR 16, NEAR THANA NAKA, KHANDAGAON, NEW PANVEL,NAVI MUMBAI-410206, Tel.: +91 22 27481247/48 * Website: www.aiktc.org

APPROVAL OF DISSERTATION

This is to certify that the thesis entitled

"FAILURE ANALYSIS OF MECHANICAL SEALS"

Submitted by

DESHMUKH SOBAN ASLAMKHAN

KHAMKAR BILAL ATHAR

PATHAN TAVIT AYUB

In partial fulfillment of the requirements for the award of the Degree of Bachelor of Engineering in Mechanical Engineering, as prescribed by University of Mumbai approved.

(Internal Examiner) **(External Examiner)**

Date: _________

ACKNOWLEDGEMENT

After the completion of this work, we would like to give our sincere thanks to all those who helped us to reach our goal. It's a great pleasure and moment of immense satisfaction for us to express my profound gratitude to our **guide PROFESSOR ZIA MOMIN** whose constant encouragement enabled us to work enthusiastically. His perpetual motivation, patience and excellent expertise in discussion during progress of the project work have benefited us to an extent, which is beyond expression.

We would also like to give our sincere thanks to **PROFESSOR ZAKIR ANSARI (Head Of Department)** and **PROFESSOR RIZWAN SHAIKH** (**Project coordinator)** from Department of Mechanical Engineering, Kalsekar Technical Campus, New Panvel, for their guidance, encouragement and support during our project.

I take this opportunity to give sincere thanks to **MR.M.LOKHANDE(HOD MECHANICAL/CIVIL/UTILITY), MR.M.N.PITALE(MANAGER)** in ***"VINATI ORGANICS LIMITED"*** for all the help rendered during the course of this work and their support, motivation, guidance and appreciation.

I am thankful to **DR.ABDUL RAZZAK HONNUTAGI** , Kalsekar Technical Campus New Panvel, for providing an outstanding academic environment, also for providing the adequate facilities.

Last but not the least I would also like to thank all the staffs of Kalsekar Technical Campus (Mechanical Engineering Department) for their valuable guidance with their interest and valuable suggestions brightened us.

DESHMUKH SOBAN ASLAMKHAN
KHAMKAR BILAL ATHAR
PATHAN TAVIT AYUB

Table of Contents

CHAPTER 4: RESULTS AND DISCUSSION

ABSTRACT

This report discusses the final draft of the chosen topic, which entitled **Failure Analysis in the Mechanical Seal**. Seal face crack propagation is one of the major causes which lead to the failure of the mechanical seal therefore analyses on the seal beforehand is important. Normally failure happens due to this crack which may happen as early as after two month after the installation of the mechanical seal.

The objective of the project is to analyze the effects on the seal face by applying multiple of different loads which are pressure and temperature based on actual condition of the pump. This project will focus more on the effects on structural stress, thermal loads, and fluid flow condition on the seal face. After the analyses on the seal face are done, some changes will be proposed in order to enhance the performance of the seal face. Later, new analyses will be performed to determine the capability on the proposed changes.

The main material that will be analyzed is silicon carbide based seal face as this is the most commonly used in the mechanical seal assembly. Therefore, the characteristics and properties of the silicon carbide will be set first in advance before the simulation of the effects of both pressure and temperature.

CHAPTER 1
INTRODUCTION

1. BACKGROUND STUDY

Mechanical seals are the most versatile type of seal for rotating shafts. Their main use is on liquid/ gas sealing, e.g., centrifugal pumps. They are also used on gas/ gas applications, e.g., compressor and agitator shaft seals, but in these cases they are usually deployed as double seals with liquid injection to provide lubrication of the seal faces. Liquid sealing duties occur in such applications as double seals where process fluid is one side and barrier fluid on the water.

Mechanical seal in principle consist of two plane faces arranged perpendicular to the axis of a rotating shaft. This gives rise to the alternative name of radial face seal. One face is fixed to the pump casing or vessel and is stationary; the other is fixed to the shaft and rotates with it. In order to keep leakage to an acceptable level, it is necessary that the two faces run with very small separation, typically less than 0.001 mm. (Mayer, 1977).

In order to keep frictional heat generation and wear within acceptable limits, a lubricating film of liquid must be maintained between the seal faces, while not exceeding the film thickness mentioned above. In most cases, where single seals are employed, this lubricant will be the sealed fluid, which may have poor lubricating properties.

In order to accommodate wear and pump build tolerances, it is necessary that one of the faces can move axially and that one of the faces has the freedom to accommodate swash (angular misalignment in the rotating component). Usually the same face has the freedom to accommodate both these movements.

The floating face may be either stationary or rotating, but for high speed applications (shaft surface speed greater than about 15 m/s) stationary mounting avoids problems arising from centrifugal forces.

Normally the seal is mounted internally, with the sealed fluid on the outer periphery of the seal. With highly corrosive liquids, however the seal can be mounted externally to minimize exposure of the seal parts to the corrosive conditions. (Lebeck, 1991).

Mechanical seals thus have a superficial resemblance to thrust bearings. However, the very close running clearance which must be maintained, poor lubricating properties of the lubricant, and very low lubricant flow combine to make the design of mechanical seals much more difficult than that of thrust bearings.

2. PROBLEM STATEMENT

Seal failed for a number of reasons and one of them may be **the result of crack propagation on the seal face** which may lead to leakage then resulting in seal failure. Usually, crack propagation will transpire on inner part of the seal face about two months after the installation of the mechanical seal. If the crack reached up to certain degree, it may spread and cause leakage in the pump.

Silicon carbide, carbon, and ceramic seal rings are always subjected to this type of failure, but other materials can also crack. This crack propagation is caused by several factors. Some of them are **over tightening clamped units, carefree handling of seal rings, improper cooling sections, dry-running conditions, or unchanged face materials** over some time of life expectancy.

Therefore, it is needed to analyze the seal face **in terms of effects of pressure and temperature** as in the real condition. From the analysis, it can be seen when the failure will start to occur and afterwards some modifications could be made to **expand the life expectancy** of the seal face and **improve the performance** of the mechanical seal.

3. OBJECTIVES

The objectives of this analysis can be divided into three parts:

i. **To investigate the cause of crack propagation which will lead to seal leakage in term of structural stress, thermal distribution, and fluid's flow.**

ii. **To propose a new modification (design) on the seal face model to encounter the previous problem and increase the life expectancy .**

iii. **To perform new analysis on the new proposed model based on the criteria mentioned in the first objective.**

3. SCOPE OF STUDY

For this analysis, only inner compression system will be analyzed as the failure always occurred on the inner rotating seal face of a single stage mechanical seal. In the compression, the type of material taken into considerations is only applied to **silicon carbide** seal faces as this type is most commonly used.

The analysis includes the study of the behavior of the silicon carbide seal face. In doing this, the study will be divided into three separate parts, in terms of the viewpoint taken. **First**, in term of stress due to high pressure applications and the effects on the seal face. **Second**, in term of the upshot of the seal face properties after being applied with thermal load and **lastly** is about the flushing fluid flows.

Besides, in view of the range and ambiguity of issues related to the subject under study, it is made in favor of the function of mechanical seals. The advantage of this study is that the discussion will not extend far beyond the limits set by practical constraints. In fact, practical constraints of the seal face properties, whether on the side of the seal face or mechanical seal itself, should not be neglected in studies of silicon carbide seal faces. The requirement for efficiency in seal faces function, to a certain extent, the possible interpretations of result and real modeling applications of seal faces.

CHAPTER 2

LITERATURE REVIEW AND THEORY

1. MECHANICAL SEAL CONCEPT

A basic mechanical seal is not a complex device. It consists primarily of a rotary seal face with a driving mechanism which rotates at the same speed as the pump shaft, a stationary seal face which mates with the rotary and is retained using a gland or in some pump models as an integral stuffing box cover, a tension assembly which keeps the rotary face firmly positioned against the stationary face to avoid leakage when the pump is not in operation, and static sealing gasket(s) and elastomers strategically located to complete the seal assembly. (Summers-Smith, 2005)

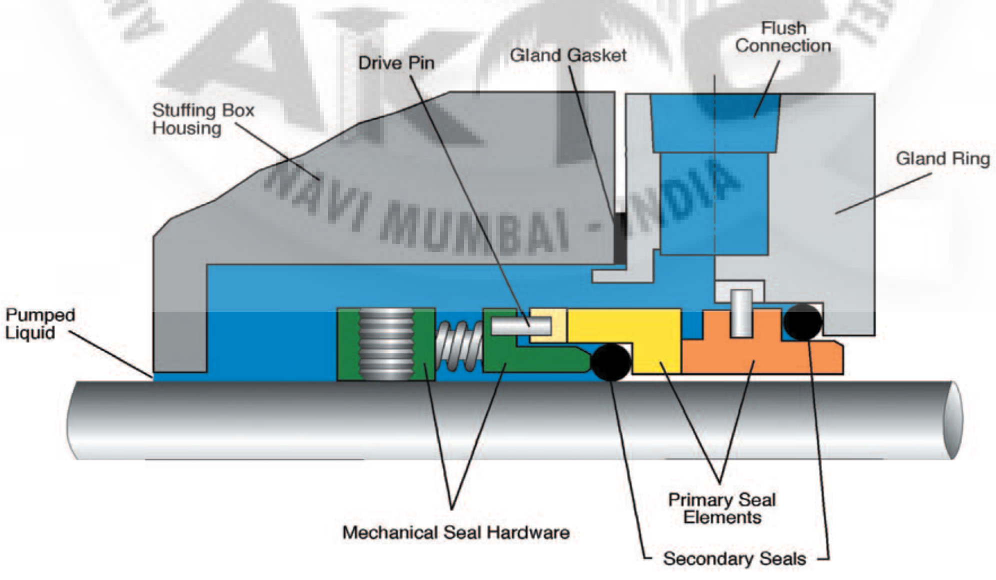


Figure 1: Basic Mechanical Seal

The rotating and stationary sealing faces commonly referred to as primary seal members, are material selected for their low coefficient of heat and are compatible with the fluid being pumped. Their extremely flat, lapped mating surfaces make it extremely difficult for the fluid to escape between them. The fluid does however, forms a thin layer or film between the faces and migrates toward the low pressure side of the faces. It is this boundary layer of fluid which is used and to cool and lubricate the seal facesTo prohibit leakage along the pump shaft through the inside diameter of the rotary and stationary seal faces, the mechanical seals assembly uses O-rings, V-rings, wedges and packing. Known as secondary seals, these components of the seal selected based on fluid compatibility, temperature, elastomeric qualities, and depending on the type and design of the seal they may perform in either a static or dynamic condition. (Heinz K. Muller, 1998).

2.SEAL FACE DESIGN

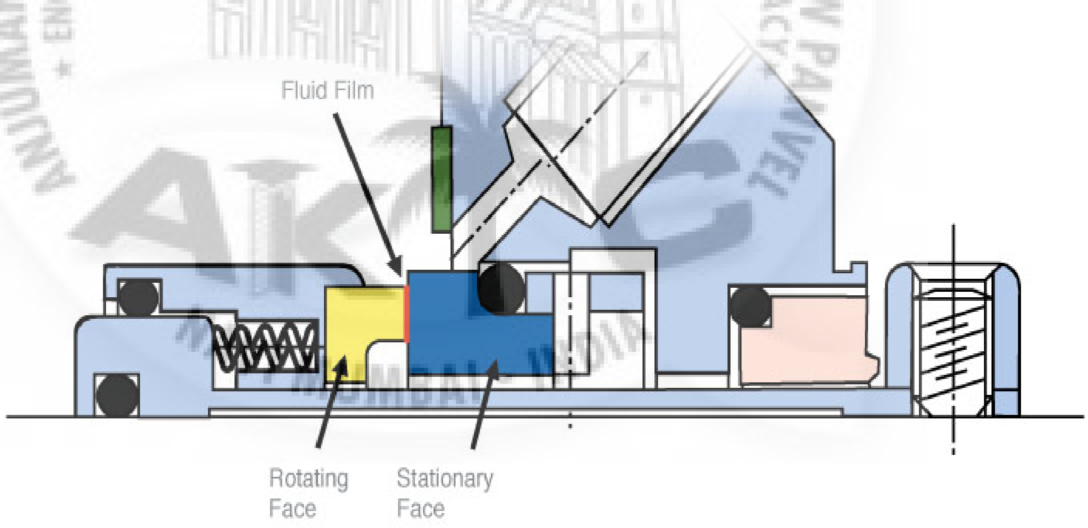


Figure 2: Stationary and Rotary Parts of Mechanical Seal

Sealing devices can be classified as static seals, such as gaskets, and dynamic seals, such as rotating interfacial axial seals, also known as end face mechanical seals. Mechanical seals are used in many types of pumps of various sizes and pressure ratings, and are used to transport a great diversity of fluids in many industries. Therefore, mechanical seals are available in a variety of configurations. However, regardless of the service conditions, all mechanical seals operate on the same basic principle. A simple mechanical seal's design is shown

.

* BASIC MECHANICAL SEAL COMPONENTS

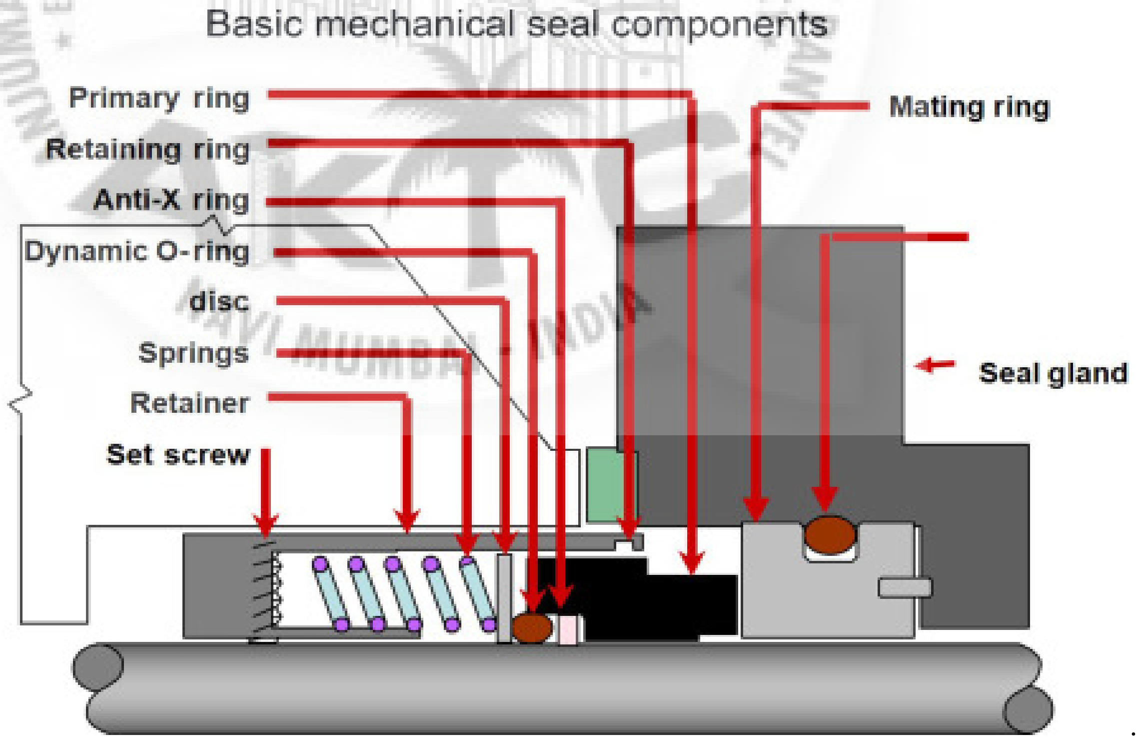


Figure: Detailed Mechanical Seal System

A seal consists of two sealing rings, either of which rotates relative to the other. One of the rings is mounted rigidly and the other is arranged so that it can move freely and align radially, axially, and angularly with the rigidly mounted ring. A dynamic seal is achieved where two rings contact perpendicularly, to the pump shaft. These rings are called seal faces, one of the rotating faces and the other one is the stationary face. The faces are lapped flat, which results in very low levels of leakage, while at the same time providing long life on the basis of normal wear.

Besides the two faces, the mechanical seal contains a set of secondary sealing dynamically and statically, loading the faces, and transmitting rotation. The secondary seals provide sealing between the seal faces and the metal parts, such as mating ring housing, sleeve, and gland. The metal parts transmit the torque and provide an axial mechanical force by means of a spring element to load the lapped faces. (Publication, 2000).

In most cases, rotating seal face is made of either silicon carbide or carbon graphite. **However, silicon carbide is the selected material in this analysis because of its properties.** Some of them are the excellent resistance from abrasion, high resistance from corrosion, and good resistance of thermal shock. In fact, if compared to carbon graphite, silicon carbide has higher maximum load capability. Therefore it always become the material of silicon for the manufacturing of the seal face.

3.GENERAL THEORIES ON THERMAL CONDIT1ON

3.1 Thermal Expansion on Ring Shaped Materials

When the temperature of a substance (solid, liquid or gas) is increased, the molecules or atoms in it vibrate faster and they tend to move away from each other, on average. This results in an expansion (increase of length and depth) of the substance as a whole. This is called as thermal expansion. Over particular temperature ranges, the thermal expansion can be described by the coefficient of linear expansion. It can be illustrated as below (Cengel, 1998):

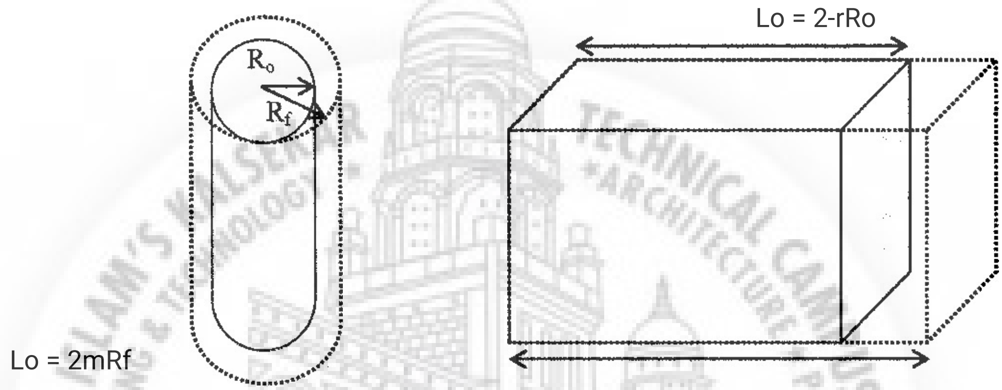


Figure 4: Thermal Expansion Diagram

Let suppose that the temperature is increased from TD to T. Thus the change in temperature is AT = (T-TD). Next, the length is changed from Lo to L, therefore the change in length is AL = (L-Lo). Thus we have:

$$\therefore \Delta L \approx Lo\Delta T$$

ALz AT

Therefore we can write as AL = uLoAT where u is called the coefficient of linear expansion.

Alternatively, it also can be written as:

(L-LO) = uL0AT

.:.L = Lo(l + CAT)

Replacing the length with radius, change in radius will be AR = (R — Ro).

To put it in the form of an equation:

$$\Delta L = \alpha L_0 \Delta T$$

$$\Delta L = \frac{Lf - Lo}{Lo} \cdot \left(\frac{1}{\alpha}\right) = \Delta T$$

$$\frac{\delta\pi}{\delta\pi}\left[\frac{Rf - Ro}{Ro}\right] \cdot \left[\frac{1}{\alpha}\right] = \Delta T$$

a -Thermal expansion coefficient

R_o =Initiat Radius

Rt = Pinal Radius (after expansion)

The thermal expansion for inner radius, outer radius, and width of the ring can be further simplified.

Inner radius: $r_f =$ ro (1 aåT) Outer radius: $R_f = R_o (1 + \alpha\Delta T)$

Width of the ring: WdWc (.1 + ccaT)

As a conclusion, on ring shaped material, the inner radius, outer radius, and width of the ring will increase. This follows the rule of linear expansion theory.

3.2 Heat Generated Due to Viscous Drag

Heat generated due to viscous drag is considerable in some cases. Specifically in high-speed applications or when dealing with large seals at normal pump speeds, it is possible that the turbulence of the liquid surrounding the mechanical seal generates as much or more heat than the faces. Experience shows that this component may be considered when the face sliding speed or velocity exceeds 25 m/s (500 R/min).

It is primarily a function of peripheral velocity and the properties of liquid in consideration. For such applications consultation with the seal manufacturer is recommended to verify appropriate design features, such as the need for a stationary spring seal. (Ortoleva, 1994).

The following illustrates the method of calculating the heat generated due to viscous drag.

First, calculate the Reynolds number based on the clearance between the seal and housing. To calculate the Reynolds number, first calculate the tangential velocity within the seal chamber.

alvelocity. Ut = $_t = \frac{\pi \cdot D \cdot n}{60}\left(\frac{m}{s}\right)$ (metric units)

Tangaltia.l $_t = \frac{\pi \cdot D \cdot n}{720}\left(\frac{ft}{sec}\right)$ Ut (US customary

tmits) Remolds $= \frac{U_t \cdot c \cdot \rho}{\mu}$ nunbef, Re (metric mits)

Remolds number, $= \frac{U_t \cdot c \cdot \rho}{g \cdot \mu}$ Re(US customary units)

ume•e:

Ut — tangeztial velocity at the OD, (mis or msec) c — clearance between the seal and housing: (m Of in) p = ofthe fluid, (kg.'m^2 or lbs:n^3) g = viscosity ofthe fluid, (Pa.s or lb- g pavitational accel«ationt 386.1 D = seal chamb« bore, (m or in) R = seal radius. (m or in)
n = shan speed. lpm
L = length of seal. Of -'clearance". (m Of in)

Next, to find out the loss coeffcient CM, use the proper equation depending on the flow regime.

Once the loss coefficient is found, then the heat generated due to viscous drag can be computed using the following equation:

$P = \frac{1}{2} C_m \rho U_t^3 \pi L R$ (W) (metric knits)

CmpUt%rLD
(Btu•Y) (US customary tnits)
4086

3.3 Thermal Deflections

Thermal deflections result from a non-uniform temperature distribution within the seal. Frictional heat, conducted through the body of the seal from the sliding interface to the seal-chamber fluid is usually the main factor, but when sealing a high-temperature fluid, additional thermal gradients occur. The direct effect of thermal expansion is a major component of the thermal deflection, but thermal stresses can induce an additional deflecting moment which must also be considered.

Calculation of thermal deflections is possible if reliable data are available for frictional heat generation, but this depends upon the face materials and the operating conditions. It is not sumcient to assume a universal value of friction coefficient. Definition of values of heat transfer coefficient, between the seal and contacting fluid can also present difficulty.

3.4 Thermal Conduction (Heat Transfer)

The majority of mechanical seals operate with at least a partial fluid film between the faces with the possible exception of unbalanced seals at high pressures where dry friction dominates and PV is most meaningful.

The frictional heat generated at the face of a mechanical seal can lower the viscosity of the fluid in the film to the point where the load bearing ability of the film is insufficient and failure results through heavy wear or face damage. It can also heat the film to such extent that the fluid boils or vaporizes at the prevailing film pressure.

The heat balance equations contain friction and heat transfer values which agree closely with e,'Qerimental values and the method is reliable when applied to general purpose seals. Its principal value here, however, is to serve to illustrate those factors which influence seal face temperatures so that choices may be made.

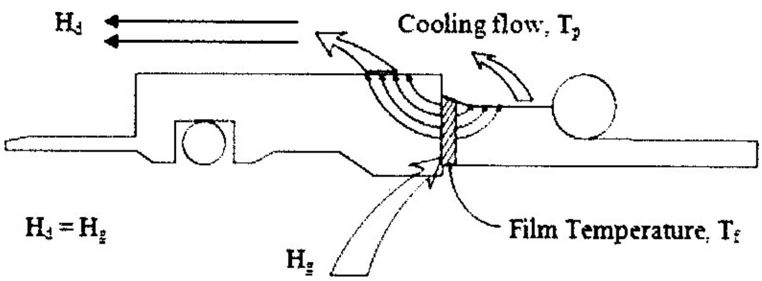


Figure 5: Heat Dissipation on Seal Face Diagram

Regarding the figure above, heat generated at the faces is as follow:

$$Hg = gDmn(Fs+ Ap \times Af \times B)$$

While for the heat dissipated:

$$Hd = m^{e} kA \tanh(m^{t} l) \times AT$$

Wherem' = Q(h'C/kA) k = thermal conductivity of seal

A = cross sectional area perpendicular to heat flow h' = heat transfer coefficient

l = axial length of seal ring heat transfer surface

C circumference of heat transfer

AT temperature rise at seal face over surrounding fluid (Tf— Tp)

Heat is dissipated from both rotating and stationary faces and hence, total heat is equal to the combination of heat from rotating face and stationary face. By equating the heat generated and heat dissipated, the temperature rise at the faces may be calculated.

4. Failure of Mechanical Seal

4.1 Seal Life

Most mechanical seal's manufacturer estimate the life of a seal is about two years at least, while some can sustain up to three years such as those specified in Standard API 682. This life span is depending on the pressure, temperature, and the application of the seal itself. The limit however, may be only reliable for steady state operation, and the life span may be changed or overstated for a cyclic operation of mechanical seal. This is because in cyclic operation, more limitations occur and the effects of dynamic state are taken into consideration which involves including vibration, fluid flow, and turbulence condition.

However, generally there are no specific or fixed rules which determine the value of pressure, temperature, and vibration a seal can tolerate along the application. An exception may be API 682 seals, which specifically address that shortcoming by including a series of cyclic tests to qualify a specific seal type for a range of services, which in result may enhance the seal life up to three years.

4.2 Type of Failures

Each mechanical seal is expected to achieve its maximum life and optimum performance when operated under its design specification. If the leakage exceeds the environmental or plant site operating limit, the seal is considered failed. This failure may occur either before or after the seal has achieved the life expectancy of its usage as specified by the manufacturer and this failure will affect the equipment in the plant and result in equipment failure and downtime which will cost a fortune.

There are many causes of seal failure such as incorrect selection of seal design, abuse of seal components before installation, erroneous installation, improper setup, improper pump selection, contamination of sealing fluid, and worn-out seal.

4.2.1 Leaking

As specified in the scope of study, this project focuses on a single stage mechanical seal. There are four possible paths for seal to leak in this type of seal as shown A seal consists of two sealing rings, either of which rotates relative to the other. One of the rings is mounted rigidly and the other is arranged so that it can move freely and align radially, axially, and angularly with the rigidly mounted ring. A dynamic seal is achieved where two rings contact perpendicularly, to the pump shaft. These rings are called seal faces, one of the rotating faces and the other one is the stationary face. The faces are lapped flat, which results in very low levels of leakage, while at the same time providing long life on the basis of normal wear.

Besides the two faces, the mechanical seal contains a set of secondary sealing dynamically and statically, loading the faces, and transmitting rotation. The secondary seals provide sealing between the seal faces and the metal parts, such as mating ring housing, sleeve, and gland. The metal parts transmit the torque and provide an axial mechanical force by means of a spring element to load the lapped faces. (Publication, 2000).

In most cases, rotating seal face is made of either silicon carbide or carbon graphite. However, silicon carbide is the selected material in this analysis because of its properties. Some of them are the excellent resistance from abrasion, high resistance from corrosion, and good resistance of thermal shock. In fact, if compared to carbon graphite, silicon carbide has higher maximum load capability. Therefore it always become the material of silicon for the manufacturing of the seal face.

- **POSSIBLE SEAL LEAKAGE POINTS**

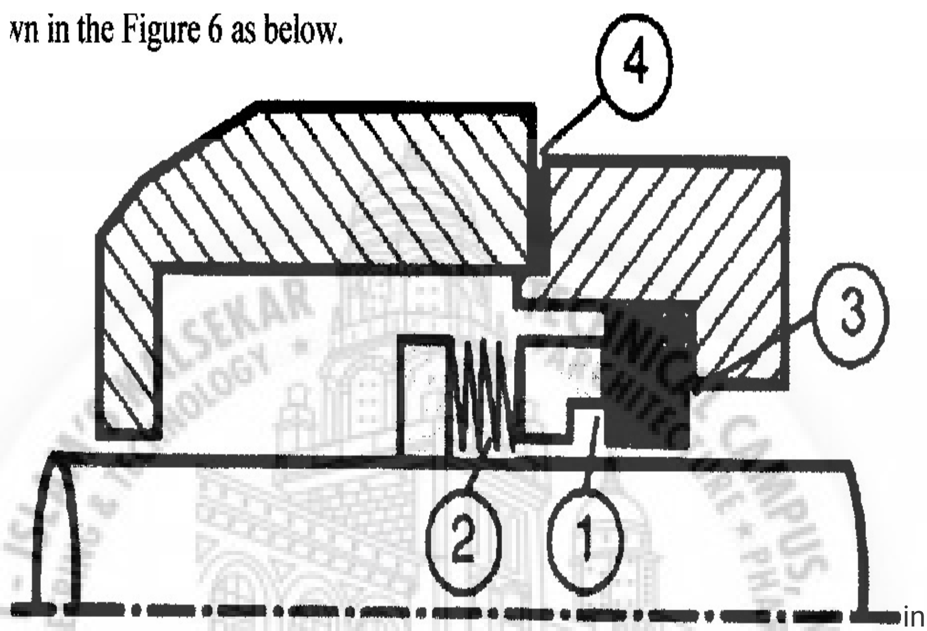


Figure 6: Possible Seal Leakage Points

1:leakage visible at the point where the shaft exits the gland or at the drain connections

2: Dynamic secondary seal leakage — also visible at the point where the shaft exits the gland or at the gland connections

3: Static secondary seal leakage — also visible at the point where the shaft exits the gland or at the drain connections

4: Gland gasket leakage — visible at the gland seal chamber interface

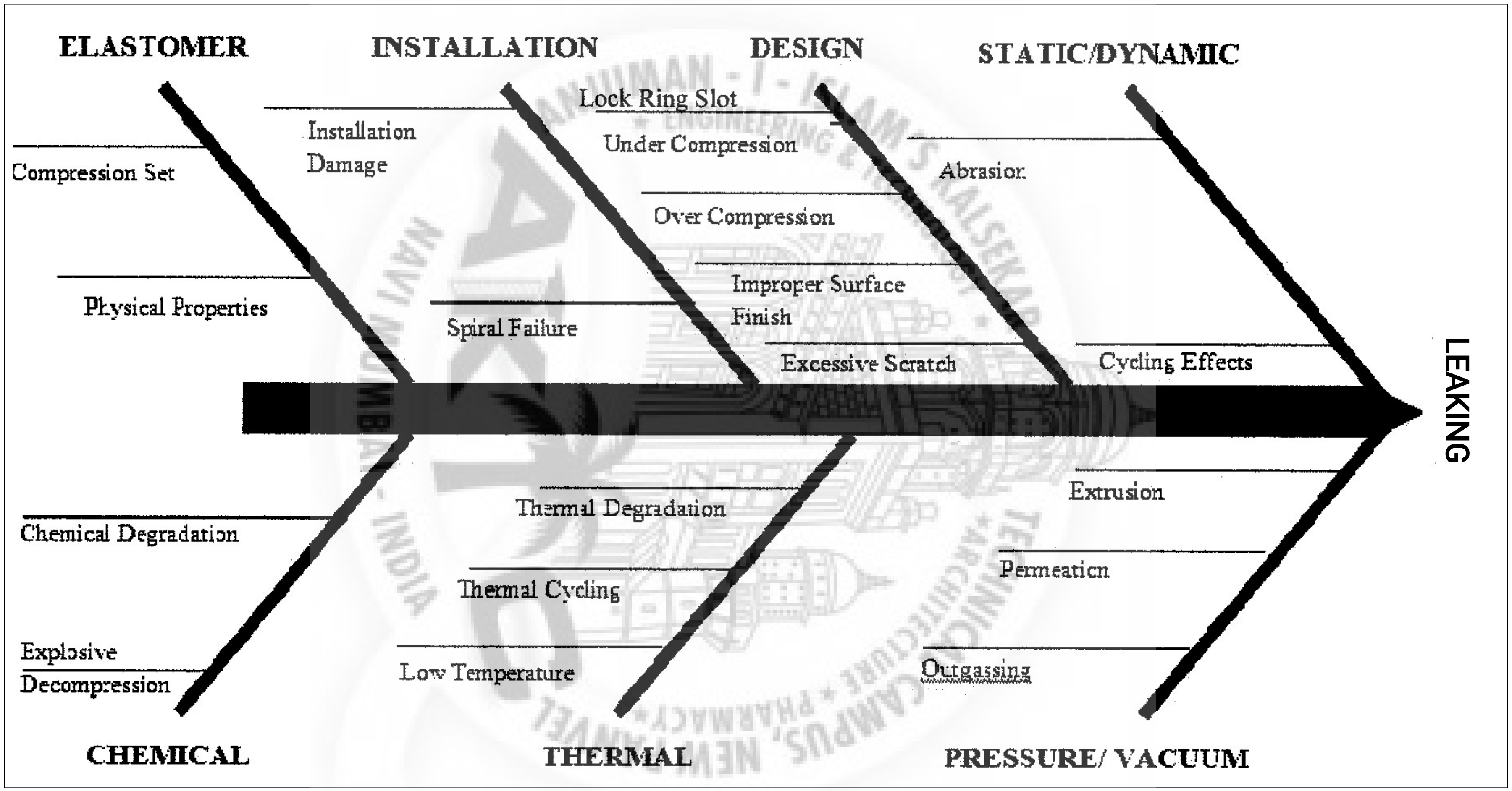


Figure 7: Fishbone Diagram on Failure Causes of Mechanical Seal (Leaking)

4.2.2 Contamination

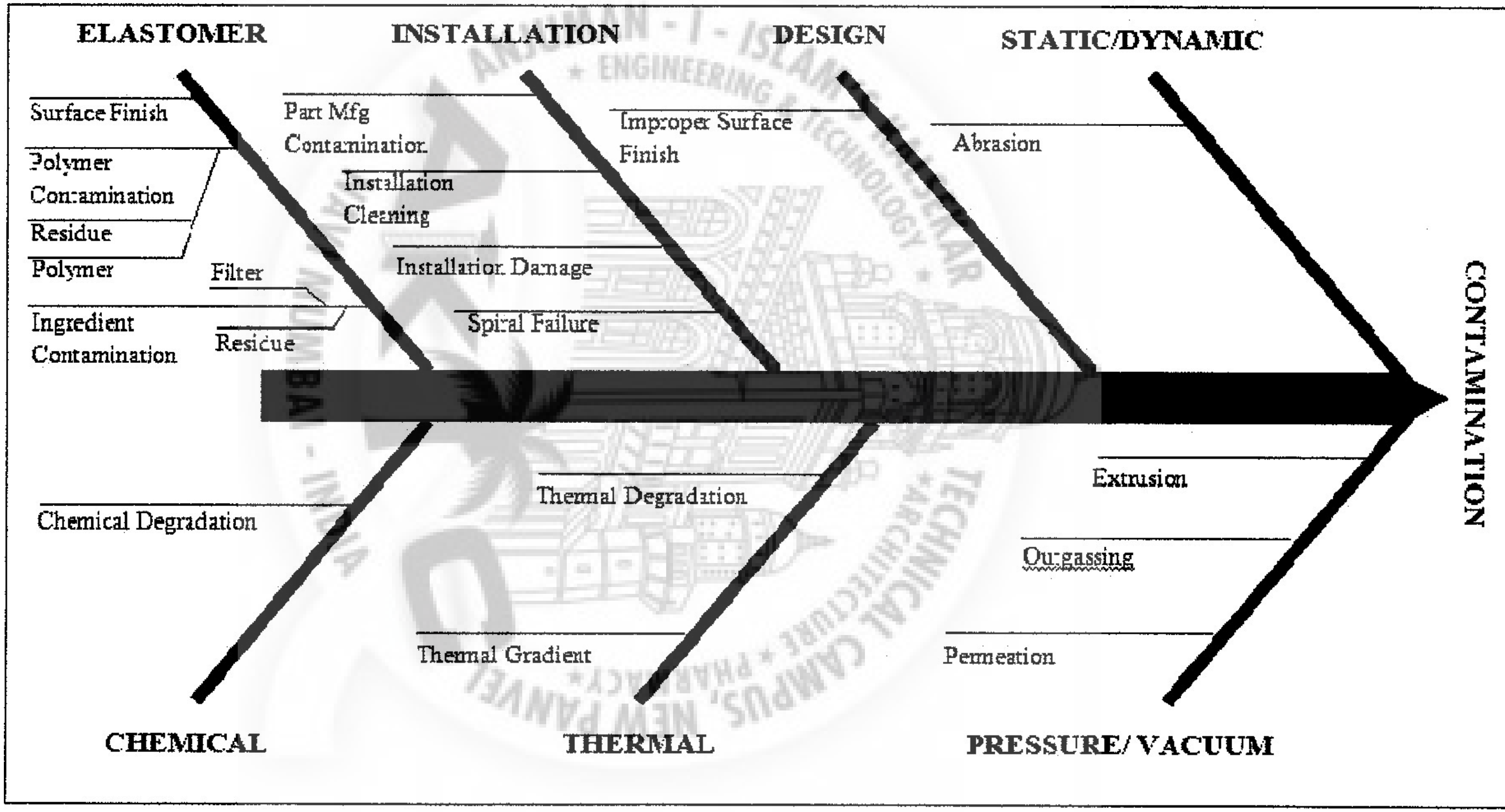


Figure 8: Fishbone Diagram on Failure Causes of Mechanical Seal (Contamination)

4.2.3 Appearance Change

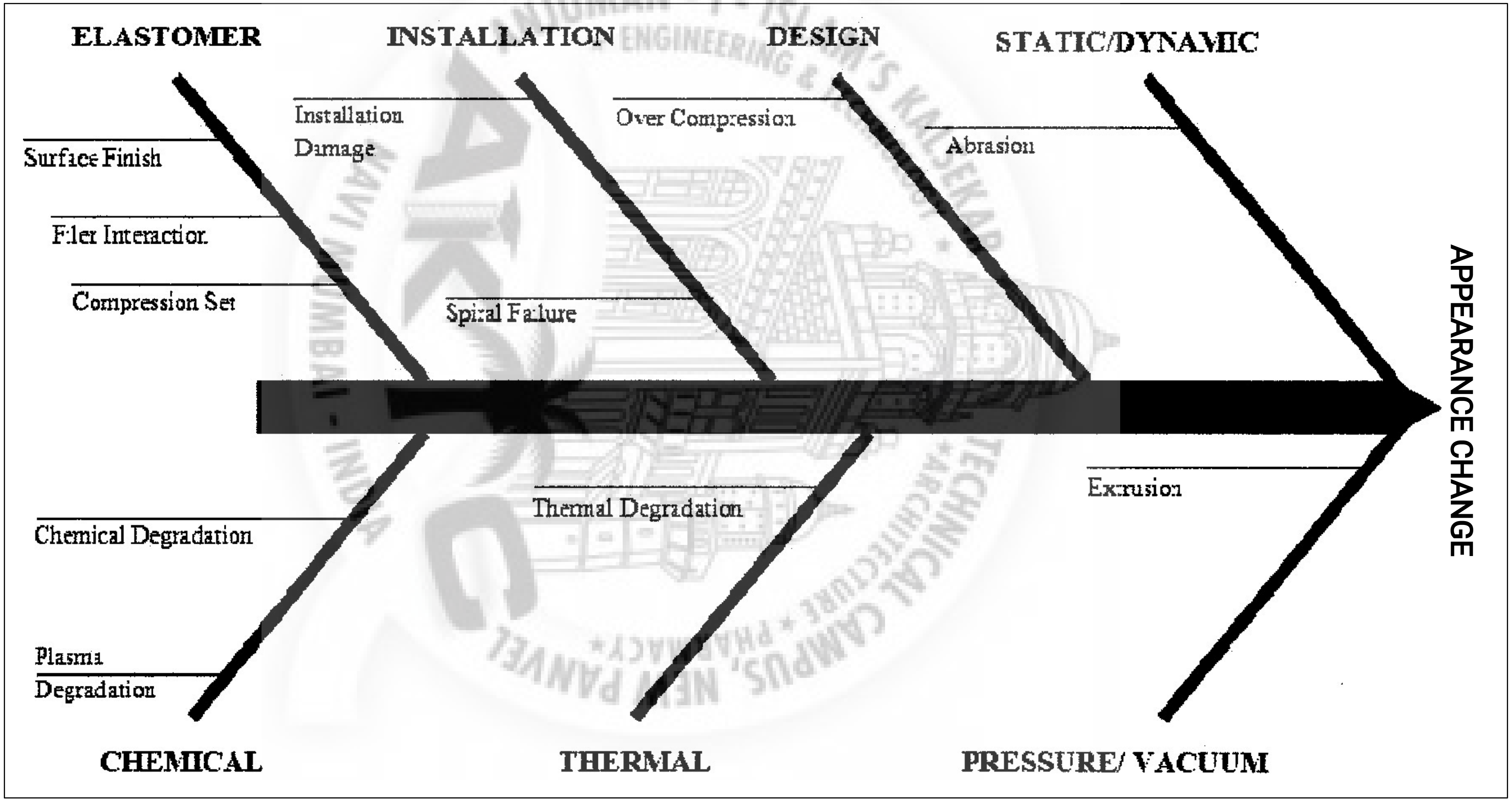


Figure 9: Fishbone Diagram on Failure Causes of Mechanical Seal (Appearance Change)

CHAPTER 3

METHODOLOGY/ PROJECT WORK

1. PROJECT WORKFLOW

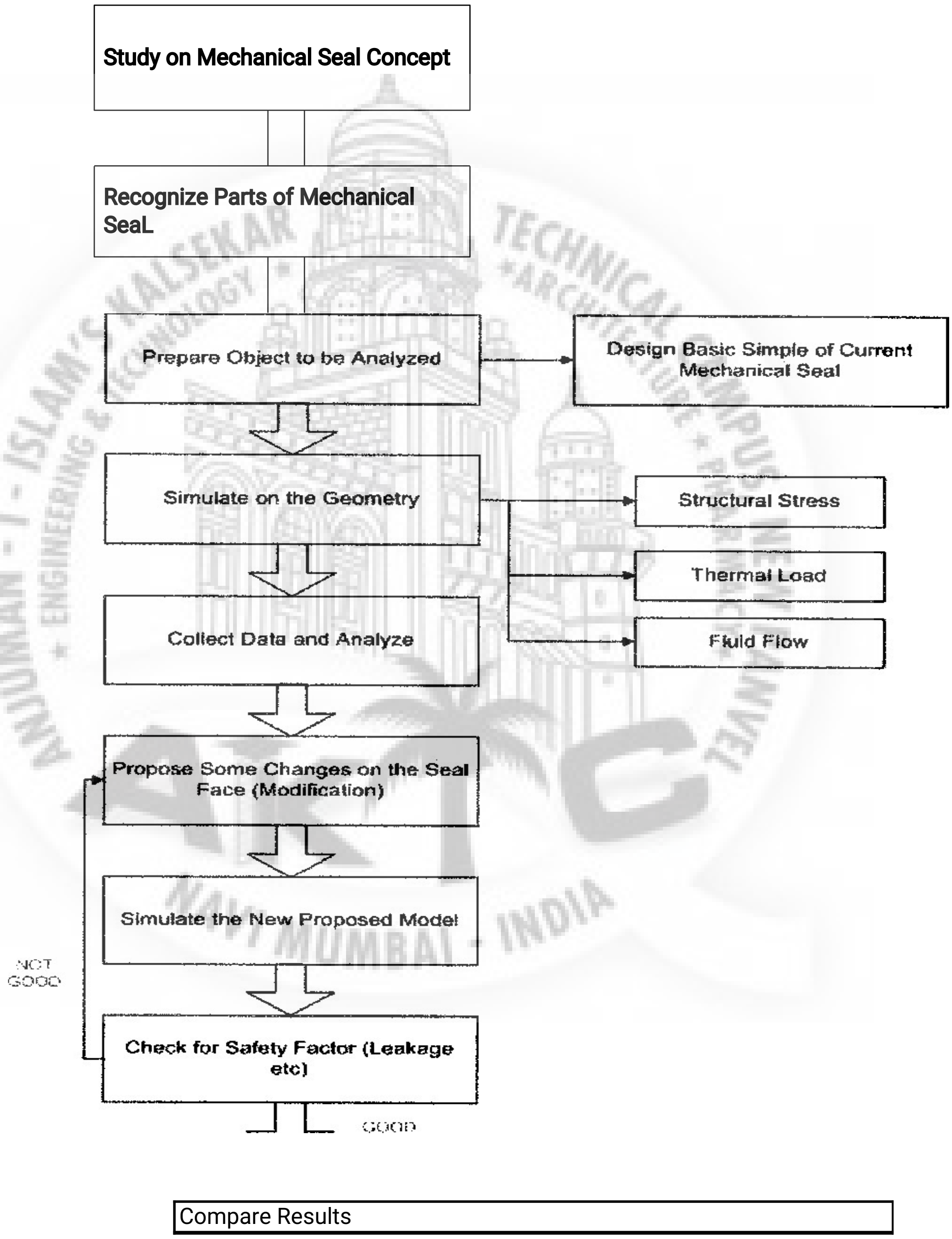


Figure 10: Project Workflow

The diagram above (Figure 10) shows the project work flow of the project. Firstly, study has to be done about the concept of mechanical seal including the function, type, and basic usage. Then, for better understanding, further research is made for recognize each part of the seal which contribute to the sealing function in their own way. These first two steps are very important because concentrating on understanding the whole concept of mechanical seal before any further analysis could be done.

After then only, a model of basic mechanical seal will be designed as the source of geometry for the analysis. From the first analysis based on several loads mentioned in the diagram, data is collected and analyzed to be as reference for the next analysis. Then, by applying several theories on stress effects and the design of the seal face itself, some modifications will be proposed to enhance the performance of the seal face. Later the second analysis will be performed and analyzed either the result shown representing a better model of seal model or not. If not good enough, another changes will be made to the model and be analyzed again until meet the requirements.

Then, the two models will be compared based on the results from the simulation analyses.

2. RESEARCH METHODOLOGY

For this research, most of the information on theories is gathered from readings. Books related to mechanical seal, pump shaft, seal design, and flushing concept do contribute to this project. From these sources, deep understanding has been achieved in order to go through this project in the right direction.

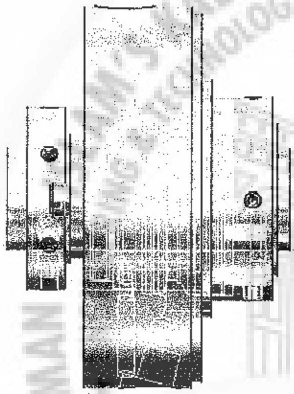
Other than written sources, many references also found from the websites. As the information is not limited to certain areas, it is easy to look for any solution regarding engineering concept of mechanical seal. Also, meetings with the supervisor are very helpful as any concern regarding this project can be asked directly in terms of technical and theoretical problems.

CHAPTER 4

RESULTS AND DISCUSSIONS

1. DESIGN OF BASIC MECHANICAL SEAL

The design of the mechanical seal has been completed based on the configuration of the single seal as it is the basic type of mechanical seal and can be used as a benchmark of analysis instead of using a double or tandem seal first which is a combination of single seal. Below is the picture of the complete seal.



PART	MATERIAL
SEAL FACE	SILICON CARBIDE
O-RING	VITON
SPRING	HAST-C
	CARBON
SPRING HOLDER	SS 316
RETAINING RING	SS 316
SET SCREW	SS 316
SHAFT SLEEVE	SS 316
FLANGE	SS 316
LOCK PIN	SS 316
THROTTLE BUSHING	BRONZE
DRIVE COLLAR	SS 316
SETTING FIXTURE	SS 316
HEXAGON BOLT	SS 316

Table 1: Part List of Single Balanced Seal

While for the stationary face:

Figure 13: Location of Stationary Seal Face

2. SETTINGS FOR THE ANALYSIS OF THE SEAL FACES

2.1 Material Properties

This section explains about the material properties of the seal faces. One is **silicon carbide for rotating face and SS 316 for the stationary face.** Below are the properties of both of them.

Properties	Value	Unit
Density	7750	kg m^a-3
Coefficient of thermal expansion	1.70E-05	CA-I
Reference temperature		
Poisson's ratio	0.31	none
Young's modulus	1.93E+11	Pa
Tensile yield strength	2.07+08	pa
Compressive yield strength	2.07E+08	pa
Tensile ultimate strength	5.68E+08	pa
Compressive ultimate strength	5.65E+08	pa
Tangent modulus	1.80E+09	pa

Table 2: Properties of Stainless Steel

Properties	Value	Unit
Density	312 7	kg m^-3
Coefficient of thermal expansion	4.00E-06	CA-I
Reference temperature		
Poisson's ratio	IUOE-OI	none
Young's modulus	4.10+1 1	pa
Tensile yield strength	1.30E+0 8	pa
Compressive yield strength	1.30E+0 8	pa
Tensile ultimate strength	3.40E+09	pa
Compressive ultimate strength	3.90E+09	pa
Tangent modulus	4.50E+09	pa

Table 3: Properties of Silicon Carbide

2.2 Structural Stress Analysis

The analysis consists of rotational and pressure condition.

22.1 General Settings

The general settings for this analysis are:

Setting	Description
Solver Type	Direct
Update Interval	2.5s
Analysis Type	
Environmental Temperature	30
Maximum Principle Stress Definition	Maximum over time
Stress Tool Definition Theory	Mohr-Coulomb Stress

Table 4: Structural Stress Analysis General Settings

2.2.2 Connections

The connection features functional to specify the contact conditions between surface before loading, and as part of the final solution to verify the transfer of loads
(forces and moments) across the various contact regions.

CurteåCMOdél,

Joint Connection	Lock Ring-Body	**Joint Connection**	Lock Ring-SiC Seal Face
Connection Type	No Separation	**Connection Type**	No Separation
Joint Connection	Body-SiC Seal Face	**Joint Connection**	SiC Seal Face-SS 316
Connection Type	No Separation	**Connection Type**	No Separation

Table 5: Connections for Current Model Stress Analysis

The differences in these contact settings determine how the contacting bodies can move relative to one another. That is the most common setting and has the most impact on what other settings are available. Most of these types only apply to contact regions made up of faces only. (Lawrence, 2007)

i. Bonded — This is the default configuration for contact regions. If contact regions are bonded, then no sliding or separation between faces or edges is allowed. Think of the region as glued. This type of contact is determined on the mathematical model, any gaps will be closed and any initial penetration will be ignored.

ii. No separation — This contact setting is similar to bonded case. It only applies to regions of faces. Separation of face in contact is not allowed, but small amounts of frictionless sliding can occur along contact faces.

iii. Frictionless — This setting models standard unilateral contact, that is, normal pressures equals zero if separation occurs. It only applies to regions of faces.

Thus gaps can form in the model between bodies depending on the loading. This solution is nonlinear because the area of contact may change as load is applied. A zero coefficient of friction is assumed, thus allowing free sliding. The model should be well constrained when using this contact setting. Weak springs are added to the assembly to help stabilize the model in order to achieve a reasonable solution.

iv. Rough — Similar to the frictionless setting, this setting models perfectly rough frictional contact where there is no sliding. It only applies to regions of faces. By default, no automatic closing of gaps is performed. This case corresponds to an infinite friction coefficient between the contacting bodies.

v. Frictional — In this setting, two contacting faces can carry shear stresses up to a certain magnitude across their interface before they start sliding relative to each other. It only applies to regions of faces.

2.2.3 Mesh

3D Mesh is the process ofdividing the set of nodes that comprise the surface mesh (or edges and faces) into a number of subsets, where each subset is also connected.

Mesh has a great influence on the solver convergence of every analysis. Several parameters that reflect the quality of the mesh are available including aspect ratio and centroid/face angle. For this structural stress analysis, the aspect ratio must be as low as 15.

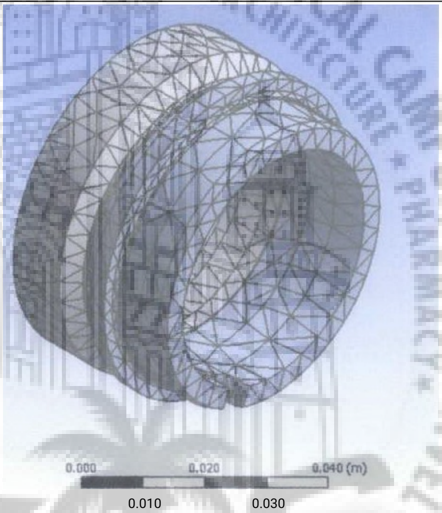
Model	Current model
Mesh Figure	
Number of Nodes	61334
Number of Elements	28545

Table 6: Current Model Meshing Element

2.2.4 Fixities

Based on the overall local and global design, the following fixities should be assumed in the analysis:

Fixities	Purpose
Cylindrical support	Fixed in radial and axial direction but free in tangential direction for rotational purpose.
Cylindrical support	Represent a pin function between carbon and flange. Provide support on radial, axial and tangent direction.
Displacement	0 displacement on x component but free on y and z component.

Table 7: Type of Fixities Used in FEA Analysis

22.5 Applied Load

Current Model	
	
Load Type: Rotational Velocity	Load Type: Pressure
Magnitude: 308.4 rad/s	Magnitude: 7.2 MPa

Table 8: Applied Load on Current Model Structural Stress Analysis

2.3 Thermal Analysis

The purpose of the analysis is to study the thermal response of Seal Face structures due to Heat Flux and Convection.

2.3.1 General Settings

The general settings for this analysis are:

Setting	Description
Initial Temperature Value	95C
Solver Type	Direct
Analysis Type	3-D
Thermal Strain Effect	YES
Reference Temperature	By Environment

Table 9: Thermal Analysis General Settings

2.3.2 Connections

The connection features functional to specify the contact conditions between surface before loading, and as part of the final solution to verify the transfer of loads
(forces and moments) across the various contact regions.

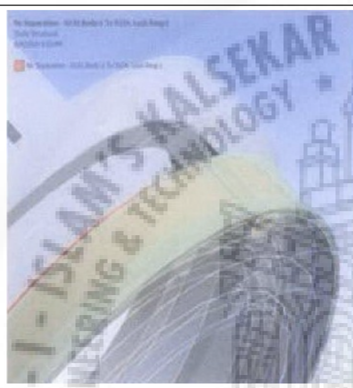
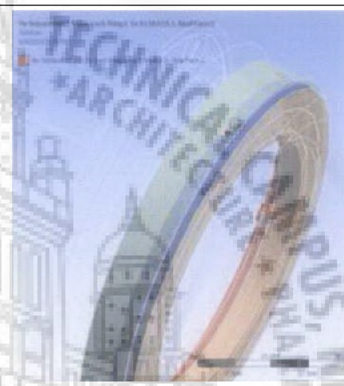
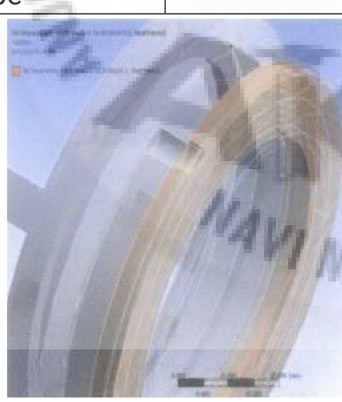
Current Model			
			
Joint Connection	Lock Ring-Body	Joint Connection	Lock Ring-SiC Seal Face
Connection Type	No Separation	Connection Type	No Separation
			
Joint Connection	Body-SiC Seal Face	Joint Connection	SiC Seal Face-SS 316
Connection Type	No Separation	Connection Type	Frictional

Table 10: Connections for Current Model Thermal Analysis

2.3.3 Mesh

3D Mesh is the process of dividing the set of nodes that comprise the surface mesh (or edges and faces) into a number of subsets, where each subset is also connected. For this analysis, the meshes are the same with structural analysis in

Table 6.

23.4 Applied Load

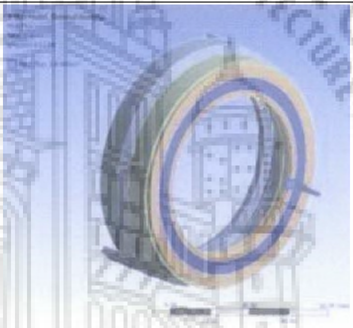
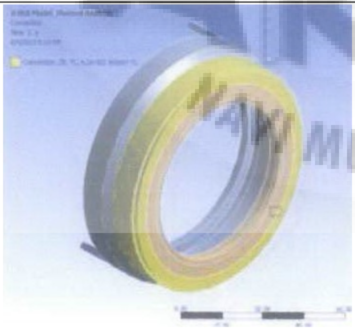
Current Model	
	
Load Type: Thermal Condition	Load Type: Heat Flux
Magnitude: 95^{0}C	Magnitude: 3.8W/mm^2
	
Load Type: Convection	
Magnitude: 4.2E-2 W/mm^{20}C	

Table 11: Applied Load on Current Model Thermal Analys

2.4 Fluid Flow Analysis

The purpose of the analysis is to study the circulation of flushing fluid on the Seal Face.

2.4.1 General Settings

There are some settings configurations in order to simulate the real conditions of the analysis as shown in the table below.

Setting	Description
Ambient Temperature	950C
Advection Scheme	High Resolution
Analysis Type	
Turbulence Numeric	First Order
Convergence Residual Type.	RMS
Equation Class	Continuity

Table 12: Fluid Flow Analysis General Settings

2.4.2 Fluid Flow Model

The model of the fluid flow is made based on the cavity of the seal as the fluid will go through the hollow spaces in the seal.

Figure 14: Fluid Flow Cavity Model

2.4.3 Mesh

Based on the cavity model, the meshes are generated to divide the model into nodes and elements.

In this type of CFD simulation especially, mesh quality is very important. Therefore, CFX-mesh is used to generate the volumetric area of the cavity model. The aspect ratio can be up to 1500 for investigating the flow of the fluid in the seal.

Number of nodes: 204388
Number of elements: 1007174

Figure 15: Mesh on the Cavity Model

2.4.4 Boundary Conditions

The boundary conditions specified the location of the inlet, outlet, and rotating wall of the cavity model.

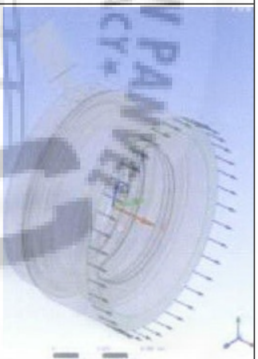
Current Model		
		
Boundary Type: Inlet Pressure	Boundary Type: Rotating Wall	Boundary Type: Outlet pressure
Magnitude: 126 Bar	Magnitude: 2950 RPM	Magnitude: 72 Bar

Table 13: Boundary Conditions for Current Model Fluid Flow Analysis

3. RESULTS OF THE ANALYSIS OF THE CURRENT SEAL FACE

As for the first step, analyses have been done for the current model of the mechanical seal. Types of analyses done are shown in the Table 14.

Type of Analysis	Function
Structural Stress Analysis	To locate the point that experiences the highest stress on the seal face due to high pressure and high temperature applications.
Deformation Analysis	To investigate the deformation effect and pattern and determine the maximum value of deformation that occurs during the application
Safety Factor Analysis	To identify the lowest safety factor applied, thus determining the safety of the seal face
Thermal Analysis (Inner Seal Face)	To determine the heat dissipations on the seal face due to the frictional force of the mating faces
Flushing Fluid Flow Analysis	To analyze the flushing fluid flow characteristic inside the mechanical seal

Table 14: Type of Analysis Done on the Current Model

3.1 Structural Stress Analysis

In this result, the stress concentration area on the analysed seal face model will be shown. As mentioned on the previous section, only Silicon Carbide (SIC) seal face will be studied, therefore the most suitable type of stresses for brittle material to be analysed are Maximum Principle Stress and Maximum Shear Stress.

For Current Model Design, two types of analyses will be performed. Those analyses are:

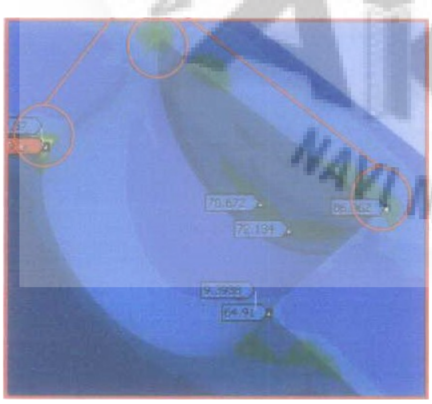
- Maximum Principle Stress Analysis
- Maximum Shear Stress Analysis.

3.1.1 **Maximum Principle Stress Analysis**

Below is the analysis done on the Current Model seal face for investigating the Maximum Principle Stress.

Figure 16: Maximum Prirhple Stress of Current Model

High stress concentration



For the Current Model of the seal face, the Maximum Principle Stress is 226.17 MPa. As shown in the figure, the highest stress location is on the lock ring slot area where the crack propagation comes from. The average principle stress concentration is only about 70.6 MPa. These sudden changes with a significant amount of stress reduction over certain localised area, in this case, it is on the corner slot, which will cause a high enormity of non-uniform deformation. This circumstance is ideal to initiate crack propagation on the inner back rotating seal face that will lead to failure.

3.1.2 **Maximum Shear Stress Analysis**

Figures below show the analysis result of Maximum Shear Stress for the Current Model existing rotating inner seal face.

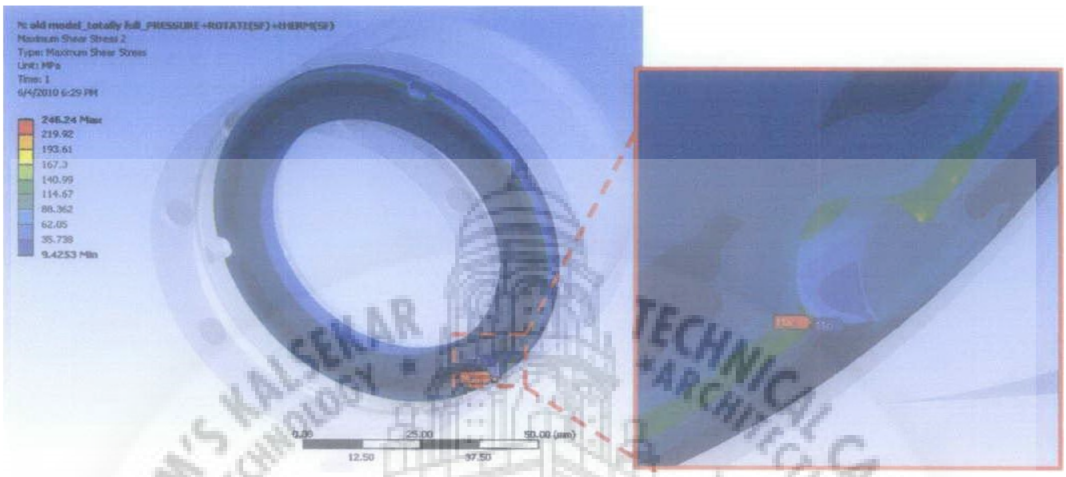
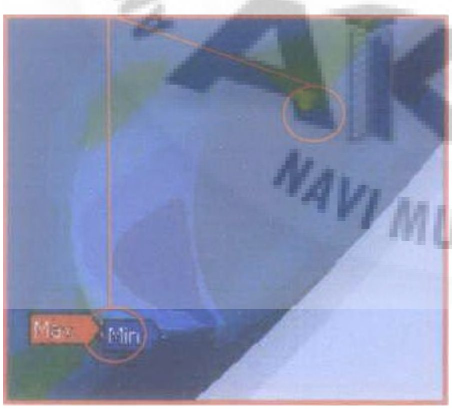


Figure 17: Maximum Shear Stress of Current Model



The maximum shear stress for the Current Model of the seal face is 246.24 MPa, which is also located on the lock ring slot area. This shows that the area, especially on the corner slot experiencing both high principal and shear stress at the same time. The

Maximum Shear Stress occurs at the corner of the slot because of its square vertices design that acts as a shear concentrating point. The fact is sharpness of the vertex has been identified as the crack starting point which is leading to the initiation of crack propagation and leads to failure of a seal.

3.2 Deformation Analysis

In this result, the analysis performed is Total Deformation Analysis. The purpose of performing the total deformation analysis is to investigate the deformation that occurs on the seal face in a 3-dimensional direction.

The figure below shows the total deformation analysis for the Current Model design seal face. The "RED" indicates the highest magnitude of deformation.

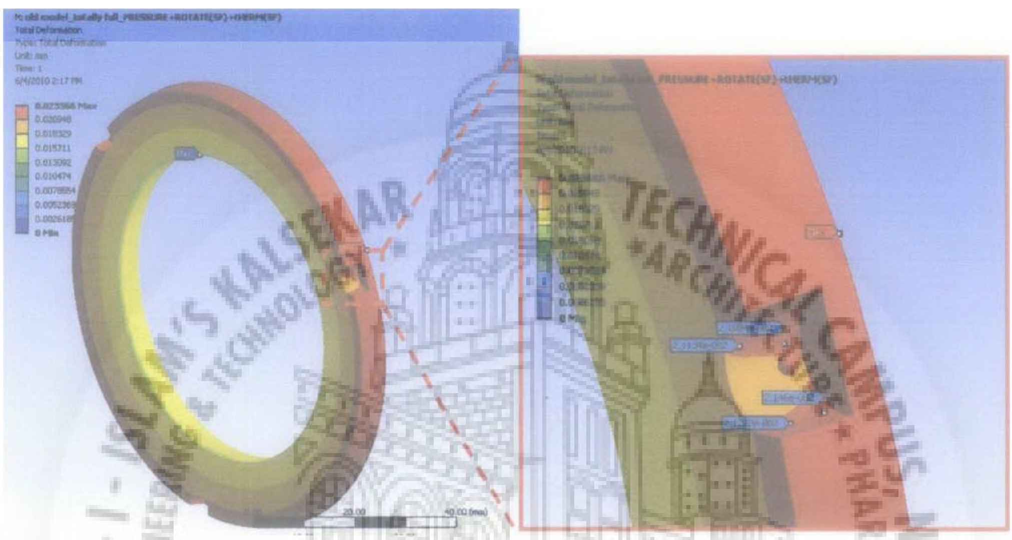
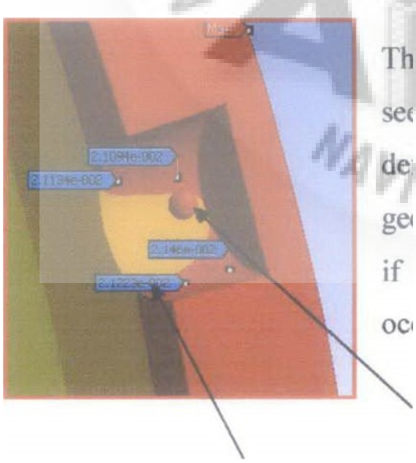


Figure 18: Total Deformation of Current Model



The maximum deformation value is 0.0236 mm. As seen from the figure besides, the non-uniformity deformations are suspected to cause a instability of the geometry's shape, that could lead to a crack initiation if there is any pressure surge or temperature spike occur.

Red Area indicates higher amount of deformation

Orange Area indicates lower amount of deforrnation

3.3 Safety Factor Analysis

In this analysis, Mohr Coulomb Safety Factor was chosen as it is the best used for brittle material.

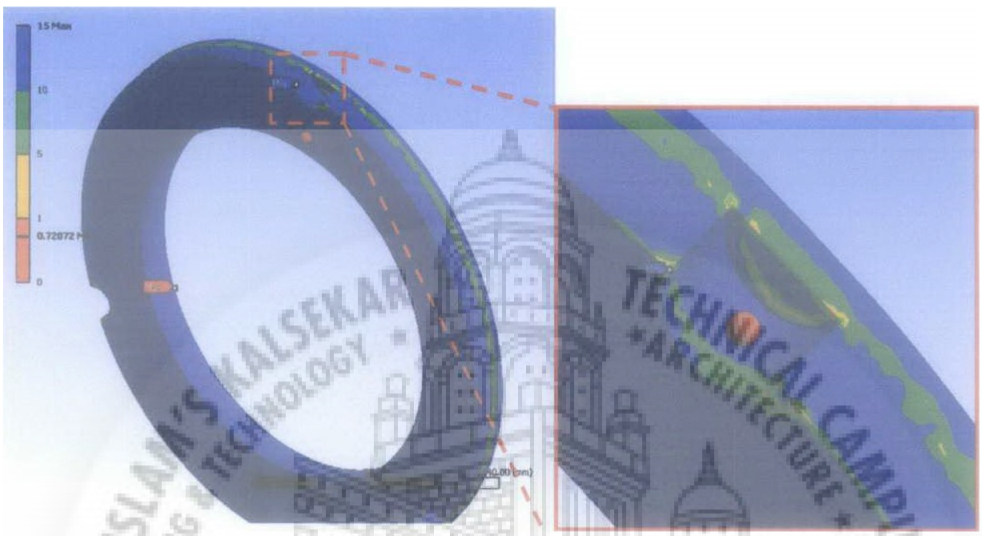
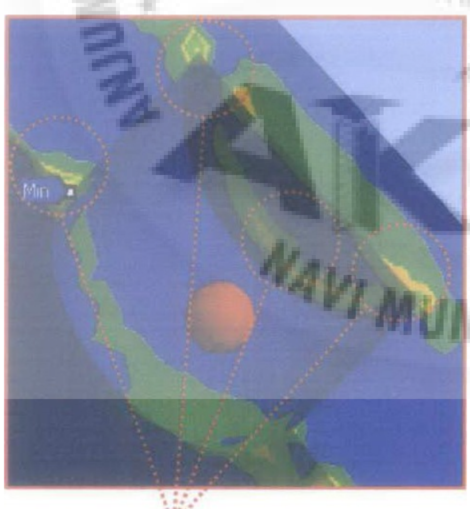


Figure 19: Mohr Coulomb Safety Factor of Current Model

As shown on the figure above, the minimum safety factor is 0.721, which is below the limit of the safe area. Although in this application, the location of the safety factor is on the end slot area, the other corner of the slot area also experiencing low safety factor (the "YELLOW" area indicates the safety factor merely above l). This pattern proves that in case there is any pressure surge, or temperature spike, the slot area will have the highest possibility to experience crack propagation. YELLOW Area indicates the area with a low safety factor.



3.4 Thermal Analysis

In this analysis, the ambient temperature is applied to analyze the heat dissipations of the seal faces due to the frictional force of the mating faces. The following figures show the result of the analysis for the current seal face.

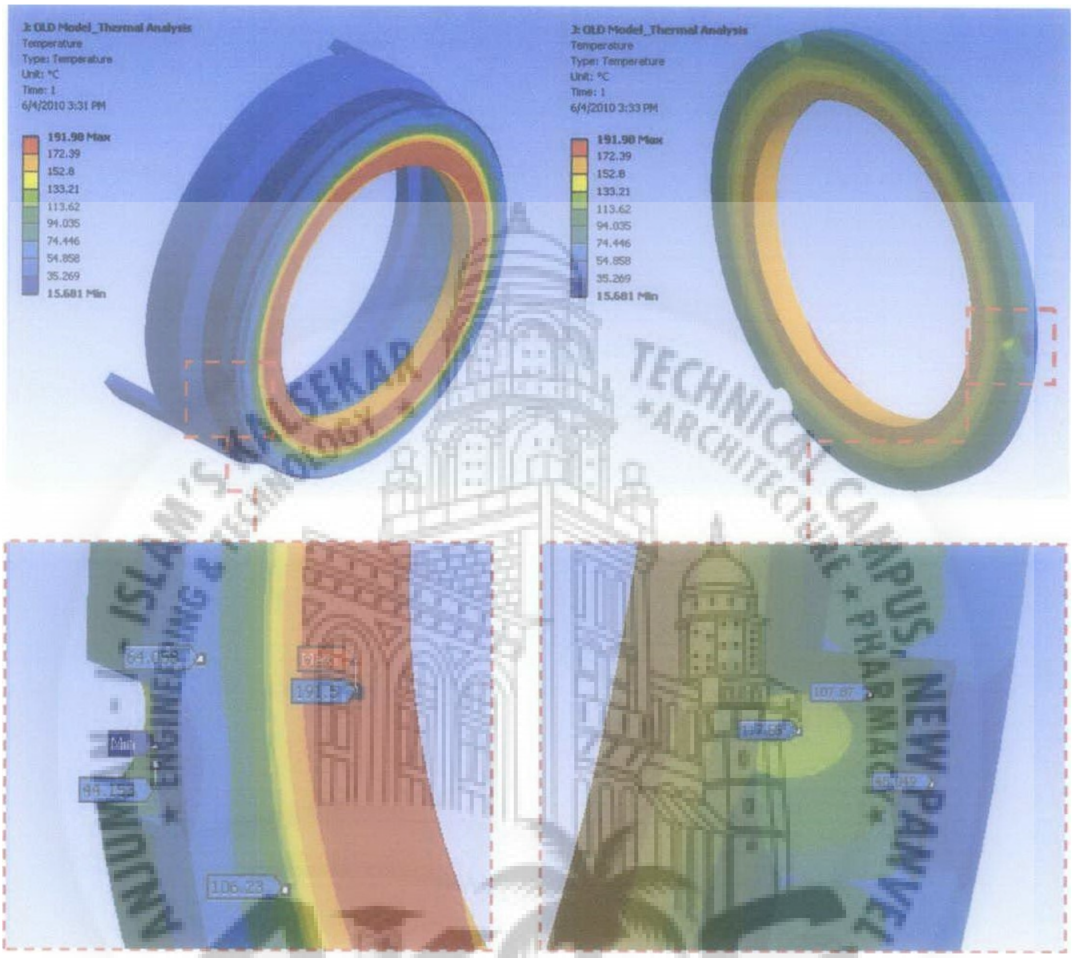


Figure 10: Maximum Temperature Point for Current Model

From the result above, the maximum temperature generated at the rotating seal face is 191.98 ^{c}c. On the slot area side, the temperature differences are extremely significance, as on the backside, it is averagely about 1 17.55 ^{0}C. While on the front side, the temperature is only about 48.05 ^{o}c. This is because of the ineffective cooling system, as the flushing fluid's flows on the backside are almost stagnant due to the blockade of the lock ring, which will be shown in the next section, making the heat trapped more. Therefore, the material expansion would be uneven as the back slot area expands more than the front slot area. This will induce crack propagation, which will lead to seal failure.

3.5Flushing Fluid Flow Analysis

The purpose of this analysis is to investigate the pattern of the flow to the inner seal face area, before performing the modification to optimize the fluid's flow to the heat dissipated around the seal faces. For the current model design, the analysis was performed by assuming that the flow's convection coefficient, h is constant as the seal face itself is rotating at a constant speed, directly hit by the constant flow of the flushing fluid. Below is the result of the flow analysis:

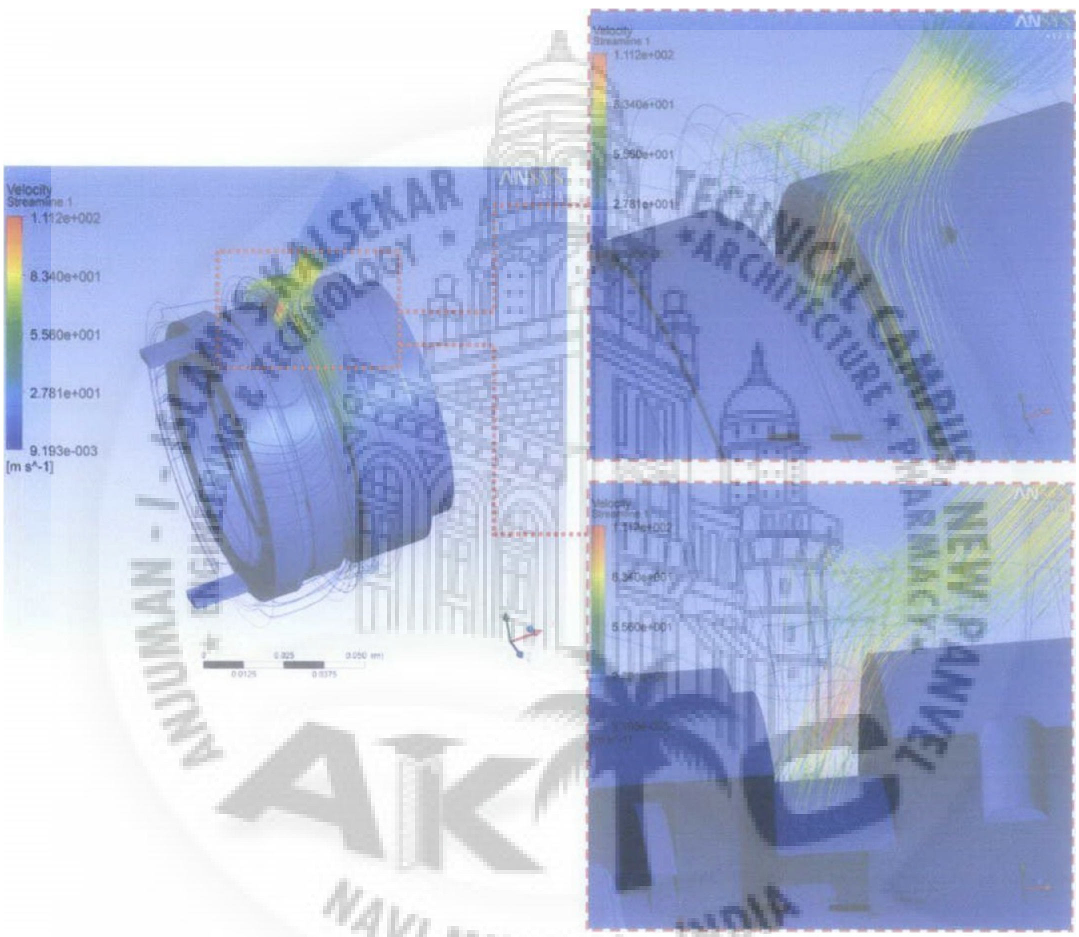


Figure 11: Flushing Fluid Flow for Current Model

As shown from the result above, the maximum velocity of the fluid's flow is about 90 m/s, which is located on the outlet of the flushing hole. The average flow's speed is 40 m/s, considerably high enough to provide good cooling. However, for this current seal face model, it is quite significant that the seal face's area was not total covered by the high velocity flushing fluids, as the back side of the rotating seal face was covered by the lock ring, making the fluid's cannot pass through it. This will certainly make the heat trapped in the backside, that will cause uneven expansion that may lead to crack propagation, as the corner of the slot area which experiencing the highest stress, is also positioned on the backside of the rotating seal face.

4. MODIFICATIONS OF THE SEAL FACE DESIGN

Next in line are the modifications of the seal face design to improve its capability. The first modification made is by thickening up the length of the seal face. As the previous seal face's thickness is ().195 in^2, the new modified seal face's

thickness is 0.354 in^2. According to Principle of Stress,c = (Y) —11-10 this will 10 linearly increase the strength ofthe seal face by 80%.

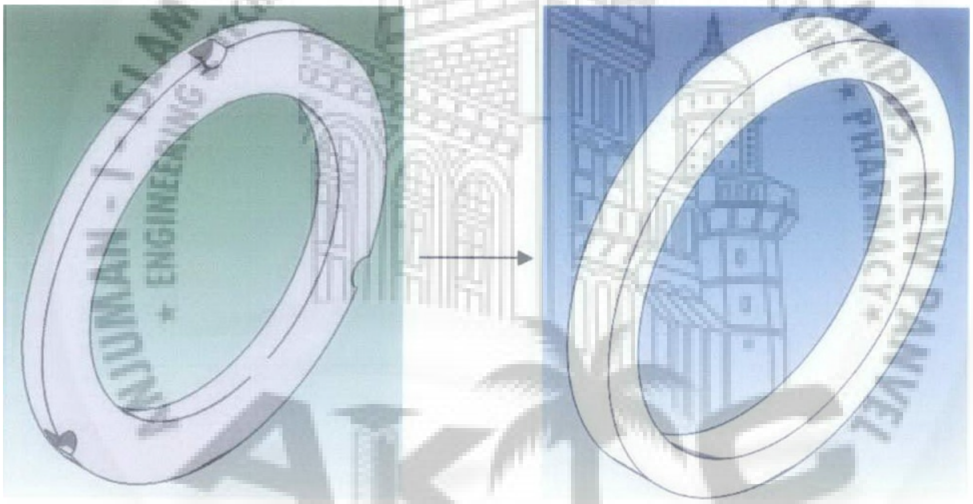


Figure 12: First Modification on the Seal Face

Next modification is the removal of the drive slot. The drive slot on the new model has been removed to enhance the strength of the seal face by eliminating the main source crack propagation. Instead, the seal face will be shrink — fitted with the metal body (SS 3 1 6), whereas the drive slot has been positioned on it.

The changes are shown in the figures below.

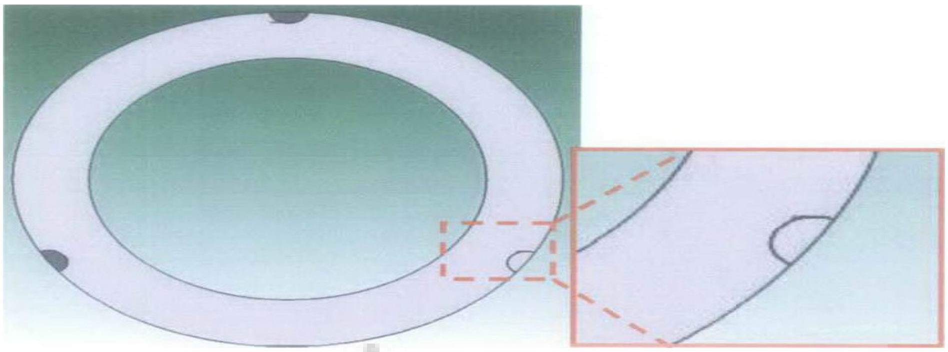


Figure 13: Current Seal Face Model with Slot Design

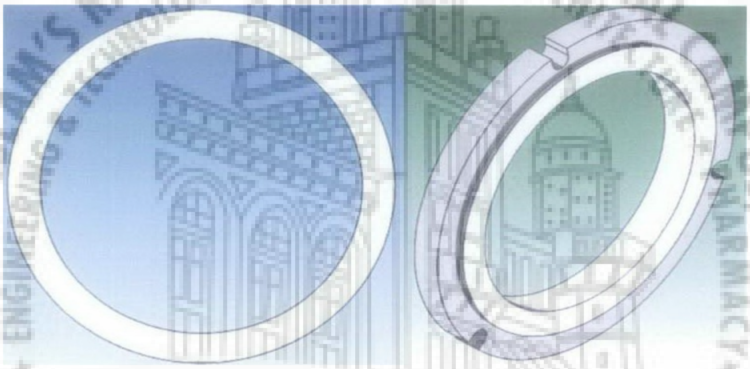


Figure 14: Assembly of New Shrink-Fitted Seal Face

To ensure that this new seal face that use shrink-fitted mechanism will able to handle high temperature, several calculations has been made to ensure that the metal body of the seal face will still hold the Silicon Carbide seal face at the highest temperature point of application.

Below is the result of the deformation of the seal face metal body. The deformation is linear to the increasing of temperature. As the temperature limit for the applications are 95 ^{0}C, and the safety factor taken for thermal deformation on the seal face are 3,the tolerance taken are 0.008 inch, where the temperature limit is 300°c

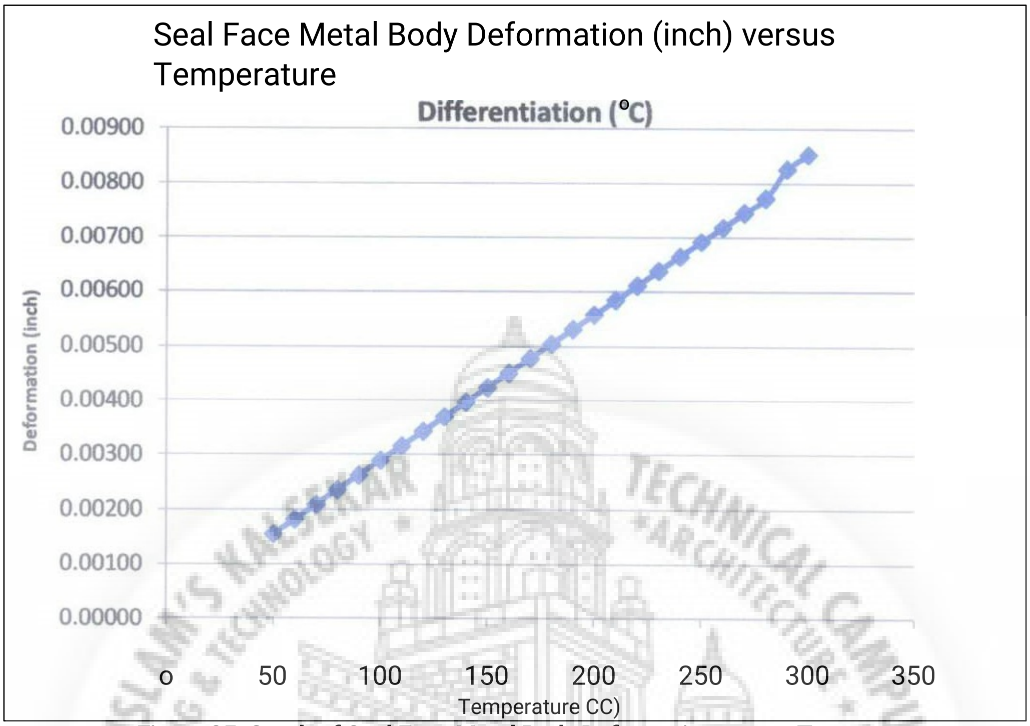


Figure 25: Graph of Seal Face Metal Body Deformation versus Temperature

There is a certain amount of stress need to be applied in order to deform the Seal Face's Metal Body. The same stress with equal magnitude will act on the Silicon Carbide (SIC) Seal Face as this seal face metal body tends to shrink during the cooling process after press fitting the SIC Seal Face. Therefore, Finite Element Deformation Analysis had performed to determine the amount of deformation made by certain amount of Stress. Graph below shows the Applied Stress versus the seal face metal body:

The result shows that the applied stress is linearly to the deformation. As the applied pressure reach to 23MPa, the maximum deformation of the metal body is only about

0.008 inch. The maximum resultant stress due to the applied pressure As the compression strength of the Sintered Silicon Carbide is about 3900 MPa, it is enough to withstand the compression load due to shrinking metal body.

5. RESULTS OF THE ANALYSIS OF THE MODIFIED SEAL FACE

After the modifications have been made, another analysis is done to ensure that the new design is fit and can serve its purpose to enhance the capability of the seal face. The type of analyses is shown in the table below.

Type of Analysis	Function
Structural Stress Analysis	To locate the point that experiences the highest stress on the seal face due to high pressure and high temperature applications.
Deformation Analysis	To investigate the deformation effect and pattern and determine the maximum value of deformation that occurs during the application
Safety Factor Analysis	To identify the lowest safety factor applied, thus determining the safety of the seal face
Thermal Analysis (Inner Seal Face)	To determine the heat dissipations on the seal face due to the frictional force of the mating faces
Flushing Fluid Flow Analysis	To analyze the flushing fluid flow characteristic inside the mechanical seal

Table 14: Type of Analysis Done on the Modified Model

5.1 Structural Stress Analysis

In this result, the stress concentration area on the analysed seal face model will be shown. As mentioned on the previous section, only Silicon Carbide (SIC) Seal Face will be studied, the most suitable type of stresses for brittle material to be analysed are Maximum Principle Stress and Maximum Shear Stress.

For the modified design, 2 types of analyses will be performed. Those analyses are:

i. Maximum Principle Stress Analysis

ii. Maximum Shear Stress Analysis.

5.1.1 Maximum Principle Stress Analysis

Below is the figure of the result of Maximum Principle Stress Analysis for the proposed modified design. The "RED" area shows the maximum magnitude of the Principle Stress.

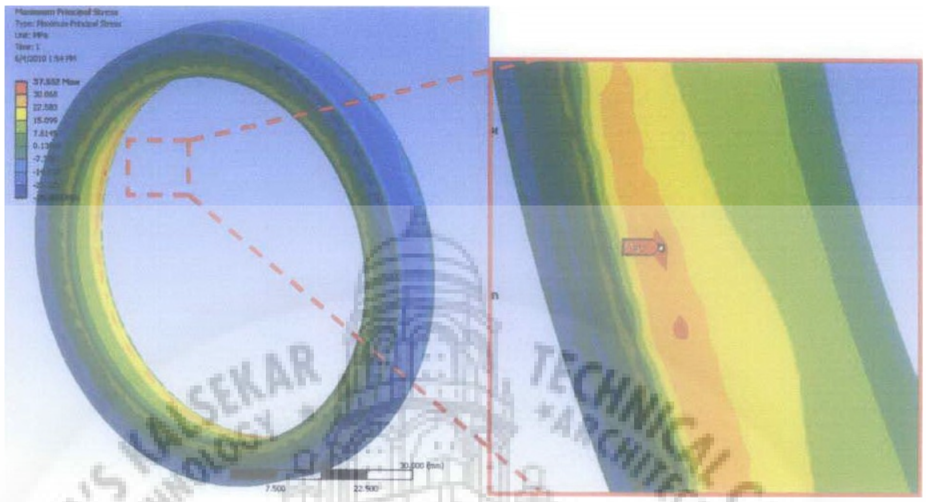


Figure 15: Maximum Principle Stress for Modified Model

For the modified design, the maximum principle stress is 37.552 MPa. The magnitude of this stress is about Six (6) times than the principle stress of the current seal face. Furthermore, the location of the maximum stress is not on the stress concentrated area, as it becomes hard to initiate the crack propagation.

5.1.2 Maximum Shear Stress Analysis

Below is the figure of the result of Maximum Shear Stress Analysis for the proposed modified design. The "RED" area shows the maximum magnitude of the Shear Stress.

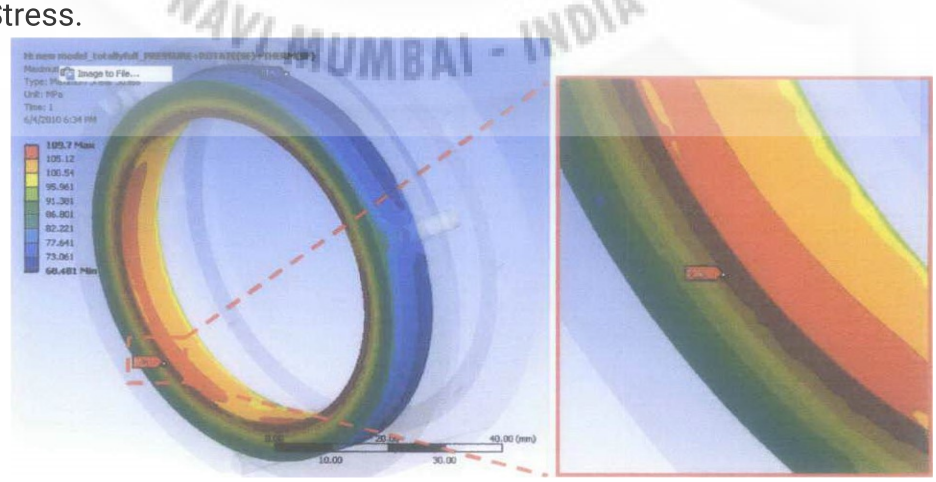


Figure 16: Maximum Shear Stress for Modified Model

The maximum shear stress for the Proposed Modified Seal Face is about 109.7 MPa, located at the edge of the seal face. Following the same case like the previous principal stress, the maximum shear stress on the proposed modified design is about half of the previous current seal face design.

5.2 Deformation Analysis

In this result, there were 2 types of analyses performed. Those analyses are:

i. Total Deformation Analysis

ii. Directional Y-axis Analysis

The purpose of performing the total deformation analysis is to investigate the deformation that occurs on the seal face in a 3-dimensional direction. However, the directional Y-axis analysis was performed to investigate the deformation that occurs on the seal face in radial direction.

Below is the figure of the result of Total Deformation Analysis for the modified design.

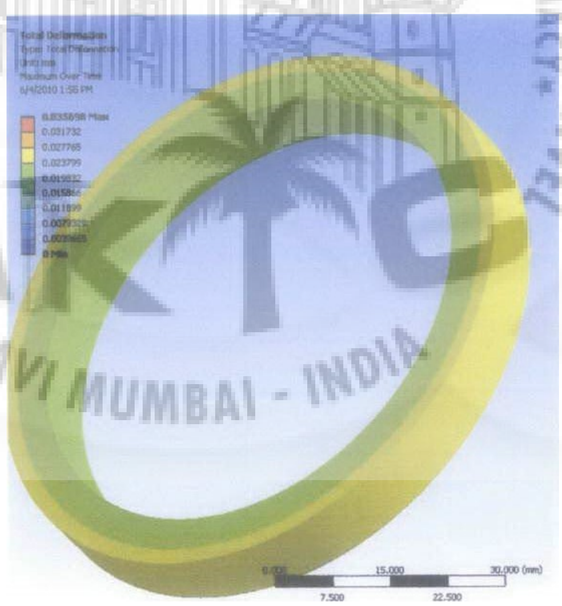


Figure 29: Total Deformation for Modified Model

The maximum deformation is 0.0225 mm which is equals to 0.0018 inches.

5.3 Safety Factor Analysis

In this analysis, Mohr Coulomb Safety Factor was chosen as it is the best used for brittle material. The result of this analysis will determine whether the design is failure or not.

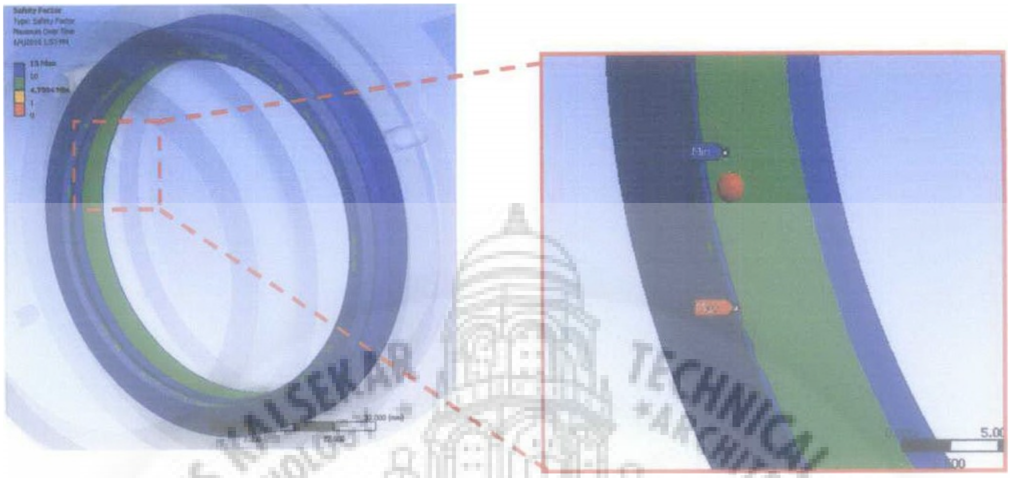


Figure 17: Mohr Coulomb Safety Factor for Modified Model

The minimum safety factor for the new design is about 4.799, which is obviously beyond the limit of the safe area, which is l .

5.4 Inner Seal Face Thermal Analysis

The purpose of this analysis is to analyse the heat generated on the mating seal face due. This analysis is performed to both current seal face model and the modified model.

After the modification has been made, this analysis has been performed to check for the improvement of the new seal face. Below are the results of the new modified design. The heat generation capacity was maintained equal to the current seal face model as the mating faces' contact area was remain the same as there was no modification made on the stationary seal face.As shown in the figure above, the maximum temperature is 132.64 ^{o}c. The temperature then reduced to 70 ^{o}c averagely. The temperature had decreased in a uniform pattern to the outside diameter of the rotating seal face. The improvement of the cooling system had also influenced the temperature of the seal face as the maximum temperature of the seal face has been reduced from 191.98 ^{0}C to 132.64^{0}C.

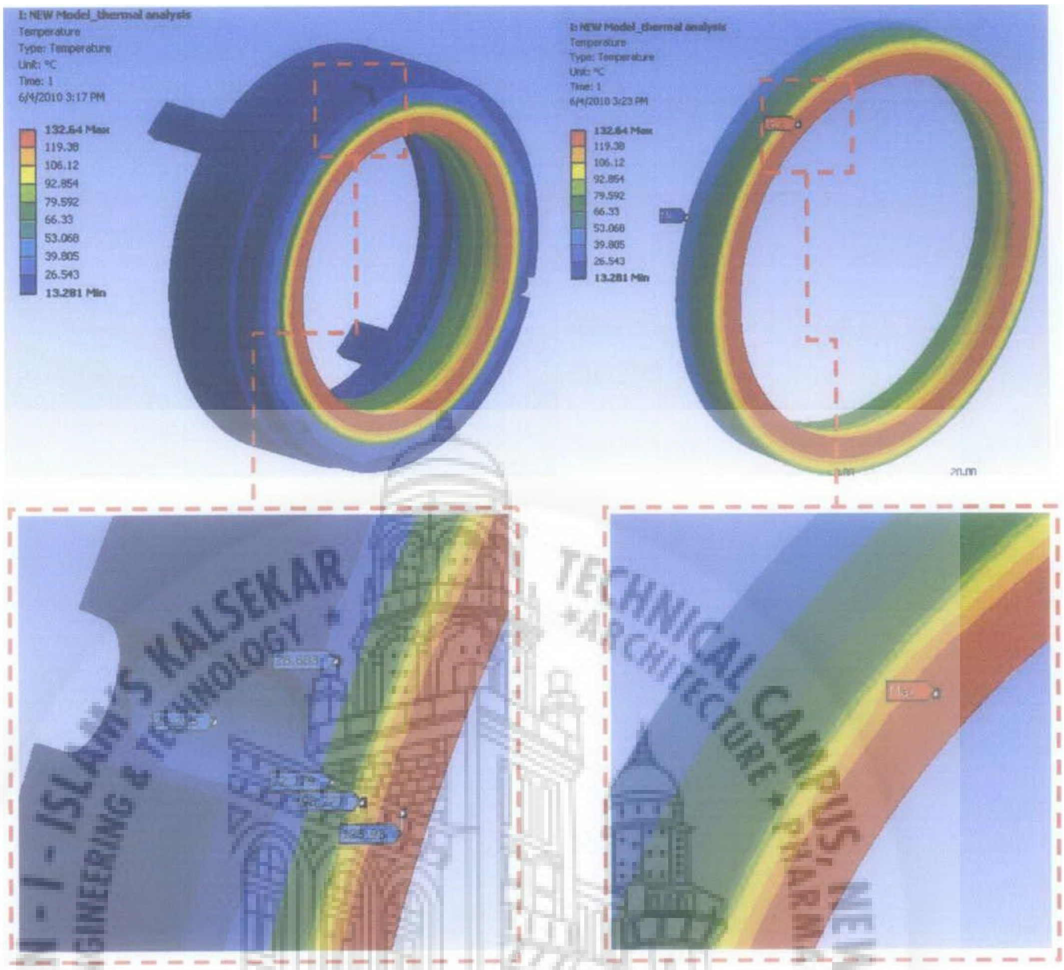


Figure 18: Thermal Analysis for Modified Design

5.5 Flushing Fluid's Flow Analysis

The analysis was performed for both current seal face design and for the modified design. The purpose of this analysis is to investigate the pattern of the flow to the inner seal face area, before performing the modification to optimize the fluid's flow to the heat dissipated around the seal faces. Some modifications have been made including by changing the locking mechanism, which is previously by using the lock ring, to by using the drive pin. This will eliminate the heat trap problem, which is located at the backside of the rotating seal face. Below is the result of the fluid's flow:

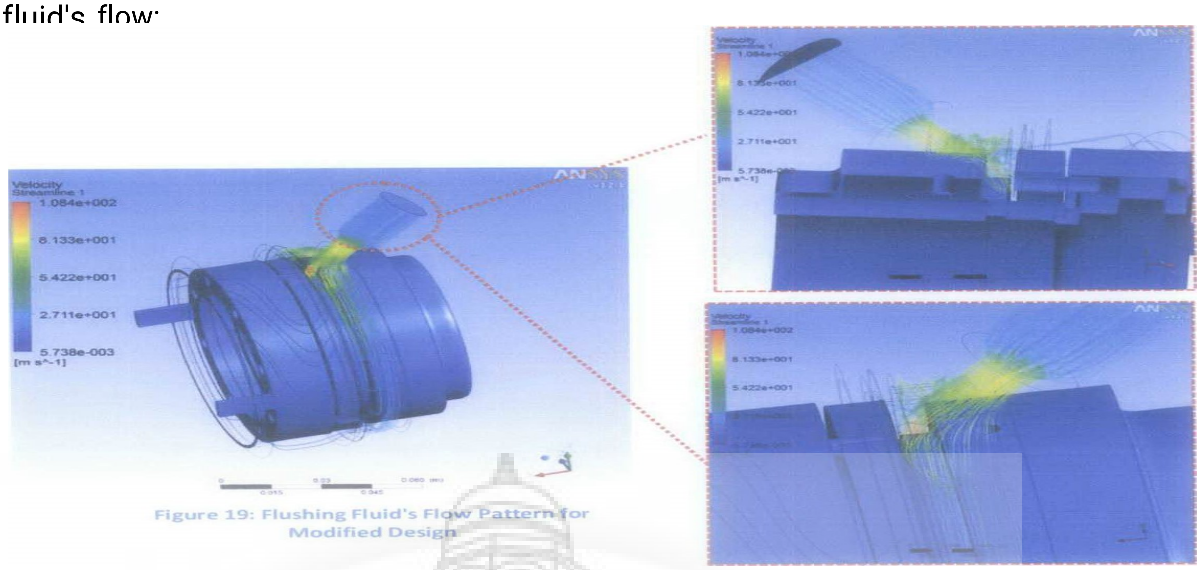


Figure 19: Flushing Fluid's Flow Pattern for Modified Design

From the result above, the maximum flow's velocity is about 90 m/s, and the average is about 50 m/s. The flow's pattern had cover up the entire seal face, including the backside of the rotating seal face, eliminates the problem of the previous current seal face model.

6. CALCULATION

6.1 Balance Ratio Calculation

The purpose of this Balance Ratio Calculation is to determine whether the new seal face model is balance or unbalanced. In this application, the pressure of 72 bar is considered as high pressure application. Therefore, the seal face must be in balanced form.

Equation of Balance Ratio:

$$B = \frac{r_b^2 - r_i^2}{r_o^2 - r_i^2}$$

New Model Specification: =0.7075 inch

r_i — 1.2500 inch $r0^2$

= 1.4488 inch

<u>B = 0.819 (Balanced)</u>

6.2 Thermal Expansion Calculation

Thermal Expansion Calculation for determining the Exact Tolerances for Shrink Fit. The purpose of this Thermal Expansion Calculation is to calculate the deformation of the metal body (Stainless Steel 316). From this calculation then, the amount of tolerances will be decided to ensure that the Seal Face (Sintered Silicon Carbide) will not slip from metal body due to high temperature applications. The calculation will be taken based on the internal diameter of the Seal Face Metal Body.

R-Ro

——— — a (AT) (Fluigent, 2010)

New Seal Face Model Specification:

Taking temperature differences, AT = I 0^{O}C

Initial Internal Diameter, I .471 inch

Thermal Expansion, a = 16.2 x 10-5 m/m.^{0}C (Stainless Steel 316)

Deformation, R-Ro= 0.00238 inch

However, the equation above only for 2-Dimensional calculation, as the actual thermal expansion involves 3-Dimensional, finite element calculation is required. Thus, Steady-State Thermal Analysis was performed using Finite Element Analysis (FEA).

6.3 Heat Generated Calculation

The purpose of this calculation is to find the amount of heat generation produced by the mating seal faces with the rotating speed of 2950 RPM.

1) Face pressure, Pf= AP (B-K) + Psp (psi)

Constant: Coefficient of friction, g = 0.1

PO = 72 bar

Pi = I bar

$- P_i = 71$bar

= 1029.76 psi

Seal area balance, B = 0.802

Pressure gradient, K 0.65

Psp = 30 psi (assumed)

∴ P_f = (1029.76) (0.802 - 0.65) + 30

= 156.52 psi + 30 psi

= 186.52 psi

2) Mean Velocity, Vm = $\frac{\pi \cdot n \cdot D_m}{60}$ (m/s)

$D_m = \frac{\text{onærn}}{2}$

$= \frac{2.953+2.441}{2} = 2.697$ in

$V_m = \frac{\pi(2950)(2.697)}{12} = 2082.92$ ft/min

3) Pressure-Velocity, PV = Pf X Vm

= 186.52 psi (2082.92 ft/min)

= 388505.34 psi-ft/min

4) Energy Consumption, HP = $\frac{\text{pv./1.A}}{33000}$

$= \frac{PV \cdot \mu \cdot [\frac{\pi}{4}(OD^2 - ID^2)]}{33000}$

$= \frac{\text{P7.ß.fi}(2.9532-2.4412)]}{33000}$

$= \frac{388.\text{sos}.34 \cdot 0.1 \cdot 2.169^2}{33000}$

= 235674 HP

5) Heat Generated = 2 x 2546

= 6501.655btu/hr

= 1905.45 w

6.4 Prandt'l Number Calculation

At 95 ^{0}C, thermal properties for crude oil are:

Thermal conductivity, k = 130 W/mK (for T > 50 K)
Density, p = 750 kg m ^{3}
Specific heat, cp= 74.1 kJ/kg
Kinematic viscosity, v 6.3 x 10 ^{7}m^{2}/s

6.3 x 10-7 Prandtl

value $= \frac{130}{(750)(74100)}$

6.3 x 10 -7

0.000002339 =

0.26

6.5Heat Convection Coefficient Calculation

I) Calculate Reynold's Number

Reynold's number, Rel) = V (Di+2t)

60 (0.062+2(0.0059))
6 x 10—7

60 (0.0738)

= 7 380 000 :.Turbulent

2) ReD.Pr= 7380000(0.26)= 1.92 x 10^{6}

Thus using a single comprehensive equation that covers the entire range of Ret) for which the data is available,

5/8 4/5

$$\overline{Nu_D} = \frac{0.62\, Re_D{}^{1/2} Pr^{1/3}}{[1 + (0.4/Pr)^{2/3}]^{1/4}} \left[1 + \right.$$

(ReD
!282000

$$= \frac{0.62\,(7.38\times10^{6})^{1/2}(0.26)^{1/3}}{[1+(0.4/0.26)^{2/3}]^{1/4}}\left[1+\left(\frac{7.38\times10^{6}}{282000}\right)^{5/8}\right]^{4/5}$$

$$= \frac{(1075.0062)}{[1+(1.3327)]^{1/4}}[1 + (7.6936)]^{4/5}$$

$$= \frac{(1075.0062)}{(1.236)}(5.641)$$

= 4878.469

3) Convection coefficient,

$$h = (\overline{Nu}_D)\left(\frac{k}{D}\right)$$

$$= 4878.469\,\frac{137\times10^{-3}}{0.0738}$$

= 8 593 509 W/m^2.K

6.6 Calculation for Heat Generation Due to Viscous Drag

I) Calculate the tangential velocity,

$$U_t = \frac{\pi\cdot D\cdot n}{60}$$

$$= \frac{\pi\,(0.075)(2950)}{60}$$

= 11.585 m/s

2) Calculate the Reynold's number,

$$Re = \frac{U_t\cdot c\cdot\rho}{\mu}$$

$$= \frac{11.585(0.0264)(750)}{(6\times10^{-7})}$$

= 7 380 000 :.Turbulent

3) Calculate the loss coefficient,

$$C_M = 0.065\cdot\left(\frac{c}{R}\right)^3\cdot Re^{-0.2} \quad \text{for Re > 10000}$$

$$= 0.065\cdot\left(\frac{0.0264}{0.0369}\right)^3\cdot(7\,380\,000)^{-0.2}$$

$$=0.065(0.7154)^3(0.042)$$

$$=0.000999561$$

4) Calculate heat generated due to viscous drag,

$$P = \frac{1}{2} C_m \rho U_t^3 \pi L R$$

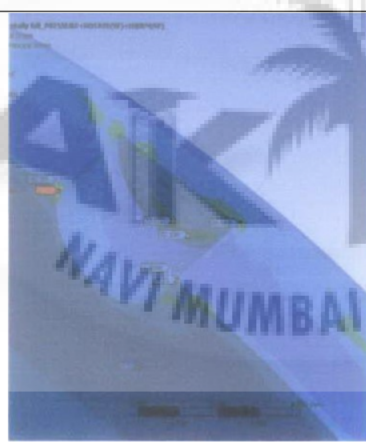
$$= \frac{1}{2}(0.000999561)(750)(11.4)^3 \pi (0.0089)(0.0369)$$

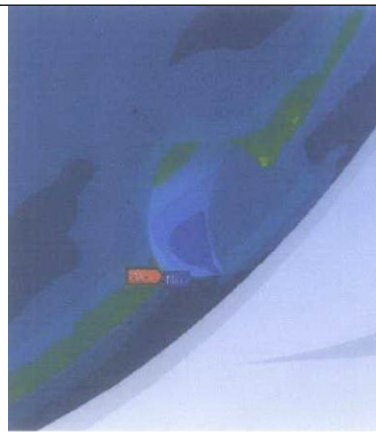
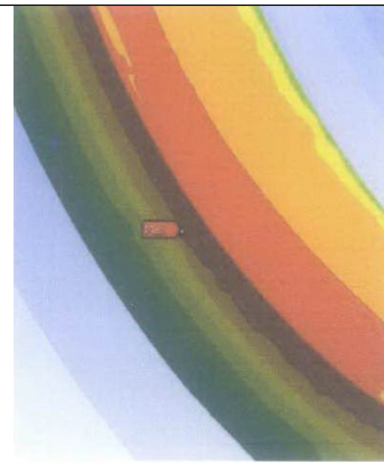
- 0.573 W < I, therefore neglected.

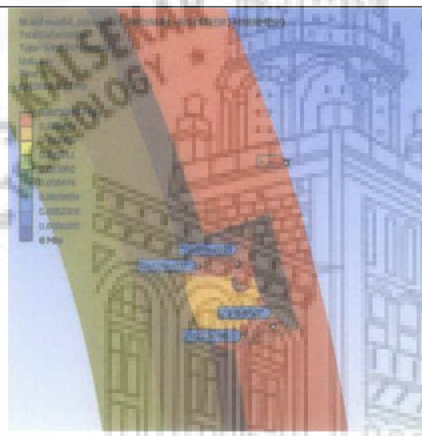
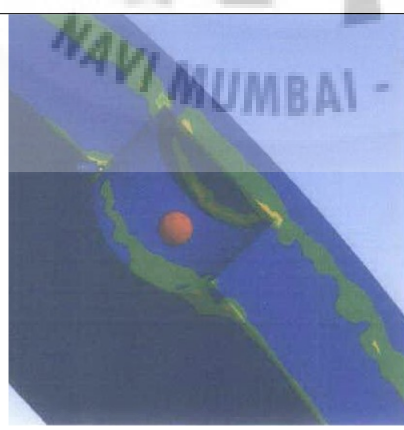
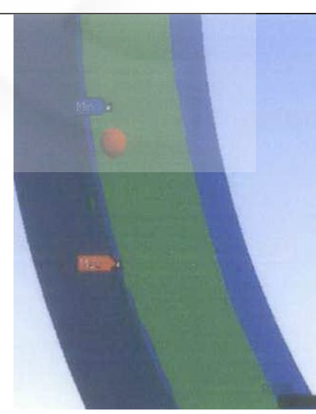
6.7 ANALYSIS SUMMARY

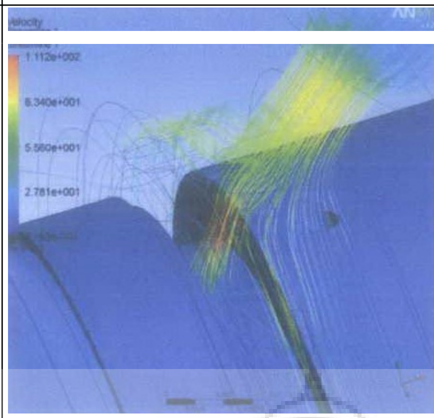
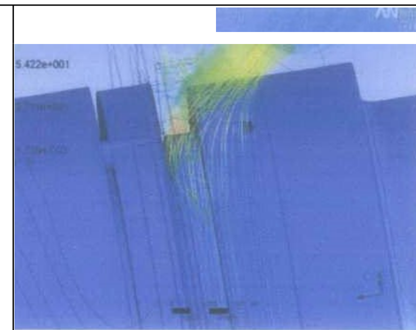
This section summarizes the analyses that were performed for both current model and for the modified model. A detail comparison can be made to evaluate the significance of improvement for the proposed modified design.

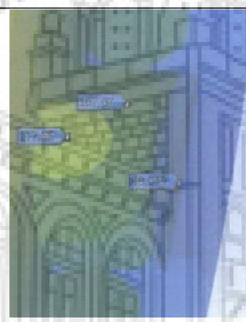
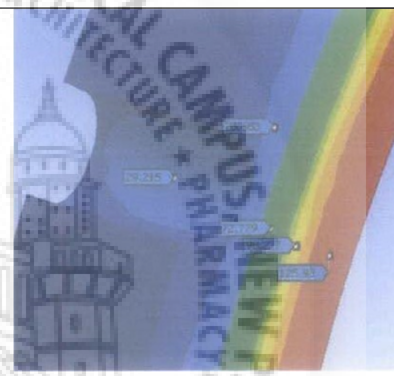
Table **IS: Comparison of Analysis between Current Seal Face and Modified Design**

Description	Current Seal Face Model	Modified Seal Face Model
Maximum Principle Stress Analysis		
Values	Max Stress : 226.17 MPa Colour Indication : Red	Max Stress : 37.552 MPa Colour Indcation : Red

Maximum Shear Stress Analysis		
Values	Max Stress : 246.24 MPa Colour Indication : Red	Max Stress : 109.7 MPa Colour Indication : Red

Maximum Total Deformation Analysis		
Values	Max Deformation : 0.0236 mm Colour Indication : Red	Max Deformation : 0.0225 mm Colour Indcation : Red
Mohrs Coulomb Stress Safety Factor Analysis		

Values	Min. Safety Factor : 0.721 Colour Indcation : Red	Min. Safety Factor : 4.799 Colour Indcation : Red
Flushing Fluid's Flow Analysis		
Values	Max Flow Velocity : 90 m/s Colour Indication : Red	Max Flow Velocity : 90 m/s Colour Indication : Red

Inner Seal Face Thermal Analysis		
Values	Maximum Temperature : 191.98 •C Colour Indication : Red	Maximum Temperature : 132.64 'C Colour Indcation : Red

CHAPTER 5

CONCLUSION

From the research I have done so far, seal face is the most common cause of mechanical seal's failure after certain period of installation, range from two months as the fastest time to failure's occurrence up to several months depending on the pump's condition. Therefore the analyses on the seal face itself are essential.

From the analysis above, on the maximum principle stress, the maximum magnitude of stress for the current seal face design is 226.17MPa and the maximum magnitude of principle stress for the modified design is only 37.55MPa. The difference is significantly large, which is about 188.62MPa. This proves that the existence of the lock ring slot area is one of the causes of seal failure due to crack propagation, caused by high stress concentration. Therefore, by eliminating the slot area, the weakness point that was suspected to be the main cause of this seal failure is also diminished.

On the maximum shear stress analysis, the highest magnitude of shear stress for the current seal face model is 246.24NfPa. The location of the highest shear stress for the current model is also on the hock ring slot area. Therefore, it was proven that by redesign back the rotating seal face by eliminating the slot area, the maximum shear stress has been reduced to 109.7MPa.

On the total deformation analysis, the pattern of deformation on the slot area was not in a uniform shape as on the centre of the slot area indicates the "ORANGE" coloured area (Figure 18), as the average deformations are about 0.0184mm, while the edge of the slot, indicates the "RED" in colour (Figure 18), averagely 0.02346mm. This non-uniformity expansion will also cause high stress concentration that may lead to seal failure. The deformation on the newly proposed modified design seems in a uniform pattern, as the colour distribution that indicates the magnitude of the deformation was evenly distributed. Therefore, this concludes that the newly designed seal face has a better uniformity of deformation which make it more stable and more durable to high stress and rapid changes of deformation.

The Mohr Coulomb safety factor for the current seal face model is 0.721, which is clearly indicates the seal failure as the minimum safety limit is l . The location of the minimum safety factor is at the seal's lock ring slot area, where the highest stress concentration also founded. For the modified model, the minimum safety factor is 4.799 which are about 5 times of the minimum safety limit. This is because the slot area for the drive pin has been positioned on the metal body, which has ductility criteria that will prevent from crack propagation initiation due sudden high pressure spike.

In the flushing fluid flow analysis, for the current seal face compression system, the cooling fluids are ineffectively convect the heat that generated from the seal face. As discussed before, it is because of the existence of lock ring, blocking the flushing fluids to flow at the backside of the rotating seal face in high speed, causing the heat to clog up. The newly modified design, had increased the cooling efficiency as the locking mechanism by using lock ring has been change to drive pin, which is not provide the blockage for the cooling fluids to enter on the backside of the rotating seal face in a high speed. This proves as in the thermal analysis, the highest temperature has been reduced from 191.98 ^{0}C to 132.64 ^{0}C. Therefore the modified design has increased the performance of a pump in terms of flow capability of the fluid while maintaining the average velocity at the outlet.

As a conclusion, the newly modified seal face design has been proven to solve problem that aroused on the current seal face design as it has a better strength, better heat circulation, and more durable.

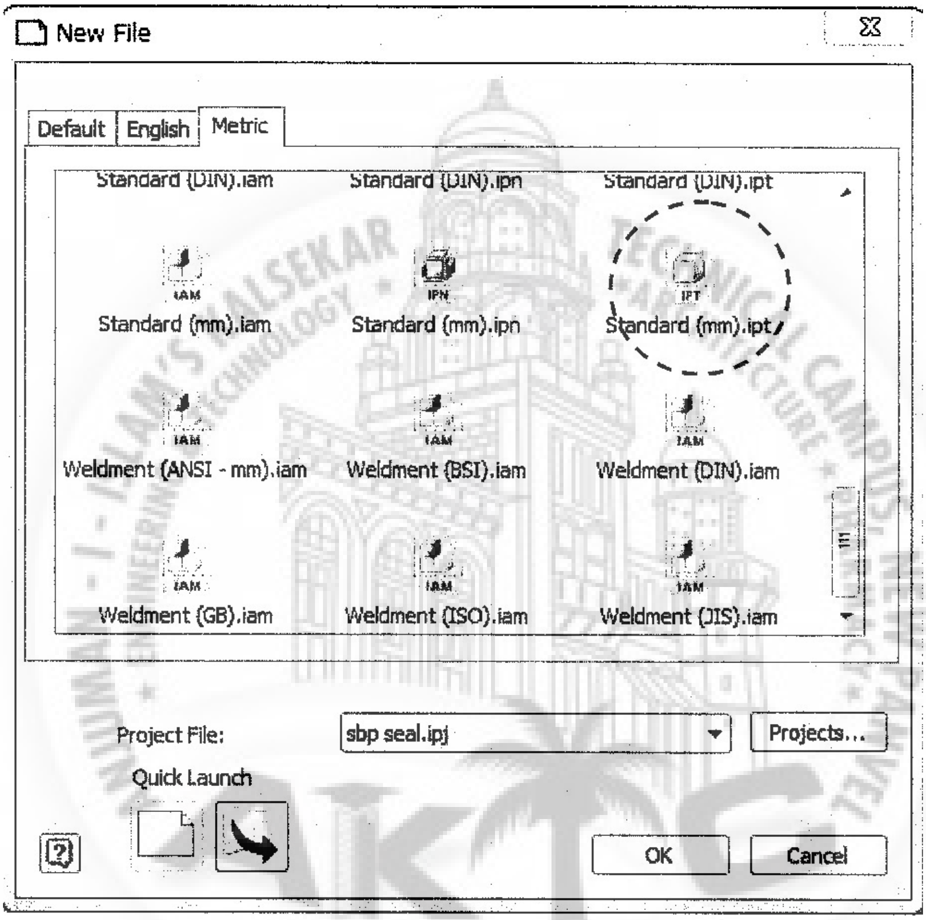
APPENDIX 6.1 :

APPENDIX A - SAMPLE OF DRAWING MANUAL

(STATIONARY SEAL FACE)

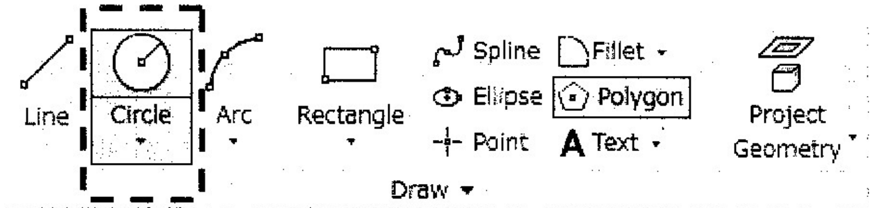
1.Open Autodesk Inventor Professional.

2.Create New file
3.Select Standard ipt.mm

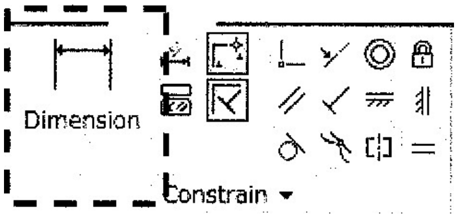


4. Click OK.

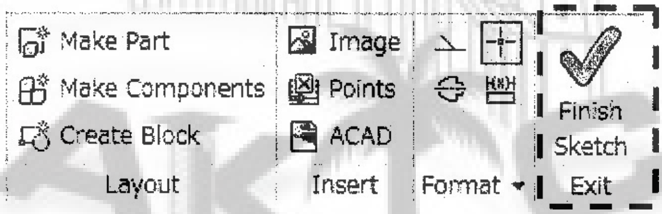
5. Select circle button from the Draw toolbar on the top-left corner.



6. Select any point as the center of the circle on the drawing and click while drag outside the center point to draw a circle.
7. Make another circle with the same center point but with different diameter.
8. To specify the diameter of the circle, select Dimension button on the Constrain toolbar at the top-center of the screen.

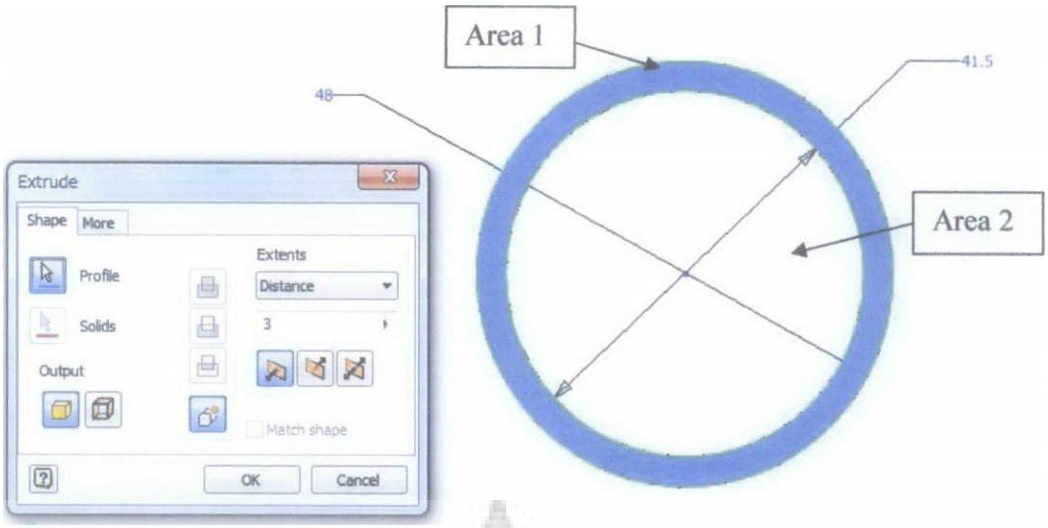


9. Select the circle and click outside once to make the dimension appear.
10. Repeat step 9 for another circle.
11. Left-click on the value for editing, and change the value to 48mm for the outer circle.
12. Press Enter.
13. For inner circle, change the value to 41.5mm.
14. Press Enter.
15. To adjust the size of the circle for better view, scroll down or up using the mouse.
16. Select Finish Sketch button to exit the sketch mode at the top-right corner on the Exit toolbar.



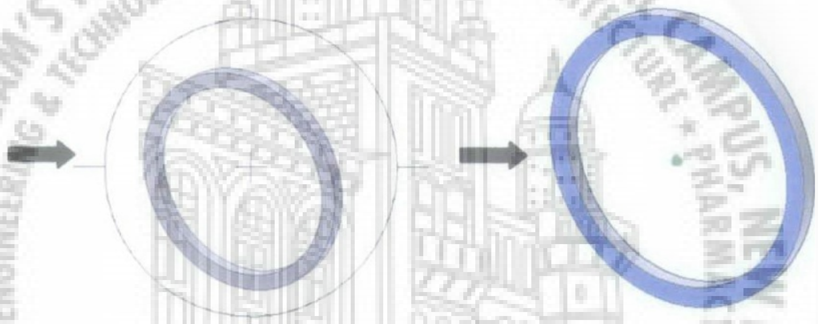
17. Select Extrude button from the Create toolbar at the top-len corner.

18. Then select Area I only and change the value to 3mm.

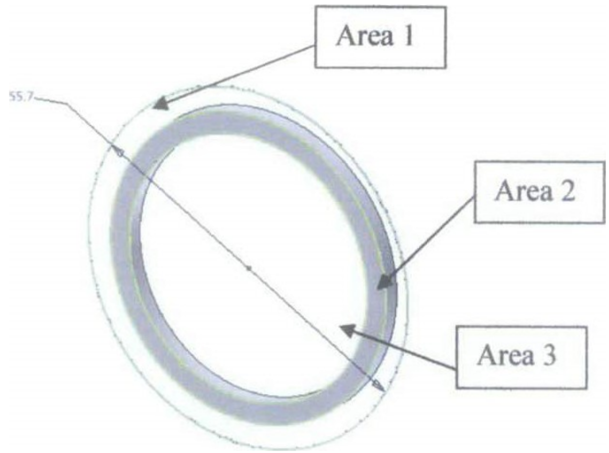


1.5

19. Select OK.
20. Select Orbit button from the right corner, then rotate the model into 3D view.
21. Select the front face of the model.

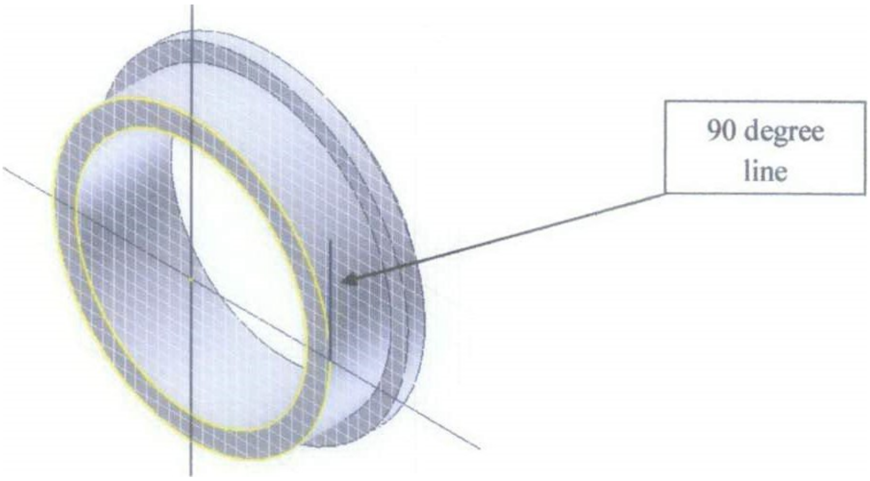


22. Enter 2D sketch mode again by using Create 2D Sketch button on the top-left of the screen.
23. Create another circle using the same center point with the dimension of 55.7mm.
24. Exit sketch mode.
25. Select Extrude and make sure Area I and Area 2 is selected, then change the value to 3mm also.



26. Select the new generated face and enter the sketch mode again.
27. Using the same center point as before, draw another circle with the diameter of 49.7mm.
28. Exit sketch mode.
29. Select Extrude and select Area I only.

30. Change the value to I Imm and select OK.
31. Select the newly generated face and enter the sketch mode.
32. Select Line button next to the Circle button and point to any corner of the circle.
33. Make a short 90 degree line perpendicular to the corner.



34. Exit sketch mode.

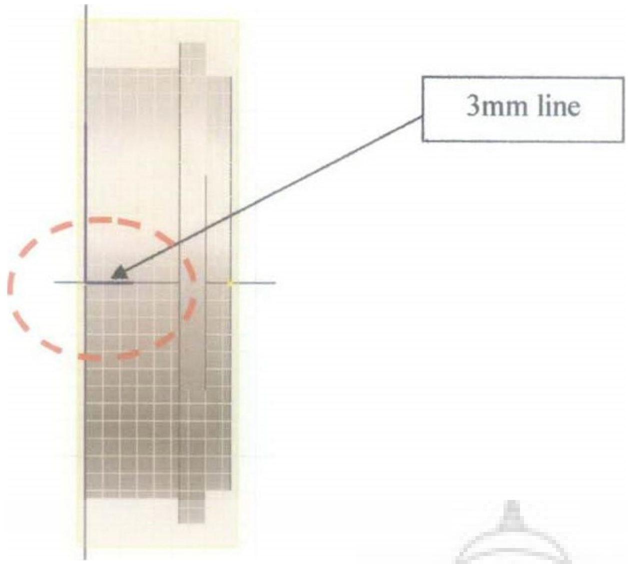
35. Select Plane button on the Work Features toolbar to create a new work plane.

36. Click on the line created before and once again to create the work plane.

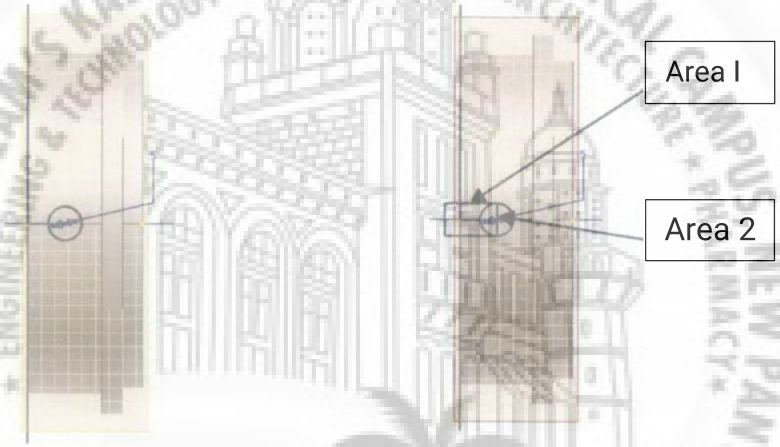
37. Rotate the model to get the side view.

38. Select the new work plane and enter sketch mode.

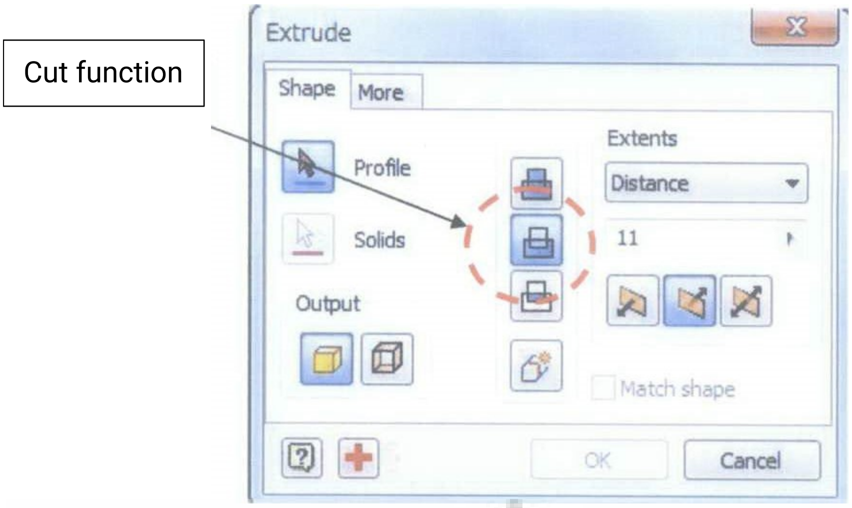
39. Make a straight line from a center of the model with the dimension of 3mm.



40. Create a circle with a diameter of 5mm using the right-end of the line as the center.
41. Then, add a rectangular using the circle diameter as the width.



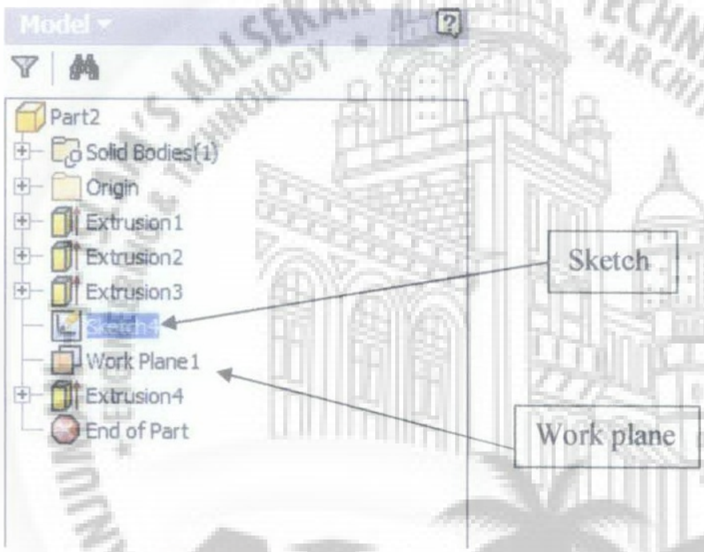
42. Exit sketch mode.
43. Select Extrude and select Area 1 and Area 2.
44. Select Cut function on the Extrude window to remove the area selected.
45. Change the value to 10mm.
46. Select OK.



47. Right-click on the Sketch on the Model panel at the left side of the screen.

48. Un-tick the visibility option.

49. Repeat Step 48 for the Work Plane on the Model panel.



50. Rotate the complete model to get the best view.

APPENDIX 6.2:

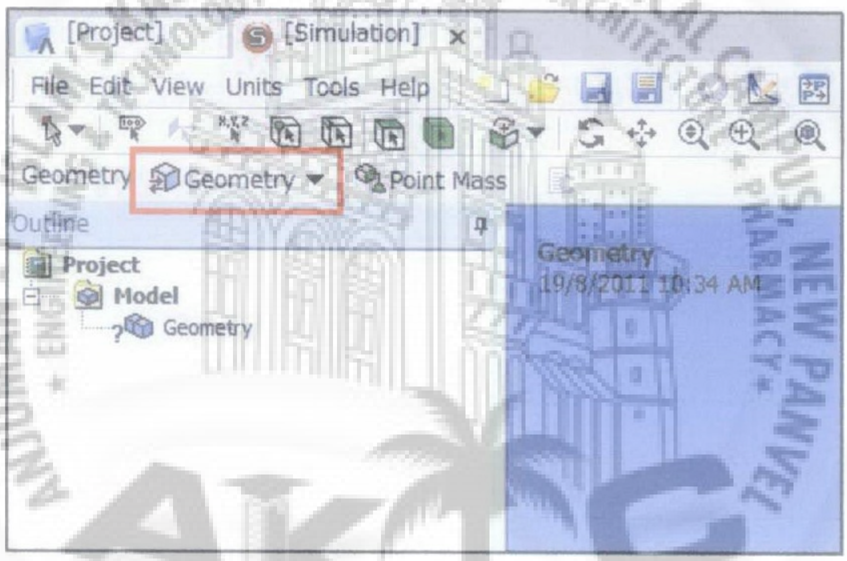
APPENDIX B - SAMPLE OF SIMULATION MANUAL

(ROTATING SEAL FACE)

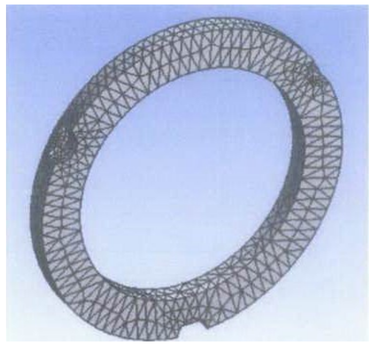
1. Run ANSYS Workbench 1 1.0

2.Crete new Simulation file.

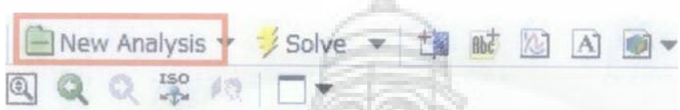
2. Import the seal face model by clicking Geometry tab at the upper right corner at the window.



3. Click on the Geometry icon and select 'Programme Controlled' under Definition characteristic.
4. Right-click on Mesh icon and select Generate Mesh.
5. For the better meshing element sizes, changes may be made by adding features such as Refinement and Method to Mesh icon before generating mesh.
6. The seal face will be meshed according to the settings applied.



7. Select New Analysis from the toolbar at the top.



tions

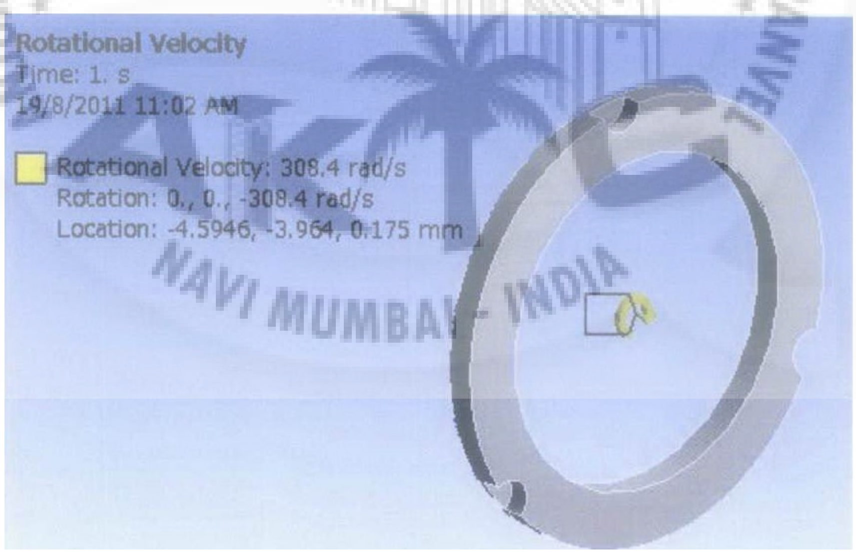
8. Select Static Structural.

9. Under Analysis Settings, make sure all the Program Controlled function is selected for the parameters in the details.

I I . Under Environment toolbar, select Inertial and insert Rotational Velocity.

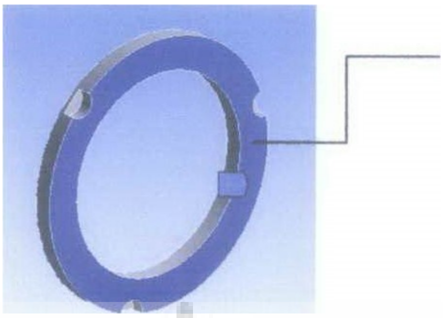
12. Select the whole body of the seal face and click apply at the Geometry under the details.

13. Specify the magnitude of the velocity.



Rotation: o.t o.t -308.4 rad/s
Location: -4.5946, -3.964, 0.175 mm

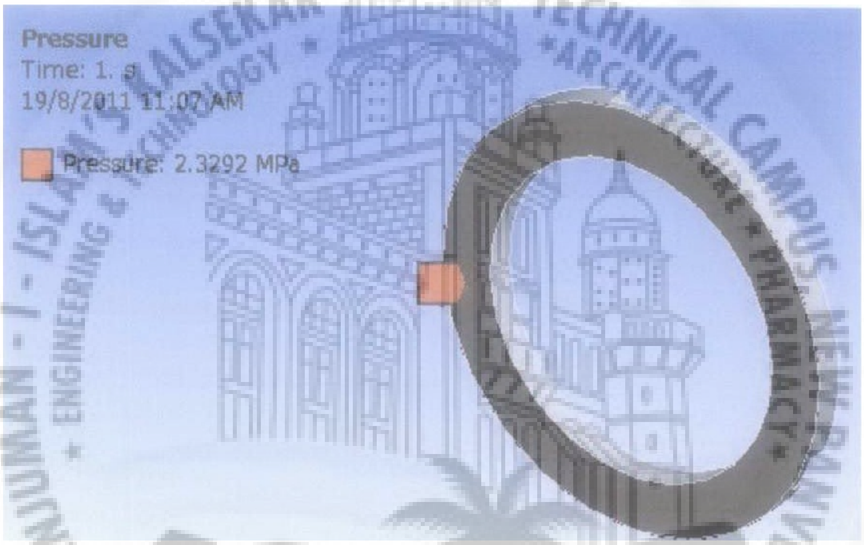
14. Select Fixed Support under Supports toolbar and apply on the front surface of the seal face.



one surface selected

15. Under Loads toolbar, add Pressure and select the face opposite of the fixed surface before.

16. Specify the magnitude of the pressure.

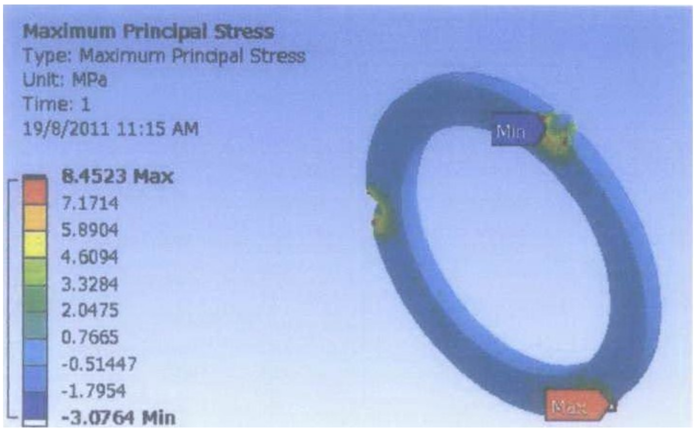


17. Click on the Solution folder icon.

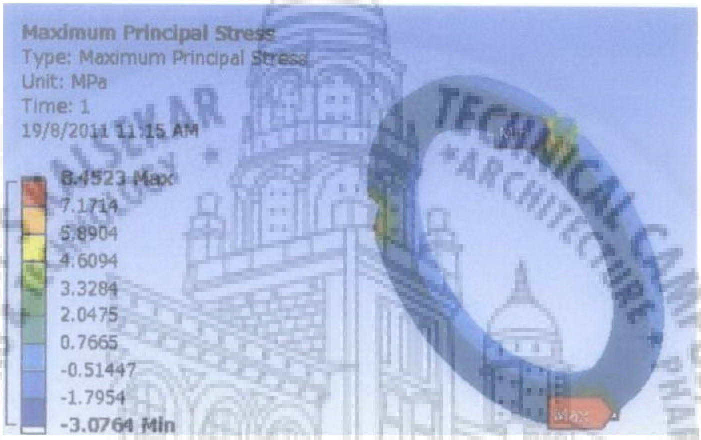
18. Under Solution toolbar at the top, select the result of interest to be shown, for instance total deformation, maximum stress, maximum strain etc.

19. Right-click the Solution folder icon again and select Solve All.

20. Result may be viewed by clicking on the respective type of solution specified before.



Maximum principal stress



Maximum principal stress

21. For better view, model may be rotated using the rotate icon above or by clicking the center wheel of the mouse.
22. To view the minimum and maximum point of the analysis as shown in the figures above, enable the Max and Min button on the toolbar above.

Result 8.7e+003 (Auto Salt .	O		DProbe

23. Another type of analysis may be done by repeating from Step 8 but using another type of analysis required.

REFERENCES

1. Lawrence, K. L. (2007). Structural and Thermal Analysis using the ANSYS Workbench Release 11.0 Environement. Texas: SDC Publications.
2. Cengel, Y. A. (1998). Heat Transfer: A Practical Approach. McGraw-Hill.
3. Heinz K. Muller, B. S. (1998). Fluid Sealing Technology: Principles and Applications. M. Dekker.
4. Lebeck, A. O. (1991). Principle and Design of Mechanical Face Seals. John Wiley & Sons.
5. Mayer, E. (1977). Mechanical Seals. Newnes-Butterworth.
6. Ortoleva, P. J. (1994). Compartments and Seals. AAPG.
7. Publication, I. S. (2000). Reliability of Sealing Systems for Rotating Mechinery. Professional Engineering Publishing Limited for the Institution of Mechanical Engineers.
8. Summers-Smith, J. (2005). Mechanical Seal Practice for Improved Performance. John- Wiley & Sons.
9. Yokota Manufacturing Co., L. (11 May, 2010). Non-external Flushing Mechanical Seal. Retrieved 10 August, 2010, from Aquadevice.com: http://www.aquadevice.com/english/05material_mechaseal.htm