Redesign and Fabrication of a Mechanical Seal for Pumps

By

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CERTIFICATION OF APPROVAL

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A project dissertation submitted to the
Mechanical Engineering Programme
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BACHELOR OF ENGINEERING (Hons)
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Approved by,

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UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

January 2008

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.

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EMIER HELMEE MD DEWA

ABSTRACT

For this project, the main objective is to redesign a new mechanical seal that will last longer and reduce the possibility of leak. The redesign process is based on the previous case histories and the "sequence-of-events" records, which categorize the possible root causes for the mechanical seal failure. The problems for this mechanical that has been identified from "sequence-of-events" are it generates excessive heat at the mating face; leak and its design do not follow the standard of American Petroleum Institute 682. The excessive heat will lead to the leak of the pump when the heat melts the o-ring. Beside that, excessive heat will vaporize the sealing fluid. The bubble produced, will reduce the efficiency of the pump. The solution for excessive heat is by redesigning the rotary face. To reduce the effect of back pressure subjected on the rotary face, proper balance ratio be used as a basis to design this rotary face. For API standard, the seal face has been changed to the solid seal face instead of inserted seal face. Beside that, the deflection of the shaft is reduced by simplified the design and reduce the weight of parts that subjected on the shaft. The main outcome of the project is the fabrication of a new mechanical seal which has been tested under a pressure test based on API 682 standard. Consequently, all the design requirements been met successfully.

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INTRODUCTION

1.0 Introduction

Whenever a rotating shaft must pass between two regions containing different fluids and it is important to keep the fluids separated, a rotating shaft seal is needed. In general the fluids may be at different pressures and they may be either a gas or a liquid. The rotating shaft seal has the function of fitting around the shaft or some part connecting to the shaft in a way such that the leakage between the two regions is acceptably small under all circumstances of operation.

The mechanical face seal forms a barrier in the shape of an annulus. Leakage is blocked or slowed either by actual intact of the surfaces or by a very thin gap between the surfaces. Sliding of the surfaces occurs in a direction normal to the leakage flow.

Examples of shaft seals abound because nearly all rotating shafts either intersect multiple fluid media and/or are supported by bearings that in turn are sealed by a shaft seal. The basic types of mechanical equipment that utilize shaft seals are as follow [1]:

1. **Pumps.** Nearly all pumps must allow the entry of a shaft into the fluid without permitting excessive leakage.
2. **Compressors.** Centrifugal compressors and many other types as well use shaft seals.
3. **Bearings.** Shaft-type seals are used to seal the lubricant in all bearings, either directly as in a sealed bearing or by sealing lubricant in an entire assembly such as a gearbox.
4. **Powered Vessels**. Nearly all powered vessels are powered by propeller whose shaft must penetrate the hull.

1.1 Background of the Project

Pumps are being used in all petroleum plants to pump the fluids. The mechanical seal is the most important part in pump in respect to failure of pump. Figure 1 shows the percentage of pump failure root causes [2].

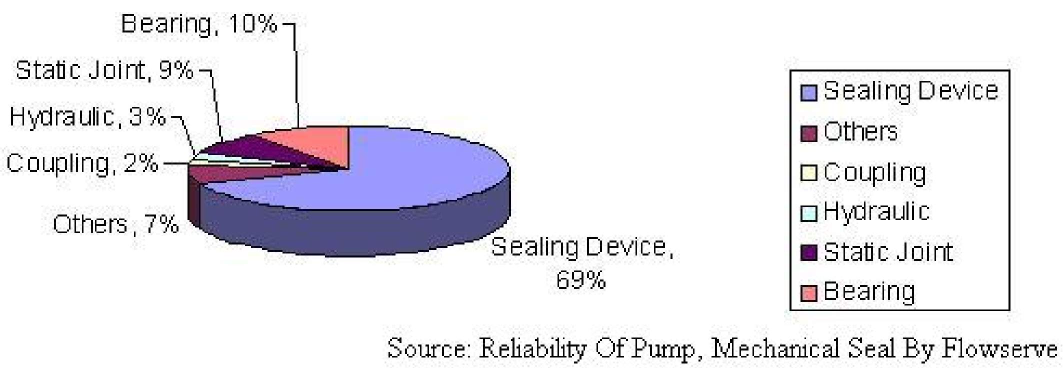


Figure 1: Percentage failure of pump [2]

1.2 Problem Statement

The failure of mechanical seals to prevent leakage in a pump is due to [5]:

1. Surface of stationary face and rotary face not exactly parallel with each other.
2. Secondary seal which is o-ring will melt if temperature is beyond the o-ring melting point.
3. Deflection of shaft
4. Improper cooling system of the mechanical seal
5. Waviness of the seal face.

The Original Equipment Manufacturer (OEM) for this mechanical seal is EKK. Eagle Industry Co. Ltd. The main problems of this design are:

1. Messy design
2. Leakage
3. The design does not follow the API standard.
4. Excessive Heat Generated at the mating Seal Face

1.3 Objectives and Scope of Study

To redesign a mechanical seal that would be used for a pump with shaft diameter 80mm. This design is referred to pump 1207 in Vinyl Chloride Malaysia (VCM) in Kerteh. The sealing liquid of the pump is hot water. All the design specifications based on ANSI / API 682 and ISO 21049. The mechanical seal will be analyzed, redesigned, modelled, fabricated and pressure tested.

LITERATURE REVIEW/THEORY

2.1. Pump Operation System

The function of impeller is to suck the product and discharge it at the outlet of the pump. The impeller is connected to motor by a shaft as shown in Figure 2. As the impeller vanes rotate, they transmit motion to the incoming product, which then leaves the impeller, collects on to the pump casing, and leaves the pump discharge. Discharge pressure will force some product down behind the impeller to the drive shaft, where it attempts to escape along the rotating drive shaft as shown in Figure 3 [2].

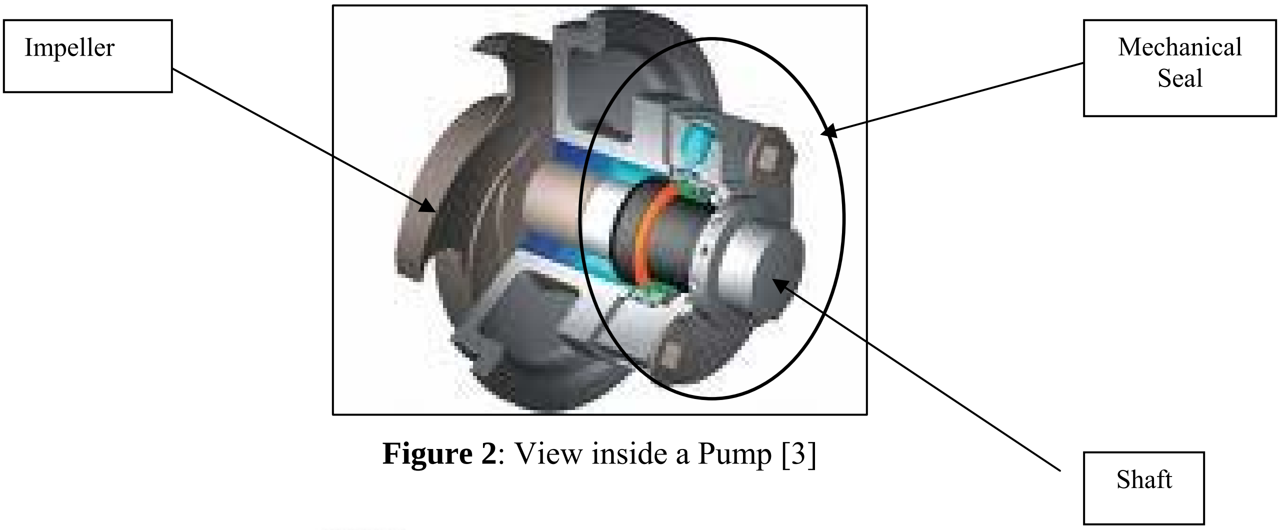


Figure 2: View inside a Pump [3]

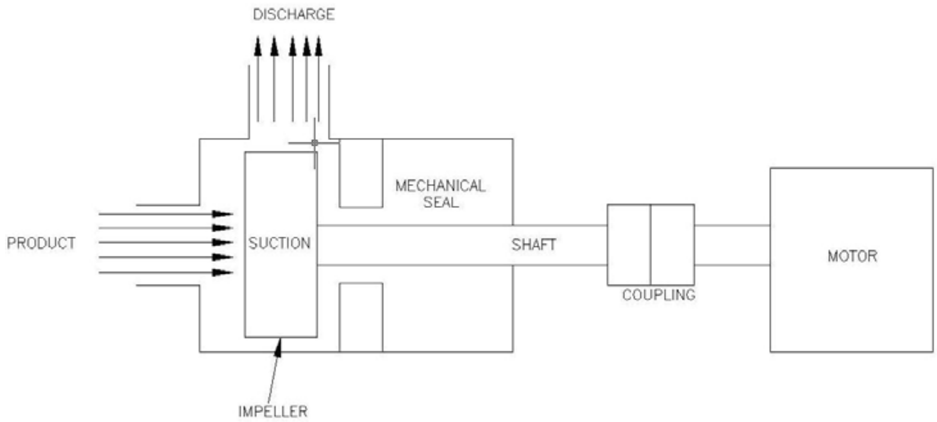


Figure 3: Operation System of a Pump

2.2. Mechanical Seal Operation System

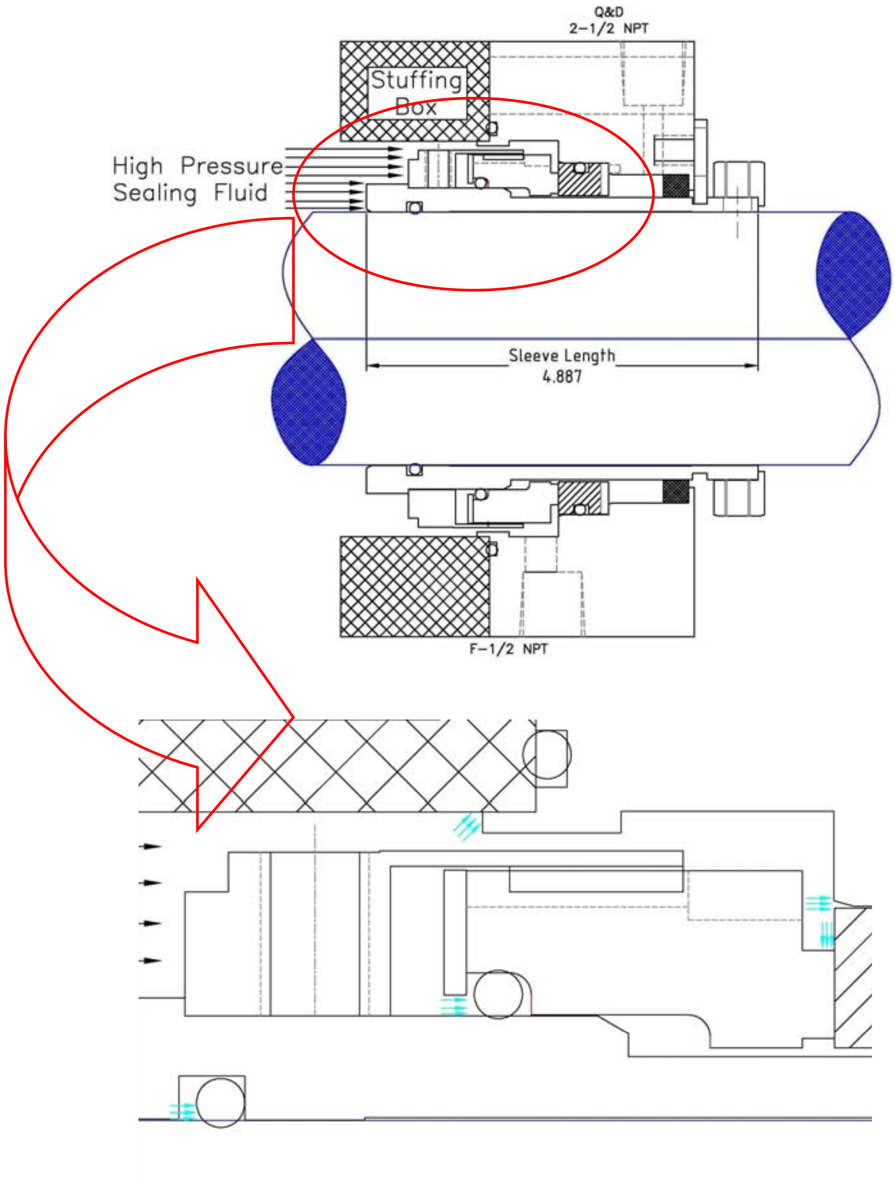


Figure 4: Operation System of a Mechanical Seal

The fluids try to escape from the operating side to the atmosphere as shown in Figure 4. Inside the pump, the discharge pressure is higher compared to atmosphere. Therefore the fluids will be forced to escape through the flat mating face.

Mechanical seal is divided into three parts. First is primary seal which seal face (stationary and rotary face). Second is secondary seal which is gasket or o-ring. The third is hardware seal such as gland, sleeve, and spring holder as shown in Figure 5.

(a) Primary Seal: Seal Face

(b) Secondary Seal: O-ring

(c) Hardware Seal: Gland and Sleeve

Figure 5: Mechanical Seal Component [4]

Mating face is the mating surface between stationary and rotary face. The sleeve is tightened to the shaft by screws at the sleeve collar. The spring holder holds the stationary face and it is tightened to the sleeve. Stationary face will be locked to the gland and it will not move at any axis. As the shaft is rotating, the sleeve, spring holder and rotary face will rotate with the shaft as shown in Figure 6. Table 1 shows mechanical seal parts' function

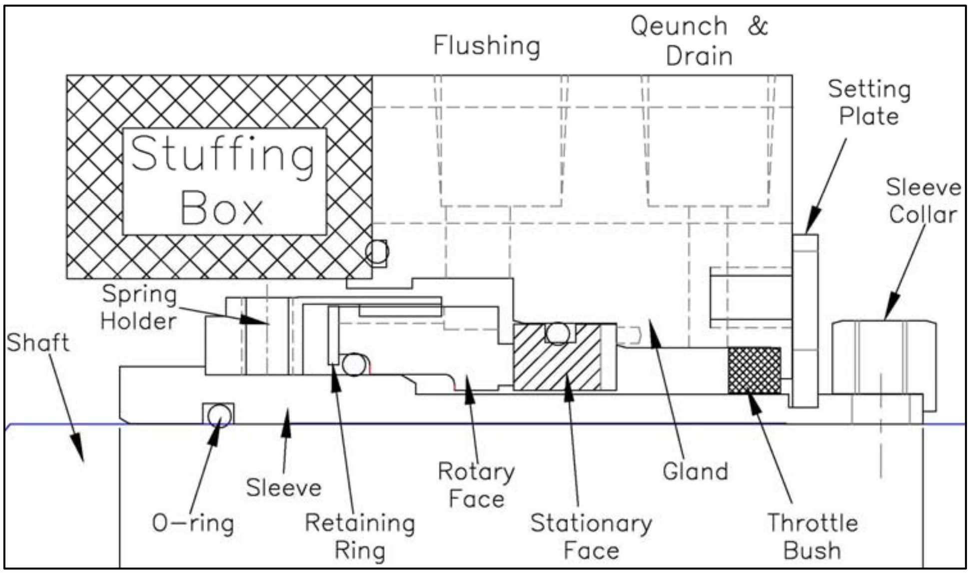


Figure 6: Mechanical Component Arrangement

Table 1: Mechanical Seal Parts' Function [1]

Components Name	Function
Sleeve Collar	Set screws of sleeve collar will tighten the sleeve to the shaft
Sleeve	As secondary shaft.
Spring Holder	Compression unit to push rotary face in place to stationary face. It is tightened to the sleeve by the set screws.
Retaining Ring	Distribute compression forces by springs to rotary face
Rotary Face	Will rotate with spring holder as it is locked to spring holder.
Stationary Face	It is locked to the gland.
O-ring	Prevent the fluid from going across the o-ring.
Gland	Providing boundary for fluid flowing out to the atmosphere side and coolant path into seal chamber.
Auxiliary Gland	Secondary gland if the mechanical seal is double seal.
Throttle Bush	Prevent the metal-to-metal contact
Setting Plate	To fix the gland when the whole mechanical seal is under compression.

Rubbing contact rotary and stationary face will minimize the leakage and will generate heat. Therefore in the mechanical seal, there will be a cooling system to reduce temperature at the mating face.

2.2.1. Various Configurations

It is useful to look at the different configurations for mechanical seals so it can be shown that all cases are very similar to the base configuration shown in Figure 7. Configuration is based on two primary factors. The first is the location of the pressure relative to the annulus. If the pressure is on the outside as in Figure 8, the seal is called an outside pressurized seal. If the pressure is on the inside of the annulus as in Figure 9, the seal is called an inside pressurized seal. While at first glance this would not seem to make much difference, this choice has considerable impact on both the mechanical design.

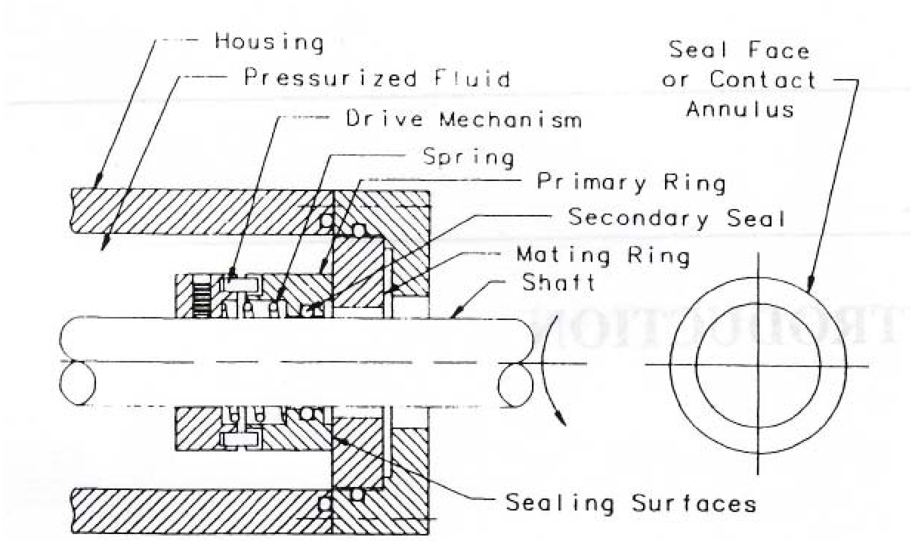


Figure 7: Mechanical Seal –outside pressurized, rotating primary ring, and fixed mating ring [1]

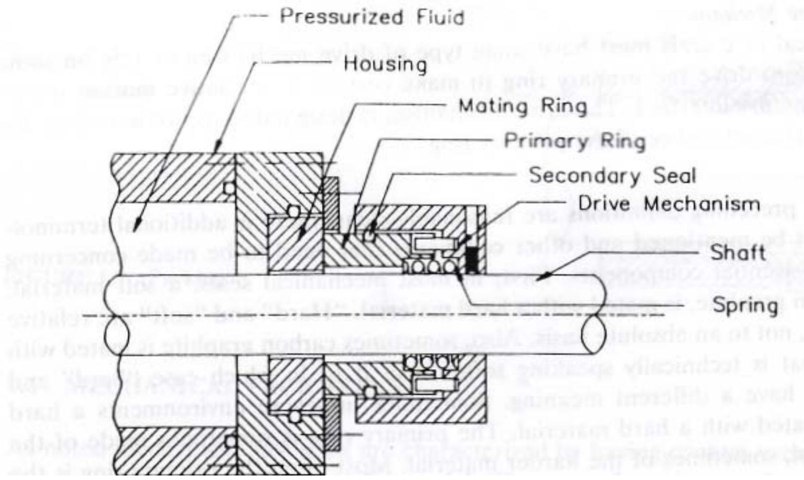


Figure 8: Mechanical Seal – inside pressurized, rotating primary ring, and fixed mating ring [1]

The other factor that influences the configuration is whether the primary ring is rotating or non-rotating. While Figures 8 and 9 both show rotating primary rings, one can design stationary face equally well where the primary ring is non-rotating (Figures 9 and 10). In summary, nearly all face seals can be classified as one of the four types shown in Figures 7, 8, 9, and 10. The design methodologies developed herein will be valid for all of these configurations [1].

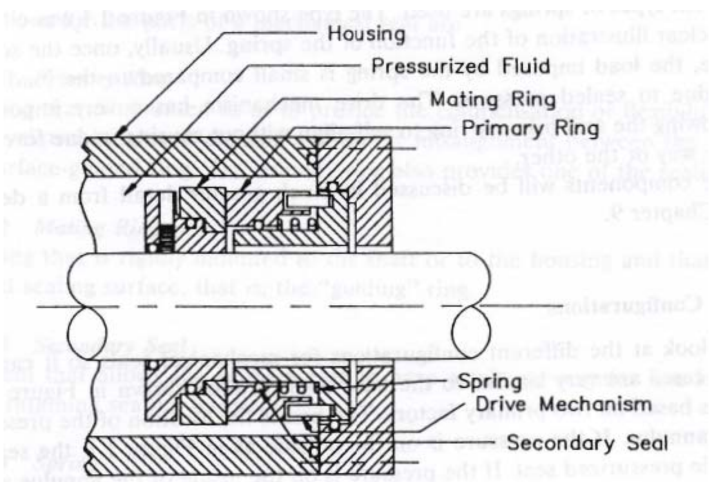


Figure 9: Mechanical Seal – outside pressurized, rotating mating ring, and fixed primary ring [1]

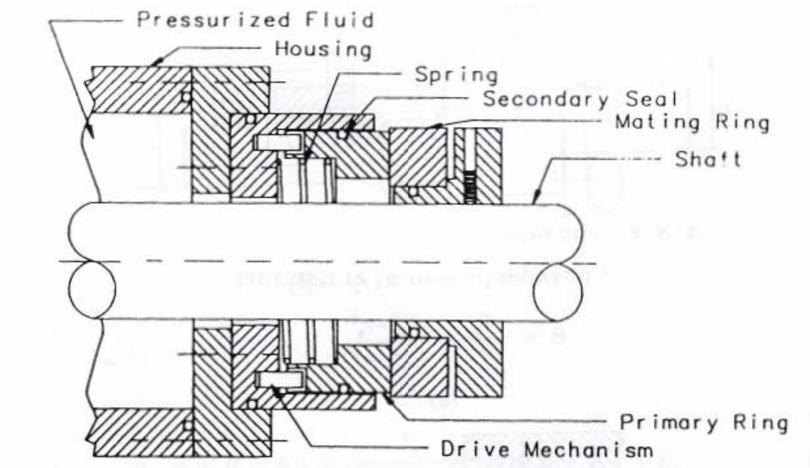
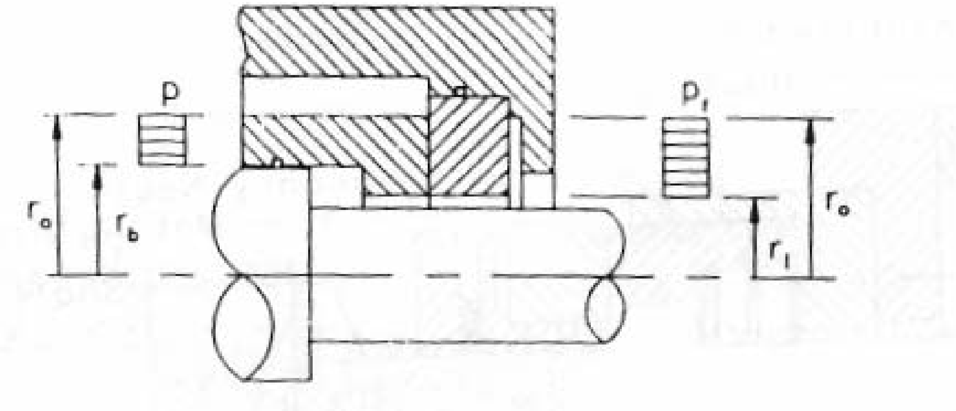


Figure 10: Mechanical Seal – outside pressurized, rotating mating ring, and fixed primary ring [1].

2.2.2. Balance Ratio

Balance ratio (B), is such an important and widely used term that its definition needs to be set straight at the outset. The definition of balance ratio is the ratio between the average load, p_f, at the mating face and back pressure that subjected to the seal face which is stuffing box pressure. Figure 11 shows how this definition is applied to inside and outside pressurized seals. The pressure p_f is determined simply by the sealed pressure times the net area over which it acts divided by the area of the face.

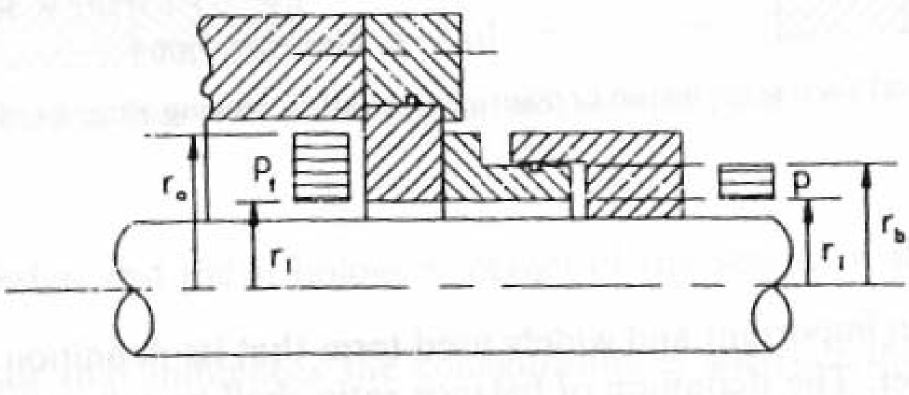
Thus, the balance ratio becomes the ratio of the net hydraulically loaded area to the face area as shown Figure 12. These definitions for *B* are used throughout this work. Now if the balance ratio *B* is greater than 1.0, the seal is termed *unbalanced.* That is, the average pressure load on the face is greater than the sealed pressure. The term *balanced* refers to the case where $B < 1.0$ or where the average pressure loads on the face is less than the sealed pressure. While most seals that operate at high pressure are of the balanced type, many low-pressure seals operate at $B > 1.0$ because of convenience of design.



(a)

$$p\,\Pi\left(r_o^{\,2}-r_b^{\,2}\right)=p_f\,\Pi\left(r_o^{\,2}-r_i^{\,2}\right) \tag{1}$$

$$B=\frac{p_f}{p}=\frac{(r_O^{\,2}-r_b^{\,2})}{(r_O^{\,2}-r_i^{\,2})} \tag{2}$$



(b)

$$p\,\Pi\left(r_b^{\,2}-r_i^{\,2}\right)=p_f\,\Pi\left(r_o^{\,2}-r_i^{\,2}\right) \tag{3}$$

$$B=\frac{p_f}{p}=\frac{(r_b^{\,2}-r_i^{\,2})}{(r_O^{\,2}-r_i^{\,2})} \tag{4}$$

Figure 11 (a): Outside pressurized seal, balance ratio. (b): Inside pressurized seal, balance ratio [1].

One can develop a simple but useful theory of operation of face seals by making some strong assumptions and simplifications. One assumes that the sealed pressure average between the faces and distributes itself in some manner such that the average value of the fluid pressure between the faces is proportional to the sealed pressure or *Kp*. The fluid pressure may support some or the entire applied load. To assure static equilibrium of the seal in the axial direction, some of the load on the seal face can be supported by some other means. Figure 12 shows a uniform pressure, p_m, that is meant to imply that this component is derived by contact; it may derive from fluid pressure such as caused by hydrodynamics effects as well. Now, considering Figure 12 and summing the forces as shown, one can solve for the pressure p_m required for equilibrium [1].

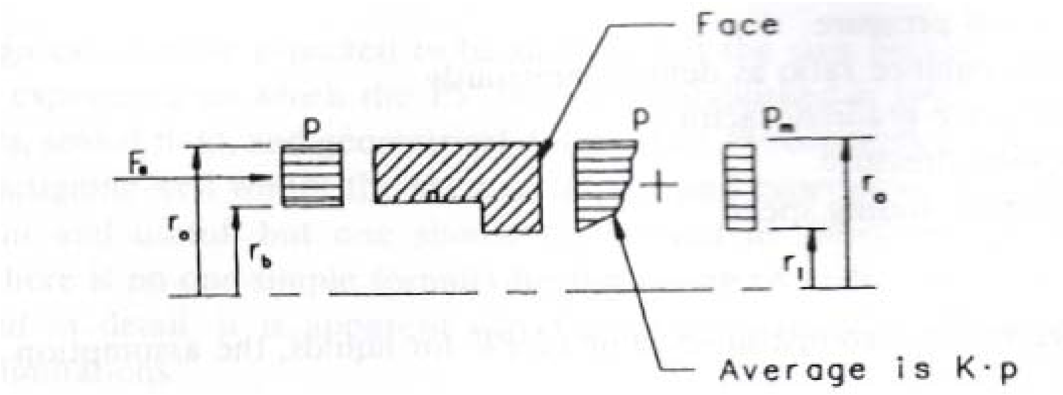


Figure 12: Elementary Theory [1].

$$p\Pi\left(r_o^2 - r_b^2\right) + F_s = Kp\Pi\left(r_o^2 - r_i^2\right) + p_m\Pi\left(r_o^2 - r_i^2\right) \tag{5}$$

$$p_m = p(B-K) + \frac{F_s}{\Pi\left(r_o^2 - r_i^2\right)} \tag{6}$$

$$p_m = p(B-K) + p_s \tag{7}$$

p_m = Mean pressure,

F_s = Spring forces,

K = Pressure gradient at the mating face

The parameter *K* is often termed the *K factor* or *pressure gradient facto.* Figure 13 illustrated the pressure gradient at the mating face of the mechanical seal. Its value greatly affects the contact pressure. If one assumes the faces are parallel and the seal gap and geometry are axisymmetric and the flow across the gap is laminar and incompressible, the fluid pressure caused by the hydrostatic pressure across the face will be approximately linear and K$\rightarrow \frac{1}{2}$.

Given the same assumptions and a compressible gas, K$\rightarrow \frac{2}{3}$ for high compressibility ratios [1].

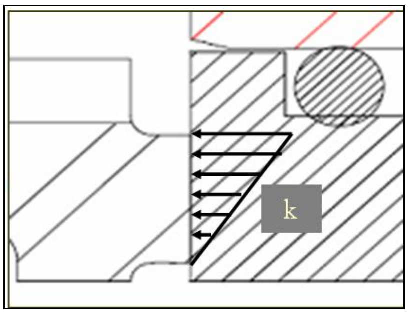


Figure 13: Pressure Gradient [5].

Equation (6) is a useful but very approximate expression for the mechanical or contact pressure that must be carried by the seal faces. The magnitude of this contact pressure is of great importance to the ability of a seal to operate without excessive friction, wear, and leakage. The equation also shows the importance of using balance ratio to reduce this component of load.

Mechanical seals are being used increasingly on fluid pumps to replace packed glands and lip seals. Pumps with mechanical seals perform more efficiently and generally perform more reliably for extended periods of time. A mechanical seal is created to prevent pumped fluids from leaking out along the drive shafts. The controlled leakage path is between two flat surfaces associated with the rotating shaft and the housing respectively. The leakage path gap varies as the faces are subject to varying external loads which tend the move the faces relative to each other.

The mechanical seal requires a different shaft housing design arrangement compared to that for the other type of seals because the seal is a more complicated arrangement and the mechanical seal does not provide any support to the shaft.

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

The seal faces are usually dissimilar materials with the softer face being the narrower surface. For abrasive applications similar hard materials are used e.g. tungsten carbide. The seal surfaces must have sufficient strength to withstand the hydrostatic fluid forces and must be able to remove the heat generated by sliding action. Carbon is often used against bronze, cast iron, stainless steel etc. Figure 14 shows the seal face that is made of carbon.

Figure 14: Seal face [10]

The seal surface must be flat, smooth and square to the shaft. Both surfaces are normally lapped to a high quality finish. The harder surface is most important because the softer surface is designed to run-in over the initial operating period.

The shaft design is critical. It must be rigid enough to support the seal in the correct position and the shaft surface finish must be suitable to ensure good sealing on the static seals (0.4 micrometers CLA or better). The shaft Total Indicated Runout (TIR) should not exceed 0.125mm. The shaft vibration should be minimum [1].

2.2.3. Cooling System for Mechanical Seal

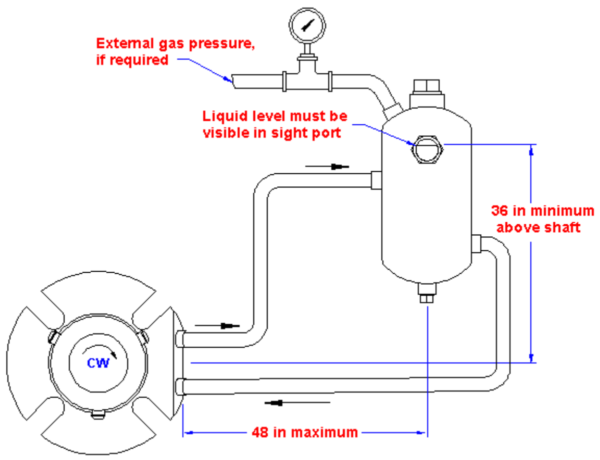


Figure 15: Cooling System for Mechanical Seal [5]

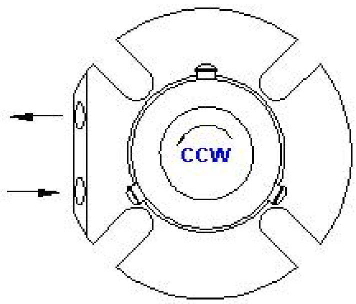


Figure 16: Direction of Coolant Flow in Mechanical Seal [5]

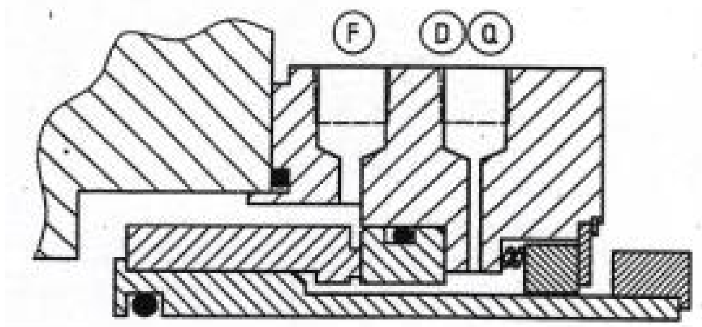


Figure 17: Path for Coolant in Mechanical Seal [5]

Rubbing contact between rotary face and stationary face will generate heat which is never desirable because it could melt the o-ring. To reduce temperature at the mating face, a cooling system is introduced. A clean liquid, from an outside source is brought into the stuffing box through a regulating valve at one atmosphere (15 psi/1 bar) higher than stuffing box pressure. The liquid should be brought in at the bottom of the stuffing box to ensure thorough cleaning. All of this liquid will eventually go into the product. The functions of flushing system are:

1. To introduce clean liquid into the stuffing box, to remove solids or any problem fluid.
2. Cool a hot liquid by flushing in a cold one.
3. Remove a liquid that is sensitive to changes in either temperature and/ or pressure.

Some seal glands have a vent or quench connection provided behind the seal so that steam or some other fluid can be used to control temperature in the seal area. A close fitting carbon (or any other non sparking material) bushing is installed outboard of this connection to provide a close clearance between the gland and the shaft. Refinery applications use a version of the quench gland and call it an American Petroleum Institute (API) gland. The functions of quench system are [5]:

1. The disaster bushing will protect the seal from hitting the inside of the stuffing box if you have a bearing failure. This is a very important feature in those applications where the product will burn or explode if overheated.
2. The disaster bushing will protect personnel if there is a massive seal failure. The majority of the leakage can be directed, down the drain connection, to a collecting tank or vent.
3. To wash away solids from the outboard side of the seal that will prevent "hang up" as the seal face wears and the seal moves forward.
4. To wash away toxic or corrosive vapors that might leak across the seal faces.
5. To control the temperature in the seal area.
6. As a back up to a heating/ cooling jacketing failure.

2.3 Fabrication of Mechanical Seal

The machining process for this mechanical seal are using the lathe and milling machine. Lathe machine is a machine where the workpiece is tightened to the jig at the spindle. The workpiece rotates and the cutting tool is static. But for the milling machine, the workpiece is static, only the cutting tool rotates. See Figure 18 for lathe machine and Figure 19 for milling machine.

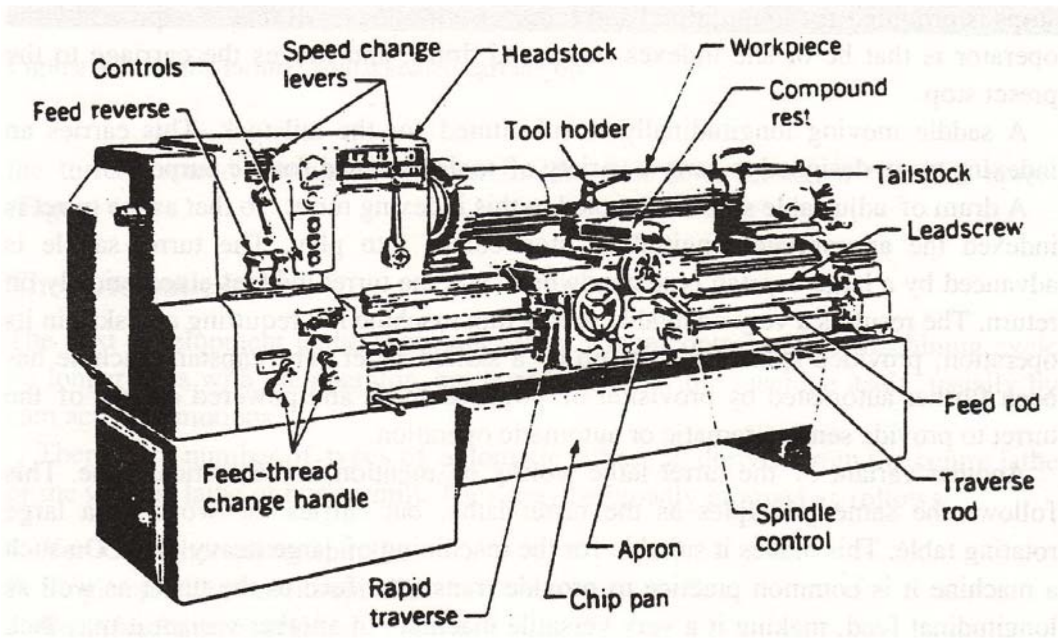


Figure 18: Lathe Machine [6]

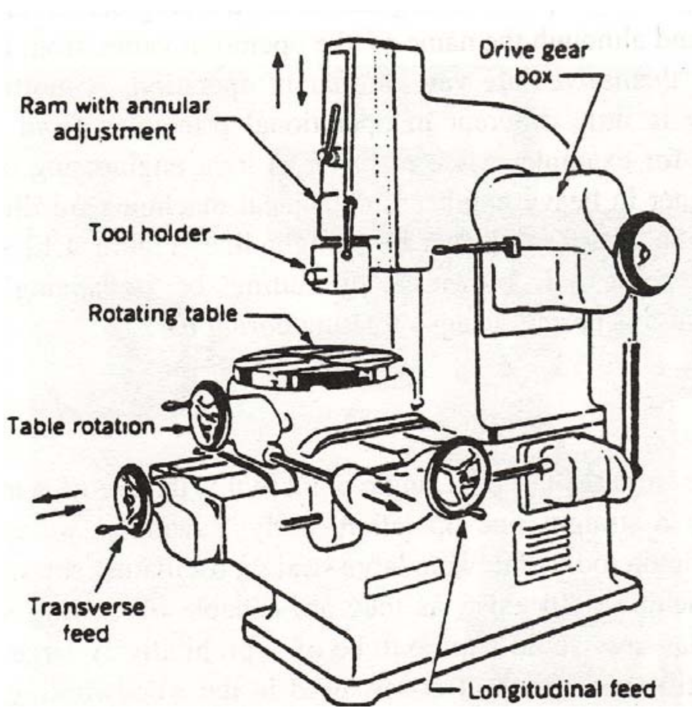
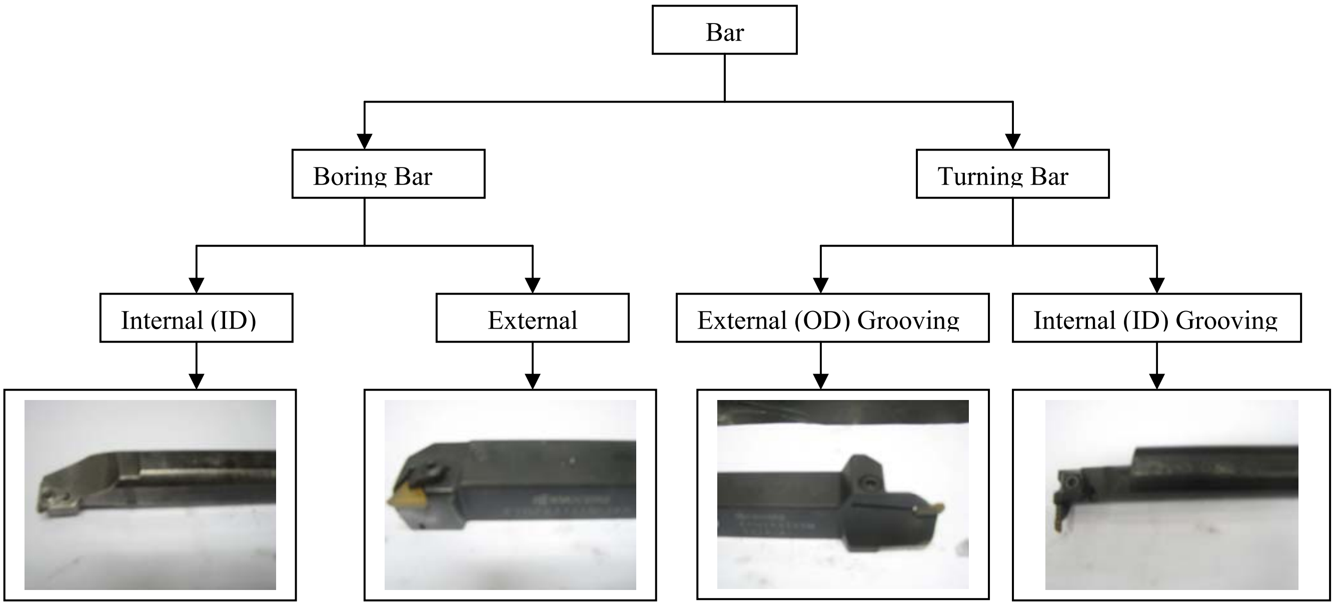


Figure 19: Milling Machine [6].

For the cutting tools, Insert is a tool that uses to cut the unwanted part at the workpiece. The insert is tightened to the bar and the bar in tightened to the tool holder. There are two kind of bar.



Internal Boring Bar External Boring Bar External Turning Bar External TurningBar

Figure 20: Type of Bar

To get surface roughness use the following formula:

$$R_z = \frac{f^2}{8R}100$$

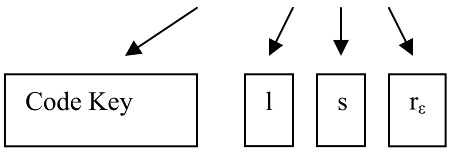
R_z = Theoretical surface roughness

f = Feed Rate (mm/ rev)

R = Corner Radius of Insert (mm)

For O-ring position, the roughness of the surface must be $\overset{32}{\surd}$ which is 32 µ.inch or 1.26 µ.m.

Say that inserts used *TPMR 16 03 04*.



l = length

s = thickness of inserts

r_ε = radius of inserts

For $\overset{32}{\surd}$ or 1.26 µ.m and R= 0.4 mm

$$R_z = \frac{f^2}{8R}100$$

$$1.26 = \frac{f^2}{8(0.4)}1000$$

$f = 0.063$ mm/rev

PROJECT METHODOLOGY

3.1 Project Done

Mechanical seal that used in this project is P-1207 that is being used at Vinyl Chloride Malaysia (VCM) in Kerteh, Terengganu. The sealing fluid for P-1207 is hot water.

The first step for this project is failure analysis of the mechanical seal. By observing at its original mechanical seal assembly drawing and sequence of event (SOE) of the pump, the weaknesses of the original mechanical seal are identified. The old mechanical seal design's weaknesses are:

1. Messy design.
2. Leakage
3. The design does not follow the API standard.
4. Excessive Heat Generated at the Seal Face.

The second step is to find the solution for the mechanical seal's weaknesses and redesign the mechanical seal with new design. The solutions for this mechanical seal are:

1. Simplified the design to a simple design. (To reduce the load to the shaft).
2. Proper o-ring's material selection, o-ring's size selection and the suitable gap for each part.
3. Change the inserted seal face to the solid seal face,
4. Redesign the seal faces with suitable balance ratio.

The third step is making a design detail for each part of the mechanical seal. Mechanical seal is modelled by using AutoCAD. Each part of the mechanical seal drawing will be assembled in AutoCAD to get the view when real assembly of mechanical seal parts.

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Fabrication flow for each part is as follow:

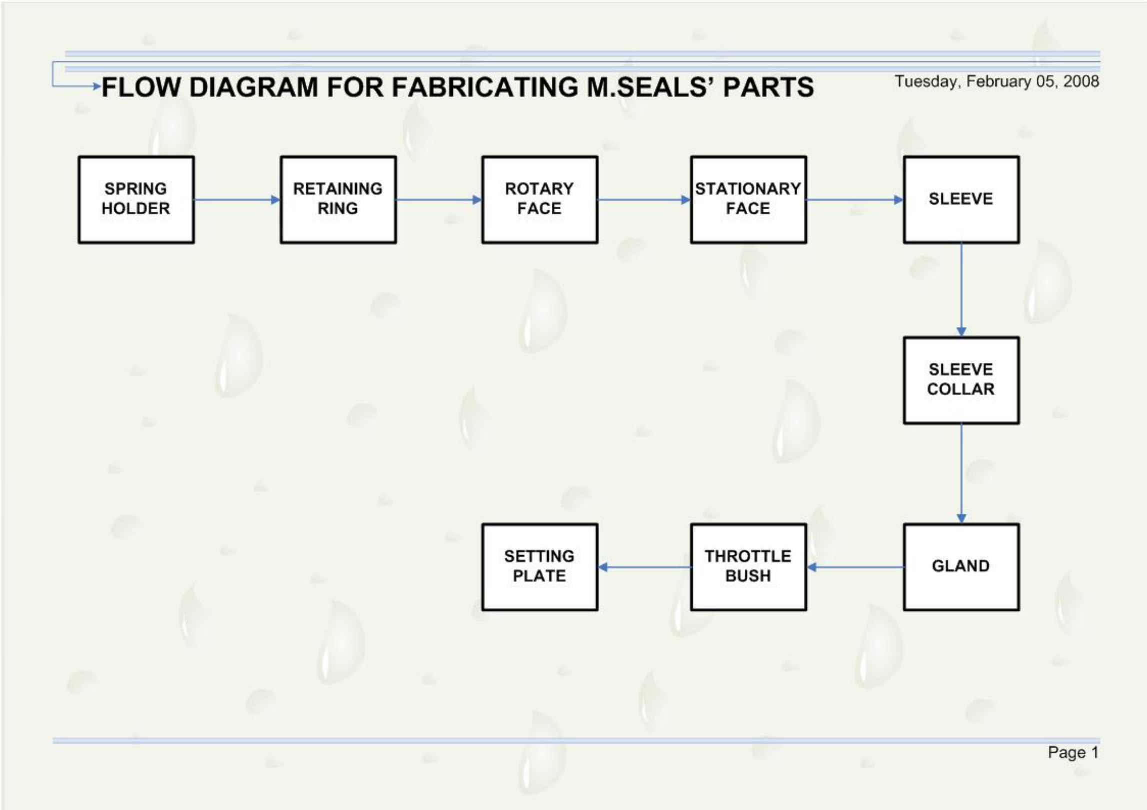

Figure 21: Flow Diagram for Fabricating Mechanical Seals' Parts

The parts that are already finished be machined:

1. Spring holder
2. Retaining Ring
3. Rotary Face
4. Stationary Face
5. Sleeve
6. Sleeve Collar
7. Gland
8. Throttle Bush
9. Setting Plate

After the fabrication process is finished, the mechanical seal has been assembled to make the Pressure Test. For the Pressure Test, it is done at ProEight Offshore Engineering's factory in Gebeng. The purpose of Pressure Test is to detect any leakage that could occur at any point in the mechanical seal. The Pressure Test setup is illustrated in Figure 22.

Figure 22: Pressure Test Equipment at ProEight Offshore Engineering Sdn. Bhd

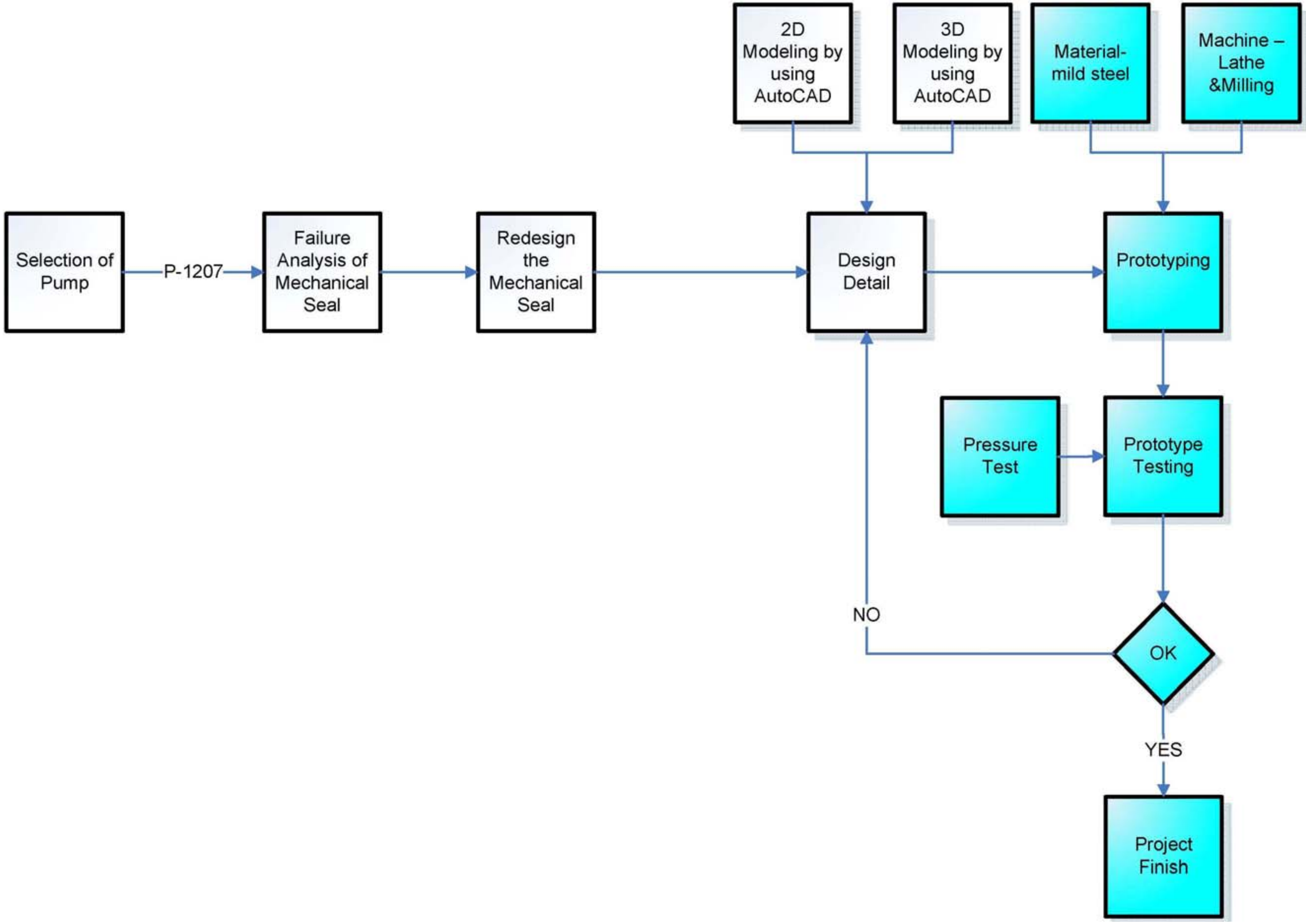

Figure 23: Flow Chart of the project

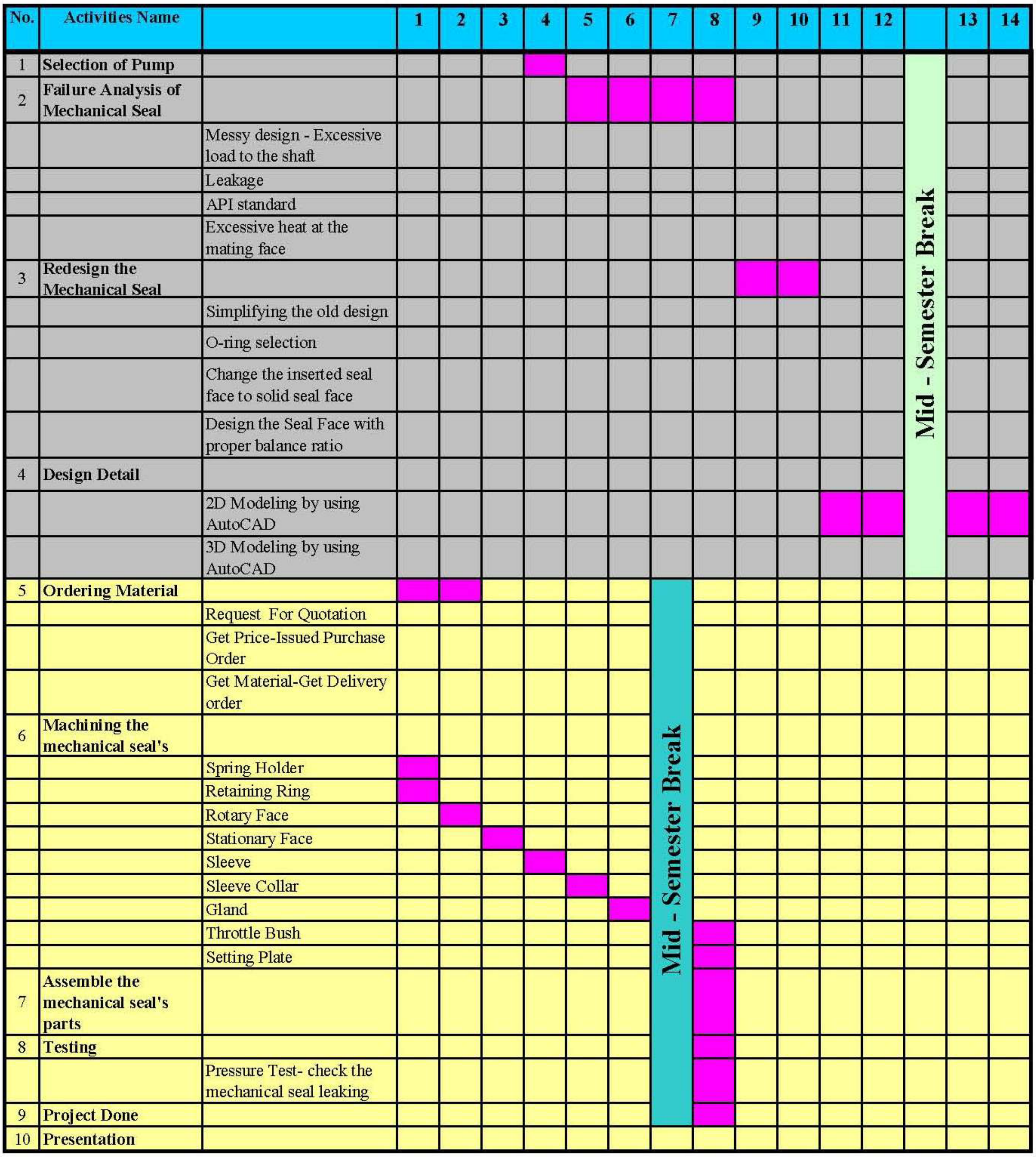

Figure 24: Gantt chart

RESULTS AND DISCUSSION

4.1. Design Limitation

Appendix 1 shows the view of old mechanical seal assembly design of P-1207. The Original Equipment Manufacturer (OEM) for this mechanical seal is EKK. Eagle Industry Co. Ltd. The main problems of this design are:

1. Messy design
2. Leakage
3. The design does not follow the API standard.
4. Excessive Heat Generated at the Seal Face

4.1.1Messy Design

Messy design which means, too many parts involved in mechanical seal operation system would cause excessive loaded to the shaft resulting deflection of shaft. Therefore this design need to resign with simple design that can operate as the old design or even better.

4.1.2 Leakage

Leakage in pump will lead to pressure drop inside the pump. Leakage are because of improper o-ring size selection, groove size, opening forces at the seal face greater than closing forces, etc.

4.1.3 API standard

In designing the mechanical seal, a standard must be followed so that it would sustain forces or pressure that compatible with its design. For mechanical seal design, American Petroleum Institute (API) 682 is a base concept in designing the mechanical seal. For P-1207, the design does not follow the API standard as shown in Figure 25. API standard do not allow the seal face inserted into the other material [1].

Figure 25: Carbon Inserted

4.1.4 Excessive Heat Generated at the Seal Face

Seal performance depends greatly on three factors that result from temperature increases. The first of these is that the tribology of the interface may become a boundary lubrication condition in the event the temperature is too high. That is, the interface may flash to a vapor. The second factor is that many materials are simply not tolerant to high levels of temperature and may be destroyed by operation at elevated temperatures. The third factor is that non-uniform heating of the seal rings produces distortions that in turn influence the interface shape.

The heat transfer problem is shown in Figure 26. Most of the heat of concern is generated by sliding friction at the seal interface itself. The seal cavity will often be cooled as shown. The heat generated at the interface must pass out through the seal ring materials until it comes to a convective boundary where it can be carried away. Usually very little surface area is available for heat transfer so that localized seal interface temperatures can get very high. The seal rings will always be hotter at the sliding end than at the free end. Thus, there is a significant axial temperature gradient as well as a radial gradient.

Figure 27 shows two types of distortion that commonly result from thermal distortion. Thermal coning or thermal radial taper is caused by the axial temperature gradient mentioned and is a common behavior in seals. Waviness is caused by any nonaxisymmetric heating. Thus, if the seal contacts heavy on one side, then the temperature distribution will vary in the circumferential direction and waviness will result.

It is essential to predict the temperatures of the various seal parts in order to assess the effect of temperature on materials and to find the extent of distortion of the seal caused by non-uniform temperatures.

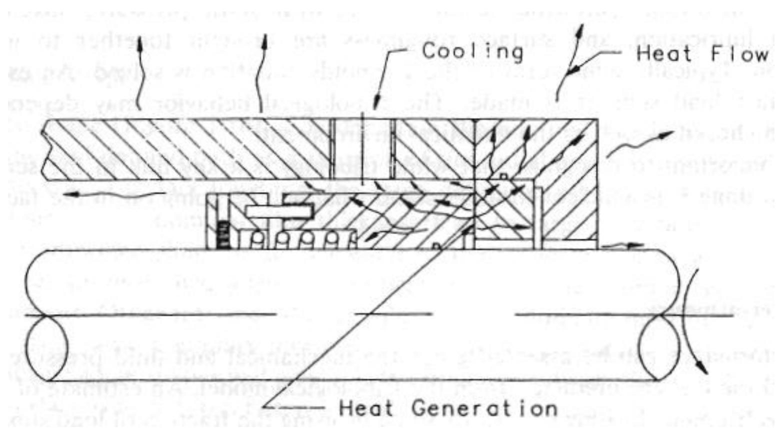

Figure 26: Heat Transfer [1].

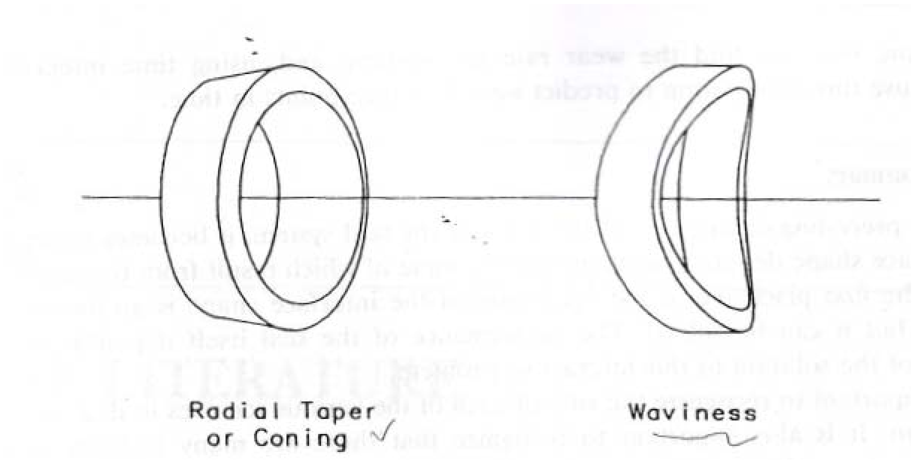

Figure 27: Common Distortions [1].

Heat is never desirable because [5]:

1. Heat means a loss of expensive energy.
2. Heat will affect the elastomer (rubber part) in the mechanical seal and reducing its' life.
3. Heat can damage some carbon faces by melting the fillers and expanding the air pockets trapped below the surface, causing pits in the carbon that will prevent it from passing a fugitive emission test.
4. Some hard faces can be damaged by a rapid temperature change.
5. Plated surfaces can "heat check" and crack due to the differential expansion between the coating and the base metal.
6. Many products can vaporize at elevated temperature, blowing the faces open and leaving solids between the lapped faces.
7. Heat will change the viscosity of many liquids. It many cases it will diminish, but in some cases the viscosity can increase.
8. Corrosion always increases with additional heat.
9. Petroleum base products can solidify between the faces.
10. Lapped faces can go "out of flat" and critical tolerances change at elevated temperature.

4.2 Design Analysis

4.2.1 Wear Analysis

Figure 28 shows most of the many types of loads that are applied to the seal rings themselves. The prediction of the stresses and deflections of the seal rings is a problem in solid mechanics. Using well-established methods one can find the exact pattern of distortion for an arbitrary set of loads. One must be concerned with the stresses in relation to the strength of the materials, and such an analysis is important to the performance assessment of the seal. Concerning the interactive problem, any tangential variation of the load on the face of the seal ring produces a wavy-type deflection behavior, as is shown in Figure 27. Any imbalance in the moments on the ring about a circumferential axis produces coning. Any nonuniform temperature distribution also produces the same types of distortions. That is why these distortions influence the interface and therefore the very pressure distribution that is causing them in the first place-thus the interactive nature of the problem.

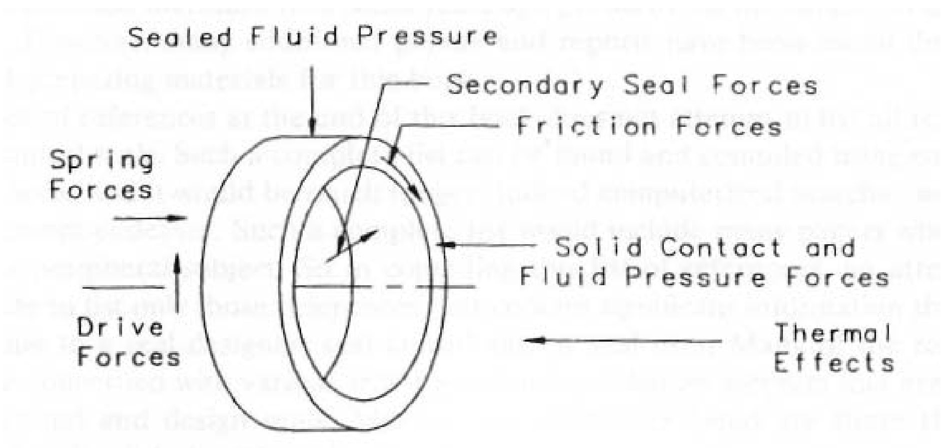

Figure 28: Solid Mechanics [1].

4.2.2. Forces Exerted on Seal Faces

On the seal faces, two forces acting on it and it is illustrated in Figure 29.

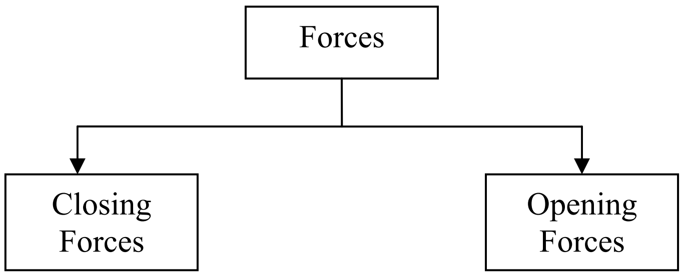

Figure 29: Forces Exerted on the Seal Face.

4.2.2.1 Closing Forces

There are at least two forces closing the seal faces [1]:

i. The mechanical spring force, F_s.
ii. The hydraulic (Fluid pressure) force caused by the stuffing box pressure acting on the seal face area, F_{pc}.

4.2.2.1.1 Spring Force

Springs force applied to the rotary face is to make sure the rotary face is pushed to stationary face when there is no pressure from the back (stuffing box pressure). The range of springs force is 5-50 lb (2.267962 kg-22.67962 kg).

4.2.2.1.2 Hydraulic Force

The hydraulic force is the back pressure distribute to the seal face area. The force is build when the pump is running. The value depends on the pressure of a pump. For P-1207, the stuffing pressure is 7.5 bars. When the pump is running, the forces from the spring is negligible because of hydraulic force is much bigger than springs force.

4.2.2.2 Closing Forces

The seal faces are subjected to three forces namely [1]:

i. A hydraulic force is created any time there is fluid between the seal faces.
ii. A centrifugal force created by the action of the fluid being thrown outward by the rotation of the pump shaft.
iii. A hydrodynamic force created because trapped liquid is, for all practical purposes, non compressible.

4.2.2.2.1 Hydraulic Forces

Testing shows that some times there is a film of liquid between the faces, some times there is only vapour, some times there is nothing at all, and some times there is a combination of all three. This means that if there is liquid or vapour between the faces, it is under pressure trying to force the lapped faces apart. The stationary face cannot move because it's being held by gland, but the spring loaded face will respond to this force.

4.2.2.2.2 Centrifugal force

Centrifugal force is acting on the spring loaded face trying to spin it perpendicular to the rotating shaft. Stationary face is not perpendicular to the shaft because it is referenced against the stuffing box face which is a casting that is not perpendicular or square to any thing. A gasket located between the gland and the stuffing box further compounds the problem.

4.2.2.2.3 Hydrodynamic Forces

Seal faces are lapped to within three helium light bands or slightly less than one micron. This slight waviness is enough to generate hydrodynamic lifting forces as we try to compress non-compressible liquid the trapped between the lapped faces. Two closing forces and three opening forces the seal faces. If the closing forces are the greater forces, the seal will generate heat that is often destructive, but always a waste of energy and pump efficiency. If the opening forces are the greater forces the seal will leak and that's never desirable.

A balanced seal, by definition, balances these opening and closing forces so that the seal will not get hot and it will not leak. Since the hydraulic closing forces were twice the opening forces, sleeve is installed inside the seal to reduce the closing area and reduce the closing force. To eliminate the hydraulic forces from acting to open or close the seal faces. This leaves only the spring force to close the seal and the hydrodynamic and centrifugal forces to try to open the seal faces

4.2.3 Waviness Effect on the Seal Face.

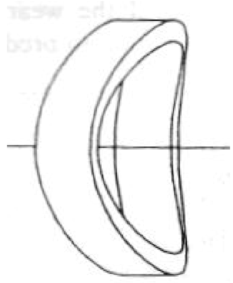

Figure 30: Waviness effect caused un-uniform spring forces

Waviness of the seal face surface is because of un-uniform spring forces acting on the seal face. For mechanical seal in P-1207, the numbers of springs be used is eight. The total spring forces are 5-50 lb [1].

k = Spring stiffness for each spring

F_s = Total spring forces acting on seal face

Spring arrangement is in parallel, so to find total stiffness of springs just summing up the stiffness of the springs. Each spring has same stiffness value, so

$$F_s = (k_1 + k_2 + k_3 + k_4 + k_5 + k_6 + k_7 + k_8)x \ , \ F_s = 8(k)x \qquad (8)$$

To reduce the effect of waviness, the numbers of springs be increased. By increasing the numbers of spring, the stiffness of the spring need to be decreased to produce same total spring's force. To find the spring stiffness,

$$F_s = (k_1 + k_2 + k_3 + k_4 + k_5 + k_6 + k_7 + k_8 + k_9 + k_{10} + k_{11} + k_{12})x$$

$$F_s = 12(k)x \qquad (9)$$

$$F_s = 12(k)x \quad \rightarrow \quad k = \frac{F_s}{12x} \qquad (10)$$

4.3 Designs Detail

In designing the mechanical seal parts' designs, there are limitations that designer must follow to make sure the ease when installation process. The limitations are:

1. Diameter of stuffing box of the pump.
2. Diameter of the shaft.
3. Diameter of the gland.
4. The dimension of circulation in, circulation out, vent and drain.
5. The type of thread for piping the cooling system.
6. The length of the sleeve.
7. Perimeter centre diameter for M-16 set screws. M-16 be used for tightened the mechanical seal to the stuffing box of the pump.

These limitations dimension must be followed exactly as the previous design. All the limitation is provided in old mechanical seal assembly drawing.

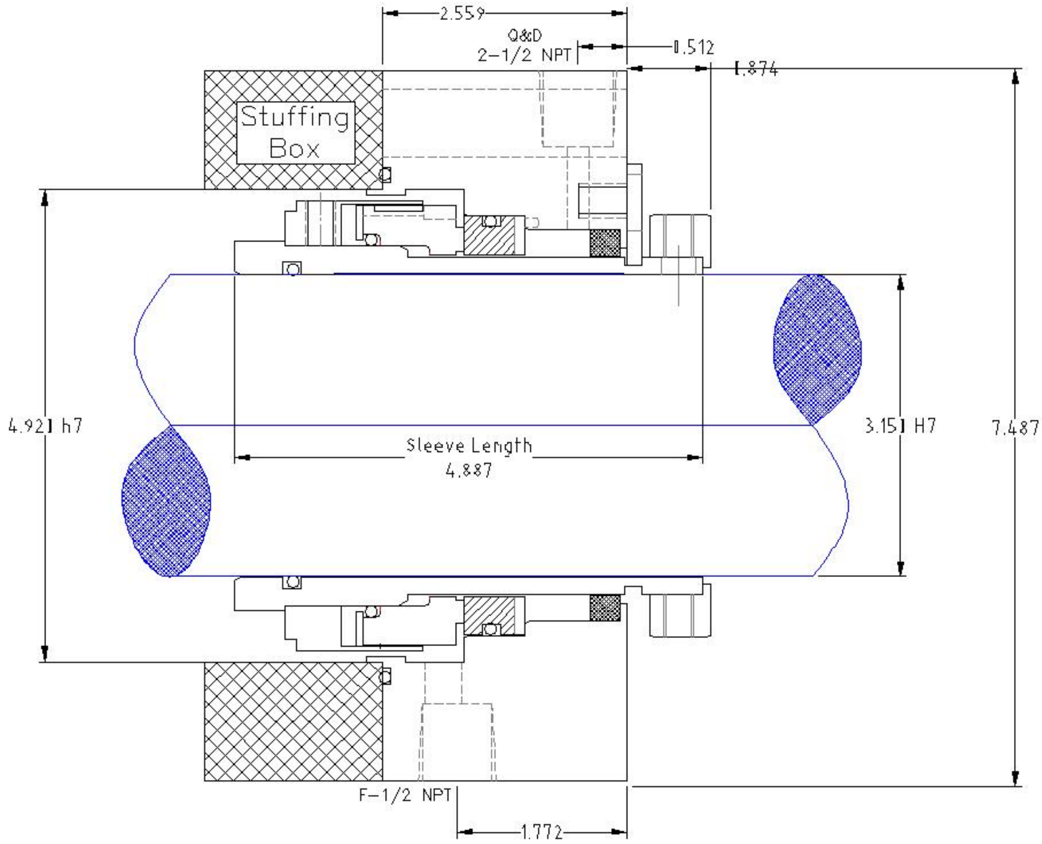

Figure 31: Design Limitation

After all the limitation is identified, the design process is started by design the rotary face and stationary face. The minimum thickness for sleeve is shown the Table 2:

Table 2:Minimum sleeve radial thickness [1].

Shaft Diameter (inches)	**Minimum sleeve radial thickness (inches)**
<2.25	0.1
2.25 to 3.25	0.15
>3.25	0.2

For P-1207, shaft diameter is 3.149 inch. Therefore the minimum thickness is 0.15 inch. The inner diameter of the seal face must bigger than (3.149+2*0.15 = 3.449 inch). Because rotary face is move laterally, add the appropriate gap with 3.449 inch. Finally the inner diameter is 3.563 in. The outer diameter could be varies as long as it will not hit the gland. At the mating face, the outer diameter and inner diameter are determined by the balance ratio. The balance ratio is formula for reducing the back pressure effect so that heat generated at the mating face will be reduced. The formula is in equation (4). For diameter 3.997 and 3.755 inches, it is determined by the o-ring size. Figure 32 is the rotary face design.

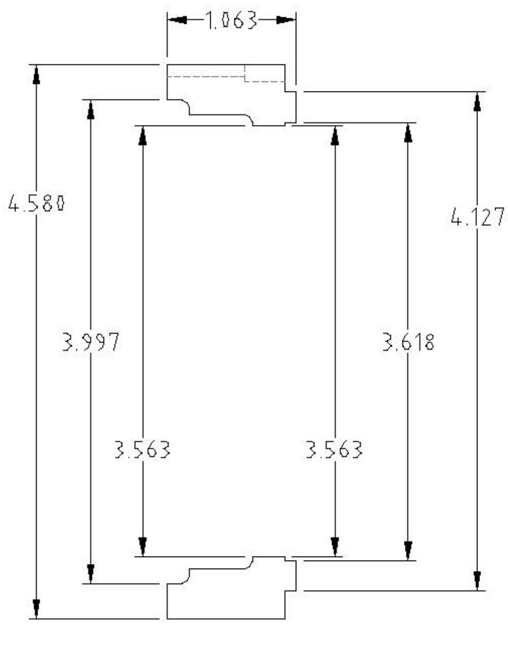

Figure 32: Rotary Race.

For stationary face, the inner diameter is the same as rotary face and for outer diameter it must be bigger than outer diameter at the rotary face's surface. The groove size depends on the o-ring size. Refer to Appendix 7 for stationary face dimension.

The sleeve length must be the same as the old design which is 4.887 inch. For the inner diameter as follow:

Inner diameter: shaft diameter + 0.0024 inch (tolerance)

: 3.149 +0.0024

: 3.151 inch

For the outer diameter of the sleeve, it is depend on minimum thickness and the gap of sleeve between seal face, spring holder, sleeve collar and throttle bush. Refer to Appendix 3 for get more detail for sleeve's dimension.

For gland dimension, outer diameter is following to previous design. For inner diameter, it is depend on the gap between gland and throttle bush, spring holder, rotary face and stationary face. The gap between gland and rotary part, it must be bigger that static part to prevent metal-to-metal contact. Refer to Appendix 8 for get more detail for gland's dimension.

For spring holder part, the changes are redesigning the number of the spring and the way the spring holder lock the stationary face. The rest of the part is the same as the previous design. Below are 3D designs before assembled and after assembled.

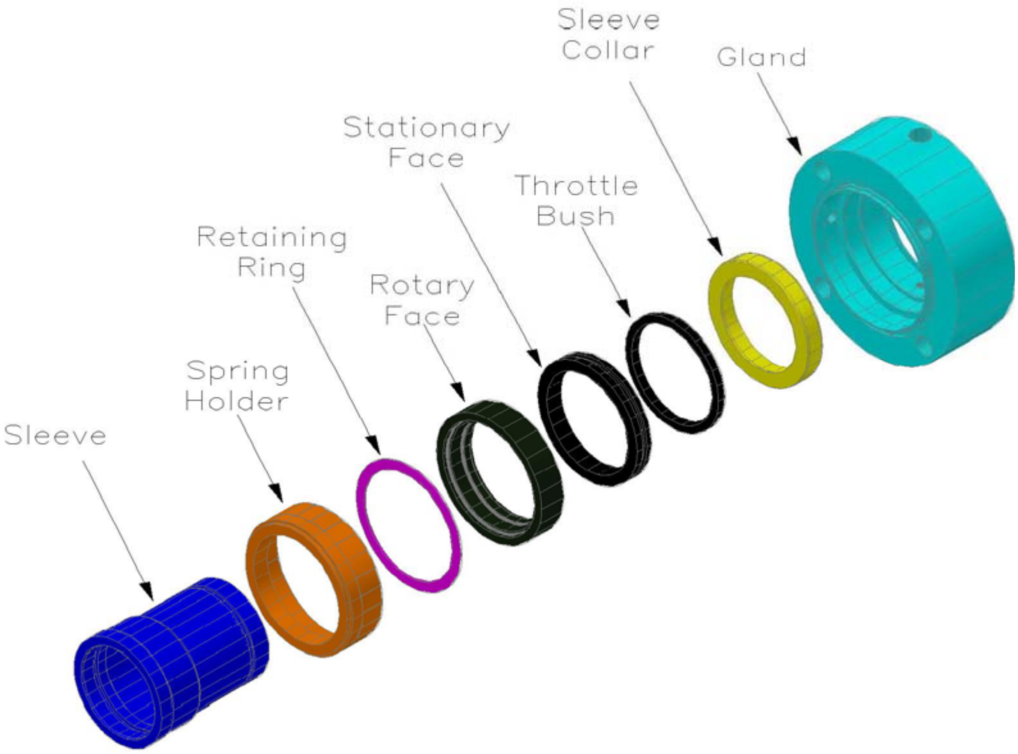

Figure 33: Disassembly of Mechanical Seal's parts

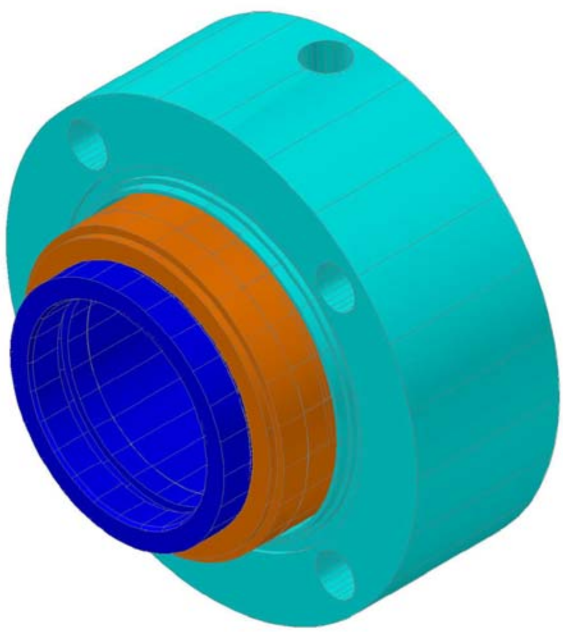

Figure 34: Assembly of Mechanical Seal's parts

4.4. Mechanical Seal's Parts Fabrication

The parts that are already finished be machined:

1. Spring holder
2. Retaining Ring
3. Rotary Face
4. Stationary Face
5. Sleeve
6. Sleeve Collar
7. Gland
8. Throttle Bush
9. Setting Plate

4.41. Spring Holder

First of all, the solid cylinder mild steel (Φ4.8" (OD) x 1.8") will be tightened to lathe machine holder. The desire dimension for spring holder is Φ4.696" (OD) x 1.437". The workpiece dimension must be bigger compare the desire dimension because of the workpiece need to be facing so that the workpiece is exactly parallel to the workpiece holder.

To get the workpiece parallel to the workpiece holder, dye gauge as shown in figure 35 be used to check to workpiece runnout. After get the wokpiece runnout zero, machining process is started by make a through hole to the cylinder. To drill the through hole, Φ 30mm drill be used and increase to 40, 50 and till 90mm which is approaching the inner radius (Φ 3.75") of the spring holder. For roughing, cutting thickness is about 0.25" per cutting.

Figure 35: Dye Gauge

For finishing for the ID, the cutting tool that will be used is internal boring bar for internal as shown in figure 36. The boring bar will be tightened to the tool holder as shown in figure 37.

Figure 36: Boring bar for internal (ID).

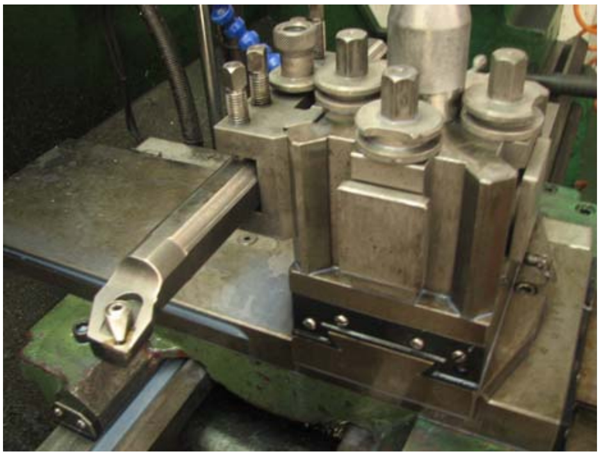

Figure 37: Boring bar tightened to the tool holder**.**

To get surface finish $\overset{32}{\surd}$ which is 1.25 μ.m, the feed rate must be $f = 0.063$ mm/rev. The insert that be used is *TPMR 16 03 04* which is radius = 4mm. The speed of spindle is about, 500 rpm. The speed of spindle it depends on the hardness of workpiece. If the harder material be used, such as stainless steel 316, the spindle speed could be 300 rpm, which is less than softer workpiece. For finishing, cutting thickness is about 0.01” per cutting.

For the OD, the internal boring bar be changed to the external boring bar as shown is figure 38. The insert be used is *TPMR 16 03 06* which is 6mm radius. The boring bar will be tightened to the tool holder, and the roughing the OD is started. For the roughing process the feed rates is 0.15 mm/ rev which is greater than finishing.

After that, the insert be changed to the *TPMR 16 03 04* and set the feed rate to 0.063mm/rev so that the surface finish value is about 1.25 µ.m. For finishing, cutting thickness is about 0.01” per cutting. The machining process continues until gets the desire dimension (Φ 3.75”).

Figure 38: External Boring Bar

After that, the internal boring bar is used with 6mm radius of insert. The desire ID is 4.608” and the depth is 0.844”. The further step is shown in figure 39.

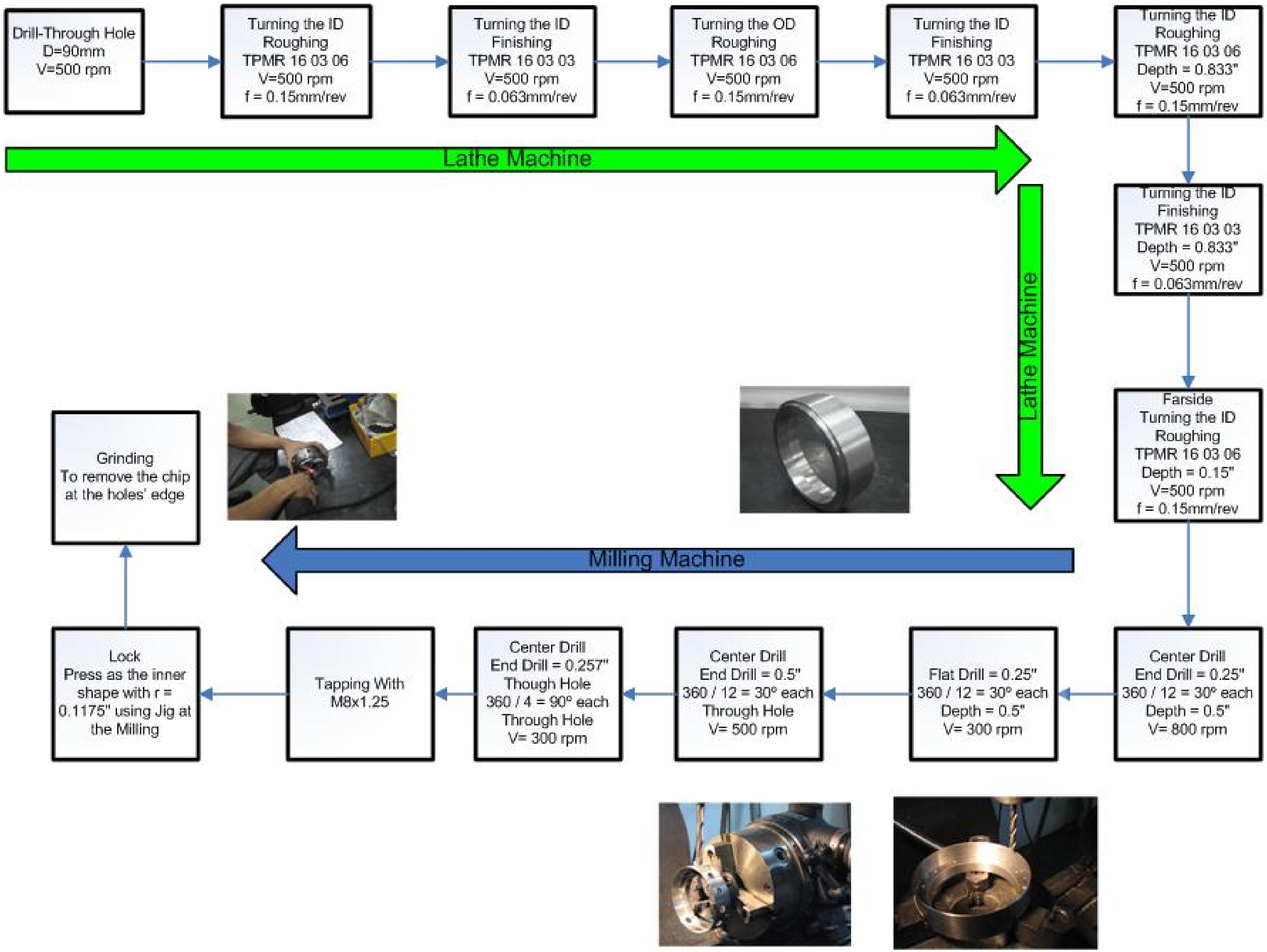

Figure 39: Flow Diagram for Machining the Spring Holder

4.4.2. Retaining Ring

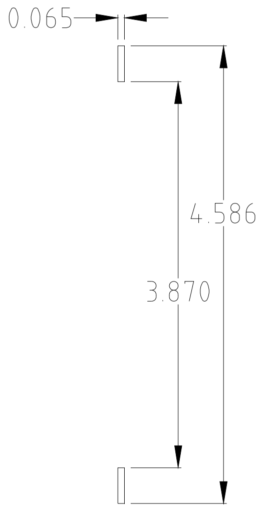

Figure 40: Retaining Ring’s Dimension

Put the workpiece in the jig of the spindle at the lathe machine. Drill a through hole, and use internal turning boring bar to get the ID of the retaining ring. It is not necessary to make the surface roughness 1.26 micrometer, because the surface at the retaining ring is not important.

Use the external turning boring bar to cut the OD of the workpiece. After that, the workpiece be faced with 0.065” thickness. After that, milling machine be used to make a hole at retaining ring. Figure 41 shown the compression set that already being assembled.

Figure 41: Compression Set Assembled

4.4.3 Stationary Face

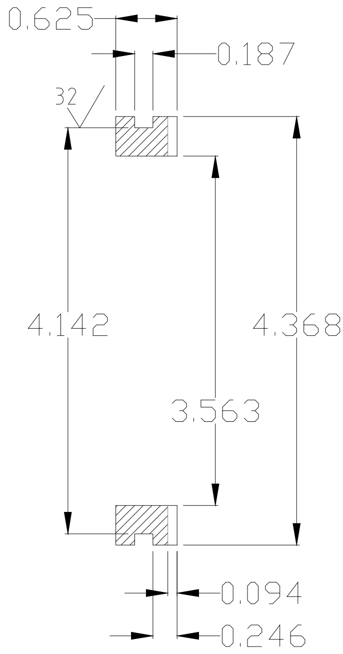

Figure 42: Stationary Face's Dimension

The material for the stationary face is Stainless Steel 316. It is harder than mild steel. Therefore, the machinist must reduce the speed of spindle. Drill a through hole with diameter of drill 90mm at the spindle speed 500 rpm. Below are next step for machining the stationary face.

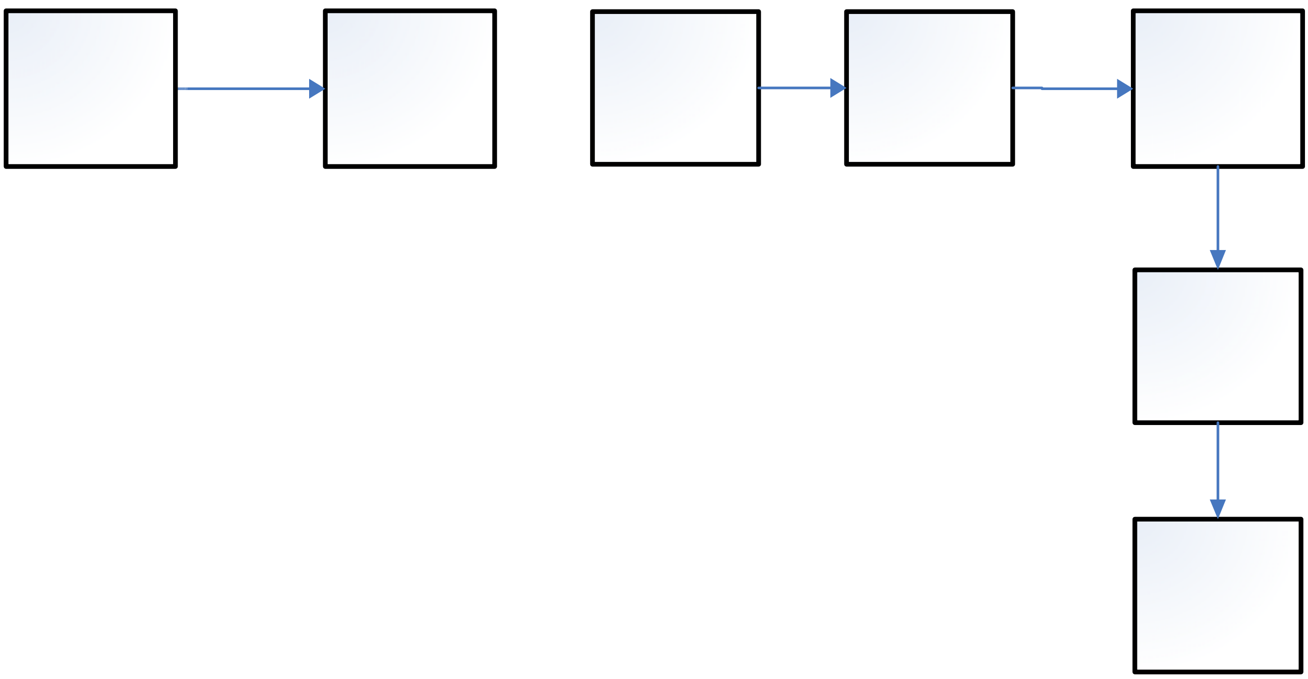

Figure 43: Flow Diagram for Machining the Stationary Face

4.4.4 Sleeve

The machining steps for sleeve are shown in figure 46. Figure 44 is the sleeve dimension for machining process.

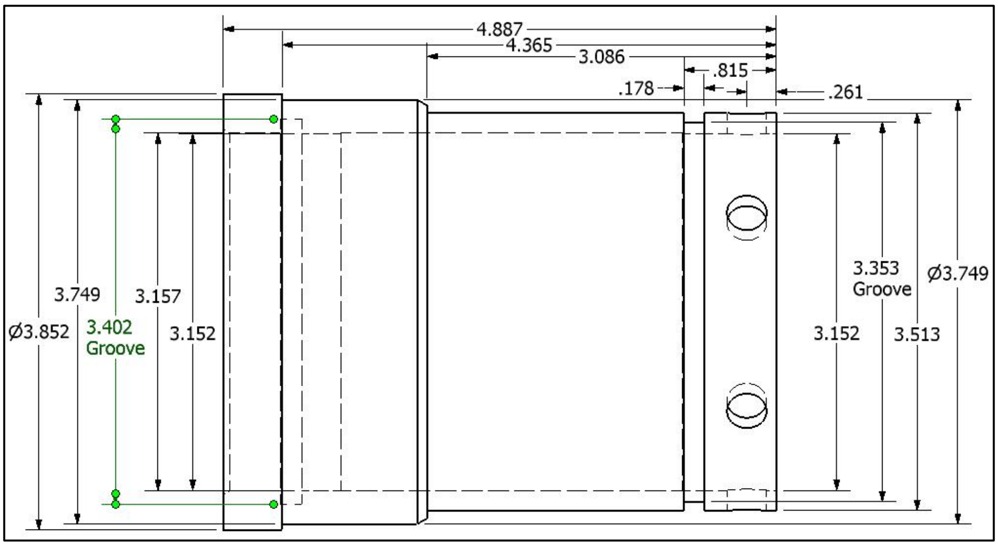

Figure 44: Sleeve's Dimension

For precaution when drill a hole by using milling machine, an adjustable rod being used to reduce the deflection effect on the sleeve. Figure 45 is the picture of adjustable rod.

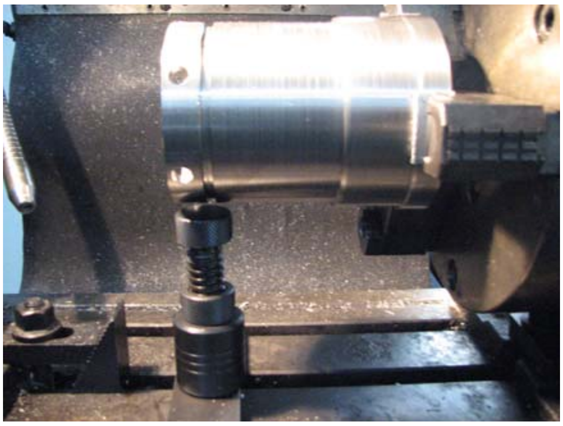

Figure 45: Adjustable rod

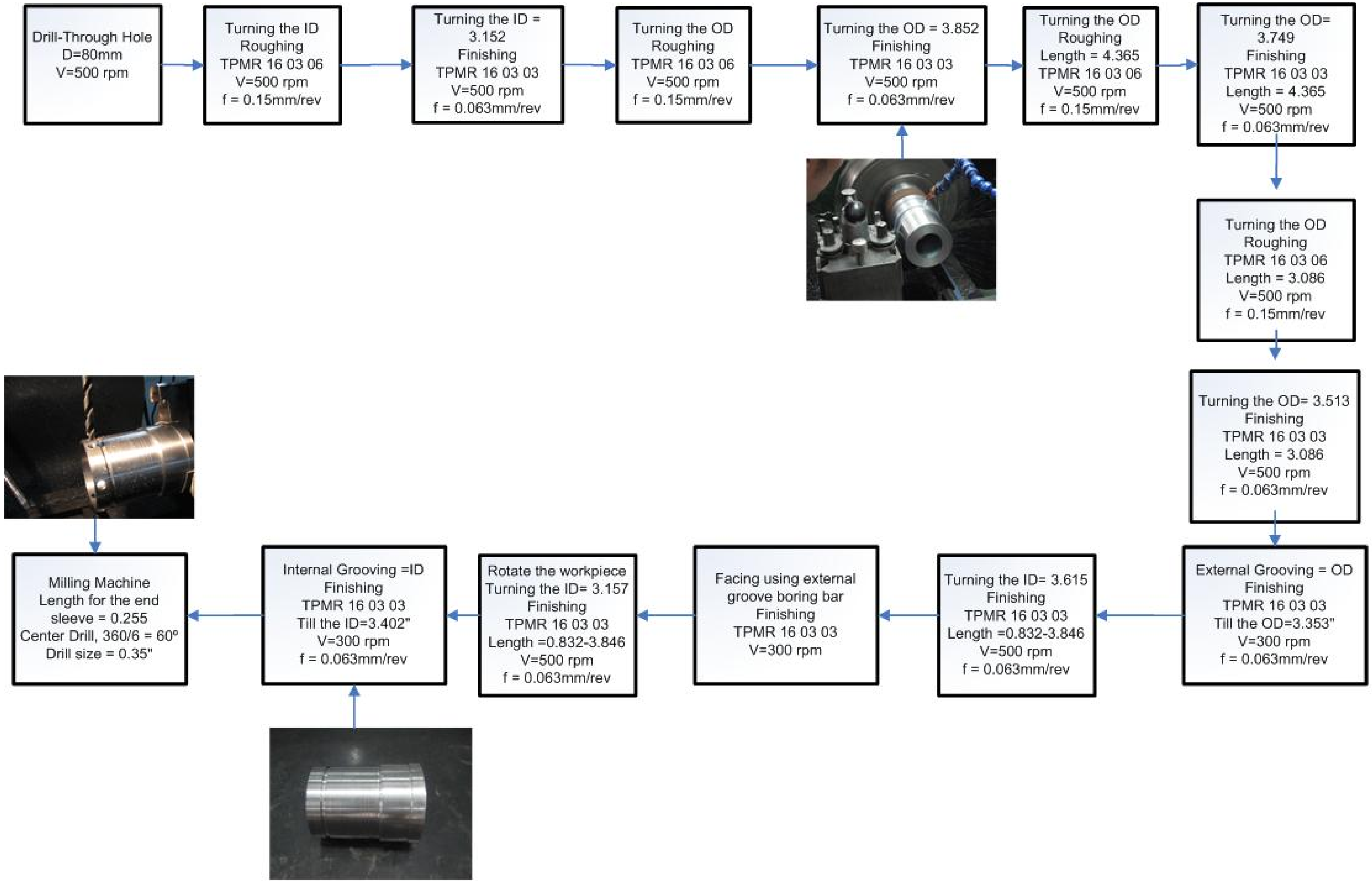

Figure 46: Flow Diagram for Machining the Sleeve

4.4.5 Sleeve Collar

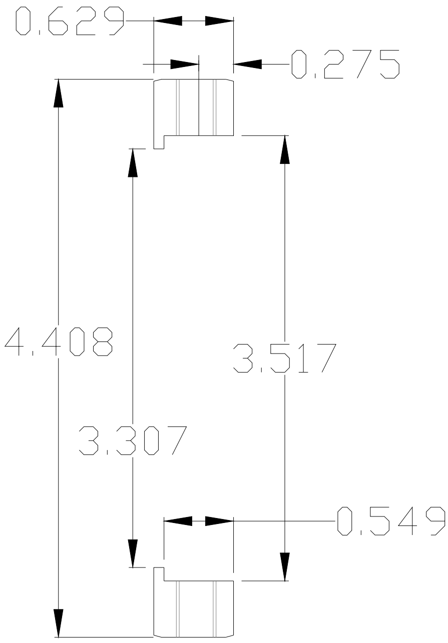

Figure 47: Sleeve Collar's Dimension

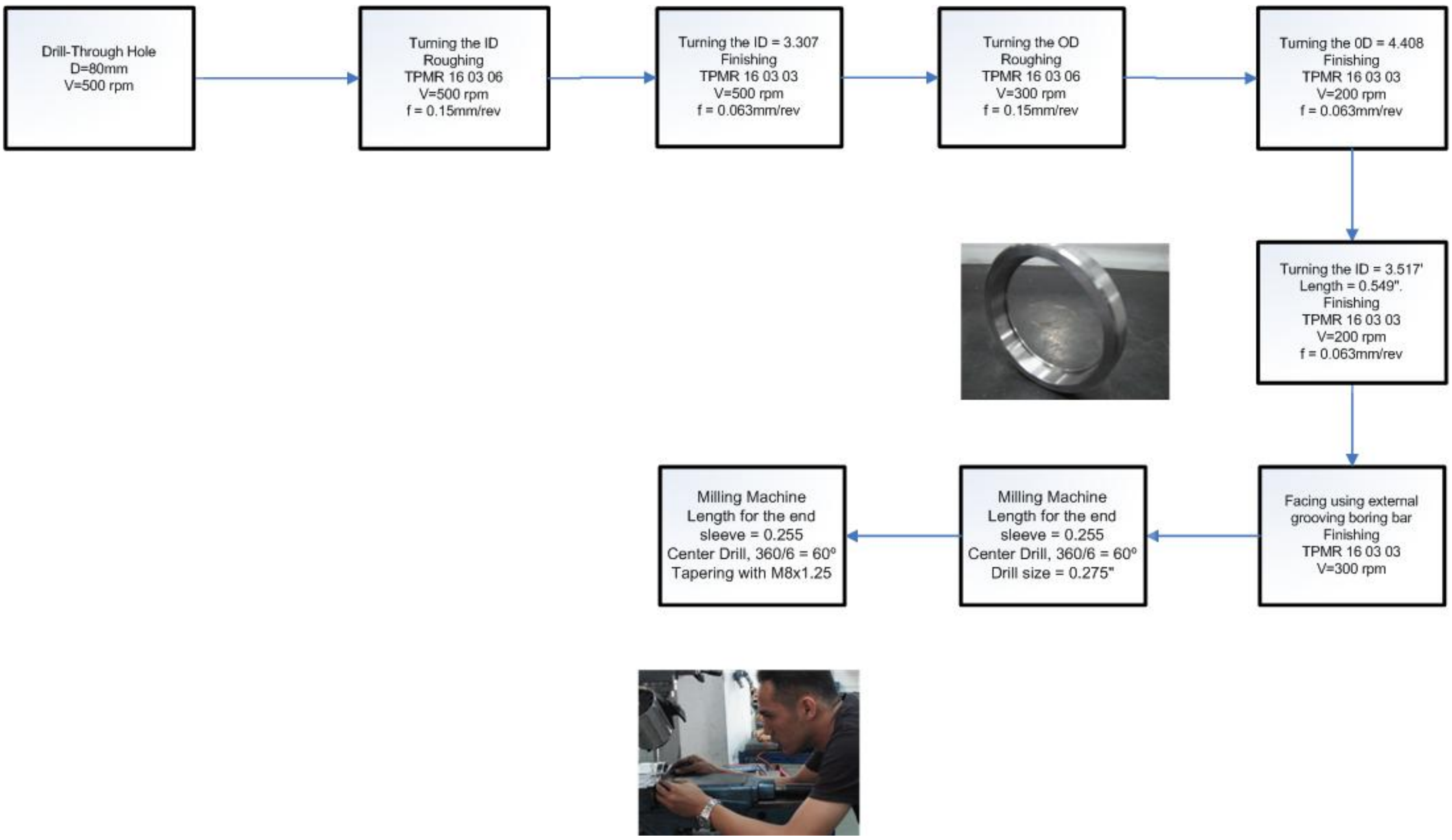

Figure 48: Flow Diagram for Machining the Sleeve Collar

4.4.6 Gland

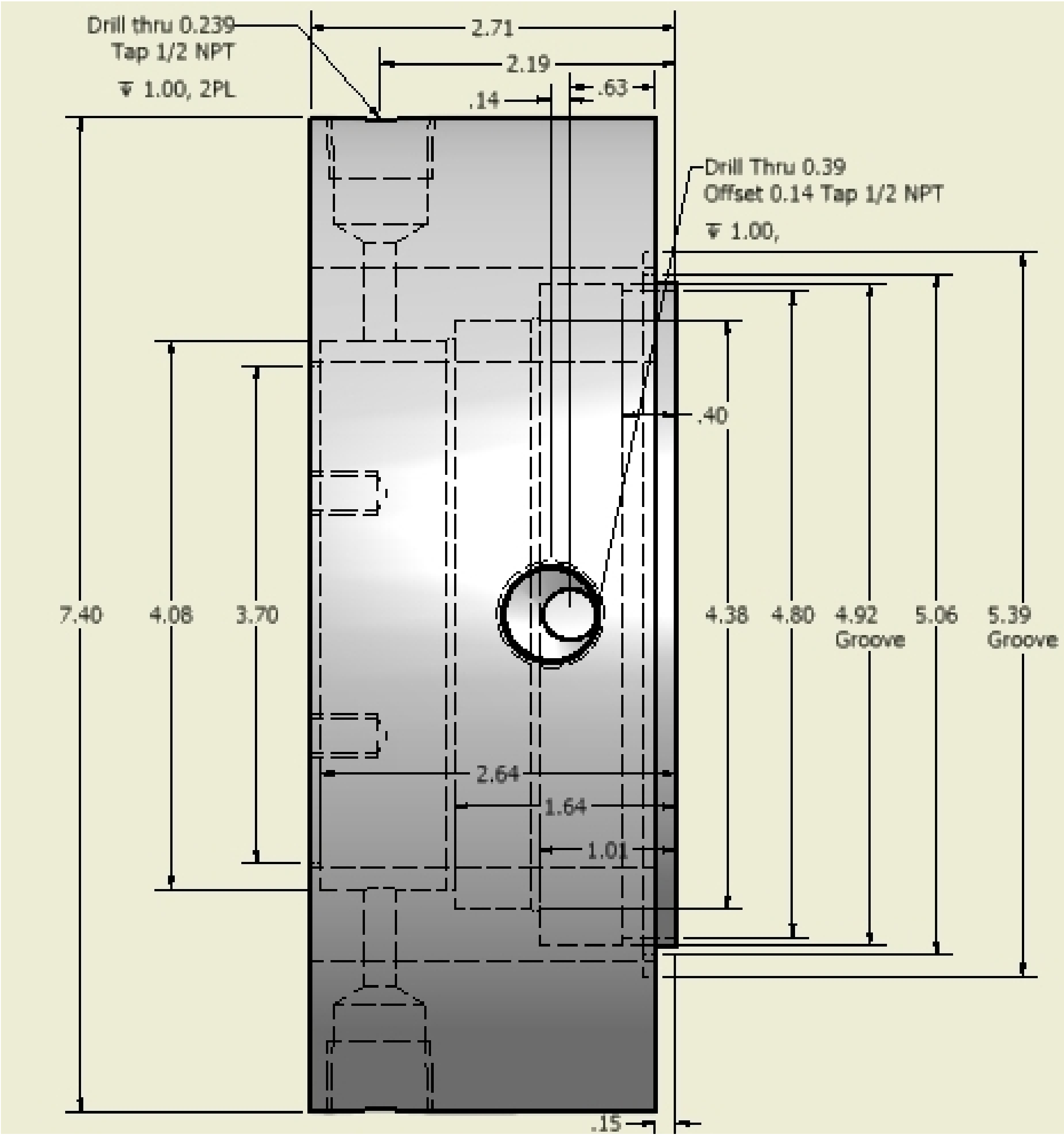

Figure 49: Gland's Dimension

For the gland, the lathe machine must have spindle size bigger enough to clamp the outer diameter of the gland. All the steps for machining the gland is shown in figure 50.

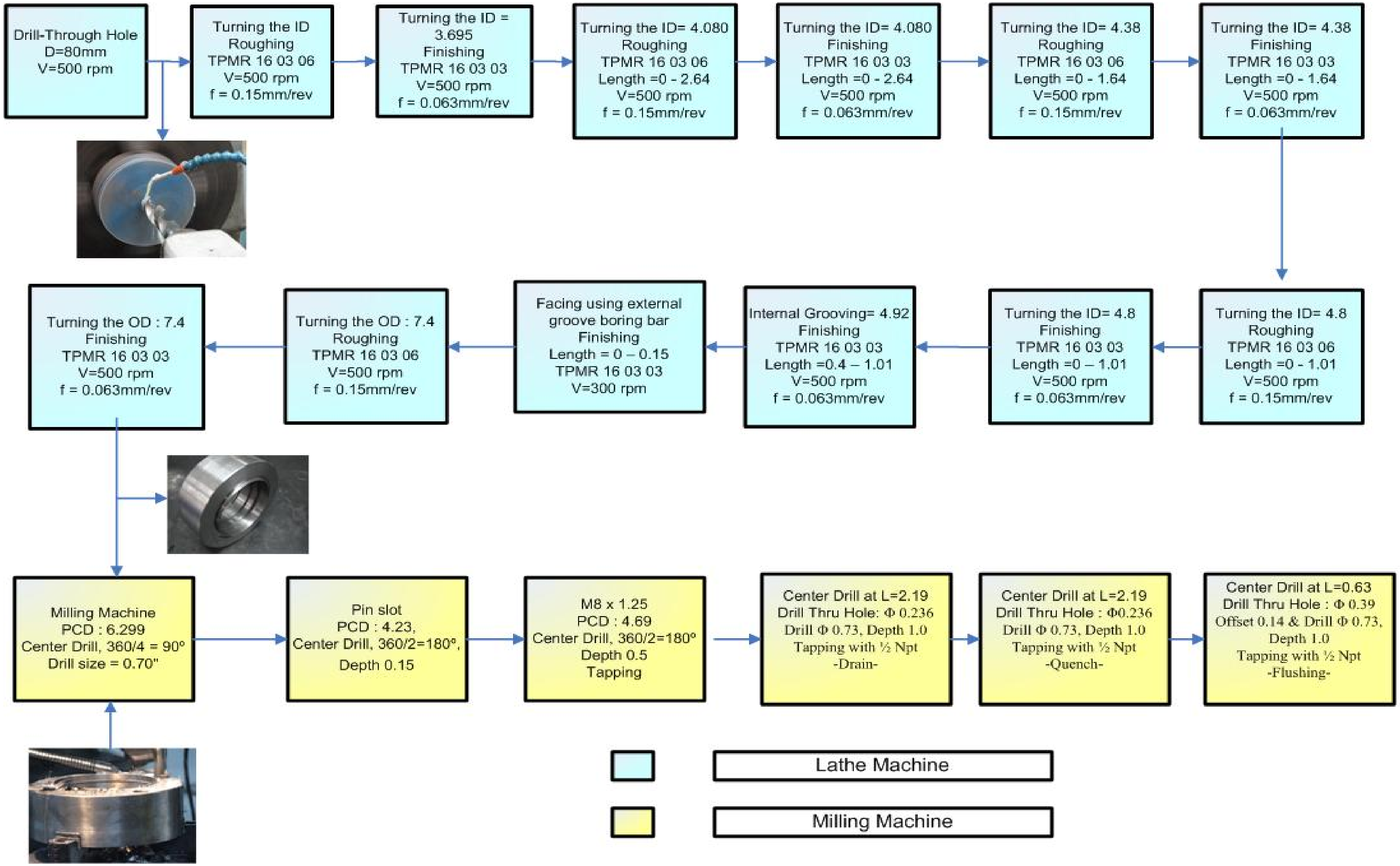

Figure 50: Flow Diagram for Machining the Gland

4.4.7 Throttle Bush

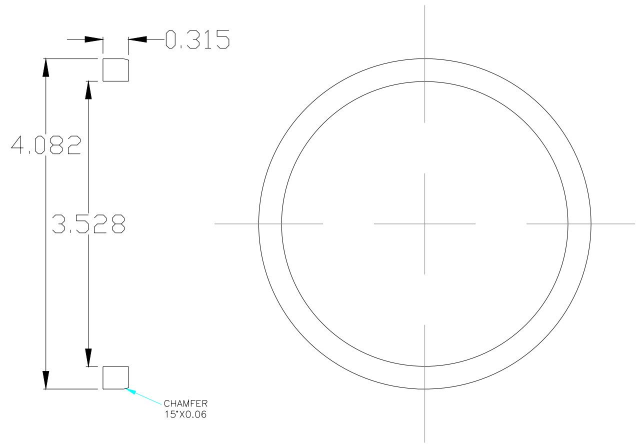

Figure 51: Throttle Bush's Dimension

Throttle bush is the easiest part to be machined. It is just involve its ID, OD and thickness.

4.4.8 Setting Plate

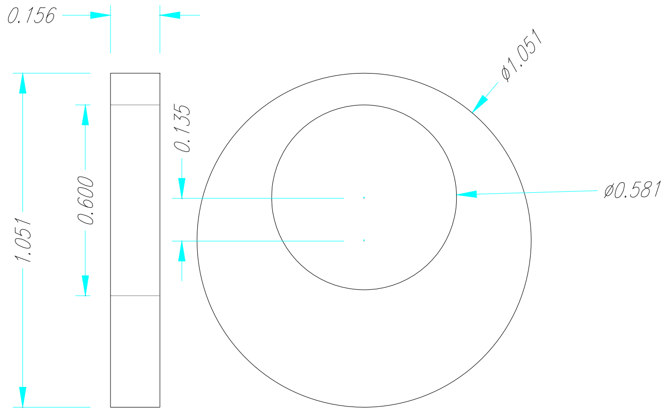

Figure 52: Setting Plate's Dimension

For setting plate, a stainless steel rod is tightened to the jig of the spindle at lathe machine. The OD for the rod is 1.051" above. External turning to get its OD by using TPMR *16 03 04* insert. After that, faced the rod with external turning bar and set the thickness 0.156". For next steps, milling machine be used to make a hole. Set the drill at the centre of setting plate, and move horizontally 0.135" and make a mark using centre drill. Drill a through hole using Φ 0.6 drill bit. For finishing, the hole is grinded to remove the exceeded bur at the edge of hole.

4.5. Mechanical Seal's Parts Assembly

After all mechanical seal's parts already are fabricated; all the parts will be assembled to form a mechanical seal. The precautions when assembly process is made are shown below:

- Swept the silicon (a waxy liquid) to the o-ring and on the o-ring location to reduce friction when assemble the mechanical seal's parts.
- Cleaning up the rotary and stationary face's surface.
- Swept the oil the rotary and stationary face's surface and rotate the parts to spread the oil evenly.

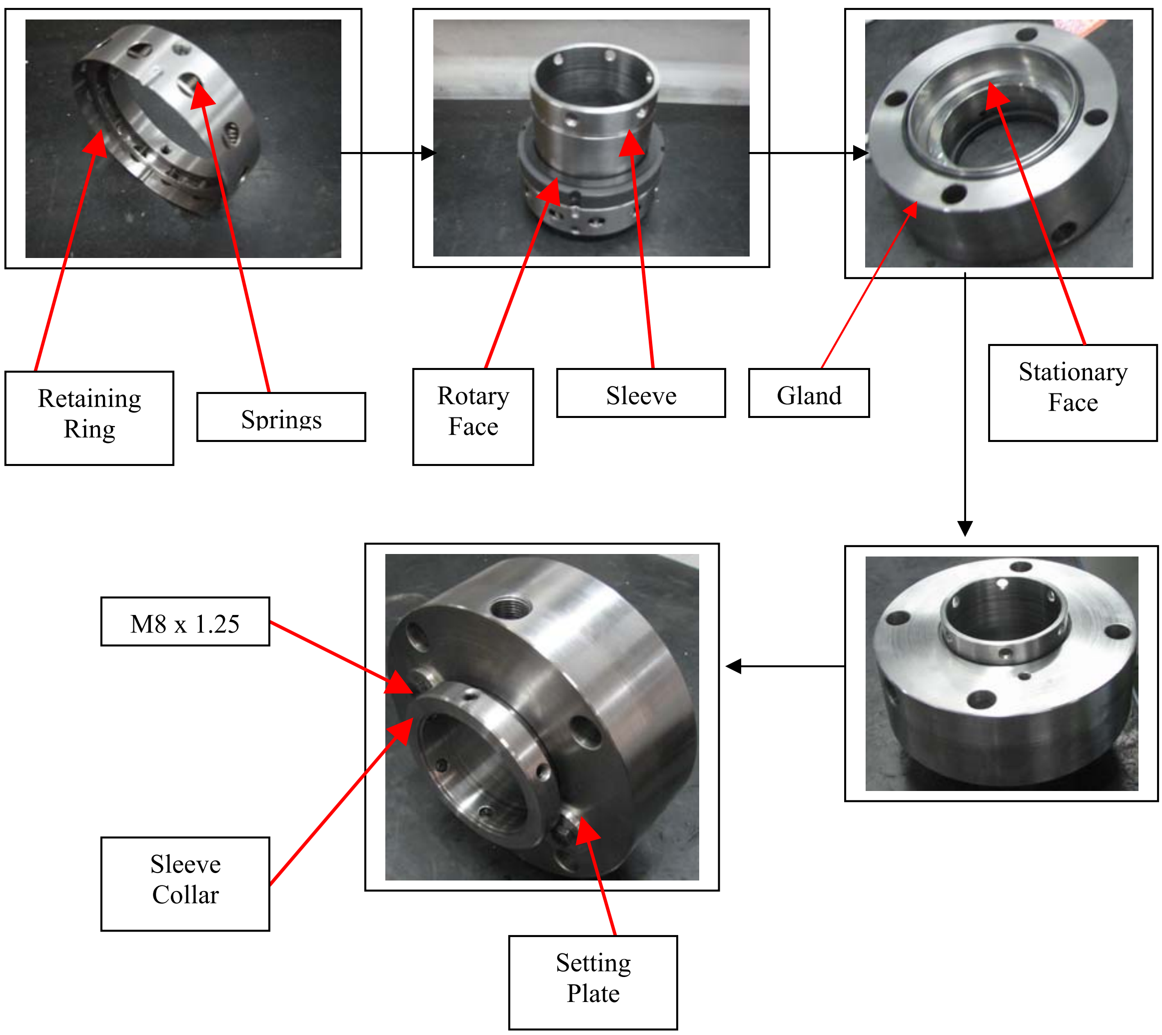

Figure 53: Steps for Mechanical Seal's parts Assembly

4.6. Pressure Test

After the mechanical seal's parts are assembled, the pressure test will be made in ProEight Offshore Engineering Sdn. Bhd.'s factory. The purpose of pressure test be made is to detect any leakage that could occurred at any point in the mechanical seal. The pressure test setup is illustrated in Figure 54.

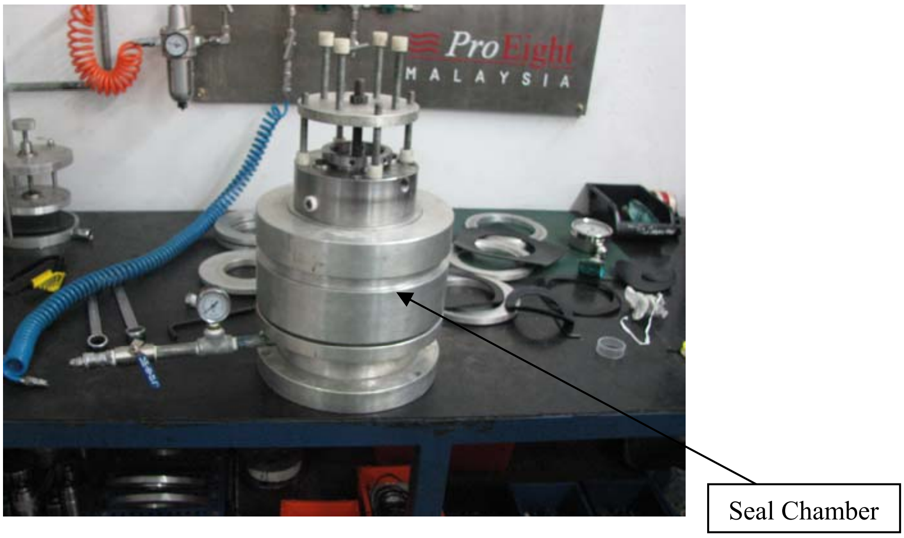

Figure 54: Pressure Test's Equipment

After mechanical seal is tightened in the seal chamber, all the pressure test equipment is set up as shown in figure 54. Open the gate valve and let's the air fill up the empty spaces in mechanical seal. After that, gate-valve is closed and read the pressure at the pressure gauge's meter as shown in figure 54.

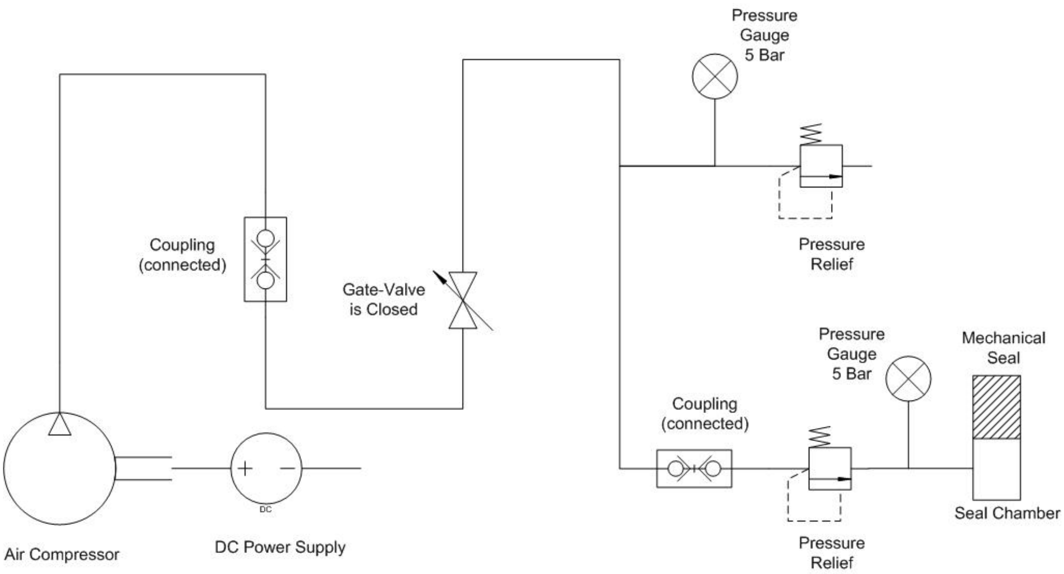

Figure 55: Schematic Pressure Test Diagram Initial Condition

After that, the pressure will be relieved at pressure relief until get the pressure 2.00 bars. For API 682 standard, mechanical seal can be tested at 2.00 bars by using air as a sealing fluid even the actual pressure is about 7.5 bars and using hot water as its sealing fluid. This is because of the molecule of air is much smaller compare to water molecules. So, decrement in pressure is related decrement of molecule of sealing fluids' size.

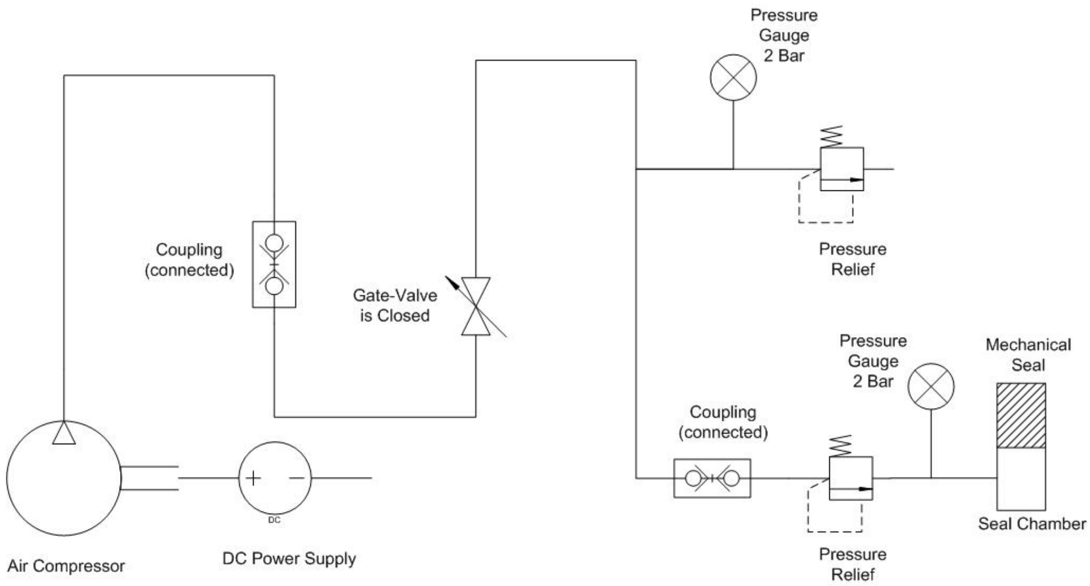

Figure 56: Schematic Pressure Test Diagram After relief the pressure

The coupling is disconnected to check is it any pressure drop at the pressure gauge. The initial pressure reading is 2.00 bars. The pressure readings at the pressure gauge have been taken every 5 minute until 30 minute.

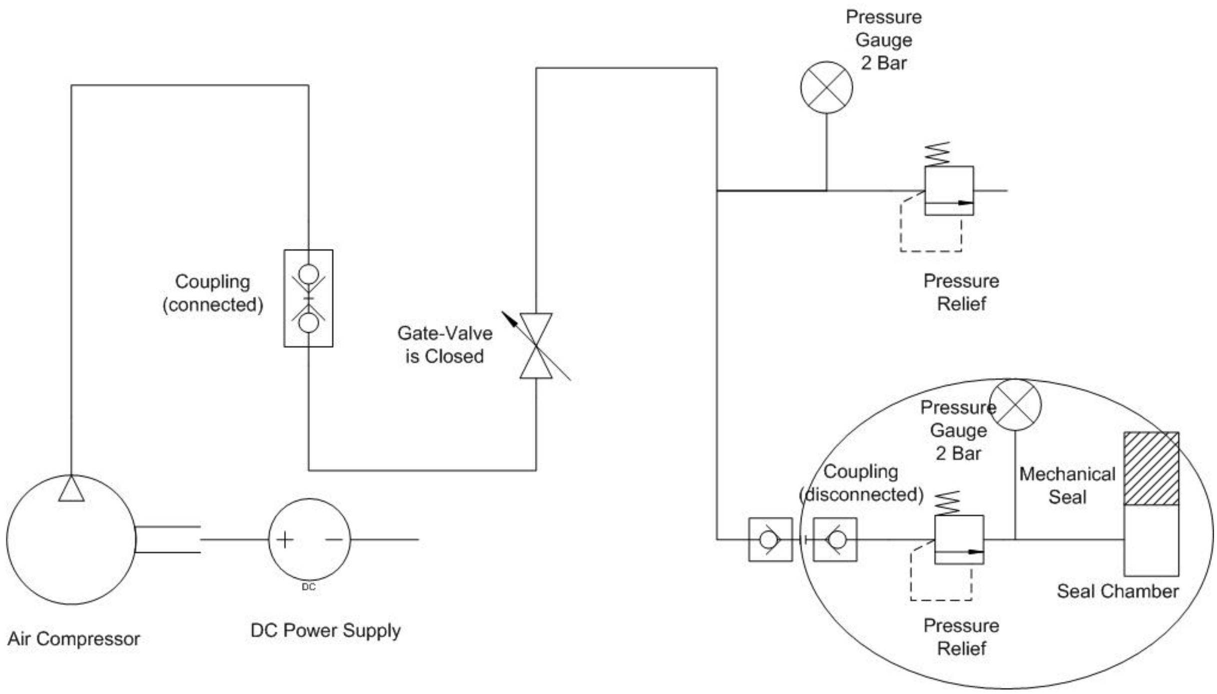

Figure 57: Schematic Pressure Test Diagram after disconnect the coupling

If there is any pressure drop, the mechanical seal is leak. For this project, there is no pressure drop at the pressure gauge. Therefore, this mechanical seal can be used in pump-1207 belong to Vinyl Chloride Malaysia safely. ProEight Offshore Engineering Sdn. Bhd has awarded certificate for this pressure test and it is attached in appendix 12.

CONCLUSION

5.1 Conclusion

The main purpose of this project is to build a new mechanical seal that could reduce the possibility of failure. The failure data history of the mechanical seal be analyzed. All the failure data history is gathered from the ProEight Offshore Engineering's technicians that already make the inspection to this mechanical seal. The root causes that lead to this failures is excessive heat at the mating face, shaft deflection cause by the overload to the shaft and this mechanical seal do not follow API 682 standard.

The excessive heat will lead to the leak of the pump when the heat will dilute the o-ring. Beside that, excessive heat will vaporize the sealing fluid. The bubble produced, will reduce the efficiency of the pump. The API 682 standard does no allow the inserted seal face. This is because of, when the expansion rates of the carbon greater that steel and it will crack the carbon. Deflection of shaft could make the stationary and rotary face's surfaces do not parallel anymore and it will open the gap at the mating face. The sealing fluid could escape atmosphere side through the opened mating face.

The solution for excessive heat is by redesign the rotary face. To reduce the effect of back pressure on the rotary face, proper balance ratio be used as a basis to design this rotary face. By reducing the force acting at the mating face, it will reduce friction force and the heat generated will decrease. For API standard, the seal face has been changed to the solid seal face instead of inserted seal face. Beside that, the deflection of the shaft be reduced by simplified the design and reduce the weight of parts that acting on the shaft.

For fabrication, all the process has be done at ProEight Offshore Engineering's factory in Gebeng, Pahang. ProEight has given their equipment and train how to machine the mechanical seal's parts. The main challenge when fabricating this mechanical is to keep surface roughness of the mechanical seal's parts especially at the location of the o-ring. If there is a sharp edge at the sleeve's groove, the o-ring may break off and it will lead to the pump leakage. Therefore, the sharp edges need to be grinded and keep the surface roughness as the theoretical value. At the same, there are some precautions when machining process is done.

For the final task, the mechanical seal has been assembled and be tested under a pressure test based on the API 682 standard. Consequently, all the design requirement has been met successfully.

5.2. Recommendations

The rotary and stationary face need to simulate with software that could check the stress concentration on the seal face's surface. By doing the stress analysis, the stress concentration at the mating face of the new design could be compare with the old design. Beside that, the simulation of temperature analysis could make the process of designing easier. At the same time, simulation of the stress and temperature can reduce the cost for prototyping the model.

REFERENCES

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[3] American Petroleum Institute. September 2004, "*Pumps – shaft Sealing System for Centrifugal and Rotary Pumps*."

[4] Durametallic Corporation. 1991, "*Dura Seal Manual, 9th edition*", Michigan USA, Durametallic Corporation

[5] Mcnally. 2007 <http://www.mcnallyinstitute.com/index>

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[7] *L.L Faulkner. 1995,* "*Fluis Sealing Technology*". Marcel Dekker, Inc

[8] J. Martini, 2001, "*Practical Seal Design*", Dekker

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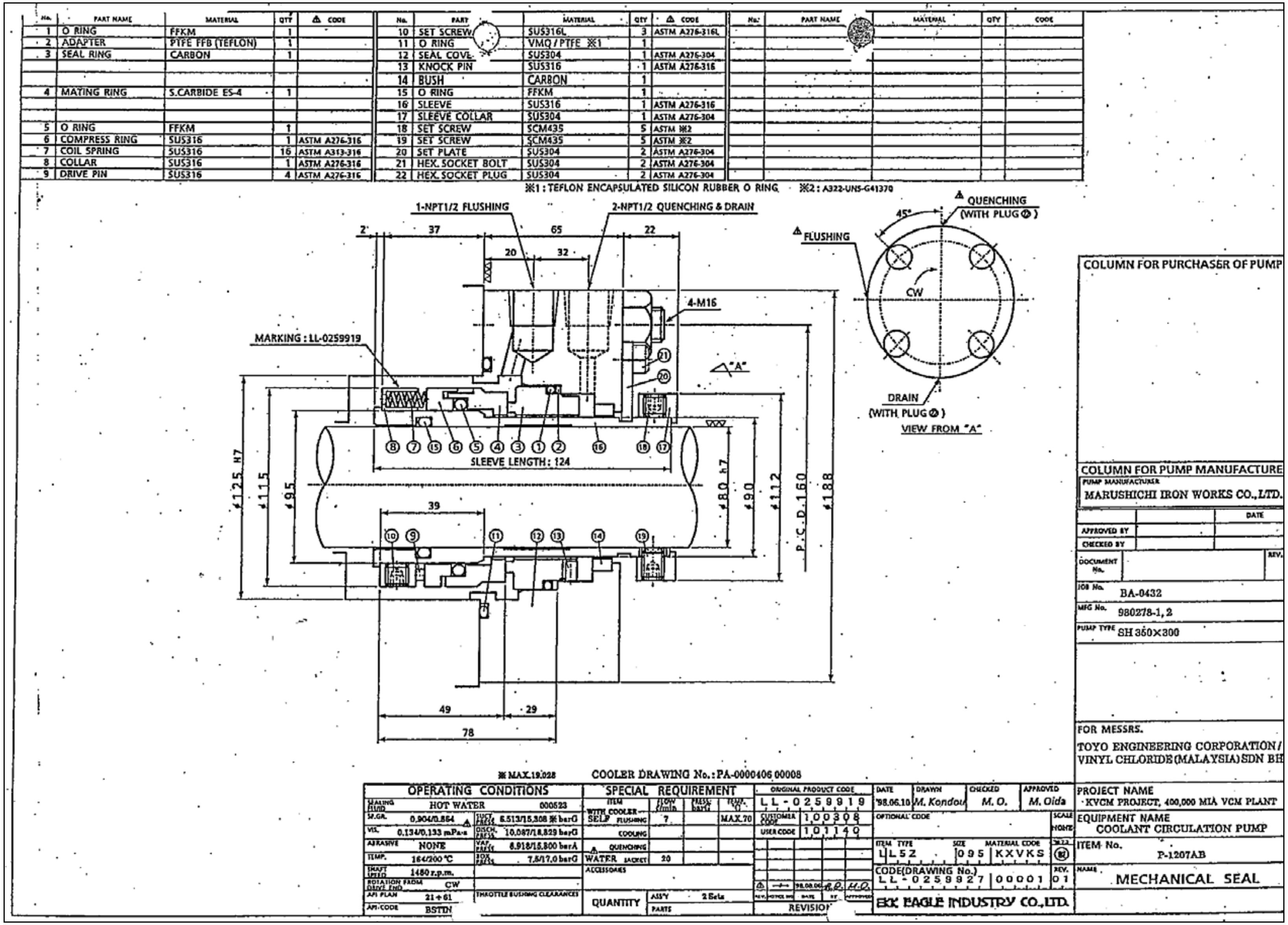
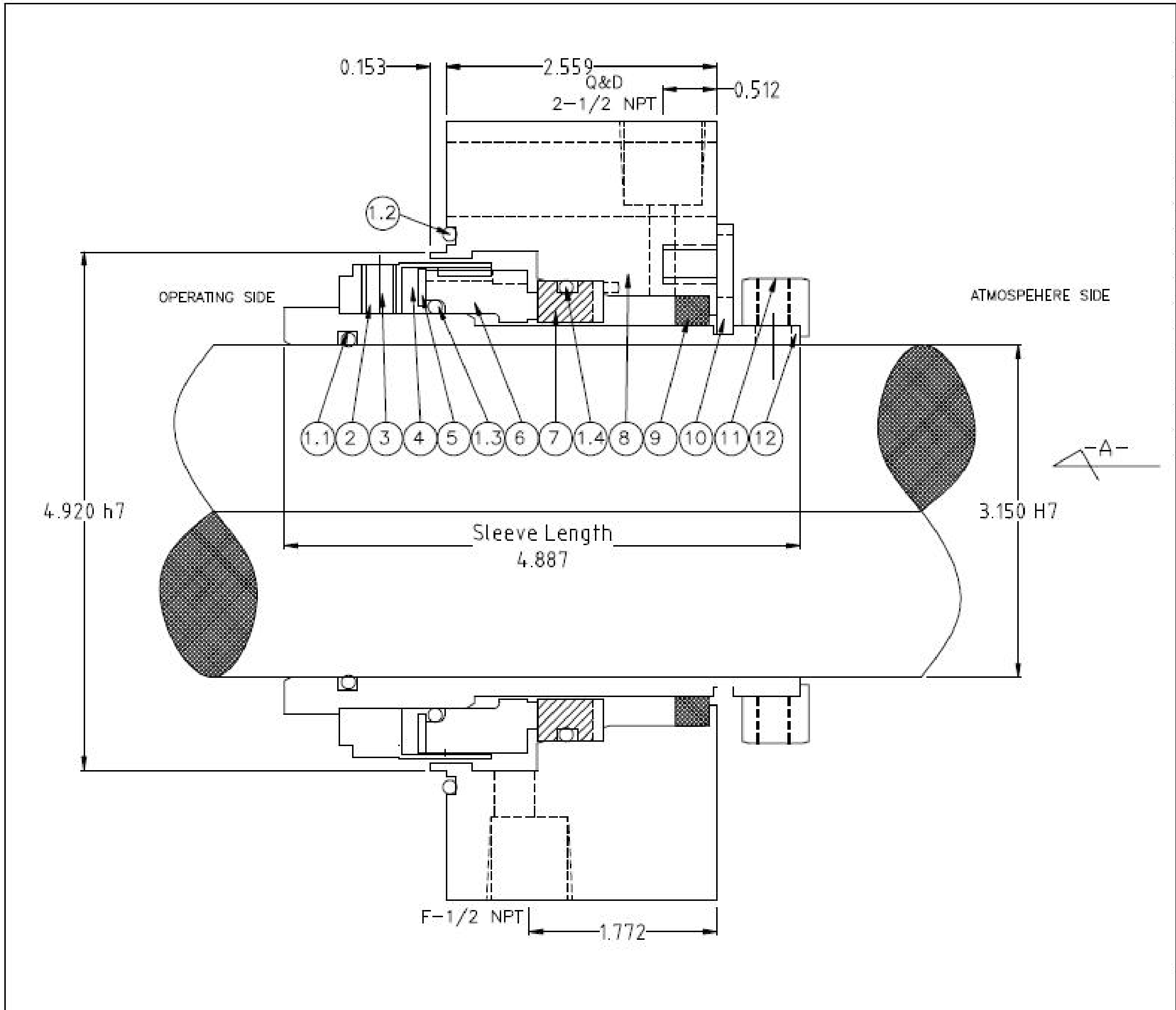
[10] Industrial Training Report by Emier Helmee Md Dewa, December 2006-July 2007.

APPENDICES

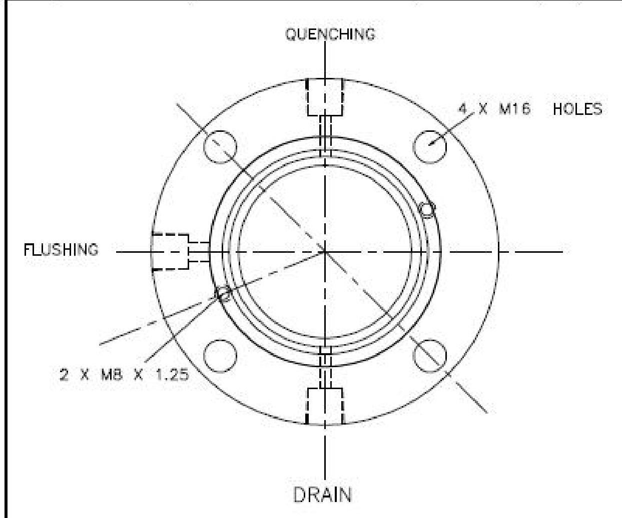


Appendix 1: Old Mechanical Seal Assembly Drawing



PARTS LIST

DWG REF.	PART CODE	DESCRIPTION	MATERIAL	QTY
1.1	226	O-RING	KALREZ	1
1.2	229	O-RING	KALREZ	1
1.3	232	O-RING	KALREZ	1
1.4		O-RING	KALREZ	1
2		SPRING HOLDER	316 SS	1
3		SET SCREW M8 X 1.25	316 SS	6
4		SPRING		6
5		RETAINING RING	316 SS	1
6		ROTARY SEAL FACE	CARBON	1
7		STATIONARY SEAL FACE	SiC	1
8		GLAND	316 SS	1
9		THROTTLE BUSH	CARBON	1
10		SETTING PLATE	316 SS	2
11		SLEEVE COLLAR	316 SS	1
12		SLEEVE	316 SS	1



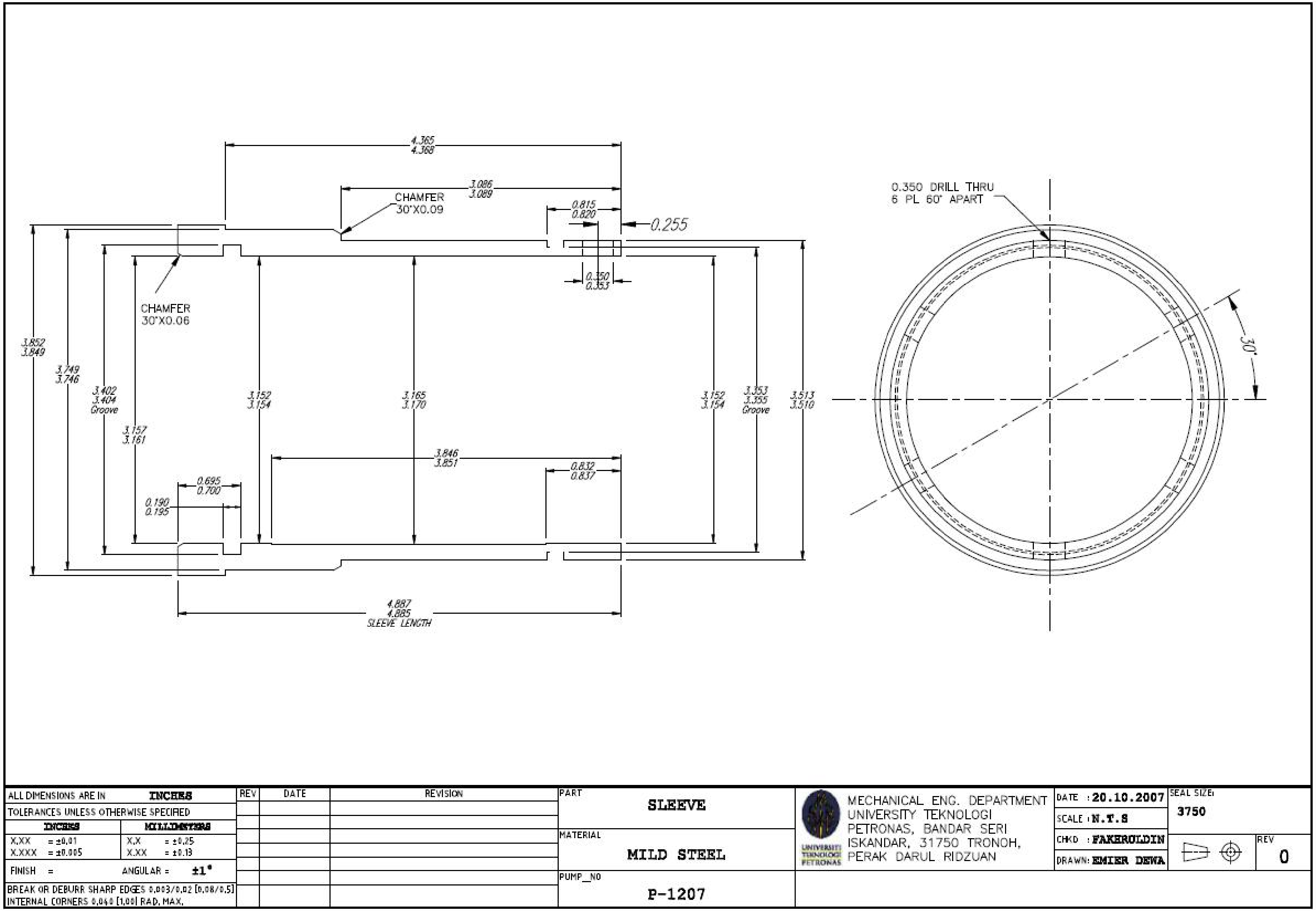
VIEW FROM -A-

CUSTOMER: VCM	* : MAX 10.010 barG		
OPERATING CONDITION:		*SEAL DATA*	*COLUMN FOR CUSTOMER & PUMP MANUFACTURER:*
SEALING FLUID: HOT WATER	SUCTION: 6.512 * barG	TYPE:	PUMP MANUFACTURER:
SPECIFIC GRAVITY: 0.904	DISCHARGE: 10.037 barG	SIZE: 3750	PUMP TYPE:
VISCOUSITY: 0.53-0.133 mCP	STUFFINGBOX 7.5 barG	ORIGINAL PRODUCT CODE:	EQUIPMENT NAME: P-1207
ABRASIVE: NONE	NO.:	CUSTOMER CODE:	LOCATION: MALAYSIA
TEMP: 164-200 °C		USER CODE:	PLANT:
SHAFT SPEED: 1450 r.p.m			
API PLAN: 21+61			
APO CODE: BSTDN			

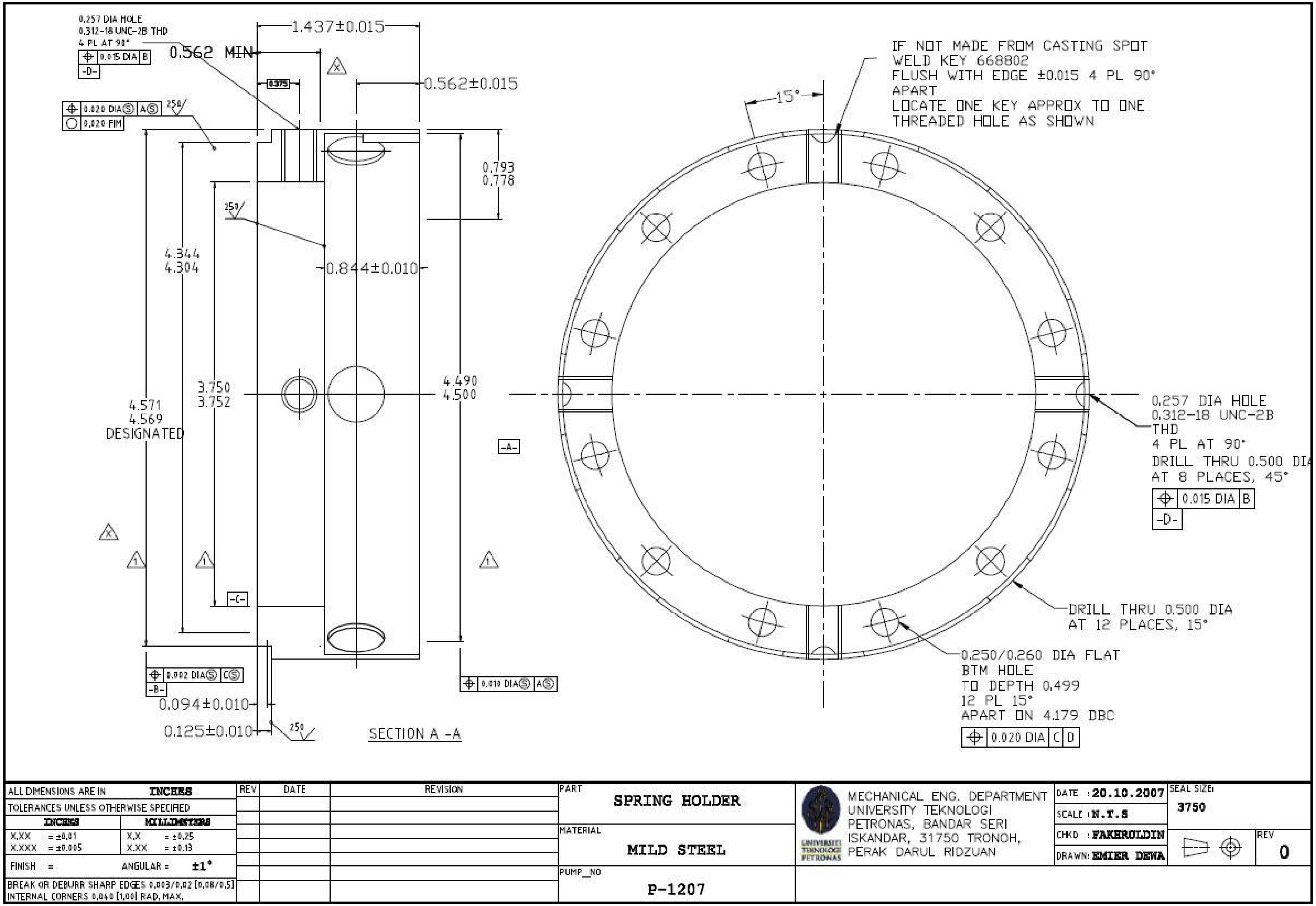
MECHANICAL ENG. DEPARTMENT
UNIVERSITY TEKNOLOGI PETRONAS, BANDAR SERI ISKANDAR, 31750 TRONOH, PERAK DARUL RIDZUAN

APPROVED BY:		DATE: 25.10.2007	REV 0
CHECKED BY:		SCALE: N.T.S	
ALL DIMENSIONS ARE IN INCHES	SHAFT DIA: 80 mm	CHKD: FAKHRULDIN	DRAWING NO:
ALL DIMENSIONS ARE REFERENCE UNLESS TOLERANCE		DRAWN: EMIER DEWA	

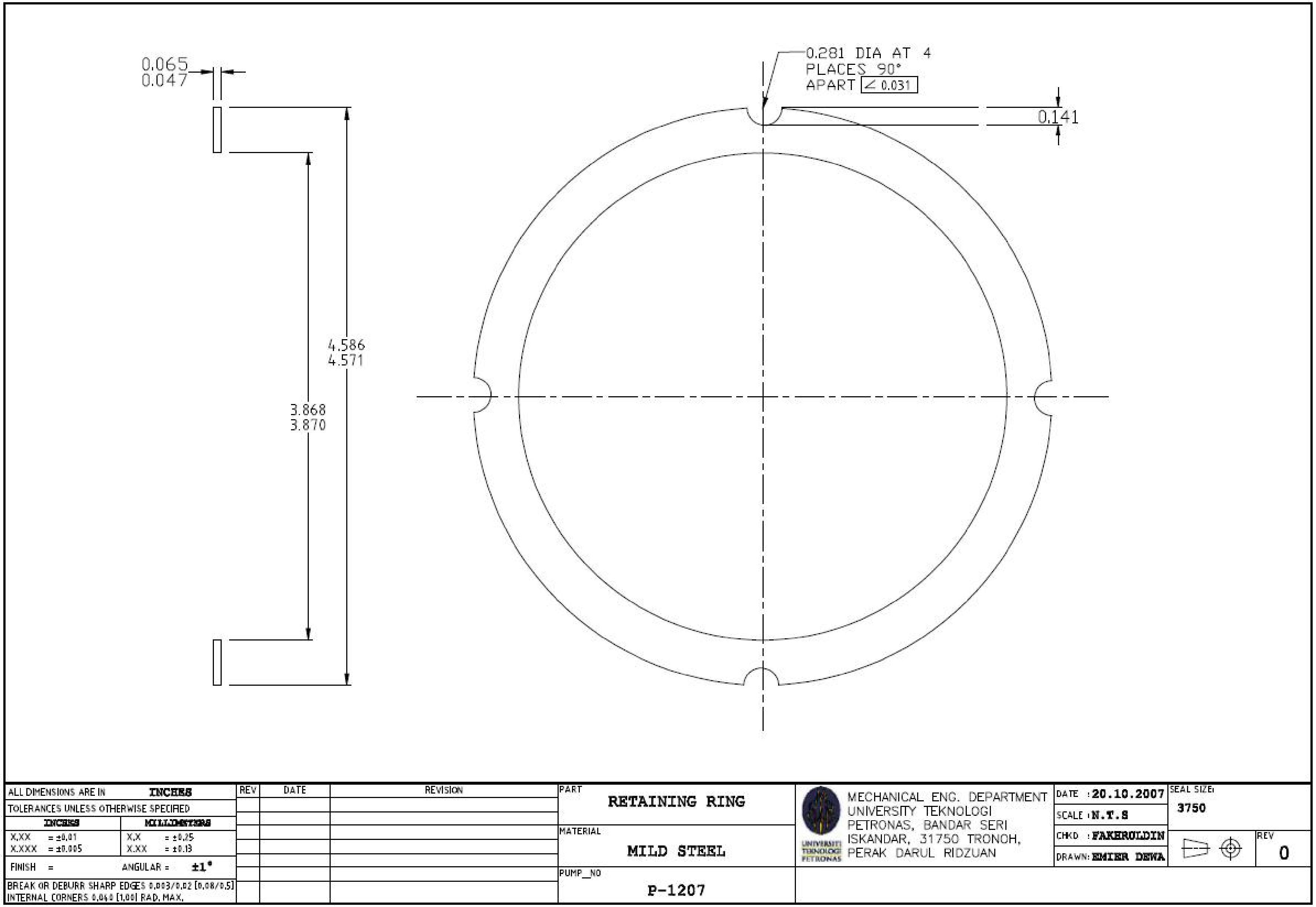
Appendix 2: New Mechanical Seal Assembly Drawing



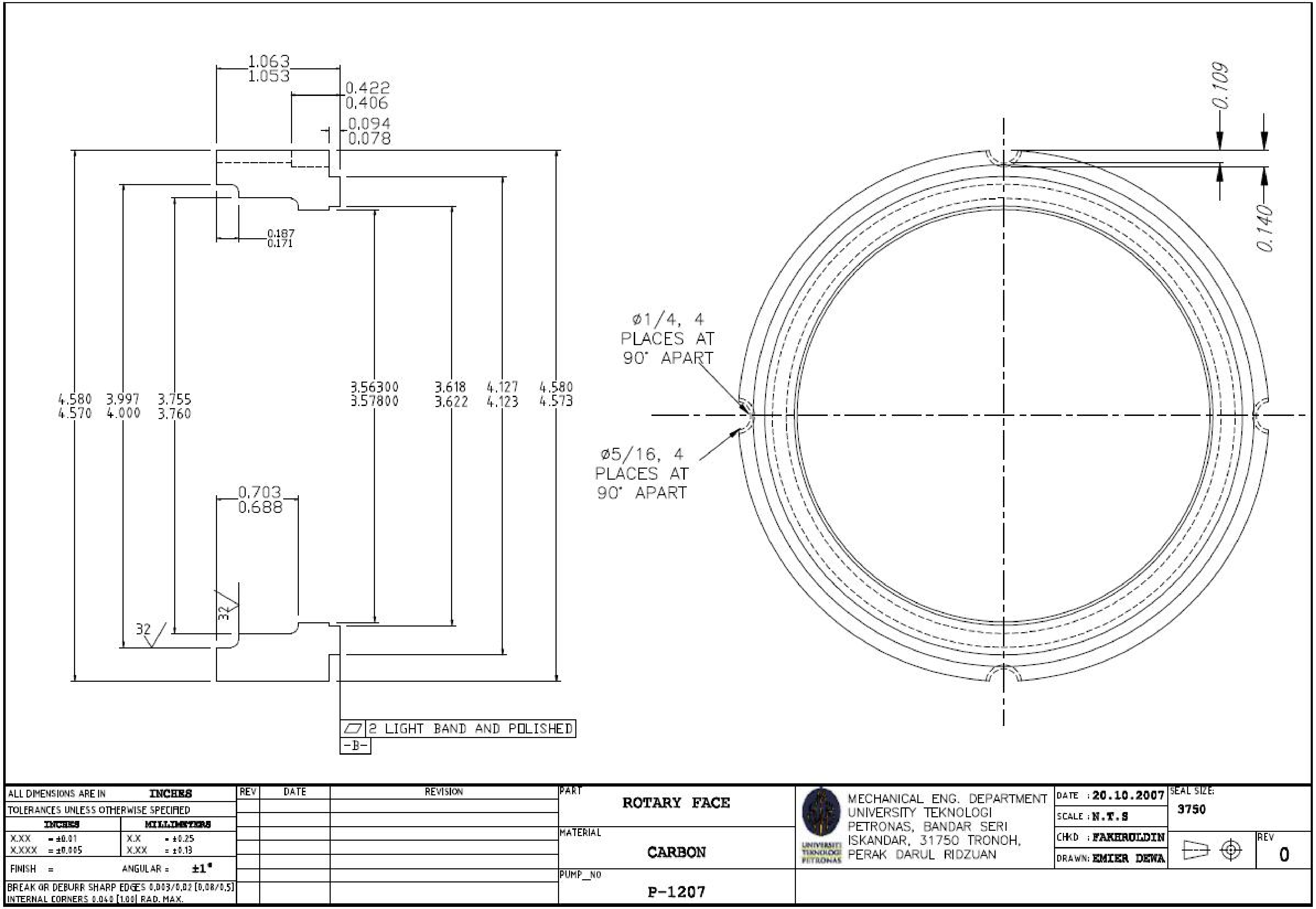
Appendix 3: Sleeve's drawing



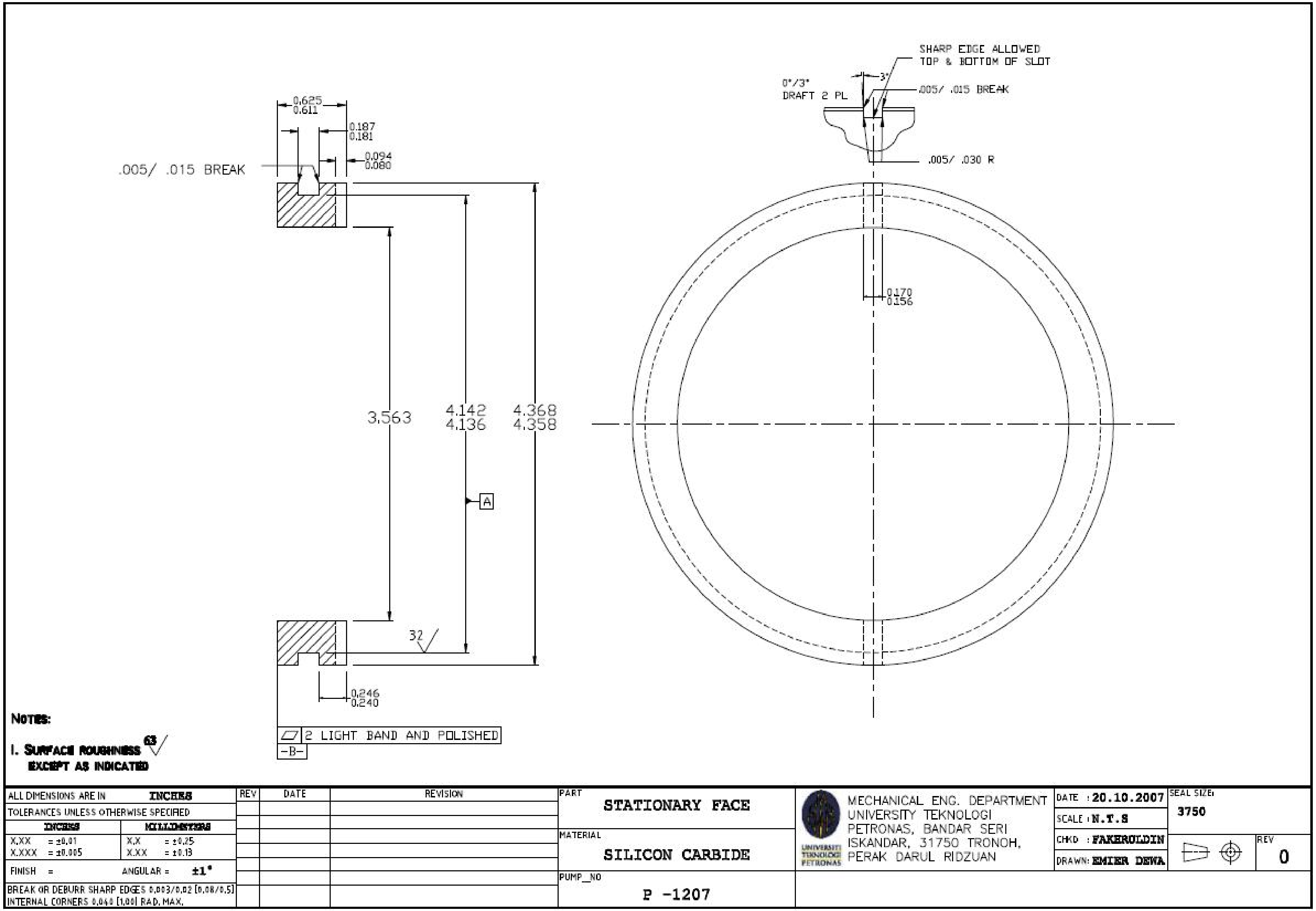
Appendix 4: Spring Holder's drawing



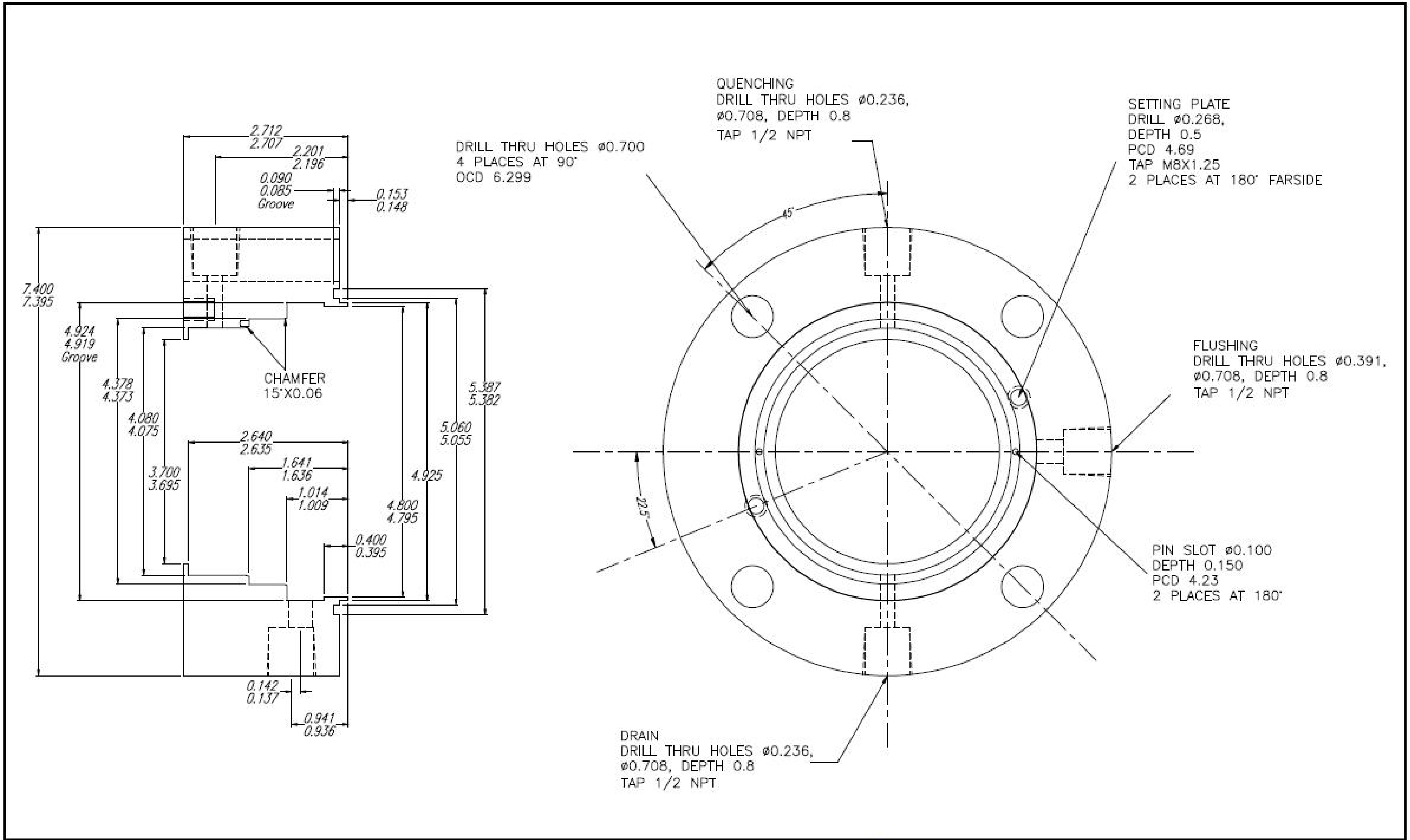
Appendix 5: Retaining ring's drawing



Appendix 6: Rotary face's drawing

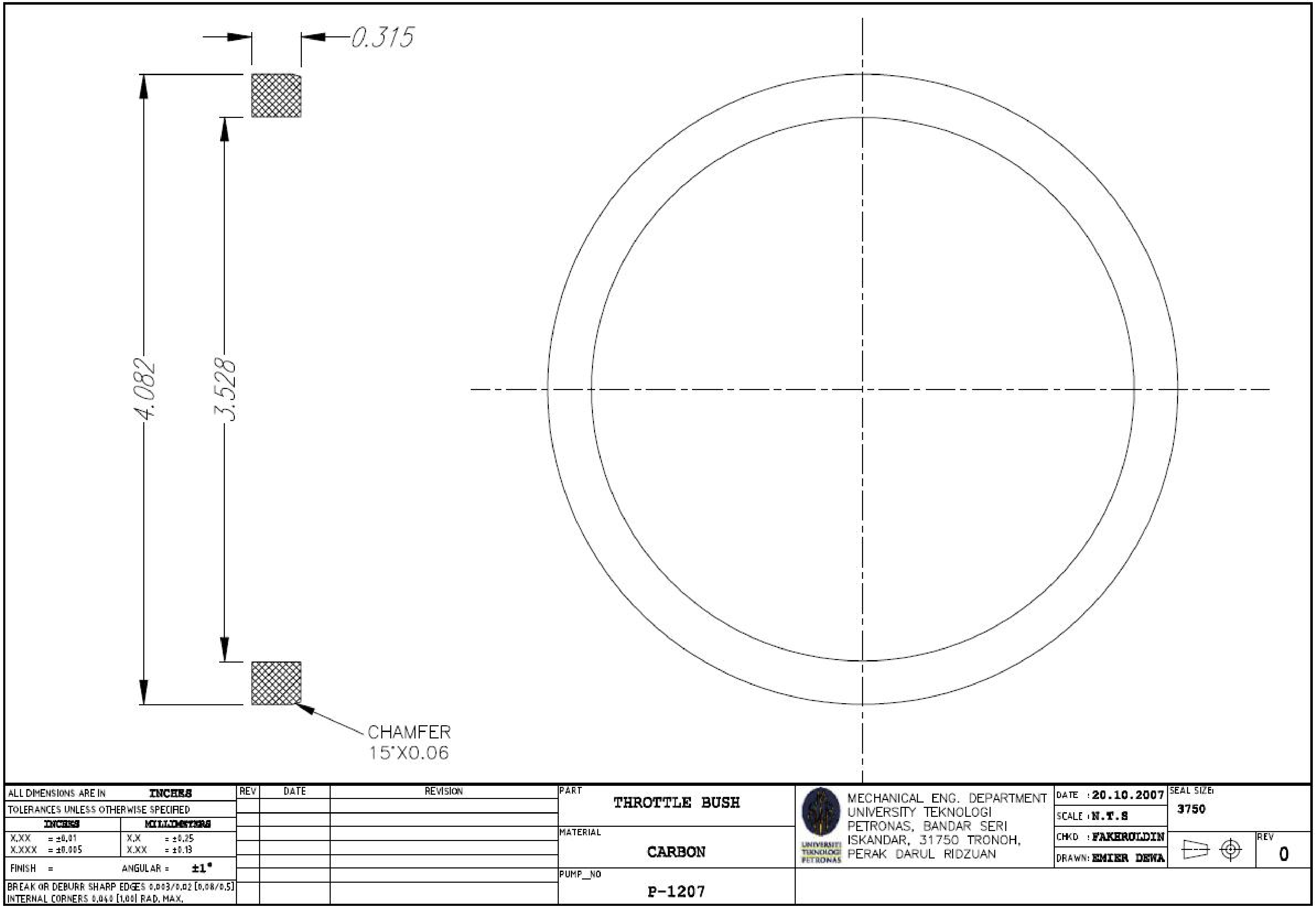


Appendix 7: Stationary face's drawing

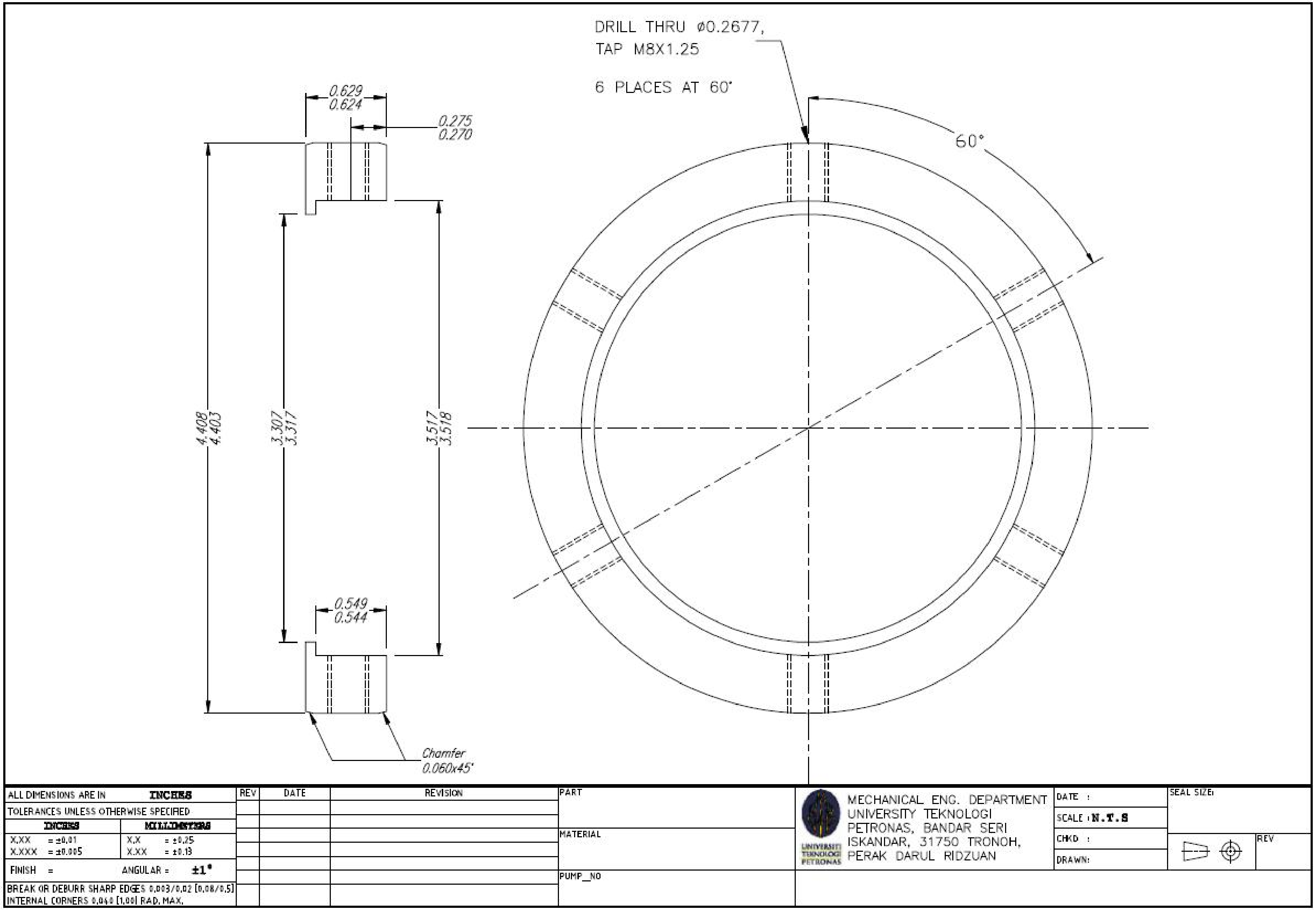


ALL DIMENSIONS ARE IN INCHES	REV / DATE / REVISION	PART: GLAND	MECHANICAL ENG. DEPARTMENT UNIVERSITY TEKNOLOGI PETRONAS, BANDAR SERI ISKANDAR, 31750 TRONOH, PERAK DARUL RIDZUAN	DATE : 20.10.2007
TOLERANCES UNLESS OTHERWISE SPECIFIED				SCALE : N.T.S
INCHES: X.XX = ±0.01; X.XXX = ±0.005		MATERIAL: MILD STEEL		CHKD : FAKHROLDIN
MILLIMETERS: X.X = ±0.25; X.XX = ±0.13				DRAWN: EMIER DEWA
FINISH = ANGULAR = ±1°		PUMP_NO: P-1207		SEAL SIZE: 3750
BREAK OR DEBURR SHARP EDGES 0.003/0.02 [0.08/0.5] INTERNAL CORNERS 0.040 [1.00] RAD. MAX.				REV 0

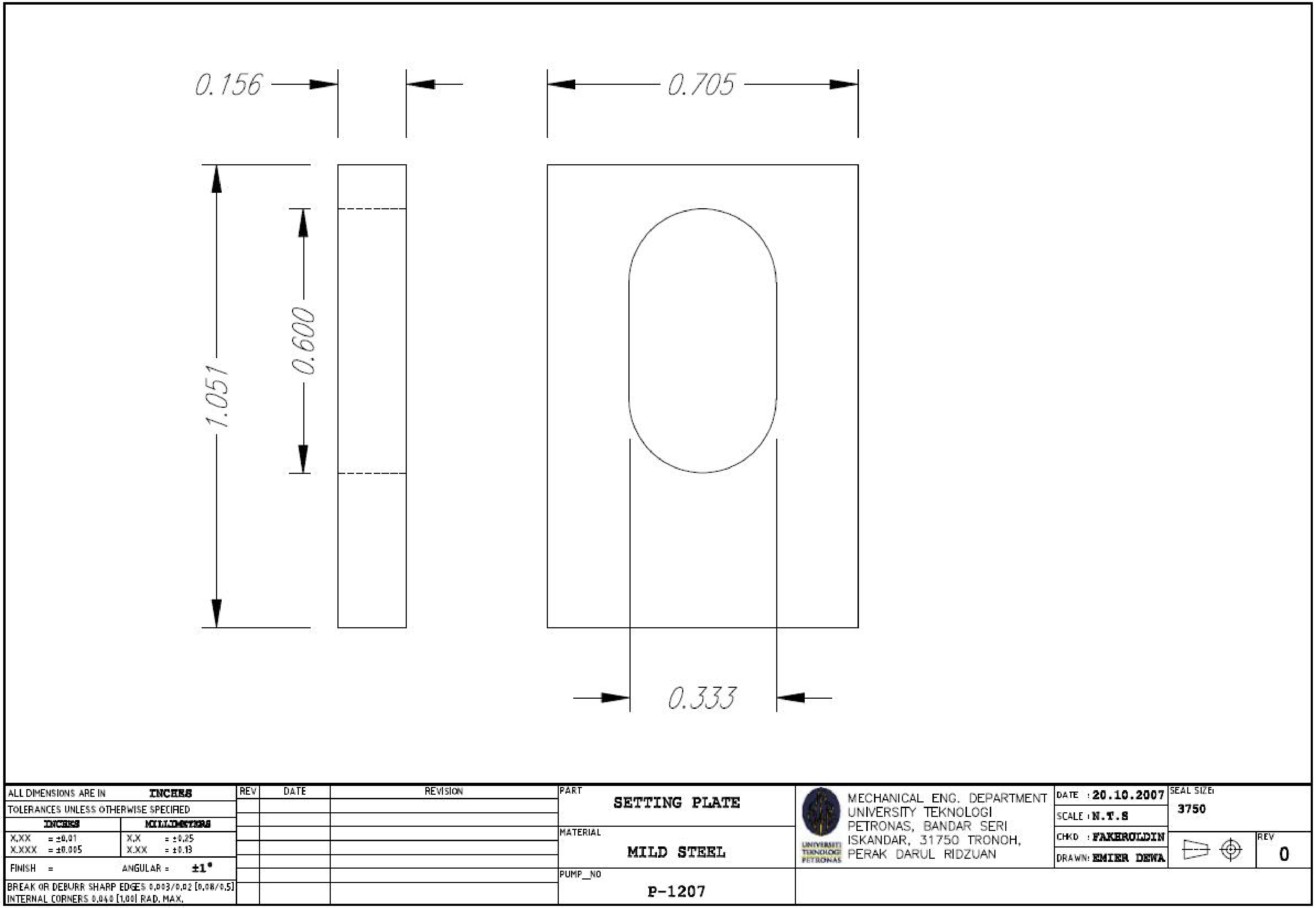
Appendix 8: Gland's drawing



Appendix 9: Throttle bush's drawing



Appendix 10: Sleeve collar’s drawing



Appendix 11: Setting plate's drawing

ProEight

ProEight Offshore Engineering Sdn Bhd (Company Number : 459320-P)
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Branch B 20, Jalan Gebeng 2/6, Kawasan Perindustrian Gebeng, 26080 Kuantan
Tel: (609) 583 5824 Fax : (609) 583 5830 Email: sales@proeight.com.my

SEAL PERFORMANCE TEST CERTIFICATE

CUSTOMER : **FINAL YEAR PROJECT (EMIER HELMEE MD DEWA)**

PUMP TAG NO : **VCM KERTEH P-1207**
DRAWING :
SEAL TYPE / SIZE : **3750 (SINGLE SEAL)**

DATE TEST : **14/03/2008**
MEDIUM USED : **AIR**
PERIOD OF TIME : **30 MIN**
PRESSURE DROP : **2 BAR**

TIME (MIN)	PRESSURE (BAR)	DROP (BAR)
0	2.00	0
5	2.00	0
10	2.00	0
15	2.00	0
20	2.00	0
25	2.00	0
30	2.00	0

TEST BY (SERVICE)	QA / QC APPROVED	WITNESS BY
Signature	Signature	Signature
Name : ABDUL MUTALIB	Name : RAJA HAIRAYRI	Name : ZULKIFLI B. MOHD NOR
Designation : SERVICES	Designation : TECHNICIAN	Designation : ProEight FACTORY MANAGER
Date : 15/3/08	Date : 15/3/08	Date :

Appendix 12: Certificate of Pressure Test Awarded by ProEight Offshore Engineering