Effective Mechanical Seal for Pump Applications at Onshore Gas Terminal

by

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Dissertation submitted in partial fulfilment of
the requirements for the
Bachelor of Engineering (Hons)
(Mechanical Engineering)

JANUARY 2009

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CERTIFICATION OF APPROVAL

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A project dissertation submitted to the
Mechanical Engineering Programme
Universiti Teknologi PETRONAS
in partial fulfilment of the requirement for the
BACHELOR OF ENGINEERING (Hons)
(MECHANICAL ENGINEERING)

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January 2009

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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AHMAD MUHAIMIN BIN ZAINAL

ABSTRACT

One of the purposes of Onshore Gas Terminal (OGT) was to export gas and condensate to the respective customers. Condensate Transfer Pumps (CTP) was used to transport condensate to the condensate metering system before being exported.

The problem that usually occurred in this operation was CTP pumps failure. The CTP pump failure was due to leaking problem that was caused by the mechanical seal failure. The failures gave bad effects on the environment issues related with the product leakage and economic issues due to product waste and disruption to condensate export operations.

There were two objectives of this study. The first objective was to examine the properties of materials that could be used for mechanical seals. The second objective was to determine the most effective material to be used for mechanical seals that would reduce the occurrence of failure in CTP pumps.

The scope of study would focus on the properties of the mechanical seal to eliminate and prevent leak. Properties such as hardness, chemical compatibility, wear resistance and compressive strength to failure were determined. The materials were tested in the lab. Among the tests were Rockwell hardness testing, corrosion testing, wear resistance testing and compression to failure testing. The materials used were carbon, tungsten carbide and silicon carbide. The results were analyzed and deductions were made.

The study concluded that the most effective material that could be used for mechanical seal was Silicon Carbide because it had the highest strength, the highest resistance to corrosion, the highest wear resistance, and high compressive strength to failure.

ACKNOWLEDGEMENT

First and foremost, the author would like to praise God the Almighty for His guidance. Though difficulties occurred, His guidance gave the author the chance to still complete this project successfully.

The author also wishes to take the opportunity to express his utmost gratitude to the individual that have taken the time and effort to assist the author in completing the project. Without the cooperation of these individuals, no doubt the author would have faced some minor complications throughout the course.

The author's utmost gratitude goes to the author's supervisor, AP Dr. Razali Hamzah who advised and guided the author with moral support throughout the project. Without his guidance and patience, the author would not be succeeded to complete the project.

To all technicians in Mechanical Engineering Department, thank you for assisting the author in completing this project. Thank you very much.

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CHAPTER 1
INTRODUCTION

1.1 BACKGROUND OF STUDY

The Onshore Gas Terminal (OGT) was located in Kerteh on the East Coast of the Malaysian peninsular in the state of Terengganu. OGT provided the separation and metering of processed well fluids received from the Resak offshore complex via the 28" 130km trunkline. Gas and condensate from the Resak OGT, and the adjacent Joint Delivery System (JDS) OGT, were exported to PETRONAS Gas Processing Plants (GPP).

Prior to export from the OGT to the GPP, the two phases of gas and condensate were metered separately. Prior to metering, the condensate pressure was boosted by the Condensate Transfer Pumps (CTP). The high pressure condensate was then metered and exported via pipeline to GPP. However, at PCSB OGT, the common problem that occurred frequently was CTP pumps failure. Normally, the failure of a CTP pump was due to the leaking problem hence affecting the daily operations at OGT.

This failure was normally caused by the mechanical seal failure. A mechanical seal was a spring-loaded device that formed a seal between the rotating and stationary parts of different types of equipment, such as pumps and mixers. Typically, this equipment would consist of a rotating shaft and a stationary housing. The purpose of the mechanical seal was to prevent leakage of fluids and gases.

A mechanical seal was an improvement on gland type sealing pumps. It was highly recommended when chemical or fluid handled was costly, hazardous, and toxic and it was used to save marginal amount of power. The function of a mechanical seal was to seal the opening where the shaft entered the pump, preventing fluid leakage. By noting that mechanical seals could be found in hundreds of various applications, this indicated the importance of these devices.

1.2 PROBLEM STATEMENT

The CTP pumps failure was one of the most common problems occurred at the OGT. This failure was due to the failure of the mechanical seal of the pumps which would lead to unscheduled plant shutdown. At OGT, the pumps operated with continually changing gas and liquid flow rates. Thus the mechanical seal had to cope with the conditions such as density and viscosity variations, temperature variations, high and low operating pressures, unpredictable process medium composition, and erosion effects, mainly by sand.

The mechanical seal failures would lead to the leaking problems of the CTP pumps, hence affecting the pumps operations where the pumps could not deliver adequate pressure which was around 90 bar before exporting the condensate to the condensate metering system. These failures would also contribute to the higher maintenance cost of the pumps mostly for repair service.

There were many factors that caused the failure of the mechanical seals. At OGT, the most frequent reason for the CTP pumps failure was mechanical seal breakdown due to wear and tear. A failure from mechanical breakdown usually showed up as fretting on the shaft, which leaded to corrosion. Normal wear and tear occurred because one seal face was held stationary while the other face rotated with the shaft. Ordinarily, the two seal faces were of dissimilar materials to prevent adhesion. Fretting also could cause the seal to leak.

Another factor was thermal degradation. The seals may exhibit radial cracks located on the highest temperature surfaces. In addition, certain elastomers may exhibit signs of softening which produced a shiny surface as a result of excessive temperatures.

This study was pertinent to rectify the problem. It was conducted to determine the most effective material for the mechanical seals to ensure the CTP pumps operated with high reliability and availability with optimum performance to eliminate the leaking problem.

1.3 OBJECTIVES

a) To examine the properties of materials that could be used for mechanical seals
b) To determine the most effective material to be used for mechanical seals for pumps

1.4 SCOPE OF STUDY

This study was conducted to determine the most effective material for the mechanical seals that could be used for the optimum operation of the CTP pumps. In order to achieve this objective, a few tasks and research needed to be carried out by collecting all theoretical details regarding the existing material for the mechanical seal especially at its normal operating condition of the pumps. Different types of materials would be investigated to determine their mechanical properties such as resistance to pressure and temperature variations, and their durability by conducting necessary experiments. Conclusion and recommendations were to be made based on the findings of this study regarding the most effective material for the mechanical seal that could withstand the conditions of the pumps during the daily operations.

CHAPTER 2
LITERATURE REVIEW

2.1 GENERAL DESCRIPTION OF ONSHORE GAS TERMINAL (OGT)

The Onshore Gas Terminal had been designed to accommodate 700 MMscfd of gas and 23 000 Bpd of condensate at an operating pressure of 90 bar. To improve flexibility and gas availability, the processing facilities consisted of 2 x 100% trains.

Gas and condensate from the 28" trunkline were received at the Onshore Gas Terminal where separation, metering and evacuation to PETRONAS Gas Processing Plants (GPP) were carried out.

2.1.1 Functional Overview of OGT

The Onshore Gas Terminal (OGT) fulfilled the following functions:

a) Provide a facility for separation of the two phase fluid received from Resak Offshore Facilities.
b) Provide a facility to meter both Gas and Condensate.
c) Provide a facility to blend gas with existing JDS to balance CO_2 ratios.
d) Provide a base for export facilities to transfer gas and condensate to Gas Processing Plant, GPP.

2.1.2 Process Overview of OGT

Resak OGT Interactive Manual [12] stated that gas and condensate from Resak were received at the Onshore Gas Terminal via a 28” trunkline from the Resak Offshore Facility. The trunkline had also been designed to accommodate fluids produced by Bunga Kekwa platform.

The new facility was integrated with existing JDS OGT. The two phase hydrocarbon fluids from Resak and Bunga Kekwa would first pass through the receiving pig station prior to separation at the two slugcatcher banks.

A crossover line to enhance flexibility between the existing JDS and the Resak OGT, had been installed upstream of the slugcatchers. This link enabled continuous processing of production fluids from Sotong Collector platform at the Resak OGT in the event the existing JDS was off line.

The larger volume of condensate generated, especially during pigging activities, which could not be handled in finger slugcatchers and GPP would be temporarily stored in hairpin loop, before being passed to downstream condensate handling facilities.

Gas from the slugcatchers was further separated within the 2 x100% cyclonic separators. The metered gas from Resak was then blended with gas from the existing JDS, before exported to GPP, via the 42” export pipeline.

Prior to metering, condensate collected in the slugcatchers was further separated by 2 x 100% cyclonic separators. The resulting Resak condensate was then mixed with JDS condensate before being pumped to GPP at a pressure of 90 bar.

Condensate held in the hairpin loop would be periodically evacuated to the new Resak OGT condensate separators or to existing JDS separators, depending on available plant capacity. The new Resak condensate separators have a capacity of 21 000 to 32 000 barrels per day (bpd).

OGT gas and condensate separation system consisted of two finger slugcatcher banks, two gas separators, two condensate separators and three Condensate Transfer Pumps (CTP). After fluid separation, condensate and gas were then routed to individual metering skids. Location of CTP pumps was shown in Figure 2.1 [12].

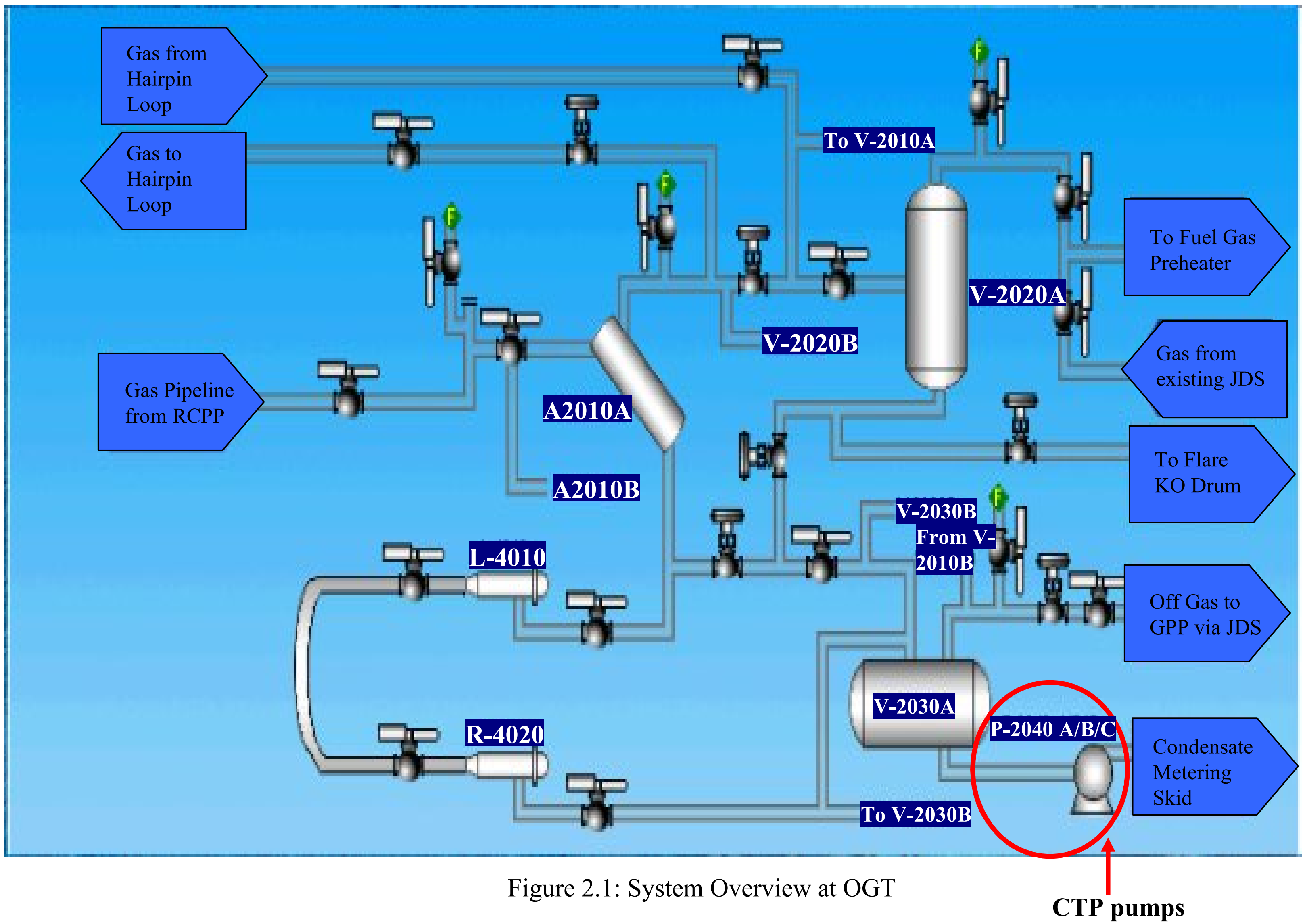


Figure 2.1: System Overview at OGT

2.2 CONDENSATE TRANSFER PUMPS (CTP)

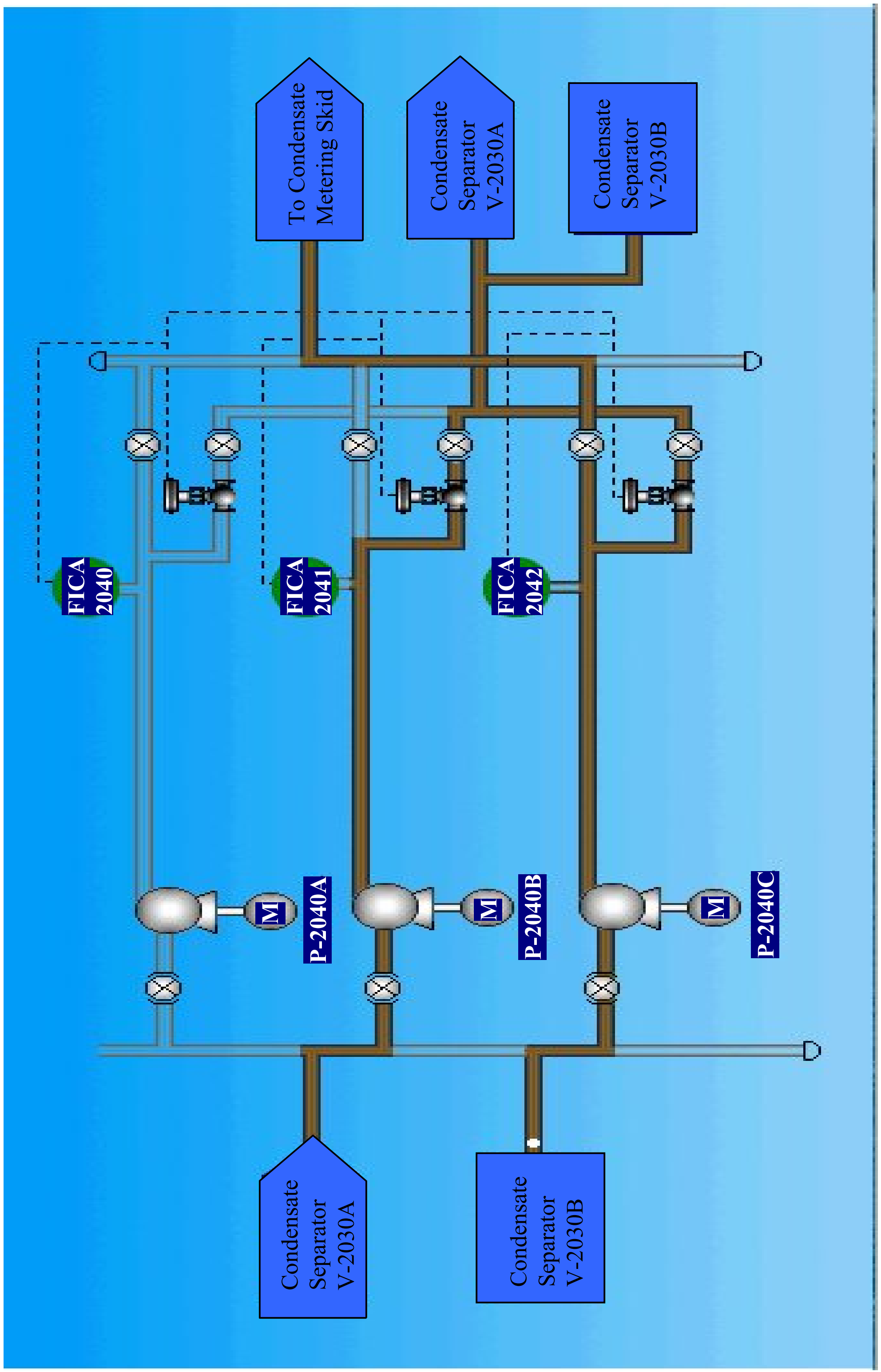


Figure 2.2: Condensate Transfer Pumps (P-2040A/B/C)

The condensate transfer pumps (P-2040 A/B/C) were horizontal multistage centrifugal type pumps with a flow capacity of 106 m^3/hr. Liquids from the condensate separator entered the pump inlet and were routed through the multistage centrifugal pump as shown in Figure 2.2 [12] above.

The multi-stage centrifugal pumps each consisted of three impeller stages. The liquid entered the eye of the impeller, and then into the wheel or impeller in the middle, was ejected at high velocity at the circumference, where it was channeled to the eye of the next impeller stage. The amount of head produced depended on the velocity imparted by the wheel and the amount of impellers. The Ingersoll-Dresser pumps were each driven by 415V, 50Hz AAB squirrel cage induction type electric motors. The condensate discharge from the pumps was transferred at a pressure of 90 bar to the condensate metering skid.

The pump was interlocked with two Shutdown Valves (SDV); SDV-2034 and SDV-2035 (V-2030 A/B liquid outlet SDV's) limit switches to ensure that any pump could not be started without at least one SDV opened. This was to prevent the pump from running dry.

The pump operating scenarios were as followed [12];

a) One pump running, one on automatic standby and the other one on manual. The standby pump would start on either of the following conditions;
    i) If the running pump failed
    ii) If higher than normal liquid levels were detected in the condensate separators (V-2030 A/B)
b) Two pumps running and one on automatic standby. In this case, the standby pump would automatically start if any running pump failed. The standby pump would not automatically start on high liquid levels as the piping was not sized for three pumps in operations.

If two pumps were running and it was observed that flow rates were sufficiently low (adequate if only one pump running), one pump could be stopped manually. For very low flow rates, the pumps may even be stopped altogether and flowed routed to bypass piping (to prevent from dry running).

The pumps were protected from running below its required minimum flow rates, by flow controllers (FICA-2040 / 2041 / 2042). If flow rates fell below minimum required, the flow controller would activate the minimum recycle control valves (FCV-2040 / 2041 / 2042 respectively), to recycle flow back to the condensate separators (V-2030 A/B) so as to maintain minimum flow rates.

From Figure 2.1 and 2.2 above, it showed that from the CTP Pumps (P-2040A / 2040B / 2040C), the condensate would be transferred to Condensate Metering Skid with the operating conditions as in appendix 10 'OGT-JDS Metering (Condensate Daily Report)'. The following Table 2.1 showed the CTP pumps operating conditions with the maximum operating limits at OGT.

Table 2.1: Operating conditions of the pumps

Parameter	Value
Maximum speed	2950 rpm
Maximum pressure	2500 psi / 172 bar
Maximum temperature	400 ^{o}F / 204 ^{o}C

At OGT, the most common problem that happened frequently was the pumps leakage. It was not desirable as it would effect the daily operations. The CTP pumps failure was normally due to the mechanical seal failure of the respective pumps. The function of the mechanical seal was to seal the opening where the shaft entered the pump, preventing fluid leakage.

2.3 GENERAL INFORMATION OF MECHANICAL SEAL

A mechanical seal was a device which helped join systems or mechanisms together by preventing leakage, containing pressure, or excluding contamination. Typical mechanical seal used in a pump was shown in Figure 2.3.

Figure 2.3: Mechanical Seal [2]

Depending on the manufacturing process and application, these materials could range from hazardous chemicals to process water to food. Mechanical seals could be found in hundreds of various applications. Any prominent industry or manufacturing operation that had a need to pump material from one place to another probably had some use for a mechanical seal, including the oil and gas business. The materials used in the design of a mechanical seal were determined by the conditions under which it would be operating.

The body of a mechanical seal was typically made from stainless steel. The wearing, or contact face of the seal, could be made from a variety of corrosion-resistant materials, such as carbon, glass-filled Teflon, tungsten carbide, and silicon carbide. The other face, the hard face, could be made from ceramic, niresist, tungsten carbide, or silicon carbide. One face rotated, while the other remained stationary. Typically, the nature of the process, the pressure and velocity of the pump, and the temperature the seal was operating in would determine the seal material.

2.4 BASIC MECHANICAL SEAL

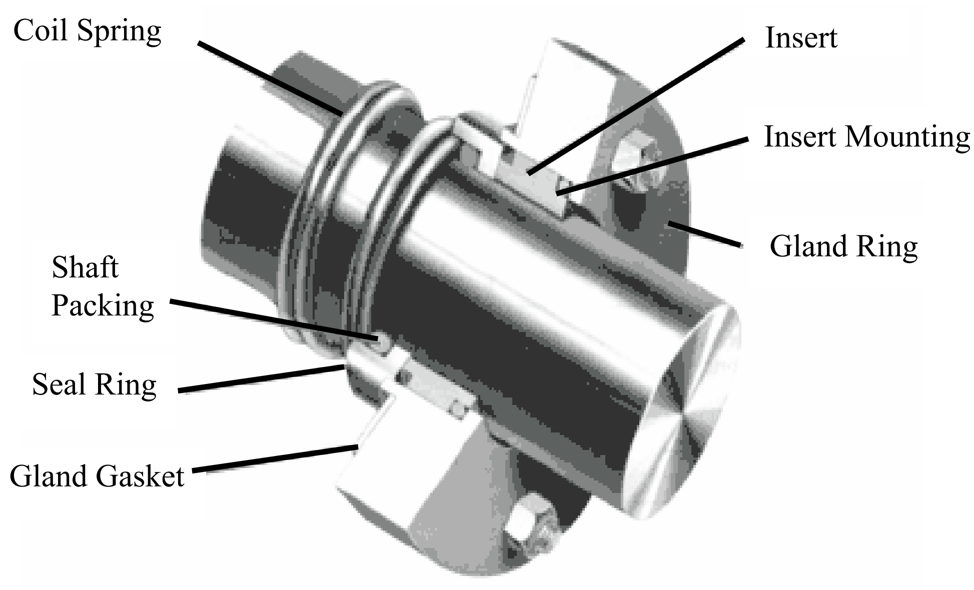


Figure 2.4: A Simple Mechanical Seal [5]

All mechanical seals were constructed of three basic sets of parts as shown in Figure 2.4:

a) A set of primary seal faces: one rotary and one stationary shown in Figure 2.4 as seal ring and insert.
b) A set of secondary seals known as shaft packing and insert mountings such as O-rings, wedges and V-rings.
c) Mechanical seal hardware including gland rings, collars, compression rings, pins, springs and bellows.

2.5 HOW A MECHANICAL SEAL WORKS

The primary seal was achieved by two very flat, lapped faces which created a difficult leakage path perpendicular to the shaft. Rubbing contact between these two flat mating surfaces minimized leakage. As in all seals, one face was held stationary in housing and the other face was fixed to, and rotated with, the shaft. One of the faces was usually a non-galling material such as carbon-graphite. The other was usually a relatively hard material like silicon-carbide. Dissimilar materials were usually used for the stationary insert and the rotating seal ring face in order to prevent adhesion of the two faces. The softer face usually had the smaller mating surface and was commonly called the wear nose.

There were four main sealing points within an end face mechanical seal as shown in Figure 2.5. The primary seal was at the seal face, Point A. The leakage path at Point B was blocked by either an O-ring, a V-ring or a wedge. Leakage paths at Points C and D were blocked by gaskets or O-rings.

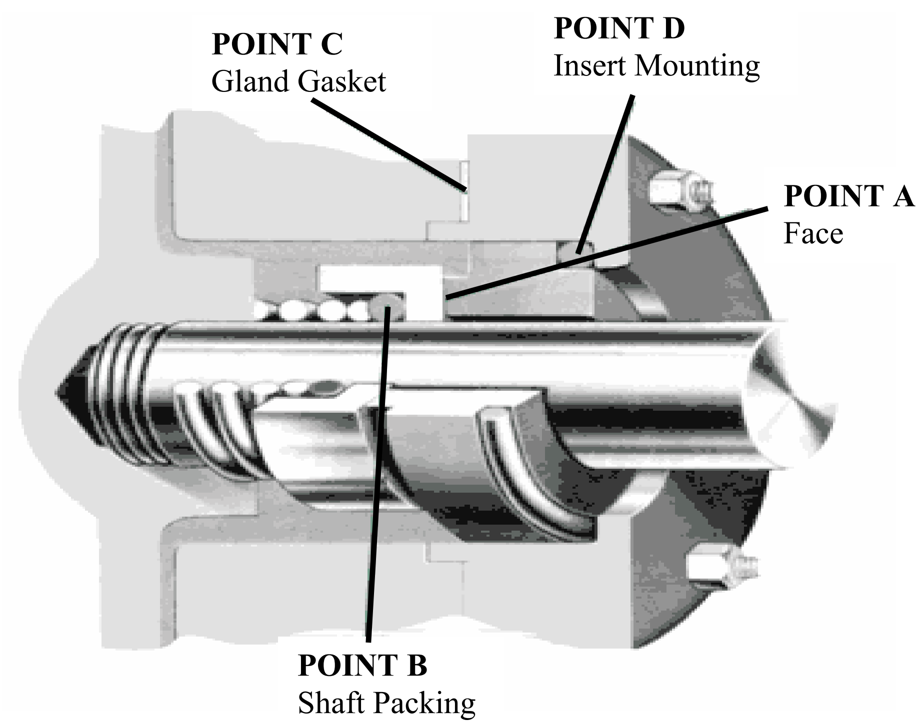


Figure 2.5: Sealing Points for Mechanical Seal [5]

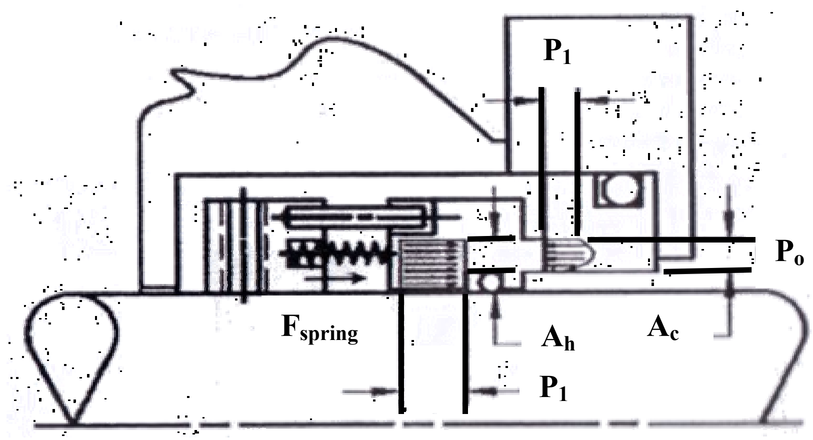
The faces in a typical mechanical seal were lubricated with a boundary layer of gas or liquid between the faces. In designing seals for the desired leakage, seal life, and energy consumption, the designer must consider how the faces were to be lubricated and selected from a number of modes of seal face lubrication.

2.6 EFFECTIVE FORCES IN A MECHANICAL SEAL

The rotating seal face was pressed against the stationary mating ring by the mechanical forces of the spring element and by the hydraulic forces generated by the pressure differential to be sealed. Together these forces were known as the closing force as shown in Figure 2.6. As the shaft rotated, the fluid under pressure penetrated the sealing gap between the sliding surfaces, reducing the sealed pressure to ambient conditions at the inner diameter of the sealing faces. The microscopic pits and asperities in the seal faces acted as lubricant reservoirs and allowed fluid to pass from one pit to the next as it crossed the face.

At the same time, the pressure in the lubricating film built up a gap opening force that counteracted the closing force. Figure 2.7 showed three simplified and idealized pressure profiles in the sealing gap. When the shaft was static, the pressure in the seal gap was nearly zero and thus the seal face load was directly supported by the stationary mating ring contact between the seal face materials. The pressure differential decreased in a linear manner as the shaft rotated. Normal wear of the faces was also taking place. In this operating state, the faces were allowed to come in contact, though fluid entered the gap at a controlled rate.

Hydrodynamic effects during rotation may cause the pressure to rise in the gap. The hydrodynamic forces were the consequence of surface imperfections and strongly depended on the sliding speed and viscosity of the sealed fluid. When the opening forces equaled the closing forces, the sealing surfaces ran on a film of liquid normally without contact and thus seal face wear would be negligible. Once the pressure differential reached approximately 2070 kPa (300 psi), the hydraulic closing force was so strong that an adequate lubricating film could not be formed between the sliding surfaces as the shaft rotated. The result was a high contact force with accelerated wear of the faces. This problem was solved by allowing the sealed pressure to act on a smaller area relative to the sliding surface area or, in sealing terminology, balancing the faces.



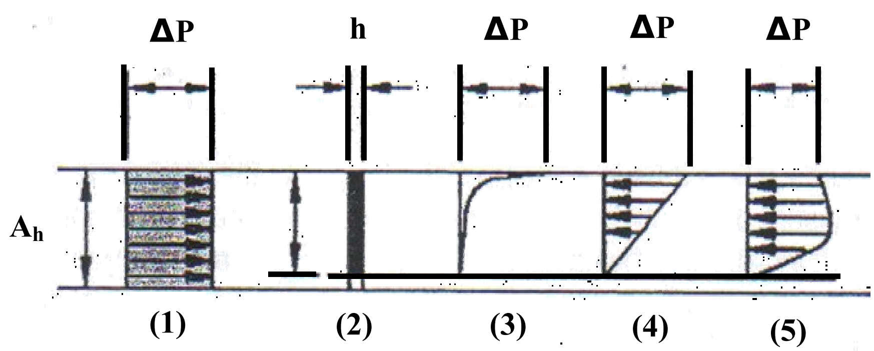
A_h = hydraulically loaded area

A_c = contact area

P_1 = sealed pressure

P_0 = pressure at the atmospheric side of the seal face

Figure 2.6: Effective forces in a mechanical seal



(1) Hydraulic closing face

(2) Lubricating film thickness h

(3) Press. curve in static condition

(4) Linear pressure drop

(5) Pressure curve with hydrodynamic component

Figure 2.7: Pressure profiles in the sealing gap

2.7 MECHANICAL SEAL SELECTION

Ron Mumbray, Rick Hull, Kevin Midgley, and Raphael Ortega (December 2003) [11] said that the proper selection of a mechanical seal could be made only if the full operating conditions were known. Details as the followings:

a) *Liquid:* Identification of the exact liquid to be handled was the first step in seal selection. The metal parts must be corrosion resistant, usually steel, bronze, stainless steel, or Hastelloy. The mating faces must also resist corrosion and wear. Carbon, ceramic, silicon carbide or tungsten carbide may be considered. Stationary sealing members of Buna, EPR, Viton and Teflon were common.
b) *Pressure:* The proper type of seal, balanced or unbalanced, was based on the pressure on the seal and on the seal size.
c) *Temperature:* In part, determined the use of the sealing members. Materials must be selected to handle liquid temperature.
d) *Characteristics of Liquid:* Abrasive liquids created excessive wear and short seal life. Double seals or clear liquid flushing from an external source allowed the use of mechanical seals on these difficult liquids. On light hydrocarbons balanced seals were often used for longer seal life even though pressures were low.
e) *Reliability and Emission Concerns:* The seal type and arrangement selected must meet the desired reliability and emission standards for the pump application. Double seals and double gas barrier seals were becoming the seals of choice.

2.8 DESIRABLE PROPERTIES FOR SEAL MATERIALS

According to Ron Mumbray, Rick Hull, Kevin Midgley, and Raphael Ortega (December 2003) [11], the seal materials of a mechanical seal provided a demanding set of requirements to ensure it could perform over an extended time period with optimum performance. The materials should have the following properties:

a) *Chemical Compatibility* – Must withstand all chemicals and combinations of chemicals without degradation, swelling, or loss of physical properties. At OGT, the potential hazard was the presence of Iron Sulphide (FeS). FeS, also known as Ferrous Sulphide, was a black crystalline compound of iron and sulphur which was often produced during operations from the reaction of hydrogen sulphide and the iron from the steel in the piping and vessels. Extreme caution must be exercised when handling this material because it ignited spontaneously when exposed to the air. This characteristic could be dangerous when any piping or vessels, including launchers and receivers were open, permitting the entry of air. Before repressuring such lines or vessels, all air must be thoroughly purged to avoid the possibility of an explosion. If FeS was discovered in any vessel or piping it must be kept saturated with water until it could be disposed of safely.
b) *Thermal Expansion* – Must be capable of compensating for different thermal expansion between components
c) *Pressure Containment* - Must be capable of sealing under the highest pressure for the application
d) *Specific Strength* - A material strength divided by its density. It was expressed in Newton meters per kilogram, and was used for tensile strength as for compressive strength. It was sometimes known as the strength-to-weight ratio.
e) *Hardness* - A characteristic of a solid material expressing its resistance to permanent deformation.
f) *High Wear Resistance* - When a mechanical seal was used in a pump, the liquid in the pump was used to cool and lubricate the seal faces. The seal faces were in sliding contact and prevented the liquid in the pump from reaching atmosphere. This contact also generated frictional heat that must be removed from the seal

faces. Failure to remove this unwanted heat often resulted in the boiling of the lubricating liquid film at the seal faces, usually leaving a deleterious residue and causing premature failure of elastomeric components in the seals (static-secondary seals). Both of these undesirable effects of elevated temperatures led to premature seal failure.

2.9 COMMON SEAL MATERIALS

According to Micheal Huebner (2003) [1], for every seal application there was an optimal pair of rubbing materials that provided the longest life and the lowest operating cost. This rubbing pair must be selected from scores of seal nose materials and an equally large number of mating face materials. The right choice of materials for the mating pair could easily increase seal life fivefold in those applications where wear was the controlling factor. This part would review some of the most common seal materials.

2.9.1 Carbon

Carbon was one of the most abundant elements on earth. It was the basis for all organic products and processes. It was also an interesting material because it took on forms from amorphous carbon to graphite to diamonds to fullerenes. Carbon was inert, stable, and could be self-lubricating.

Mechanical carbons used in seal faces were a mixture of amorphous carbon and graphite. The percentages of each helped determine the physical properties on the final grade of carbon. In addition to carbon, other elements and compounds were present that affected the properties of the grade of carbon. Some of these were the impurities from the original sources of carbon; others were specifically added to improve some aspect of the carbon's performance.

Impregnants of the carbon would increase the strength of the final material. Thus, the selection of impregnants was a critical factor in determining the properties of the final material. Impregnants included various plastic and resins, metals, and salts. The most common impregnants for mechanical seal faces were thermoset resins and antimony metal.

a) Resin Impregnated Carbon

As the name implied, resin impregnated carbon was a mixture of amorphous carbon / graphite that had been impregnated with a thermoset resin. This was by far the most common type of carbon for mechanical seals used in industry today. Most of resin impregnated carbons were capable of operating in a wide range of chemicals from strong bases to strong acids. It possessed good frictional properties and an adequate modulus to help control pressure distortions.

b) Antimony Impregnated Carbon

Metallized carbons were available with a variety of metal impregnants including copper, bronze, lead and antimony. Of these, antimony had proven to be the most successful in seal applications. The addition of antimony had a couple of beneficial effects that could improve seal performance. First, the addition of a metal impregnant increased the strength and modulus of the material. This was beneficial for high pressure applications when a stronger and stiffer material was needed. Finally, antimony impregnated carbons were more resistant to blistering in high viscosity fluids or light hydrocarbons. However, the chemical compatibility of an antimony carbon was limited by the antimony metal.

2.9.2 Tungsten Carbide

Tungsten carbide (WC) was carbide ceramic that was used in many products requiring high hardness and toughness. WC was readily available in powder form but must be processed into final shape. Tungsten carbides were most often manufactured as cemented carbides. As cemented carbide there was no attempt to bond WC to itself. Rather a secondary metal was added to bind or cement the WC particles together. This resulted in a material that had the combined properties of both the WC and the metal binder. This had been used to an advantage by providing greater toughness and impact strength than possible with WC alone. One of the primary weaknesses of cemented WC was its high density.

Cobalt bound WC proved to be one of the most successful carbides. As the mechanical seal developed over the year, Co-WC was the standard ceramic material. Cobalt though did not exhibit the range of chemical compatibilities required by the industry and was gradually replaced by nicked bound WC. Ni-WC was widely used as a seal face material especially where its high strength and high toughness properties were very beneficial. It had good chemical compatibility generally limited by the free nickel.

2.9.3 Silicon Carbide

In the simplest sense, silicon carbide (SiC) consisted of one atom of silicon bonded to one atom of carbon. This resulted in a tenacious bond that was extremely stable over a wide range of temperatures and chemical environments. It was also had other desirable properties such as a high hardness and a high modulus. Unfortunately, it was also a material that was difficult to manufacture in shapes suitable for component design. For many years, a reaction bonding process had been used to manufacture components. More recently, sintering processes have been used. Other methods such as chemical vapor deposition (CVD) or conversion processes were used in areas outside of mechanical seals.

The vast majority of SiC material was solid material free from any significant pores, voids, or imperfections. This resulted in seal faces that were lapped very flat and smooth. Under certain conditions, faces that were too flat or too smooth could prevent fluid from migrating across the seal faces, which could prevent proper lubrication of the seal faces.

The most common varieties of these materials involved free graphite particles dispersed throughout the SiC material. On a lapped surface, this resulted in small patches of graphite surrounded by a hard SiC surface. The graphite did not provide additional lubrication by providing graphite into the sliding interface. Rather the soft graphite material created small depressions in the surface that enhanced lubrication with the process fluids. As an alternate approach, SiC was available with small controlled voids throughout the material, which also created small depressions on the lapped surfaces.

2.10 COMPARISON OF SEAL MATERIALS

By reviewing the material available for seal materials, it was clear that each had its own set of strengths and weaknesses as indicated by Table A.3, Table A.4, and Table A.5 in the appendixes. It was possible to compare physical data that could help explain the differences between the materials. It was also possible to list the general strengths and weaknesses of each material type that could give guidance in its use.

Table A.3 showed comparison of the properties of the seal face materials. From Table A.3, silicon carbide possessed the highest hardness value compared to the tungsten carbide and carbon. But for the modulus and strength properties, tungsten carbide was the material that had the highest value in both properties followed by silicon carbide and carbon.

Table A.4 showed strength and weaknesses of the seal face materials. From Table A.4, it was clearly indicated that silicon carbide had excellent performance in the abrasive services, blister resistant, chemical resistance, and thermal shock resistance. For tungsten carbide, it was excellent in the abrasive services, and impact resistance. For carbon, it possessed excellent ratings in thermal shock resistance property.

Table A.5 showed chemical compatibility of the seal face materials. From Table A.5, silicon carbide and tungsten carbide have excellent performance in the light hydrocarbons condition. Silicon carbide also possessed excellent ratings in overall chemical resistance especially with respect to the acidic environment.

By comparing the data and information from the respective tables in the appendixes, the material with the most excellent performance was silicon carbide, followed by tungsten carbide and carbon. Silicon carbide possessed the desired properties that could ensure the effectiveness of the mechanical seal such as high hardness, high thermal shock resistance and high chemical resistance. However, tungsten carbide also showed excellent performance in the impact resistance and abrasive services. Carbon had the excellent performance in the thermal shock resistance.

From the tabulated information, each material had its own advantages and disadvantages. Hence, it was very crucial to determine the most effective material for the mechanical seals for the pumps applications while the seal materials provided a demanding set of requirements to be fulfilled. Generally, the seal materials must be stable, conduct heat, be chemically resistant and provide good wear characteristics. Nowadays, more exotic materials have been developed including ceramics and various grades of mechanical carbons. Therefore, proper selection of the material from the most common seals which were carbon, tungsten carbide and silicon carbide would help eliminating the problems in the future.

2.11 COMMON CAUSES OF CTP PUMPS FAILURE AT OGT

The failure of mechanicals seals used in difficult sealing applications could also be perilous as they provided a safeguard against hazardous materials escaping into the environment. The biggest problem faced during the use of mechanical seals, was the constant threat to the seal face due to several factors that caused wear and tear, making the seals unfitted for use.

According to A Tyldesley (2000) [3]

> Operating experience across the gas industries suggested that early failures, from poor selection, installation or operation dominated the overall failure data, and that very few mechanical seals actually wore out. Some American data analyzed failure modes as followed:
>
> i) 40 % operating problems (for example operating outside the design intent)
> ii) 24% assembly errors, misalignment
> iii) 19% faulty design of the lubrication circuit, this led to overheating and degradation of the seal faces
> iv) 9% poor selection of seal components, for example failure of polymeric static seals by chemical attack
> v) 8% miscellaneous

2.11.1 Wear and Tear of Mechanical Seal

At OGT, failure of the mechanical seal was frequently caused by wear and tear of the seal thus making it as a common source of premature CTP pumps failures and subsequent repairs.

Thus, these high temperature and pressure would give significance effect on the mechanical seals which based on sliding contact between the faces of two rigid rings. The seal face was being distorted by either temperature or pressure, where the lapped hard faces were especially sensitive to either change in temperature or pressure excursions.

This condition would lead to the mechanical breakdown problem. The face would get chipped or pitted, leading to the leakage of the fluids and gases indicating the seal was not performing its intended purpose. The failure from mechanical breakdown usually showed up as fretting on the shaft, which led to corrosion. Fretting was one of the most common types of mechanical attack. It caused leakage past secondary seals and left the shaft or sleeve beneath the secondary seal corroded and damaged. This area would appear pitted or bright and shiny compared with the overall finish of the shaft or sleeve as showed in Figure 2.8.

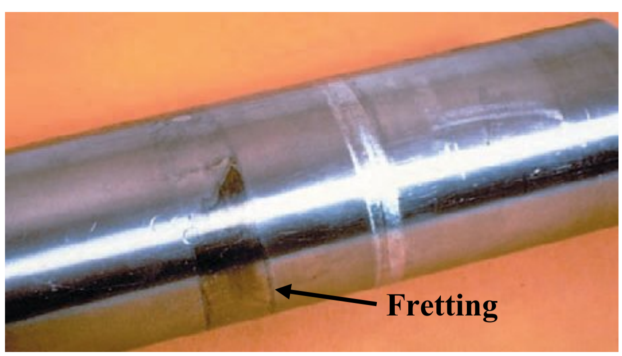


Figure 2.8: Results of Fretting [3]

Fretting resulted from reciprocating movement of secondary seals over a shaft sleeve, with consequent removal of the passive coatings intended to protect the sleeve from corrosion. Constant vibration of the shaft packing over this surface also removed the passive coating and allowed further corrosion to occur.

The wear and tear of the mechanical seal also led to the seal face distortion. Symptoms of face distortion included excessive leakage at the seal. Visual examination of the seal faces showed a non-uniform wear pattern, which was sometimes difficult to detect. By lightly polishing the seal faces on a lapping plate, high spots would appear at two or more points. By referring to the Figure 2.9, this indicated uneven wear.

Seal face deflection was typified by uneven wear at the seal face. The concave or convex wear pattern was continuous for 360° around the seal faces. A concaved seal face may result in excessive seal-face torque and heat, while one that was convexed resulted in abnormally high leak rates. Seals with either condition generally would not be stable under cyclic pressure conditions.

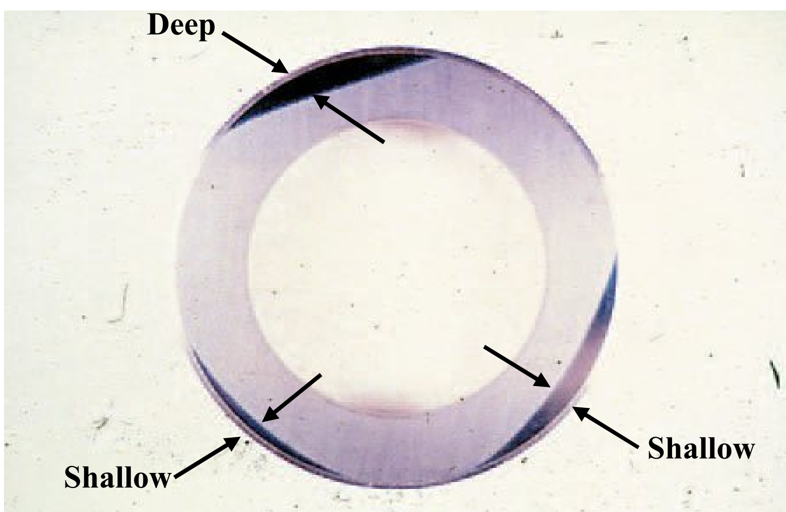


Figure 2.9: Uneven Wear Patterns [11]

As a major contributor of CTP pumps failure at OGT, wear and tear of the mechanical seal also would lead to the misalignment of the seal thus provided a space for leakage to happen. Rapid seal wear would result in premature seal – pump failure, making the seal unfitted for use. Therefore, using the right material for the mechanical seal was the key factor to prevent the problem.

2.11.2 Thermal Degradation

Another factor that led to the mechanical failure at OGT was the thermal degradation. Eagle Burgmann and John Crane (2004) [2] stated that one symptom of thermal attack was the appearance of fine-to-large radial cracks that emanated from the center of the metal or ceramic ring. Characterized as heat checking, these cracks acted as cutting edges on the carbon-graphite and other seal face materials, and the consequent scraping action rapidly wore out the seal as shown in Figure 2.10.

Common causes of heat checking were vaporization at the seal faces and excessive pressures and velocities. Combination of these factors could result in higher friction and heat at the seal faces. The excessive thermal stresses that develop would result in fine fractures.

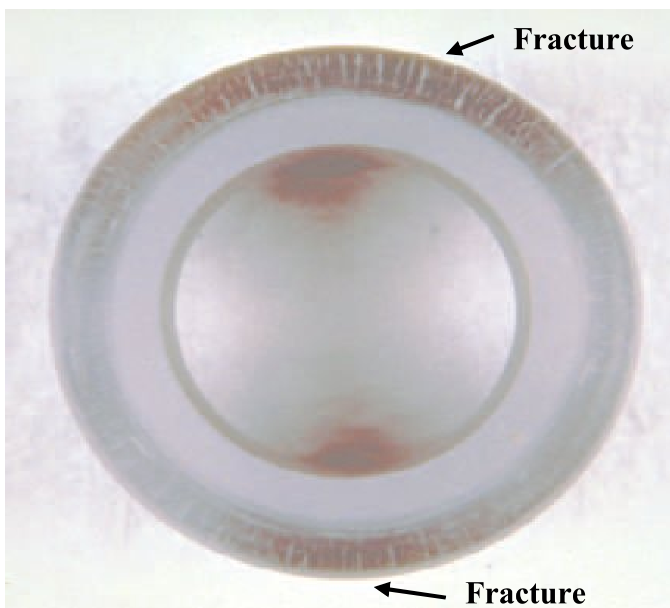


Figure 2.10: Ceramic Ring with Heat Checking Cracks [2]

Popping, puffing, and blowing of vapors at the seal face, known as vaporization, was another symptom of thermal attack that resulted in excessive leakage and damage [2]. If vaporization did not cause failure, it usually shortened seal life and impaired seal performance. Inspection of the seal faces usually showed chipping at the inside and outside diameters, and pitting over the entire area.

Vaporization occurred when heat at the seal faces could not be adequately removed, and the liquid between them rapidly evaporated or flashed. It also could be caused by operating the seal too near the flash temperature and flash pressure of the product in the seal cavity. Other operating conditions that caused vaporization were excessive pressure for the seal and excessive seal face deflection.

Spalling was another symptom of thermal attack characterized by small circular sections that occurred on surfaces away from the seal face, such as the outside diameter and the back of the seal [2]. With regards to Figure 2.11, spalling was caused by excessive thermal stresses in a carbon-graphite seal. It seemed to occur with virtually any fluid, and was the result of moisture suddenly being driven off when the seal was overheated.

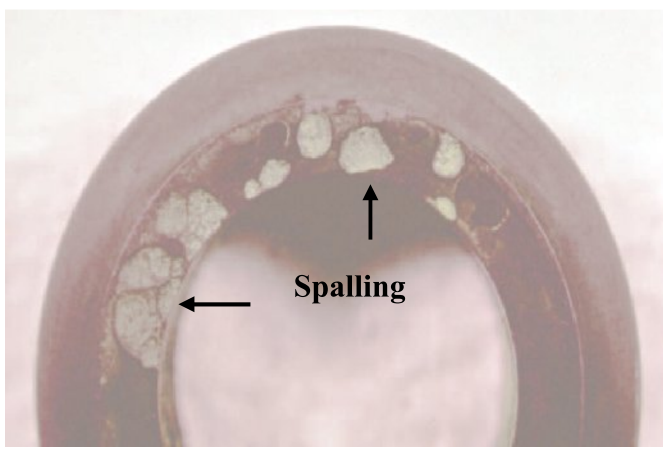


Figure 2.11: Carbon – Graphite Seal Exhibiting Spalling [2]

CHAPTER 3
METHODOLOGY

Figure 3.1 showed the flow chart of the procedures that had been implemented to complete this study. Based on the topic which was the effective mechanical seal for Condensate Transfer Pumps (CTP) at Onshore Gas Terminal (OGT), literature review on the topic was done first. The literature review was focusing on the background of the study such as the function of the CTP pumps and the mechanical seal itself. Important information about the mechanical seal such as the factors needed to be considered before selecting a mechanical seal and the types of materials commonly used for the sealing purposed were gathered and analyzed throughout the study. The materials used were carbon, tungsten carbide and silicon carbide. After relevant hypotheses, theories and facts were identified, lab tests and experiments were done. Several tests such as Rockwell hardness test, wear test and corrosion test were implemented in this study. Results from the tests were analyzed and deductions were made. Discussion on the properties of each material such as the wear resistance, hardness, strength, and chemical compatibility were done to clarify the achieved data and findings. From the discussion, conclusion was made by determining the most effective material for the mechanical seals of CTP pumps at OGT to ensure high reliability of the pumps in the daily operations. Furthermore, some recommendations were suggested for the effectiveness of the mechanical seals. All data, results and findings were recorded and documented in the final report for reference in the future. Upon submission of the final report, this study would be completed and terminated. Gantt chart of the project was shown in Appendix 4 and Appendix 5. Summary of these activities and methods was shown in Figure 3.1 as followed.

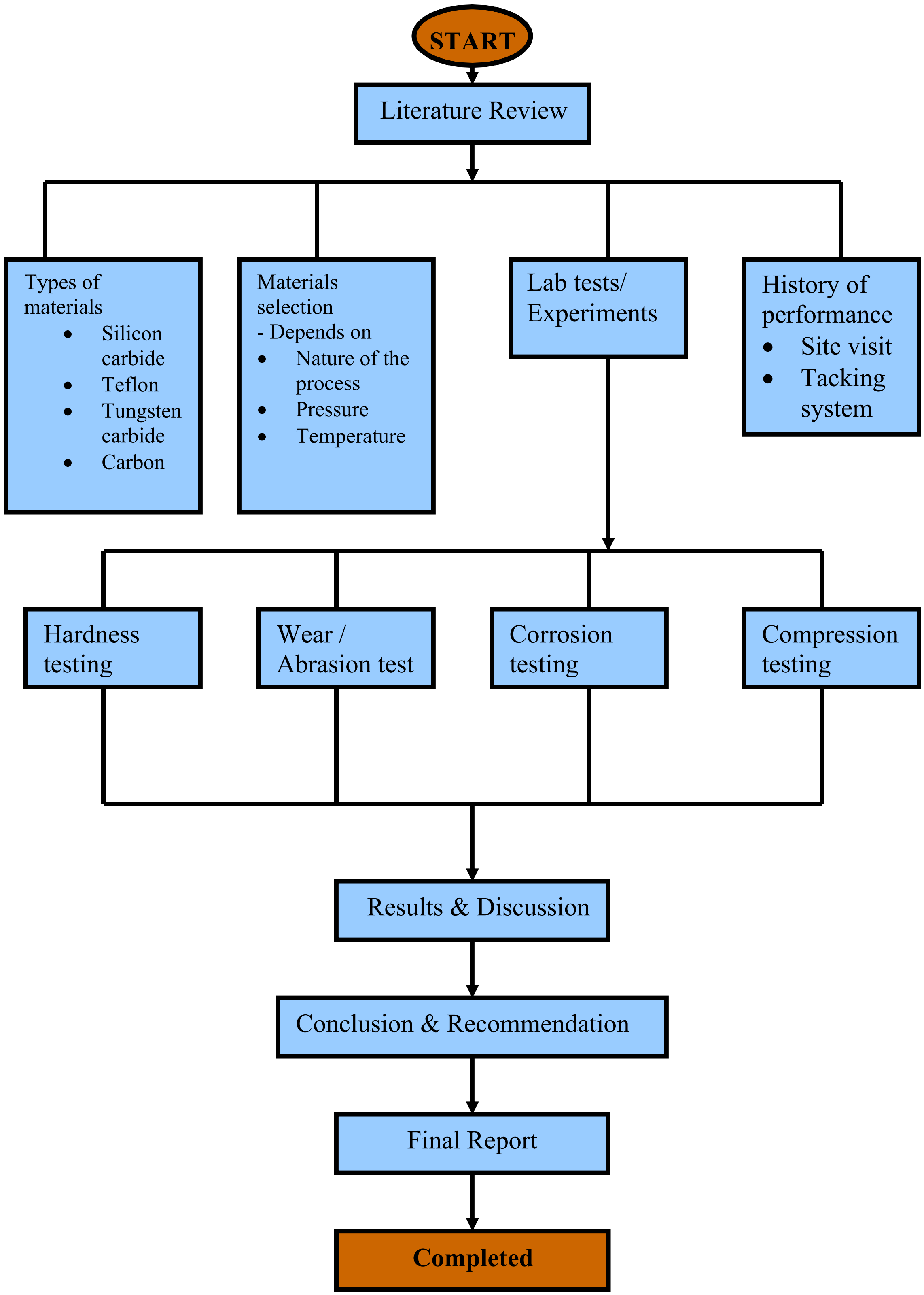


Figure 3.1: Flow Chart

3.1 HARDNESS TEST (ASTM E – 18)

Hardness was a measure of a material's resistance to localized plastic deformation. Although hardness testing did not give a direct measurement of any performance properties, hardness of a material correlated directly with its strength, wear resistance, and other properties [16]. Hardness tests were performed more frequently than any other mechanical test because of it is simple, inexpensive, and non-destructive. Other mechanical properties often may be estimated from hardness data, such as tensile strength.

a. Rockwell Hardness Test

The Rockwell Hardness test was a hardness measurement based on the net increase in depth of impression as a load was applied. Hardness numbers have no units and were commonly given in the R, L, M, E and K scales. The higher the number in each of the scales meant the harder the material [17].

b. The test procedures were as followed:

i) The indenter was set into the sample at a prescribed minor load (10 kg).

ii) A major load (140 kg) was applied and held for a set time period. The force on the indenter was then decreased back to the minor load.

iii) The depth reading was taken while the minor load was still on. The hardness number was read directly from the scale. Five (5) consistent readings were obtained from each sample, and then average values were calculated.

iv) The readings were taken from three different locations of the sample. This was done to check the hardness on the surfaces of the sample that might be different from each location due to the manufacturing and service history.

v) The Rockwell hardness number was calculated from the depth of permanent deformation of the indenter into the sample according to the specifications of ASTM E-18. The test was shown in Figure 3.2.

Figure 3.2: Rockwell Hardness Testing

3.2 CORROSION TEST (ASTM G 111)

Autoclave corrosion tests were a convenient means for laboratory simulation of many service environments for the purpose of evaluating the corrosion resistance of materials and for determining the effects of metallurgical, processing, and environmental variables on corrosion processes. The reason for such tests was to more closely recreate the high temperatures and pressures commonly occurring in commercial or industrial processes [18]. This test was according to ASTM G 111, 'Guide for Corrosion Tests in High Temperature and/or High Pressure Environments'.

This test method would determine the corrosion rate of the materials to chemicals which were hydrogen sulphide, H_2S and carbon dioxide, CO_2 gasses that existed in the condensate as shown in appendix 11 'OGT-JDS Gas Analyzer System (Daily Report)'.

a. The test procedures were:

i) The test was conducted using Autoclave machine as shown in Figure 3.3. Some parameters needed to be recorded before the test was conducted. The data that needed to be recorded and measured were the weight and the surface condition of the samples.

ii) To start the test, two liter of ionized water was loaded into the autoclave tank. 30 gram of sodium chloride, NaCl was then added into the solution. It was assumed that 1.5% of seawater was presented in the solution.

iii) Each material was immersed in the tank containing the solution. The lid of the tank was tightened to prevent leaking of H_2S and CO_2.

iv) The pressure of the total mixture was set up according to the operating condition as shown in appendix 10.

v) The equipment was left for seven days for the corrosion to take place. After this period, any changes on the specimens were recorded and evaluated.

Figure 3.3: Autoclave Corrosion Test

3.3 WEAR TESTING (ASTM C808)

Wear testing (Abrasion test) was done to evaluate the abrasion or wear resistance of the mechanical seal. This test method covered the determination of the resistance of seal faces to abrasion produced by the Taber Abraser on its surfaces. During the test, Taber Abraser would cause a rub-wear action (sliding rotation) on the surfaces of the test pieces.

a. The test procedures were as followed:

i) The test specimens were weighted to the nearest 0.1 mg and the weights were recorded.

ii) The test specimen was secured to the equipment platform, which was a motor driven at a fixed speed.

iii) Two abrasive wheels were lowered onto the specimen surface, and as the platform rotated, it turned the two wheels.

iv) The test was ended after certain specific time or specific number of cycles.

v) Any remaining loose abrading was removed from the test specimen by light brushing. The test specimen was reweighed.

vi) Abrasion resistance was calculated as loss in weight at a specified number of abrasion cycles.

3.4 COMPRESSION TESTING (ASTM E 9)

There were several forces operating in a mechanical seal. The forces were axial and radial forces, closing and opening forces, and hydrostatic and hydrodynamic forces as explained in part 2.6. These forces would react against each other, where the seal face would eventually being pressed and compressed. Compression test was a method of determining behavior of materials under crushing loads. Specimen was compressed and deformation at various loads was recorded.

In this experiment as shown in Figure 3.4, the material was subjected to an increasing axial compressive load. Both load and strain were monitored either continuously or in finite increments, and the compressive properties were determined. The data obtained from the compression test may include yield strength, yield point, Young's modulus, the stress-strain curve and the compressive properties. This testing was according to ASTM E 9 standard.

Figure 3.4: Compressive Test (Universal Testing Machine)

3.5 HEAT (TEMPERATURE) TEST

a. By using lapping machine as shown in Figure 3.5 below, heat are generated. This heat would determine which material was generating the highest temperature during the process which was undesirable heat.

b. The test procedures were:

i) The air pressure for lubricant was set to 10 psi.

ii) The air pressure for diamond slurry was set to 10 psi. The diamond slurry would add resistance to the sliding surface.

iii) The material was placed in the ring for rigidity during the process.

iv) A constant weight (2 kg) was placed on the ring.

v) Every 10 minutes, the temperature generated by sliding process between the material surface and the sliding contact was taken.

vi) The steps were repeated by using different materials.

vii) The data were analyzed to determine which material was generating the highest temperature.

Figure 3.5: Lapping Machine

CHAPTER 4
RESULTS AND DISCUSSION

EXPERIMENTAL DATA

4.1 Hardness Test (ASTM E – 18)

The hardness test was conducted to indicate the strength of materials and its resistance to scratching and to wear. This test would indicate the strength of each material and indicated the effective material that could be used as mechanical seals.

The test was done in three different locations on each sample. This test would confirm whether each different location would have the same or different values of strength. Figure 4.1 showed the locations where the hardness test was performed on the mechanical seal which were indicated by P, Q and R respectively.

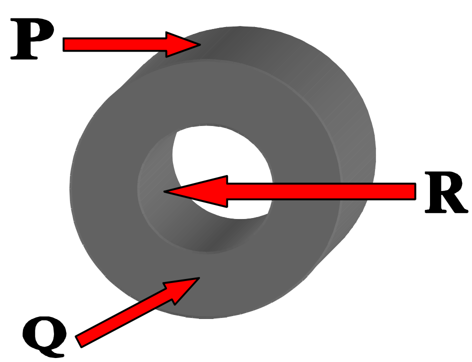


Figure 4.1: Locations on mechanical seal where the hardness test was performed

For carbon material, the Rockwell hardness test was done by using B scale (HRB). In this scale, load of 100 kg was applied onto its surface at the respective different locations as stated above. Table 4.1 below showed the hardness values for carbon with five consistent readings and the average reading for each location.

Table 4.1: Hardness Value for Carbon

No. of Readings	**Rockwell Hardness Value (HRB)**		
	P	**Q**	**R**
1	**117.2**	**116.4**	**117.7**
2	**117.1**	**120.3**	**118.8**
3	**120.2**	**119.9**	**118.5**
4	**118.3**	**120.3**	**118.2**
5	**119.2**	**118.1**	**118.1**
Average	**118.4**	**119.0**	**118.1**

For tungsten carbide and silicon carbide, the test was done by using C scale. For this scale, the applied load was 150 kg, which was 50 kg higher than the applied load in B scale for carbon. Table 4.2 and Table 4.3 below showed the hardness values for tungsten carbide and silicon carbide and the average reading for each sample, respectively.

Table 4.2: Hardness Value for Tungsten Carbide

No. of Readings	**Rockwell Hardness Value (HRC)**		
	P	**Q**	**R**
1	**86.2**	**76.0**	**80.7**
2	**87.8**	**78.6**	**82.3**
3	**77.9**	**87.8**	**84.6**
4	**78.7**	**85.1**	**79.7**
5	**83.4**	**88.6**	**86.2**
Average	**82.8**	**83.2**	**82.7**

Table 4.3: Hardness Value for Silicon Carbide

No. of Readings	Rockwell Hardness Value (HRC)		
	P	Q	R
1	90.2	90.5	91.2
2	92.7	91.5	90.8
3	92.3	93.2	92.3
4	91.6	92.5	91.3
5	92.2	92.4	91.4
Average	91.8	92.0	91.4

Rockwell hardness C scale was used for silicon carbide and tungsten carbide because the materials were considered as hard metals (hardened steels, hard irons, deep case-hardened steels, and titanium). Meanwhile, carbon was a malleable material therefore Rockwell hardness B scale was used for this type of material (copper alloys, soft steels, aluminum alloys, and malleable iron). The strength for C scale materials was higher than B scale materials hence the applied load for these two scales were different [16].

From these three tables (Table 4.1, Table 4.2, and Table 4.3), the standard deviations between three locations for each material were calculated. The result showed that the readings have low standard deviation which ranges from 0.25% to 0.42%. This indicated that the strengths of the materials were almost the same at any position.

Comparison between each material showed that silicon carbide had the highest strength and carbon had the lowest strength. From the tables above, silicon carbide had an average reading of 91.74 HRC. Tungsten carbide had average reading of 82.91 HRC while carbon had average reading of 118.5 HRB. Carbon had the lowest strength because it fell under B scale (mild steel). It was also because the other two materials were composites that have higher strength compared to carbon.

Based on this experiment, it could be concluded that silicon carbide was the effective material that could be used for mechanical seal since it had the highest strength compared to the other materials which were tungsten carbide and carbon.

4.2 Corrosion Test (ASTM G 111)

This test was the laboratory simulation of service environments for the purpose of evaluating corrosion resistance of the materials. It was according to the ASTM G111 'Guide for Corrosion Tests in High Pressure and/or High Temperature Environments'.

To run the test, the service environment of the real process was created where total pressure of 60 bar was used. It contained hydrogen sulphide, H_2S and carbon dioxide, CO_2 which were very corrosive elements. The partial pressure of the H_2S was 0.002 bar which meant 100 ppm of H_2S was presented in the solution.

Table 4.4 below showed the initial and final weight of each sample before and after the test was done. The changes in weight may indicate the corrosion process that might take place during the test.

Table 4.4: Measured weight for corrosion test

Type of Material	Sample 1		Sample 2	
	Before	After	Before	After
Silicon Carbide	58.959 g	58.959 g	194.591 g	194.591 g
Tungsten Carbide	75.609 g	75.601 g	523.023 g	523.021 g
Carbon	25.781 g	25.769 g	189.564 g	189.549 g

Table 4.5 showed the weight loss and its respective percentage for every material after the test was conducted. These values were the indicator for the corrosion rate of each material under the aggressive environments based on the weight loss corrosion coupons.

Table 4.5: Weight loss after corrosion test

Type of Material	Weight Loss (g)					
	Sample 1		Sample 2		Average	
	Weight	Weight %	Weight	Weight %	Weight	Weight %
Silicon Carbide	**0**	**0**	**0**	**0**	**0**	**0**
Tungsten Carbide	**0.008**	**0.0106**	**0.002**	**0.0038**	**0.005**	**0.0072**
Carbon	**0.012**	**0.0504**	**0.015**	**0.0079**	**0.0135**	**0.02915**

From Table 4.5, it showed that every material had a very low weight loss percentage. From the results, silicon carbide showed no weight loss, tungsten carbide had 0.0072 % of weight loss and carbon recorded 0.02915 % of weight loss. These values indicated that the corrosion rate of each material was very slow hence making the materials as the suitable candidates for the mechanical seal purposed. It could be concluded that all materials have high resistance to corrosion and chemically stable in the hazardous environment.

However, by comparing the weight loss values among the three materials, it showed that carbon had the highest weight loss followed by tungsten carbide and silicon carbide. This was due to the fact that carbon was more reactive towards the environment (H_2S and CO_2) compared to the other materials. The reaction caused the surface of the material to corrode hence losing its original weight. Since carbon was more reactive, it corroded faster compared to the other two materials. Silicon Carbide and Tungsten Carbide were more stable since they were composite materials.

Silicon Carbide did not have any weight loss since the composite made out of a non metal material which was silicate that did not corrode in such condition/environment.

Tungsten Carbide had moderate corrosion rate compared to carbon but not as good as silicon carbide. The weight percentage loss also indicated the same findings where carbon had the highest weight loss and silicon carbide did not have any weight loss percentage.

From the weight loss corrosion coupons, it could be concluded that Silicon Carbide had the lowest corrosion rate hence the most effective material that could be used as mechanical seal.

Other than calculating the weight loss, visual inspection also had been done in order to evaluate the surface condition on each sample before and after the test was conducted. Figure 4.2 and Figure 4.3 below showed the surface condition of the materials before and after the test was conducted, respectively.

(a) (b) (c)

Figure 4.2: Surface condition of the mech. seal before corrosion test was conducted: (a) Silicon Carbide; (b) Tungsten Carbide; (c) Carbon

(a) (b) (c)

Figure 4.3: Surface condition of the mech. seal after corrosion test was conducted: (a) Silicon Carbide; (b) Tungsten Carbide; (c) Carbon

From Figure 4.2, it showed that there was no defect or corrosion occurred on the surface of the mechanical seal before the test was conducted. This condition would be used as a comparison with the condition after the test was conducted. Figure 4.3 showed that there were some oxides on the surface of Tungsten Carbide and Carbon but not on Silicone Carbide. The amount of oxide found on Carbon mechanical seal was the highest compared to other materials. This showed that carbon had the highest corrosion rate among the three materials.

The visual inspection results were similar with the weight loss analysis which concluded that silicon carbide was the most effective material to be used as mechanical seal.

4.3 Wear Testing (ASTM C808)

The purpose of wear testing was to determine the resistances of the materials to the abrasion wear. It could help to determine the material that had the highest resistance to wear that was due to the frictional contact with another surface as in real operation.

Table 4.6, Table 4.7 and Table 4.8 below showed the results of the wear testing for 500 cycles, 1000 cycles and 1500 cycles, respectively.

Table 4.6: Taber Abrasion Test Results for 500 cycles

Test sample	Load applied (g)	Initial weight (g)	After 500 cycles		
			Weight (g)	Weight loss (g)	Wear index (g/rev)
C	10000	25.769	25.701	0.068	1.36×10^{-4}
TC	10000	75.601	75.579	0.022	4.40×10^{-5}
SiC	10000	58.959	58.941	0.018	3.60×10^{-5}

Table 4.7: Taber Abrasion Test Results for 1000 cycles

Test sample	Load applied (g)	Initial weight (g)	After 1000 cycles		
			Weight (g)	Weight loss (g)	Wear index (g/rev)
C	10000	25.769	25.653	0.116	1.16×10^{-4}
TC	10000	75.601	75.551	0.050	5.00×10^{-5}
SiC	10000	58.959	58.935	0.024	2.40×10^{-5}

Table 4.8: Taber Abrasion Test Results for 1500 cycles

Test sample	Load applied (g)	Initial weight (g)	After 1500 cycles		
			Weight (g)	Weight loss (g)	Wear index (g/rev)
C	10000	25.769	25.589	0.180	1.20×10^{-4}
TC	10000	75.601	75.538	0.063	4.20×10^{-5}
SiC	10000	58.959	58.922	0.037	2.47×10^{-5}

From these tables (Table 4.6, Table 4.7, and Table 4.8), graphs of weight loss versus cycles and wear index versus cycles were constructed as shown in Figure 4.4 and Figure 4.5 below.

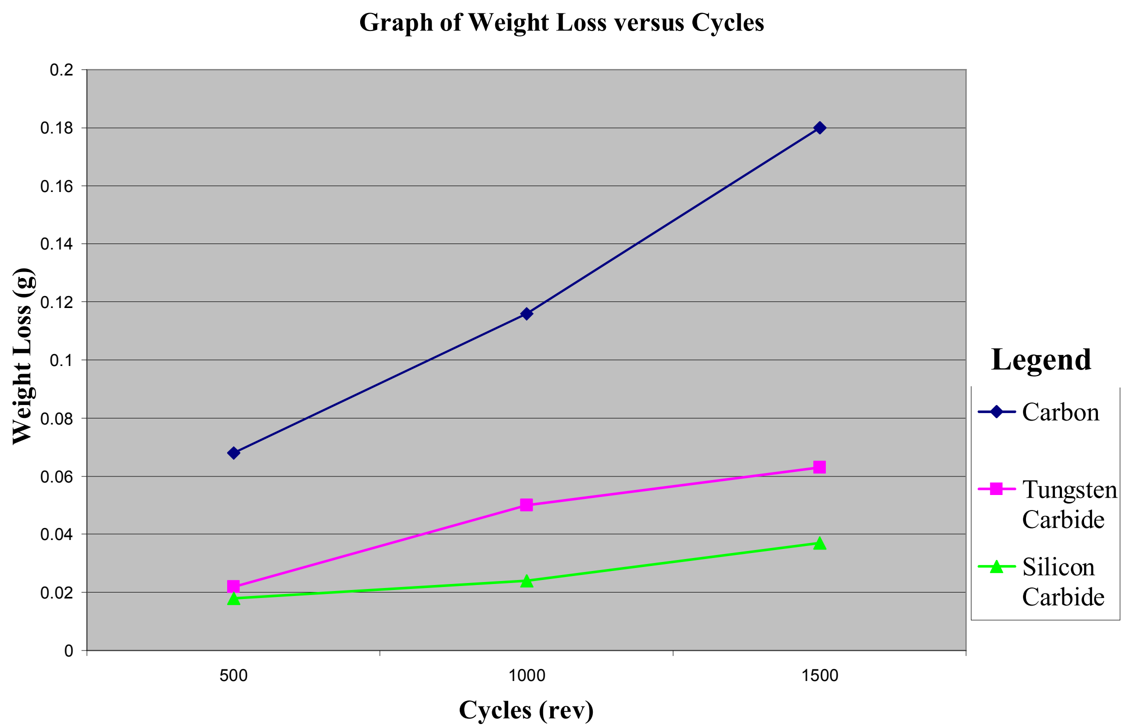


Figure 4.4: Weight Loss versus Cycles

From the above graph in Figure 4.4, it showed that Carbon had the highest weight loss among the three materials followed by tungsten carbide and silicon carbide. For Carbon material, it had percentage weight loss of 0.26 % for 500 cycles, 0.45 % for 1000 cycles and 0.70 % for 1500 cycles. For Tungsten Carbide, it had 0.03 % of weight loss for 500 cycles, 0.07 % for 1000 cycles and 0.08 % for 1500 cycles. As for Silicon Carbide, it had percentage weight loss of 0.03 % for 500 cycles, 0.04 % for 1000 cycles and 0.06 % for 1500 cycles.

From these values it showed that as the number of cycles increased the weight loss from its original weight would also be increased. This was because as the number of cycle increased, the amount of contact and friction that occurred on the surface of the mechanical seal would also be increased thus increasing the weight loss.

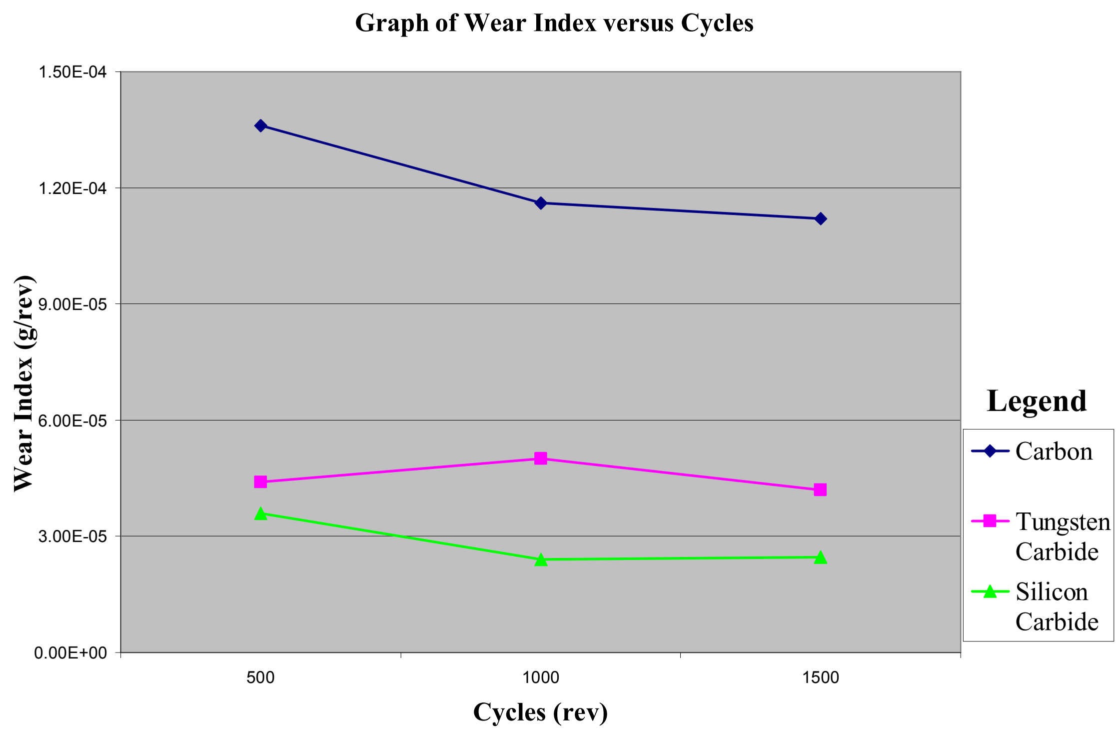


Figure 4.5: Wear Index versus Cycles

Figure 4.5 showed that carbon had the highest wear index among the three materials followed by Tungsten Carbide and Silicon Carbide. The result from this figure indicated that the weight index was greatly affected by the weight loss. Initially, there were great significant of wear indexes for the first 500 cycles for every material. This was due to the soft layer formed at the outermost surface of each material which could easily wear. After that, the wear indexes for the next two cycles were reduced steadily.

From this experiment, it showed that Silicon Carbide had the highest resistance to wear and abrasion followed by Tungsten Carbide and Carbon. Based on this experiment, it showed that Silicon Carbide was the most effective material to be used as mechanical seal.

4.4 Compression Testing (ASTM E 9)

Compression testing was conducted to determine the compressive strength to failure of the materials. There were several forces operating in a mechanical seal. The forces were axial and radial forces, closing and opening forces, and hydrostatic and hydrodynamic forces. These forces would react against each other, where the seal face would eventually being pressed and compressed.

Table 4.9, Table 4.10, and Table 4.11 below showed the results of compression testing for carbon, tungsten carbide and silicon carbide, respectively.

Table 4.9: Result of compression testing for carbon

Time (s)	Stroke (mm)	Load (kN)
57.000	1.12	58.28
57.281	1.13	58.92
57.547	1.14	59.58
57.813	1.15	60.21
58.094	**1.15**	**60.33**
58.359	1.16	59.93
58.625	1.17	59.78
58.906	1.18	56.57
59.172	1.18	56.07
59.438	1.19	55.28
59.719	1.20	54.90

Table 4.10: Result of compression testing for tungsten carbide

Time (s)	Stroke (mm)	Load (kN)
497.531	7.79	1319.46
497.812	7.79	1319.61
498.093	7.79	1319.54
498.375	7.79	1319.79
498.656	7.79	1319.69
498.937	**7.79**	**1319.82**
499.218	6.72	678.58
499.5	5.06	15.98
499.781	5.05	15.98
500.047	5.05	15.99
500.328	5.05	15.92

Table 4.11: Result of compression testing for silicon carbide

Time (s)	Stroke (mm)	Load (kN)
485.094	9.63	462.08
485.36	9.64	463.34
485.625	9.64	464.34
485.891	9.65	465.49
486.172	9.66	466.39
486.438	**9.66**	**467.11**
486.703	9.75	348.89
486.969	9.95	0.19
487.235	9.91	0.27
487.5	9.88	0.28
487.766	9.86	0.25

Data from these tables were gathered and a graph of load versus stroke was plotted for carbon, tungsten carbide, and silicon carbide as shown in Figure 4.6 below. The graph showed the highest load every material could sustain before plastic deformation occurred (rupture).

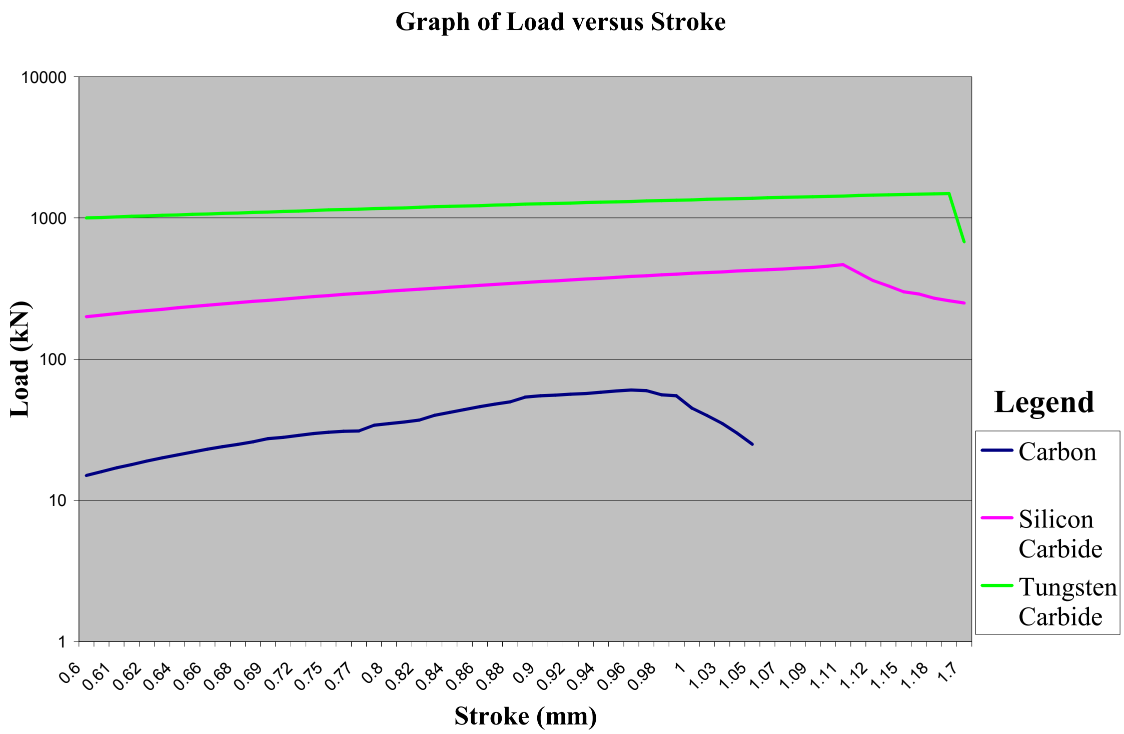


Figure 4.6: Load versus Stroke

The figure and the tables above indicated the load and the stroke when the experiment was conducted. The highest load values in the tables and the graph were the maximum load that the materials could withstand before it ruptured. These load values also indicated the compressive strengths to failure of each material when subjected to the compressive force carried out by the piston of the testing equipment (stroke).

Figure 4.6 above showed the graph of load versus stroke for every material. From the graph, it indicated that tungsten carbide had the highest compressive strength to failure where it could withstand a maximum load of 1319.82 kN followed by silicon carbide with the maximum load of 467.11 kN and carbon with 60.33 kN of maximum load.

From these values, it showed that tungsten carbide had the highest compressive strength to failure when subjected to compressive force followed by silicon carbide and carbon. Also, the material which had the higher maximum load would take longer time before the ruptured occurred. Tungsten carbide was the material which took the longest time before deformation happened because of the bond between the atoms was the strongest compared to silicon carbide and carbon. This bond would try to keep its initial state until it could no longer sustain its shape thus plastic deformation would take place.

From this experiment, it could be concluded that the effective material with the highest compressive strength was tungsten carbide where it could withstand the compressive load up to 1319.82 kN followed by silicon carbide and carbon.

4.5 Heat (Temperature) Test

For the heat (temperature) test, the heat was generated when the materials surfaces came in contact with the rotating face of the lapping machine. For every 10 minutes, the heat generated was recorded by using the digital thermometer. The results of the experiment were shown in Table 4.12, Table 4.13 and Table 4.14 below.

Table 4.12: Generated heat for silicon carbide

Time (min)	Temperature (^{o}C)
10	28.0
20	29.2
30	28.8
40	28.0
50	28.0
60	28.4

Table 4.13: Generated heat for tungsten carbide

Time (min)	Temperature (^{o}C)
10	28.8
20	28.2
30	28.0
40	28.0
50	29.2
60	28.4

Table 4.14: Generated heat for carbon

Time (min)	Temperature (^{o}C)
10	29.2
20	29.2
30	28.4
40	28.0
50	29.2
60	29.0

Initial findings from this experiment showed that the recorded temperatures of every material had no big difference with each other. To support these findings, a graph was constructed as shown in Figure 4.7 below.

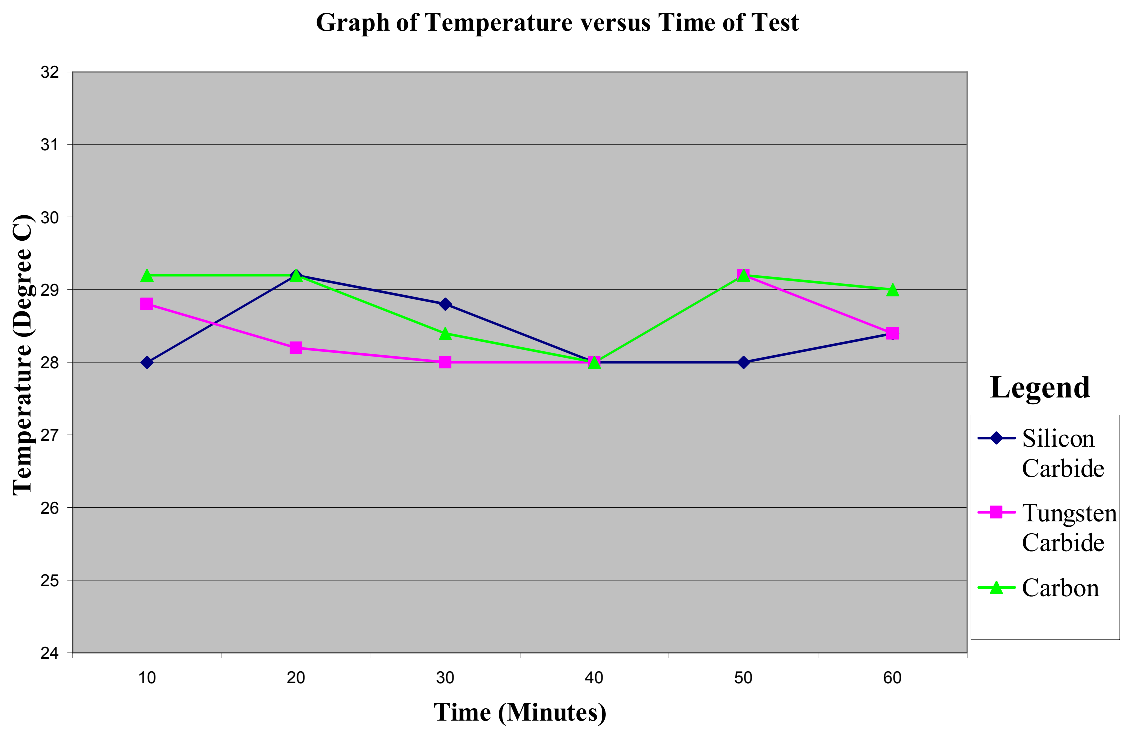


Figure 4.7: Temperature versus Time of Test

Figure 4.7 showed the graph of temperature versus time for the heat test. From the tables and figure above, it showed that the generated heats for every material were almost the same. The values were fluctuating within the temperature ranges of 28^{o}C and 29^{o}C and never went higher although the time was increased. There was no significance difference between the gathered data.

This situation happened due to the lubricant usage. The function of the lubricant was to cooling down the lapping machine's surface to avoid excessive heat that would damage its surface. Hence, the temperature would remain the same throughout the process to maintain the equipment's life span. Due to this limitation, the experiment was considered unsuccessful because the desired outputs were not achieved. However, the procedures would be adjusted in the future to get the accurate results.

CHAPTER 5
CONCLUSION AND RECOMMENDATION

5.1 CONCLUSION

The main objective of this project was to determine the most effective material that could be used for mechanical seals to ensure high reliability of the CTP pumps at OGT. The effective material could reduce the occurrence of failures in the pumps. The material was determined based on the characteristic and properties that the material possessed. Lab tests like hardness testing, corrosion testing, wear testing, compression testing and heat (temperature) testing were conducted to evaluate the material properties. Based on these testing it could be concluded that:

a) Silicon carbide had the highest strength with the highest hardness value of 91.74 HRC (Rockwell hardness C scale) followed by tungsten carbide (82.9 HRC). Carbon had the lowest strength with 118.5 HRB (Rockwell hardness scale B) which was considered as malleable iron. Material with C scale was hardened compared to material with B scale. It showed that silicon carbide was a very hard material that was suitable to be used as mechanical seal.
b) All materials showed a great resistance to corrosion in the corrosion testing where very low weight losses were recorded. However, silicon carbide had the highest resistance to corrosion because there was no corrosion occurred on the material in the testing and no weight loss was recorded. Tungsten carbide had 0.0072 % of weight loss followed by carbon with 0.02915 % of weight loss. This showed that silicon carbide could withstand highly corrosive environment in the real applications where hydrogen sulphide and carbon dioxide existed.

c) Silicon carbide had the highest wear resistance where it recorded the least value of weight loss percentage (0.04 %) and the smallest wear index of 28.2 μg/rev, followed by tungsten carbide (0.06 % and 45.3 μg/rev) and carbon (0.47 % and 124 μg/rev). This showed that silicon carbide had a great resistance to abrasive wear that usually occurred in the real process. This also indicated that silicon carbide would have longer life span compared to the other materials.

d) Tungsten carbide had the highest compressive strength to failure with the maximum compressive load of 1319.82 kN followed by silicon carbide (467.11 kN) and carbon (60.33 kN). The compressive strength to failure showed the ability of the materials to withstand the compressive forces that were exerted on the mechanical seals.

From these findings, it concluded that the most effective material that could be used for mechanical seal was silicon carbide because it had the highest strength, the highest resistance to corrosion, the highest resistance to wear, high compressive strength to failure and it could withstand high pressure condition. The application of this material as mechanical seal in the CTP pumps would help in preventing leaking problem of the pumps.

5.2 RECOMMENDATION

One of the issues arise during this study was the usage of used mechanical seal. The accuracy of the results using used mechanical seal would not be as good as using a new one because the properties of the material had change and gradually decreased when it was used for a certain period of time. The usage of new mechanical seal would improve the outcomes of the study but it may need some additional cost.

The other problem faced during completing the study was technology issue. In the Corrosion Testing where Autoclave machine was used, the temperature was at room temperature because the temperature gauge of the equipment was not functioning. In this testing, it was suggested to simulate the corrosive environment in high temperature and high pressure conditions as in the real applications. The results could be improved by using the machine that could provide these both conditions.

This study could be continued by evaluating other type of materials that could be used as mechanical seal in stead of the materials that were evaluated in this project which were carbon, tungsten carbide and silicon carbide. Other materials that could be studied were aluminum oxide and bronze where these materials may have desirable properties such as excellent wear characteristics and could withstand high pressure and high temperature conditions.

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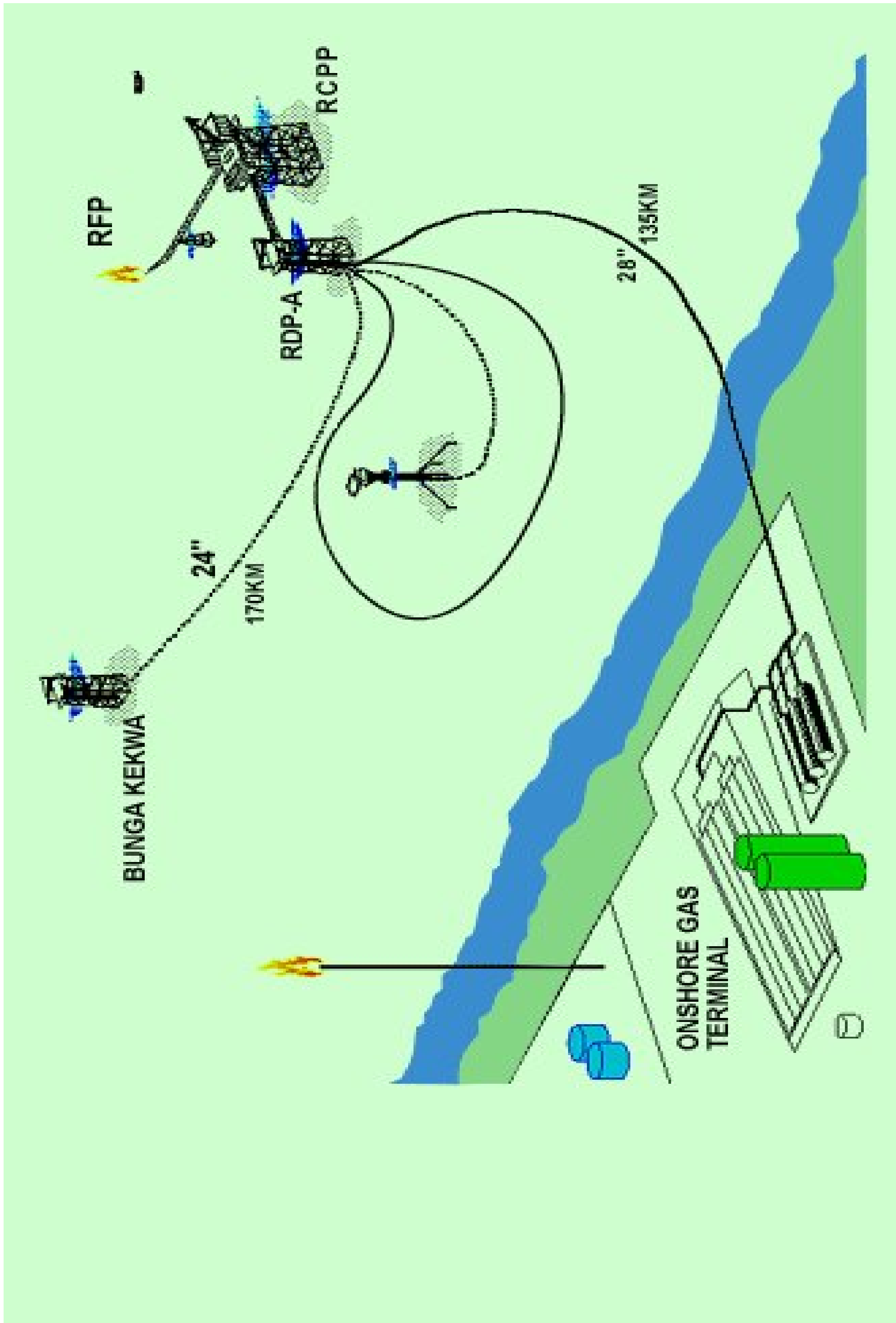

Figure A.1: Facilities layout of OGT

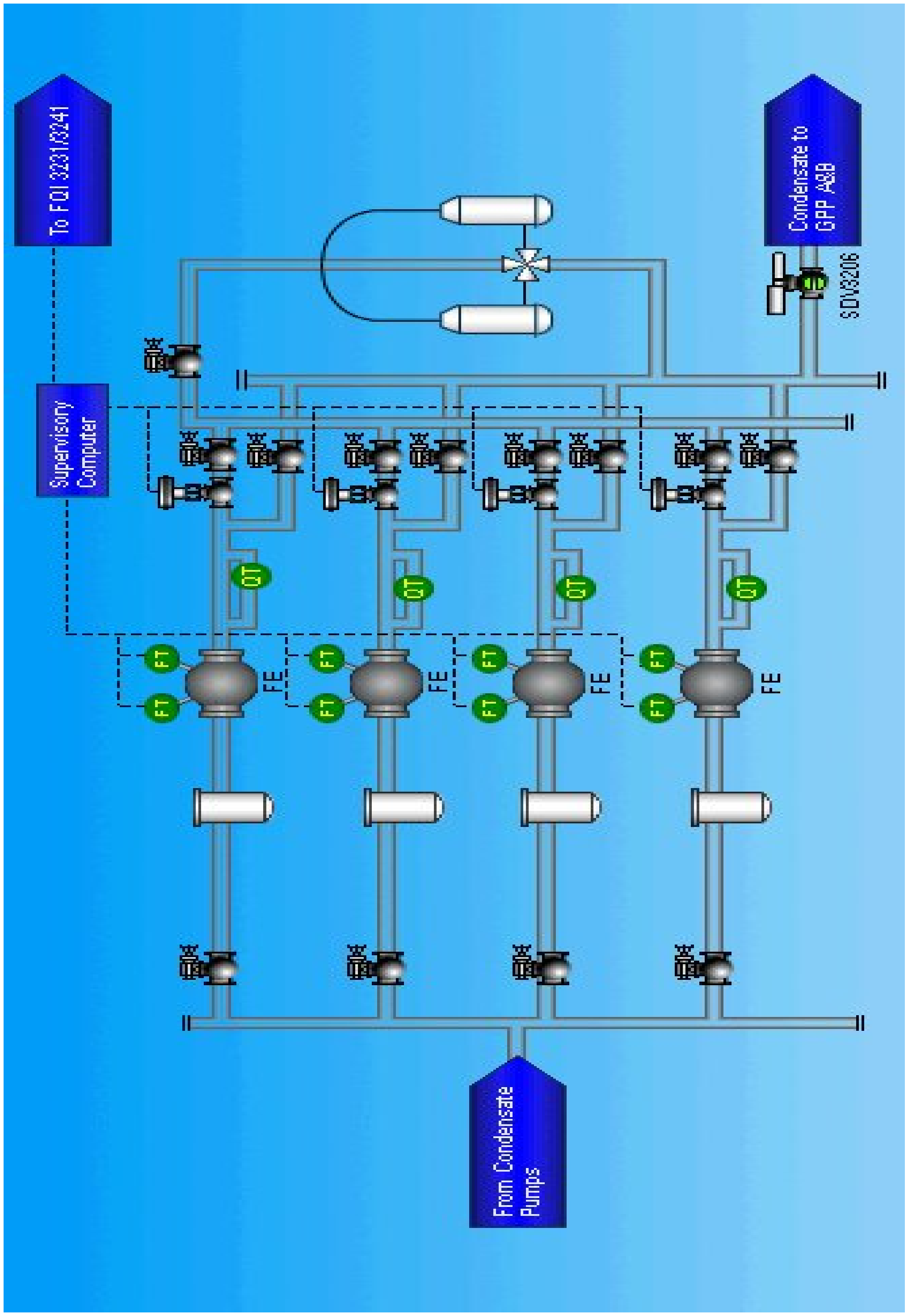

Figure A.2: Overview of condensate metering system

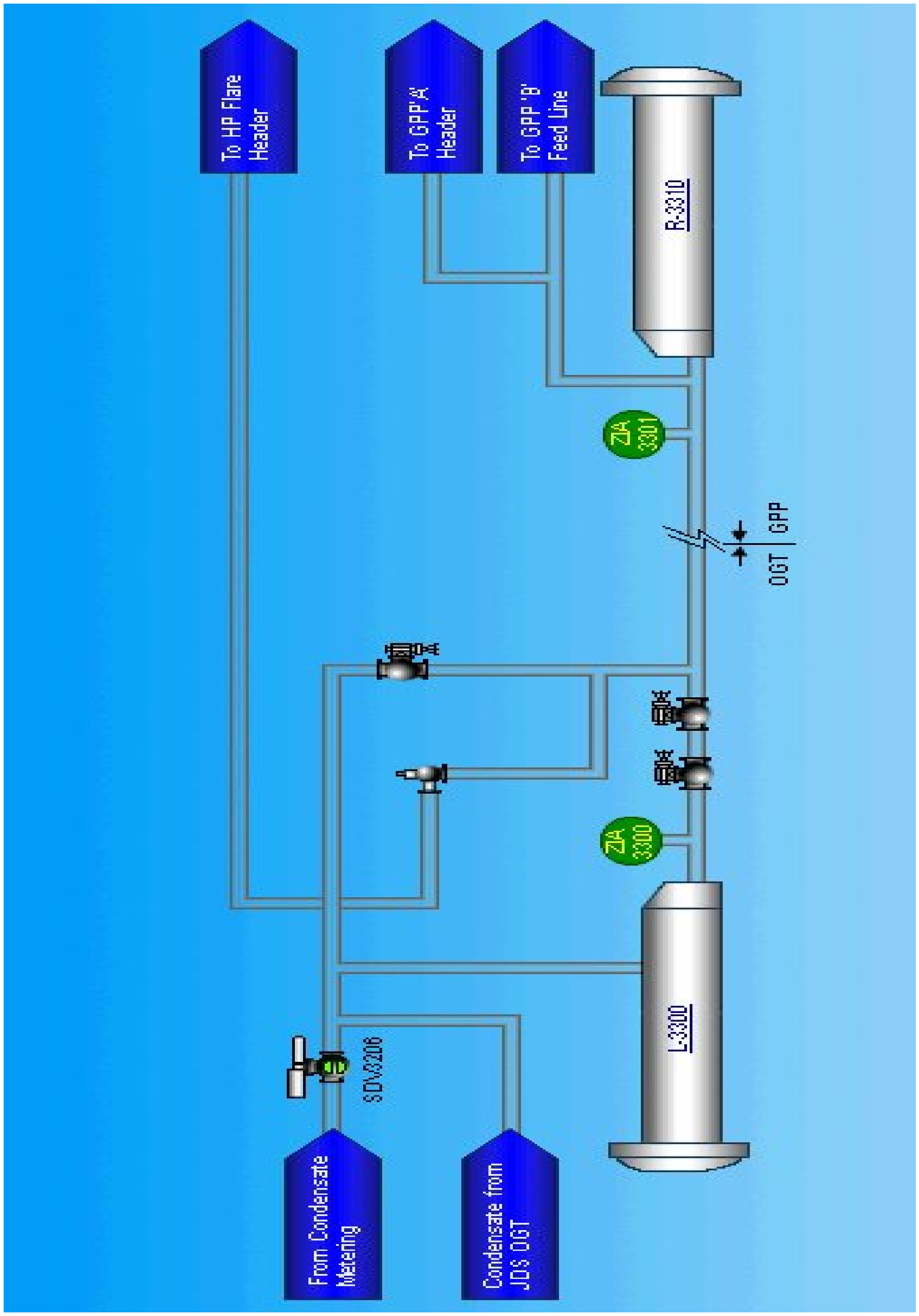

Figure A.3: Overview of condensate import/export

Appendix 4

Table A.1: Gantt chart of First Semester of 2-Semester Final Year Project

No.	Detail/ Week	1	2	3	4	5	6	7	Mid-semester break	8	9	10	11	12	13	14
1	Selection of Project Topic	■	■													
2	Preliminary Research Work		■	■	■											
3	Submission of Preliminary Report				●											
4	Project Work					■	■	■								
5	Submission of Progress Report									●						
6	Seminar (compulsory)									●						
7	Project work continues									■	■	■	■	■	■	■
8	Submission of Interim Report Final Draft														●	
9	Oral Presentation															●

● Suggested milestone

■ Process

Appendix 5

Table A.2: Gantt chart of Second Semester of 2-Semester Final Year Project

No.	Detail/ Week	1	2	3	4	5	6	7		8	9	10	11	12	13	14
1	Project Work Continue	■	■	■					Mid -Semester Break							
	- Temperature (heat) test is conducted at vendor's lab		●													
2	Submission of Progress Report 1				●											
3	Project Work Continue				■	■	■	■								
	- Changes on test parameters. Hardness test is done					●										
4	Submission of Progress Report 2									●						
5	Seminar (compulsory)									●						
5	Project work continue									■	■	■	■			
	- Corrosion test and wear testing are conducted										●	●				
6	Poster Exhibition											●				
	- Compression testing is conducted												●			
7	Submission of Dissertation (soft bound)													●		
8	Oral Presentation														●	
9	Submission of Project Dissertation (Hard Bound)															●

● Milestone

■ Process

Table A.3: Comparison of Properties of Seal Face Materials

Properties \ Materials	Carbon – Resin Impregnated	Carbon – Antimony Impregnated	Carbon – Dry Running	Carbon – Acid Grade	Silicon Carbide – Reaction Bonded	Silicon Carbide – Self Sintered	Tungsten Carbide – Nickel Bonded	Alumina Oxide
Density – g/cc	1.8	2.4	1.9	1.9	3.08	3.1	14.9	3.89
Hardness*	95	90	80	95	2400	3000	1600	1500
Modulus – psi	3.5×10^6	3.6×10^6	2.0×10^6	2.6×10^6	50×10^6	60×10^6	88×10^6	55×10^6
Transverse Strength – psi	11500	12500	8000	10000	40000	65000	250000	47000

*Note 1: All values are representative values and will vary depending upon vendor and specific grade.

Note 2: Hardness for carbon materials is given in Shore Scleroscope; hardness for ceramic materials is given in Vickers.

Table A.4: Strengths and Weaknesses of Seal Face Materials

Properties \ Materials	Carbon - Resin Impregnated	Carbon - Antimony Impregnated	Carbon - Dry Running	Carbon - Acid Grade	Silicon Carbide - Reaction Bonded	Silicon Carbide - Self Sintered	Tungsten Carbide - Nickel Bonded	Alumina Oxide
General Duty	Excellent	Fair	Poor	Poor	Excellent	Excellent	Good	Fair
Dry Running*	Good	Poor	Excellent	Fair	Fair	Good	Poor	Good
Abrasive Services	Poor	Poor	Poor	Poor	Excellent	Excellent	Excellent	Fair
Blister Resistance*	Fair	Excellent	Fair	Fair	Excellent	Excellent	Fair	Fair
Chemical Resistance	Good	Fair	Good	Excellent	Fair	Excellent	Fair	Excellent
Impact Resistance	Good	Excellent	Good	Good	Fair	Fair	Excellent	Fair
Thermal Shock Resistance	Excellent	Excellent	Excellent	Excellent	Excellent	Excellent	Good	Poor

*Note: Ratings may indicate properties when run against other common seal face materials

Table A.5: Chemical Compatibility of Seal Face Materials

Chemicals \ Materials	Carbon - Resin Impregnated	Carbon - Antimony Impregnated	Carbon - Dry Running	Carbon - Acid Grade	Silicon Carbide - Reaction Bonded	Silicon Carbide - Self Sintered	Tungsten Carbide - Nickel Bonded	Alumina Oxide
Light Hydrocarbons	Fair	Excellent	Excellent	Excellent	Excellent	Excellent	Excellent	Excellent
Acids	Good	Poor	Good	Excellent	Fair	Excellent	Good	Excellent
Bases	Good	Poor	Good	Excellent	Poor	Excellent	Good	Excellent
Overall Chemical Resistance	Good	Fair	Good	Excellent	Good	Excellent	Good	Excellent

*Note: Chemical compatibility ratings are for general categories of chemicals. The user must confirm the compatibility for each specific application.

Sample of calculations

1) Corrosion testing

Sample calculation for silicon tungsten carbide (Sample 1);

a) To calculate weight loss; Initial weight = 75.609 g

Final weight = 75.601 g

Weight loss = (Initial weight – Final weight)

= (75.609 – 75.601) g

= **0.008 g**

b) To calculate weight loss percentage;

Weight loss (%) = (weight loss / original weight) x 100

= (0.008 / 75.609) x 100

= **0.0106 %**

2) Wear testing

Sample calculation done for Carbon (applied load of 10000 g) after 500 cycles;

a) To calculate weight loss; Initial weight = 25.769 g

Final weight = 25.701 g

Weight loss = (Initial weight – Final weight)

= (25.769 – 25.701) g

= **0.068 g**

b) To calculate wear index; Weight loss = 0.068 g

Cycles = 500 rev

Wear index = (Weight loss / Cycles)

= (0.0.68 g / 500 rev)

= **1.36×10^{-4} g/rev**