Experimental Analysis of Mechanical Seal Design with Enhanced Thermal Performance

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ACKNOWLEDGEMENTS

I would like to say thanks to God for guiding and helping me every step of the way. Without his blessing none of this would be possible. Thanks to my friends for their support and encouragements throughout this period, without them it would have been much harder.

Thanks to my advisor Dr. M.M. Khonsari for the guidance, advice and support he had given me over the years. Thanks to the members of our group, for your support and guidance.

To Mr. Buck, Mr. Allan Dupree, and Mr. Adam Hoover from John Crane, I thank you guys for the support and assistance you have shown toward this research over the years. Special thanks to Mr. Jim Layton from the engineering machine shop and Mr. J. Campbell from Metal solution in Lafayette for helping me whenever it was needed.

Finally, Thanks to the Louisiana Board of Regents, the US Department of Energy and Louisiana Department of Natural Resources without financial support this research would not be possible.
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NOMENCLATURE

$A_f$ – seal face area

$B$ – geometric balance ratio, dimensionless

$C_p$ – specific heat

$D_m$ – mean diameter

$F_c$ – coefficient of friction

$H_m$ – heat generation rate for the seal

$K$ – pressure gradient factor, dimensionless

$K/H$ – wear coefficient

$M$ – mass flow rate

$N$ – shaft speed

$P_m$ – contact pressure

$P_s$ – spring pressure

$P_v$ – parameter combining seal face contact pressure and the rotational velocity

$T_{in}$ – coolant inlet temperature

$T_{out}$ – coolant discharge temperature

$w$ – wear rate (time rate of wear)

$W_i$ – load supported by fluid

$W_m$ – load supported by contact surface

conven- conventional

surf- surface

temp- temperature
ABSTRACT

For industries that use pumps and mixers in their process operations, it is of paramount importance to control the leakage from these equipments. The leakage presents safety hazards and economic loss. It is widely believed that the majority of downtime associated with a pump or a mixer is the result of a mechanical seal failure. Therefore, it is important for these industries to have a mechanical seal that is a reliable performer and enables a longer operational life than what exist in the market at present.

The aim of this research was to design and test a mating ring with superior thermal performance that could be used in the conventional seal arrangement without modification to the existing arrangement. The new design was called Fin Mating Ring. The design had radial fins on its circumference and it could replace an existing conventional mating ring without modification to the gland.

This thesis contains both an analytical and experimental phases associated with the performance of the new design. Different designs were created and optimized by using Finite Element Analysis (ANSYS). The performances of the new designs are compared with the heat transfer characteristics of the conventional design.

Experimental measurements of temperature, flow rates, and pressure in both the new mechanical seal design and the existing conventional seal were performed. Both were tested using the same flow, pressure, rotational velocity and coolant (flush). The test results revealed that the temperatures measured in
the new design (Fin Mating Ring) were lower than the conventional ring. Therefore, the fin mating ring design had superior heat transfer characteristic that the existing conventional mating ring. Due to the lower surface temperatures, the fin mating ring design is expected to have a lower wear rate than the conventional ring and sustain a longer seal life.
CHAPTER 1: INTRODUCTION

Mechanical seals are used to control leakage from centrifugal pumps and mixers. They also isolate the shaft housing from the process fluid and reduce leakage to the surrounding environment. Before mechanical seals were invented, packing was used to control the leakage of process fluid into the environment. However, packing leakage rate was too high and if the packing was not maintained properly, it could result in scouring of the shaft and loss of packing material. This would accelerate the leakage rate and cause the seal to fail. The inherent problems associated with the packing cause industries to look for a more reliable and efficient way to control process fluid leakage. These necessitates led to the invention of the mechanical seal.

Figure 1: Mechanical Seal Components (Somanchi, 2004)
The introduction of mechanical seals to control process fluid leakage made packings less desirable. Packings became increasing unacceptable when the leakage rate of the process fluid from packing is compared to that from a mechanical seal. The ratio between packing and mechanical seal leakage rate is 800:1 respectively (PPc pp1-3). Furthermore, packing increases the power requirement by a ratio of 6:1 when compared with mechanical seals. With the restrictions placed on process industries factories and manufacturers by the Environmental Protection Agency (E.P.A.), the Occupational Health and Safety Association (O.H.S.A.) and the economic implications of the process fluid loss coupled with downtime as well as workers safety, have been instrumental in the development of a mechanical seal in favor of packings. the development of a mechanical seal that could be more efficient in leakage prevention and have a longer service life had thus become extremely crucial.

1.1 What Are Mechanical Seals?

A seal is a mechanically loaded device consisting of rotating and stationary members, having lapped faces operating in close proximity under hydraulic load, used to minimize the leakage between a rotating shaft and a stationary housing (PPc pp 2-1).

1.2 Types of Seals

There are two distinct groups of seals, surface guided seals and fixed clearance type seals.

1) Surface guided seals are those where one of the seal faces is flexibly mounted with respect to the shaft or housing and is thus entirely supported and guided by
the second seal face, one of which is sliding relative to the other. A surface guided seal has certain characteristics common to bearings. The leakage gap is determined by the character of the surfaces and seal design. In the case where there is no lift off, the seal is termed as contacting seal and when lift off happen during operation they are term non-contacting. However, in both cases, under static condition and zero pressure all surface guided seals contact (Lebeck, 1991).

2) Fixed clearance seals are seals that by design do not touch throughout the life of the seal

1.2 (A) Fixed Clearance Seals

Visco Seal: The effectiveness of a visco seal depends on the viscosity of the liquid it is using for the sealing action and the clearance. This seal uses the liquid itself to seal against it or a separate supply of liquid. It may also seal against a gas.

Figure 2: Visco seal (Lebeck, 1991)
**Labyrinth Seal:** This is a mechanical seal that fits around a shaft to prevent the leakage of oil or other fluids. This seal is composed of many straight threads that press tightly inside another shaft, or stationary hole, so that the fluid has a very difficult path to escape into the surrounding. Some designs have threads that exist on the outer and inner section of the seal. These threads interlock, to produce the long characteristic path to slow leakage. However, for these seals to be effective on a rotating shaft, a very small clearance must exist between the tips of the labyrinth threads and the running surface.

![Figure 3: Labyrinth seal](image)

**Bushing Seal:** These seals limit leakage while allowing a limited lubricant flow rate accompanied by a pressure drop, the lubricant flow at a pressure slightly higher than the sealing pressure. The flow area for this seal is the clearance between the bushing and the shaft. Its effectiveness in reducing process fluid leakage is determined by the length and the clearance. The clearance should be large enough to allow for all shaft rotation.

![Figure 4: Bushing seal (Lebeck, 1991)](image)
**Floating Ring Seal**: These seals are used in the buffer gas assembly so that the mating surfaces can never touch. The probability of excessive wear for the seal assembly is extremely low. This minimizes the probability of seal failure. However, due to the relatively large clearances required, floating ring seals exhibit a high leakage rate.

![Floating Ring Seal Diagram](image)

**Figure 5**: Floating ring seal (Lebeck, 1991)

**Ferrofluid Seal**: This seal uses the response of a magnetic fluid to an applied magnetic field to accomplish its sealing action. The basic seal components include the ferrofluid, a permanent magnet, two pole pieces and a magnetically permeable shaft. The magnetic circuit is completed when the stationary pole pieces and the rotating shaft, concentrates magnetic flux in the radial gap under each pole piece. The fluid which is applied to this gap assumes the shape of a liquid O-ring and produces a hermetic seal.
1.2 (B) Surface Guided Seals

**Lip Seal:** These seals have a flexible lip that makes contacts over a small axial length on the shaft. This lip rubs against the shaft or housing to prevent leakage or to stop dirt and other undesirables. The seal lip should point toward the medium being contained.

**Circumferential Seal:** These are control gap seals that are not fluid tight but rather restrict and control the flow by means of an engineered radial gap between the shaft and the seal insert. It makes contacts over a definite axial length on a
cylindrical surface. It is designed with segmented rings that operate in contact with the shaft. These segmented rings have complex joints that allow it to clamp around the cylinder entirely and thus minimize leakage.

![Image of a seal](image)

**Figure 8:** Circumferential seal (Lebeck, 1991)

**Packing:** This represents the method of stuffing packing material into an enclosed circular area to restrict the process fluid leakage commonly referred to as the stuffing box. It is pack tight into this box as this is the mechanism by which it restricts leakage. It is usually packed over a large area in order to create more leakage resistant.

**Mechanical Face Seal:** A Mechanical face seal is characterized by having contact over the area of a face or annulus. Compared to its cylindrical surface counterpart, the circumferential seal, a mechanical face seal is somewhat simpler because the seal need not to be made in segments. The term face indicates that the seal contact is over an area rather than having line contact or it may indicate that the contact is on the face of housing or a shaft. The term mechanical implies that a device rather than soft packing is used. the essential
characteristic of the seal. Mechanical designation also implies touching as well so as to allow distinction of a mechanical seal from a clearance seal. Often the mechanical face seal is referred to as an end seal or a radial face seal which indicates the form of the sealing surface (Lebeck, 1991).

The essential components of a mechanical seal are

A) Primary ring: this ring is mounted to the shaft and rotates with it. This primary ring is designed so that it provides the flexibility to allow for small axial and angular motion for misalignment between it and the mating ring. It is also called the rotational ring and makes the primary sealing surface.

B) Mating ring: this ring is rigidly mounted to the gland housing and provides the secondary sealing surface. It is usually referred to as the stationary ring.

C) Spring: it provides the force to hold both the primary and mating ring together in the absence of a fluid pressure so that they seal.

D) O-Ring: it prevents the process fluid from leaking as it presents an obstruction in its path, as a secondary sealing element it also ensures the sealing integrity of the mechanical seal.

E) Drive mechanism: this mechanism provides the rotational drive that makes the primary ring rotates. This drive mechanism is designed in the fashion that it does not reduce the self aligning characteristic of the primary ring.

1.3 Equipment in Which Mechanical Seals Are Used

Any rotating equipment which is used to process a fluid requires leakage prevention. However different types of seal are used depending on the
fluid that is to be sealed in the housing or where on the equipment the seal is to be placed. For example a lip seal is used to seal the oil in the power train of pump and a shaft (mechanical seal) is used to seal the fluid being pump. The major industrial areas, which use these mechanical seals, are:

**Compressors:** Shaft seals are used in the compressors to reduce leakage from the power train.

**Pumps:** Mechanical seals are used to reduce the leakage of the process fluid from the pump housing.

**Process Industries:** Centrifugal pumps used in the petroleum, chemical, textile, drugs and manufacturing industries generally utilized mechanical seals.
Automotive engines, transmission, gearboxes, compressors, air conditioners and other devices that require sealing use different types of shaft to prevent leakage.

**Aerospace:** The engines in rockets and turbo jets use shaft seals to restrict leakage.

**Agricultural:** Shaft seals are used by pumps used in irrigation, pumping of fertilizers and insecticides.

**Household Appliances:** Mechanical seals are used by pool pumps, garbage disposal pumps, dishwashers and washing machines pumps.

**Power Generation:** Mechanical seals are used in water turbines and steam turbines.

### 1.4 Sealing Parameters and Arrangements

Sealing configuration depends on two primary factors. The first is the location of the pressure relative to the annulus. If this pressure is on the outside as depicted in the figure 10, it is called an outside pressurized seal.

![Figure 10: Mechanical seal: outside pressurized, rotating primary ring, fixed mating ring (Lebeck, 1991)](image)
However sometimes the pressure is on the inside as shown in figure 11 this is called an inside pressurized seal.

![Diagram of a mechanical face seal: inside pressurized, rotating primary ring, fixed primary ring (Lebeck, 1991)](image)

**Figure 11:** Mechanical face seal: inside pressurized, rotating primary ring, fixed primary ring (Lebeck, 1991)

There are advantages and disadvantages associated with each arrangement. Most seal designs undergo thermally-induced radial taper that tends to open the seal faces at the outside diameter as thermal heating increases. If a seal is outside-pressurized, this convergent radial taper increases the fluid pressure load support so as to reduce the friction and the thermal load. Therefore, this configuration has stable thermal radial taper. For an inside-pressurized seal, the thermal taper is unstable and it develops a thermally induced divergent taper which results in poor lubrication. However, if one is using a large diameter small cross section seal element that is unsupported diametrically, an outside pressurized ring will buckle in the diametrical plane while the inside pressurized ring will not. Sealing arrangement can either have the primary or mating ring
rotating or stationary. If the primary ring is rotating then the secondary springs are less susceptible to clogging caused by deposits due to the rotating motion. Since in most cases the rotating ring is cheaper and it can be mounted directly on the shaft, it is more convenient to have it rotating. Also it is difficult to mount the primary ring to the housing for it to be stationary. Another advantage is that the rotating primary ring is placed inside the stuffing box where it is exposed to the fluid for easy cooling (Lebeck, 1991).

The Balance Ratio

This is the ratio between the average load $P_f$ expressed as a pressure imposed on the face by the action of the sealed pressure, and the sealed pressure $P$.

For an outside pressurized seal the balance ratio ($B$) is

$$ p \pi (r_o^2 - r_p^2) = p_f \pi (r_o^2 - r_i^2) $$

$$ B_o = \frac{p_f}{P} = \frac{r_o^2 - r_p^2}{r_o^2 - r_i^2} $$ (Lebeck, 1991)

For an inside pressurized seal

$$ p \pi (r_p^2 - r_i^2) = p_f \pi (r_o^2 - r_i^2) $$

$$ B_i = \frac{p_f}{P} = \frac{r_p^2 - r_i^2}{r_o^2 - r_i^2} $$ (Lebeck, 1991)
Figure 12: Outside pressurized seal and balance ratio (Lebeck, 1991)

Figure 13: Inside pressurized seal, balance ratio (Lebeck, 1991)
The PV Parameter

The PV parameter is defined as the product between the sealed pressure and the average sliding speed.

\[(PV)_{n} = [p(B - K) + p_{s}]U\]  (Peterson and Winer, 1980)

\(B = \text{Seal balance ratio}\)

\(K = \text{Pressure gradient factor}\)

\(p_{s} = \text{Spring pressure}\)

\(n = 1\)

The PV is a very important factor for determining the heat generated by rubbing.

It is also important for categorizing the severity of service.

Table A provides information on the effect of material properties on the PV limit. It highlights the different environment that face arrangements are suitable for and how which material used for primary and mating ring impact on the mechanical seal performance.

Figure 14 is experimental data that shows how the PV limit, material properties for the primary and mating ring influence the wear characteristic for the mechanical seal. The relationship between the choices for the rings material has an impact on the coefficient of friction for the mechanical seal. The higher the PV limit, the greater is the rings wear rate. However, the choice for the material use for primary and mating ring can improve the PV limit thus improving the seal ability to reduce the wear rate and prolong its life.
## Table 1 Seal Face Material and their PV Limitation (PPc Handbook)

<table>
<thead>
<tr>
<th>Sliding Materials</th>
<th>Mating Ring</th>
<th>PV Limit (Lb/in² Ft.min)</th>
<th>Comment</th>
</tr>
</thead>
<tbody>
<tr>
<td>Primary Ring</td>
<td>Ni-Resist</td>
<td>100,000</td>
<td>Better thermal shock resistance than ceramic</td>
</tr>
<tr>
<td>Carbon Graphite</td>
<td>Ceramic (99% AL₂ O₃)</td>
<td>100,000</td>
<td>Poor thermal shock resistance. Better corrosion resistance than Ni-resist</td>
</tr>
<tr>
<td></td>
<td>Tungsten (6% Ni)</td>
<td>500,000</td>
<td>Ni binder for better corrosion resistance</td>
</tr>
<tr>
<td></td>
<td>Siliconized Carbon</td>
<td>500,000</td>
<td>Good wear resistance. Thin layer of siliconized Carbon makes relapping questionable</td>
</tr>
<tr>
<td></td>
<td>Silicon Carbide (solid)</td>
<td>500,000</td>
<td>Better corrosion resistance than tungsten carbide</td>
</tr>
<tr>
<td></td>
<td>Carbon Graphite</td>
<td>50,000</td>
<td>Low PV</td>
</tr>
<tr>
<td></td>
<td>Ceramic</td>
<td>10,000</td>
<td>Good service on sealing. Certain acid application</td>
</tr>
<tr>
<td></td>
<td>Tungsten Carbide</td>
<td>120,000</td>
<td>PV is up to 185,000 with two grades that have different binders</td>
</tr>
<tr>
<td></td>
<td>Silicon carbide</td>
<td>500,000</td>
<td>Excellent abrasion resistance. Good corrosion resistance, moderate Thermal shock resistance</td>
</tr>
</tbody>
</table>
Figure 14: Wear rate as a function of $PV$ level (Lebeck, 1991)

1.5 Factors That Influence Seal Life And Performance

To ensure longevity, it is important that the contact surface of the static and rotating ring component of the mechanical seal operate at the lowest temperature possible. Also, the temperature at the contact surface should be uniform. This is necessary to reduce waviness (deviation from flat) radial taper
(deviation from flat in the radial direction) and thermo-elastic instabilities (Lebeck, 1991).

Figure 15: Common distortions (Lebeck, 1991)

Figure 15 shows how thermo-elastic instabilities affect the rings. The heat generated due to rubbing cause the rings shape to distort which increase leakage and ultimately seal failure. The effectiveness and efficiency of a mechanical seal depends on its ability to restrict leakage. However the amount of leakage permitted by a seal depends on its wear resistance, thermal conductivity of the primary and mating surface, orientation, geometry, and the chemical compatibility of the rotating and static ring. Researchers have done tests to ascertain how seals made out of different materials will behave under working conditions (Paxton, 1980). The tests ascertain the working environment for which the mechanical seals with certain materials are suitable. Earlier seal manufacturers made seals with the basic components, i.e. springs, bellows, o-rings, sleeves, and the static and rotating ring. However, they found out that under different operating conditions depending on temperature, pressure and
orientation seals made out of different material behave differently. The $PV$ value is important in ascertaining how much heat will be generated at the contact interface during operation. For mechanical seals the $PV$ value can cause the friction coefficient to increase, decrease or stabilize at a low value after steady state is reach. When the rings are at steady state and the $PV$ value is within a certain range the mating ring will have a transfer film deposit on its surface which will not break down unless the $PV$ value changes (Lebeck, 1991). These transfer films keeps the coefficient of friction at a low value and reduce the wear on the rings. This transfer film is made of the softer material of the primary ring being deposited on the harder material of the mating ring. The chemistry behind its formation depends on the temperature at the interface. This temperature depends on the $PV$ value.

**Lubrication:** Mechanical seals can operate in any of the three lubrication regime, boundary, mixed and full.

![Lubrication regimes](image)

*Figure 16: Lubrication regimes (Lebeck, 1991)*
In the full-film mode, the entire load is supported by the fluid pressure and there is almost no touching of the surface asperities. The coefficient of friction is very low and the wear will be very small, thus the rings last longer. However, full film lubrication will have increase leakage of the process fluid. The mixed lubrication regime is the most common in mechanical seals. Mixed friction occurs as some of the load is carried by the fluid pressure and the rest by mechanical contact due to the surface asperities. The load supported by the mechanical contact is an important consideration in seal design development as this impact on the friction heating and the resulting wear. In the mixed lubrication regime, the wear will be higher than in the full-film mode but leakage of the process fluid will be less. Whenever boundary lubrication is not present, it is because the operational speed is so low that the fluid pressure have not develop or the level of lubricant present in the interface is so small it cannot develop lubrication film. This will result in high frictional heating and wear and the seal life would be short. However, leakage in this mode would be much less than the other two modes.

**Interface Shape:** The geometry of the interface shape affects the operational life of the mechanical seal. The interface shape is defined by the gap between the primary and secondary surfaces with respect to the radial and angular alignment between the surfaces. The interface shape does not remain constant but changes with time due to wear. However, this shape is an indication of how the mechanical seal is performing and the type of lubrication present at the interface. Figure 17 shows different interface shapes. These shapes correspond to the type of lubrication present at the interface and the seal wear. The interface shape is
fundamental to the seal’s performance. Some of the shapes shown provide good hydrodynamic lubrication like the first shape shown. The seventh shape shown is likely to give rise to excessive leakage, though it has good hydrodynamic lubrication the third, fifth, and sixth shape will cause a thermally distressed surface on the rings. It will also cause the coolant to vaporize giving rise to hot spots. These hot spots occur because the rings are touching off in these areas and they are carrying in the contact force. These hot spots have a higher wear rate than the other areas of the contact surface resulting in uneven wear.

![Interface Shape](image)

**Figure 17:** Interface shapes (Lebeck, 1991)
CHAPTER 2: LITERATURE REVIEW

To design for an enhanced seal life, it is necessary to study the factors that influence its performance. It is important to understand how the operating conditions, design and flush arrangement applied to cool the interface affect seal life. Another important factor is the materials used in the seal rings. To determine the optimum performance and to prolong seal life, both experimental and theoretical analyses are needed.

2.1 Estimating the Heat Transfer Coefficient

In order to estimate how much heat is transfer between the fluid and the rings in contact within the sealing chamber, a study was conducted by Doanne (1990). This study comprised of a series of experimental measurement combined with numerical computations to obtain the local and average Nusselt number for the wetted area of the mating ring and the primary ring. This average Nusselt number was used to calculate the average heat convection coefficient. Emphasis was placed on showing how the wetted area influences the amount of heat transfer that take place. It was noted that the heat transfer for the seal is greater in the axial and radial by a ratio of twenty to one with respect to the circumferential direction (Doanne, 1990). This is an important concept for seal designers to take into consideration. The wetted area i.e. the area in contact with the flush fluid plays a vital role in removing the frictional heat generated between the contact face. Thus it is necessary to expose as much area as possible of the seal to the flush to improve the heat transfer characteristics. This would reduce wear and improve the seal life. The greater the exposure to the flush fluid the
greater is the dissipation of the heat generated at the contact interface.

Numerical and computational analyses were performed by Luan and Khonsari 2006. This research showed the flow pattern emerging from the flush fluid for the purpose of cooling of the rings. Both laminar and turbulent flows were considered. Determination of an average approximate value for the convection coefficient is another important research needed to establish how much heat transfer takes place between the rings and the coolant. Analyses were performed to establish correlation relations for predicting the heat transfer coefficient based on the Nusselt number (Luan and Khonsari, 2006). It is important to have a value of the heat transfer coefficient for rings design and performance evaluation.

Correlations for the Nusselt numbers from which the average heat transfer coefficient develop by Luan and Khonsari, 2006 are as follows

$$Hc = \frac{Nu K_f}{D}$$  \hspace{1cm} (1)

With \(Nu_{\text{rotor}} = 72.62 \Pr^{0.89} \Re_{ro}^{0.07}\) \hspace{1cm} (2)

$$Nu_{\text{stator}} = 121.51 \Pr^{0.89} \left( \frac{Re_{\text{flush}}}{Re_{ro}} \right)^{0.56}$$ \hspace{1cm} (3)

Where \(Pr\) is the Prandtl number with the restriction that \(Pr>0.6\), \(Re_{ro}\) is the Reynolds number (turbulent flow) with the velocity being the rotational speed of the motor. \(Re_{\text{flush}}\) is the Reynolds number using the radial flush flow velocity and \(K_f\) is the thermal conductivity. The frictional heat generated at the contact interface is conducted through the rings to their surfaces where convection takes
place with the coolant to dissipate the heat. The maximum heat flux occurs on the primary ring surface near the contact face.

Philips (1997) experimentally analyzed the impact of thermal distortion of mechanical seal faces has on the seal performance. The seal face torque, thermal gradients, and fluid flow patterns under the normal operating conditions of the seal were measured. The analysis showed that one can ascertain an average value for the heat convection coefficient from the Nusselt number and use this value to calculate the heat generated due to friction at the seal interface. Using this heat generation value, the coefficient of friction for the seal could be estimated. Phillips used this value (average heat convection coefficient) to analyze how a particular seal would perform within a specific environment.

2.2 How Surface Finish Influence Seal Performance

The rings surface finish can have a pronounced influence on the performance of a mechanical seal. Micro asperities on the rubbing surfaces contribute to the forces holding the seal faces apart (Anno, 1990). Seal face materials vary in their grain structure and since the original smooth planar surface finish is quickly worn off, then it is expected that the size and concentration of the micro asperities on the rubbing faces be a function of the rubbing material themselves. Therefore, load support capability, coefficient of friction and torque should be dependent upon the materials, all other variables being equal. Batch, (1990) assumed that part of the force keeping rubbing faces separate is due to the surface waviness. However, to some extent, this force depends on the stiffness or the compressive modulus of the rubbing faces.
Waviness was assumed to be generated by the wearing in process which takes place at start up. It was clear that the coefficient of friction increases as more loads are carried by the asperities and less by the fluid between the contact interface. Therefore, the surface finish is an important factor in evaluating a mechanical seal performance as surfaces with lower asperities have less to carry the load. This provides more area for fluid to be between the contact surfaces which result in a higher fluid load.

2.3 Heat Generation and Evaluation of Coefficient of Friction

Another critical factor in the evaluation of seal performance is to determine the heat generated at the interface. Buck, (1999) proposed a simple analytical model that approximates the seal as a fin and provided an equation for solving the heat generated at the seal interface. The equation used for heat generated at the interface is

\[ H_m = P_m V A_f f \]  

\[ H_m \] is heat generated at the contact surface

\[ P_m \] is the contact pressure

\[ V \] is the rotational velocity

\[ A_f \] is the contact face area and \( f \) is the coefficient of friction

The amount of heat transfer to the flush is calculated by using

\[ H = m C_v \Delta T \]  

\( H \) is the heat

\( m \) is the mass flow rate (coolant)
$C_p$ is the specific heat and

$\Delta T$ is the temperature change

This equation is important in determining the flush rate necessary to keep the seal temperature at a minimum level during operation to prolong the seal life.

The contact pressure $P_m$ plays an important role in determining the frictional heat generated and the seal subsequent wear rate. Therefore care must be taken when its value is being estimated. The contact pressure is evaluated using this formula (Buck, 1999)

$$P_m = (P[B - K] + P_s)$$

(6)

Where $P_s$ is the spring pressure on the face of the seal, $B$ is the balance ratio and $K = 0.5$ and $P$ is the fluid pressure.

The balance ratio $B$ depends on the sealing arrangement and how the seal is pressurized and if $B$ is greater than one (1) the seal is term unbalanced. If the average pressure load on the seal was greater than the seal pressure it is unbalance while when the average pressure load is less than the seal pressure it is balance.

$$B = B_i = \frac{P_f}{p} = \frac{r_h^2 - r_i^2}{r_o^2 - r_i^2} \quad \text{for inside pressurize}$$

(7)

$$B = B_o = \frac{P_f}{p} = \frac{r_h^2 - r_i^2}{r_o^2 - r_i^2} \quad \text{for outside pressurize}$$

(8)

It is important to determine the coefficient of friction for different material combination for a mechanical seals. An investigation was done using different
materials for the mating and primary ring (Paxton, 1990). The results showed that material properties and the closing pressure (contact pressure) influence the value obtained. Though seals operate with a gap between the stator and rotor which allows the fluid to lubricate the interface, the gap distance was important in determining the performance of the seal. If the gap is too large then the friction coefficient is low but the leakage rate increases. The interface surface roughness and the surface finish for both rings influence the coefficient of friction during operation and its subsequent wear rate. The micro asperities at the rubbing surface (contact surface) determine, the frictional heat generated and the type of surface contact.

Many mechanical seals operate in the mixed friction where the load support is made up of both fluid and contact pressure. Lift off occurs when then the surfaces (rotating, mating) are not touching and the fluid support the load on the seal. However for mechanical seal surfaces lift off is not achieved. Contact pressure plays a major role in determining a seal wear rate and affects the friction coefficient that exists between the contact surfaces. The mechanical seal must be in static equilibrium axially and the applied load to it equal to \( W_m \) plus \( W_f \). \( W_m \) is the load supported by the contact surfaces and \( W_f \) the load supported by the fluid.

\[
W_f = \int_A p dA \quad \tag{9}
\]

\[
W_m = \int_A p_m dA \quad \tag{10}
\]

\( p_m \) is the contact pressure and \( p \) is the pressure of the fluid in the seal chamber.
2.4 Lubrication Impact on Heat Transfer from Mechanical Seals

Clark, (1990) used Computational Fluid Dynamics (Fluent) using commercial software Fluent to perform virtual prototype tests and laboratory testing in order to evaluate the effectiveness of coolants used in dual mechanical seals. Their research showed that axial circulation of fluid to and from warmer region of the seal promotes heat removal. They also showed that the highest flow fields are associated with the area around the rotating ring which had the highest turbulence as there are radical geometric changes in radial cross section between mating and primary ring. These simulations also showed that to increase the net heat removed by the coolant one can increase the radial gap as this causes the fluid to circulate more effectively in the axial direction. It is important to have an adequate flow rate of the coolant entering the seal chamber. If the flow rate is too low, then the coolant will vaporize at the contact surface (Buck, 1999). This would result in a higher coefficient of friction, which would increase the frictional heat develop. The amount of coolant necessary for optimum seal performance depends on the seal size, its speed of rotation, the discharge pressure and the materials that make the rings. Computational simulations using Fluent showed that increasing the flow rate beyond the optimum point does not improve the heat transfer characteristics of the mechanical seal (Luan and Khonsari 2006). Lubrication of the contact surface influences the interface shape of the rings in contact (Lebeck, 1991). If the interface shape is non-uniform due to insufficient lubrication the resulting hot spots would accelerate seal failure.
2.5 Wear

Deviation from form such as waviness and radial taper are cause by thermo elastic instabilities. These thermal stresses caused by the temperature developed at the interface due to rubbing contact would accelerate the deformity of the ring geometry. This deformity ultimately could lead to failure of the mechanical seal. Wear occurs fastest in areas where contact occurs, so if the mechanical contact pressure is known, the wear rate can be predicted, though this would only be an estimated value. For seals different type of wear can occur, these include adhesive wear, abrasive wear and chemical wear. Adhesive wear normally occurs in seals and it is the process where small particles of material are removed from a material by having made contact or being sheared. As long as materials touch at the asperity tips there will be adhesive wear. The only way to reduce or to eliminate this is to make sure that the surfaces are separated by a fluid. The time rate of wear

$$W = \frac{KP_mV}{H_d}$$

(11)

$K$ the wear coefficient of the material and $H_d$ its hardness, $V$ is the sliding velocity and $P_m$ is the contact pressure.

Three properties that are important to seal performance are coefficient of friction, wear rate and the $PV$ limit. The friction coefficient is specific to the system consisting of the two face materials, their precise geometry, the fluid, the speed, the pressure and environmental factors such as contamination. Friction values for seals are specific to the condition under which the seal was tested.
However the friction coefficient is based on the net amount of load that is not being supported by the average fluid drop across the seal. Wear rates for seals are dependent on the $PV$ level. Beyond certain $PV$ level the wear rate would dramatically increase causing accelerated seal failure. Wear data may not repeat for different seal test as the lubrication for the tests may differ and seal wear is path dependent. Some of the factors that shorten seal life are discontinuous operation, vibration, corrosive and abrasive fluids. However as the coolant between the surfaces vapor concentration increases, the coefficient of friction increases. Buck (1999) concluded in a study that seals have longer life based on the fraction of their face for which a liquid is present in the sealing interface.
CHAPTER THREE: DESIGN CONCEPTS

The aim of this research was to design a mating ring that would provide improved thermal performance over an existing conventional ring design. An evaluation of the different factors that influences a mechanical seal performance was done and designs were made based on these analyses. Some of the factors necessary are as follows (Buck, 1989)

a) The amount of surface area exposed to the coolant

b) The width of the mating ring that forms the face area, as the larger the ring the thermal resistance to heat conduction

c) Effective fins to improve the heat transfer

d) Consideration was given to the optimum face area so that the primary ring do not overlap during operation

e) Slots and grooves on the mating ring that takes advantage of the heat flow in the axial and radial directions to enhance heat transfer.

All the designs were optimize using Finite element software ANSYS coupled with analytical solutions. If the heat generated at the interface is controlled, then the life span of the mechanical seals can be extended. However, material properties influence the performance of a mechanical seal as it set the coefficient of friction, which has a direct impact on the wear rate. If thermo-elastic instabilities are minimized then the seal life is prolonged. If seals have a more uniform temperature profile, waviness and irregular wear profile are reduced. Also, if seals operate at a low surface temperature within the operating environment, its life expectancy would be longer.
Experimental tests conducted by Doanne (1990) have shown that heat transfer for a mechanical seal takes place mostly in the axial and radial direction. The heat generated at the seal interface is dissipated by convection to the seal chamber coolant flow by the rotor and the stator. The largest magnitude of heat flux occurs on the rotor surface near the interface between the rotor and the stator. The thermo-elastic instabilities that build up in the mechanical seal during operation are caused by thermal stresses. Therefore, if the thermal stresses are reduced, then seal life could be prolonged. Thermo-elastic instabilities occur due to poor liquid lubrication, high speeds, high loads and if the seal material is prone to heat check. These instabilities are hot spots on some regions of the interface that developed a much higher temperature than the average causing some type of thermal damage. Hot spots expand relatively more than the adjacent material, thus causing a higher pressure to act on it which results in more friction heating (Lebeck, 1991). All the heat leaving the seal is through convection with the coolant surrounding it in the stuffing box and conduction through the gland. Coolant fluid impacts on the seal in two ways: one as a result of the process fluid and the other through a flush port through the gland.

For a seal design to be successful the seal material should have the following properties (Buck, 2001)

1) They must be wear resistant

2) They should have a low coefficient of thermal expansion

3) They should have high overall strength
4) They should have good thermal properties, such as high thermal conductivity, to remove heat generated from the rubbing surfaces

5) They should have good resistance to corrosion from both inside and outside environment and

6) They should be easy to manufacture and have low cost

The material used for the primary and mating ring in seal assembly is usually different so that the resulting friction and wear is minimized. Therefore the selection of material pairs should be made with the following consideration (Buck, 2001).

a) A hard face and a soft face are often used. The hardness difference is usually 20%

b) Low friction coefficient between rotating material and stationary material is needed to decrease the heat generation at the interface and thus reduce thermal expansion and

c) The two materials should have modulus of elasticity difference so that the stiffer material will be able to run into the softer one to make good sealing.

One of the most important aspects of seal life is how the rings are cooled. Any design that improves the cooling characteristics of the primary and mating ring prolong the seal life. Mechanical seals cooling system can be a closed loop for the modified gland and an open loop for the conventional gland. The coolant which is called the flush is an external flush if it is taken from a source which is not the process fluid. However, If the flush is taken from the process fluid it is called an internal flush. Whenever the flush flow is passed over the leakage side
of the seal it is called quenching. Quenching provide cooling by supplying a fluid of known temperature around the leakage side of the seal rings and it washes away the any foreign particles that may exist. The mating ring for this design will be cooled by using an internal flush. This ring was design to be used in a conventional gland.

### 3.1 Fin Mating Ring and Design Optimization

![Figure 18: Drawing of Fin Mating Ring](image)

The fin mating ring was made of 17-4-PH and its dimensions are as depicted in fig 18. 17-4-PH is a precipitation hardening finish steel making the
properties throughout the material more homogenous. The mating ring dimensions and holes were done before it was heat treated. Since hardness is an important characteristic in reducing the wear rate, the ring was heat treated so that it had a Rockwell C hardness of 45. Afterwards, its face was lapped to a surface finish between 1-2 helium light bands. One helium light band measures approximately 0.00012 inch (.000304 m). The fin mating ring can be use in the conventional seal without the need to modify the gland. This is an open loop system in which the coolant (flush) flow over the mating ring, the seal interface, over the primary ring and through the stuffing box into the process fluid to be discharged. However, for every mating ring made, its shape can improve its heat transfer characteristic. The larger the diameter of ring the higher the convective heat transfer area. Also this increases the conductive heat transfer resistance and causes a net reduction in the heat transfer efficiency (Buck, 1989). The vast majority of heat transfer takes place within a distance approximately two face width from the contact interface in the radial and axial direction and the ring with the greater thermal conductivity transfers the majority of the heat. The face area is the area for both rings that are in contact and heat generation takes place over this area.

From the three dimensional drawing of the mating ring shown in figure 19, it is obvious that this design takes advantage of the facts stated above. The face area is kept to a minimum in order to reduce thermal resistant to conduction and slots are cut in areas to maximize the amount of heat transfer possible. The slots are important in improving the heat transfer characteristics of the mating ring.
The greater the surface area close to the interface that facilitates interaction with the coolant the more efficient the heat transfer rate.

**Figure 19:** Solid depiction of fin mating ring
Figure 20: FEA ANSYS simulation for conventional mechanical seal

Figure 21: FEA ANSYS simulation for fin ring mechanical seal
Before the mating ring was made Finite Element Analyses using ANSYS were performed to optimize the design. Simulations with regards to the temperature profile using the same boundary conditions were done and the results are shown in figures 20 and 21. Figure 20 represent the temperature profile for an existing design for a conventional ring while figure 21 represent the temperature profile for the Fin ring. The ANSYS simulations showed that the fin ring will have a surface temperature approximately eight degrees Celsius cooler that the existing conventional ring.

**Figure 22:** Fluent simulation for a conventional ring temperature profile (Luan and Khonsari, 2006)

Figure 22 shows the axisymmetric temperature contours for the heat distributed through the stator and the rotor using the Fluent C.F.D. package. It is obvious that the heat is conducted in the axial and radial direction of the two rings.
Figure 23: Comparison of temperature magnitudes of computational result and experimental (Luan and Khonsari, 2006)

Figure 23 shows how the measured experimental heat values taken at three different radial location match up against the simulated results. The measured values are depicted in squares while the simulated results curve shown. Doanne (1991) found that the axial temperature gradient was at least 28 times larger than the circumferential temperature gradient. If the coolant was delivered, as close as possible to the surface interface without compromising the structural integrity, then the amount of heat transfer that takes place should increase. The surface area that is exposed to the coolant is very important as the heat transfer is a function of area. The more area exposed to the coolant the greater the heat transferred.

The heat transfer rate could be improved by increasing the surface area. This was accomplished by making fins on the mating ring. However for a fin to be
useful it has to have effectiveness greater than two. To evaluate the effectiveness, an infinite fin approximation was used with \( E_f = \left( \frac{kp_A}{hA_c} \right)^{0.5} \). \( k \) is the conduction coefficient, \( p_A \) is the perimeter of the fin, \( h \) is the convection coefficient and \( A_c \) is the cross sectional area (Incropera, 2002). However, this equation does not represent a true approximation of the fins made for the fin mating ring. It provides a close approximation of the fins effectiveness. Fins are more effective in an environment when the convection coefficient is small. However, the smaller the cross sectional area of the fins, the more effective would be the fin designs. Having fins on the mating ring would improve its heat transfer capability therefore reducing the heat at the interface and prolong the seal life. Using the equations (Luan and Khonsari 2006)

\[
Nu_{stator} = 121.51 Pr^{0.89} \left( \frac{Re_{flush}}{Re_{rov}} \right)^{0.56}
\]

for the Nusselt number and

\[
Hc = \frac{NuK_f}{D}
\]

for the convection coefficient, the approximate convection coefficient was calculated. Calculations were done to optimize the cross sectional area to give largest fin effectiveness possible with the manufacturing processes acting as a constraint. With the required dimensions for the fin calculated, the other concepts drawn from the theory was incorporated. The fin mating ring would allow the coolant to go as close as possible to the contact interface and take advantage of the axial and radial heat transfer characteristic. The mating ring would be able to be used in a conventional mating ring setup with no additional O-rings or parts. Therefore the flush (coolant) should leave the sealing chamber by mixing with the process fluid, pass the pump wear rings, into the pump scroll to the pump discharge.
For this design, emphasis had to be placed on the diameter on the mating ring where the fins will start. It was important that enough face area exist so that the primary ring does not overlap during operation. This was critical especially at start up because the axial movement of the motor causes the primary ring to be slightly misaligned until it aligns itself. With this in mind, the diameter at which the fins start was larger than the diameter of the primary ring.

3.2 Conventional Gland

A conventional gland has an open loop cooling system. The coolant (flush) enters the gland as shown in figure 25 by the black arrows. It flows into the sealing chamber and make contact with both the primary and mating rings. Heat is transferred from both of these rings by conduction through the rings to its surface and by convection to the coolant. The coolant then leaves the sealing chamber through the wear rings into the pump scroll. This coolant then mixes with the process fluid and is discharged. O-rings are inserted on the pathways
between the mating ring and the gland as well as the primary ring and the shaft to prevent leakage to the surroundings. The pressure of the coolant entering the gland is important as it provides a hydrodynamic effect to the seal faces in contact adjusting to the spring pressure. It also acts as a lubricant at the contact face to reduce the friction. If the pressure is too low; it will cause the coolant (lubricant) to vaporize. This would reduce the lubrication effectiveness of the device and cause the wear rate to increase thus reducing the life of the seal. If the coolant pressure is too low, for example if it is below the sealing chamber pressure, then the coolant would not flow and the heat transfer will not take place. The gland is a housing for the mating ring and in conjunction with the stuffing box makes the sealing chamber complete.

3.3 Testing Procedure

To analyze the effectiveness of new seal designs, it is important to compare their performance with existing designs. To do this, mechanical seals
are tested in accordance with API Standard 682. The test standards are as follows
1) Dynamic phase:
Tests are done at constant temperature and pressure at 3600 rpm. The test has to be done for over 100 hours and at base point conditions.
(2) Static phase:
Tests are done at constant pressure and temperature at zero rpm for 4 hours.
(3) Cyclic phase:
i) Tests are done at base point conditions at 3600 RPM until equilibrium established
ii) Drop seal chamber pressure to 0 psi to vaporize all fluid. Re-establish base point pressure
iii) Drop the seal chamber fluid temperature to 20°C/70°F. Then re-establish base temperature
iv) Raise seal chamber fluid temperature to 80°C/180°F.
v) Turn of seal flush for 1 minute.
vi) Shut down test (0 RPM) for 10 minutes
Before the mating rings are tested they are lapped to a surface finish between one and two helium light band. A helium light band is 0.0000116 inch or 0.29 micron. This is necessary as surface finish has an impact on a seal performance. The larger the asperities that are in contact the more the heat generated by rubbing. The mating ring was bored with 0.08 inch (0.00203 m) diameter holes to a depth 0.002 inch (0.000051 m) from the contact interface. These holes are at
different radial distances. J-thermocouples are placed in these holes and are sealed in these holes using copper oxide cement and left for twenty-four hours so that the cement can dry properly. Afterward the mating (static) ring is placed in the gland and the thermocouple wires are run through gland in 0.1 inch (0.00254 m) diameter holes.

**Figure 26:** Face area dimensions between diameters 2.30” - 2.625” (5.842 cm – 6.668 cm)

The thermocouple holes in the gland are then sealed using epoxy and given twenty-four hours to cure. The thermocouples used are ¼” (0.00635 m) J-type and they have a degree of accuracy of ± 0.75%. The gland containing the mating ring was then mounted on the shaft and the primary ring was placed in
front of it on the shaft. The simulated pump housing/ stuffing box is then mounted on the shaft and the sections are fitted together until it seal without leaking. The completed seal chamber is then tested for leakage by turning on the circulatory pump, which causes the coolant to flow through the seal chamber at a specific flow rate and pressure. If there are no leakages the thermocouple wires extending from the back of the gland are connected to Webdaq thermocouple reader. This thermocouple reader is connected to the internet system and its reading can be accessed by punching in the correct IP address.

The mechanical seal was tested according to API Standard 682. First the static test is done where the sealing chamber was pressure up to 20 psi (137900 pa), at room temperature and left for four hours. Afterward checks are done to see if there are any leaks. The mechanical seal is test put through a cyclic test with base pressure at 20 psi (137900 pa). The temperature of the seal chamber was lower to approximately 20 °C by using ice water. It was left for a period of about 30 minutes for it to attain base point temperature of 36 °C. The flush was then turned off for one minute and then the shut down test was done. However the cyclic test was not done according to API Standard 682 because the mechanism to get the hot water to the test rig was not installed. The final test is the dynamic test for this test the mechanical seal sealing chamber was pressurized up to 10 psi (68950 pa) and a rotational speed of 3600 rpm. The coolant (flush) through the seal chamber has a flow rate of 10 gpm (.000633 m³/s). This test was run for over 100 hours. The temperature data from the transient to the steady state period was recorded for analysis.
3.4 Operational Features of the Test Rig

A test rig was design so that it could test mechanical seals in accordance with the API Standard 682. API Standard 682 has three criteria for testing mechanical seals.

(1) Dynamic phase:
Tests are done at constant temperature and pressure at 3600 rpm. The test has to be done for over 100 hours and at base point conditions.

(2) Static phase:
Tests are done at constant pressure and temperature at zero rpm for 4 hours.

(3) Cyclic phase:
i) Tests are done at base point conditions at 3600 RPM until equilibrium established

ii) Drop seal chamber pressure to 0 psi to vaporize all fluid. Re-establish base point pressure

iii) Drop the seal chamber fluid temperature to 20°C/70°F. Then re-establish base temperature

iv) Raise seal chamber fluid temperature to 80°C/180°F.

v) Turn of seal flush for 1 minute.

vi) Shut down test (0 RPM) for 10 minutes

The test rig used is self contained and rest on a stand. It can be moved from one location to another wherever a test needs to be done. The data collection system is housed on a separate cart and it can be moved with the test rig.
This test rig has a safety feature that would shut it down automatically in case there is leakage of the coolant to the surroundings beyond a certain limit. The Webdaq data gathering system of the test rig is adequate as it can collect uninterrupted data for up to three million data points. The coolant, used to cool the seal during testing was fed through a closed loop system in which the coolant is being cooled and recycled to do work. The design of the test rig is simple; the reservoir (seal pot) provides the water supply for the simulated pumping housing and the coolant for the mechanical seal. This reservoir is also the heat exchanger for the system. Inside the reservoir are copper coils through which water flows in a counter flow direction. This system provides a medium for heat exchange. A small pump is affixed to the system to provide head that fluid flow at a certain pressure can be maintained throughout the system. The function of the variable speed drive is to change the speed at which a seal is being tested. The reservoir
Table 2

Legend for the Mechanical Seal Test Rig

<table>
<thead>
<tr>
<th></th>
<th>Description</th>
<th></th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Reservoir with heat exchanger</td>
<td>9</td>
<td>Motor</td>
</tr>
<tr>
<td>2</td>
<td>Pressure relief valve</td>
<td>10</td>
<td>Variable speed drive</td>
</tr>
<tr>
<td>3</td>
<td>Level switch</td>
<td>11</td>
<td>Webdaq and computer</td>
</tr>
<tr>
<td>4</td>
<td>Circulating pump</td>
<td>12</td>
<td>Shaft</td>
</tr>
<tr>
<td>5</td>
<td>Flow meters</td>
<td>A</td>
<td>Thermocouples</td>
</tr>
<tr>
<td>6</td>
<td>Filter</td>
<td>B</td>
<td>Air intake valves</td>
</tr>
<tr>
<td>7</td>
<td>Simulated pump housing</td>
<td>C</td>
<td>Heat exchanger coolant intake</td>
</tr>
<tr>
<td>8</td>
<td>Gland with mechanical seal</td>
<td>D</td>
<td>Heat exchanger coolant discharge</td>
</tr>
<tr>
<td>O</td>
<td>Valves</td>
<td>Φ</td>
<td>Pressure gages</td>
</tr>
</tbody>
</table>

was designed so that it can be pressurized from an external source. This was important as it allows one to increase the pressure delivered to the simulated pump housing. The valves, flow meters and pressure gages are used to adjust or change the operational point to what is desired. The level switch is a major safety aspect of this test rig. In the event that a leak is sprung during operation, the level switch is calibrated so that whenever the water level in the reservoir falls below the centerline of this device. It causes the entire rig to shut down by shutting off the power supply.

Due to the simulated pump housing used, the rig can test seals from 44.5 to 76.2mm internal diameter. The assembly of the stuffing box and the gland is done in the common way, where the primary ring position is marked in relation to
the gland and the static ring. Adjustments are made to how much spring force between the faces during operation is needed. Usually the primary spring is moved 4.76 to 6.35mm toward the power train from where it sits in relation to the gland and the stuffing box. The data collection systems are the J-thermocouples attached to the static ring behind the contact face at a distance to ensure the structural integrity of the ring. The distance from the contact face varies for different thermocouple, depending on the radial, circumferential and axial temperatures that are being measured. These thermocouples are fed into a Webdaq data collection system. The Webdaq can be connected to the internet and the data can be accessed to retrieve the information remotely via an IP address.

Flow meters and pressure gages are used to read and record the data for later analyses. Temperatures for the coolant at inlet and discharge were measured along with the temperature of the static ring at (different) two locations. It is important that the surface temperature for the static ring is measured in more than one place in the circumferential direction so that one can determine if the temperatures throughout the surface are uniform.
CHAPTER 4: RESULTS, ANALYSIS AND DISCUSSION

4.1 Computational Simulations

Figure 28: Coolant flow direction in conventional gland

Figure 28 shows how the flow direction of the coolant (flush) in the conventional gland. However for this flow to take place the pressure of the coolant entering the sealing chamber has to be above the pressure within the sealing chamber. The gage pressure of the coolant was measured prior to entry into the sealing chamber to ensure that the coolant flow was maintained.

This flow pattern applies to both the new and the conventional designs as they use the conventional gland. To compare and evaluate the performance of the new design against the existing conventional design, experimental tests and Finite Element Method analysis using ANSYS were done.
A) ANSYS

Finite element analysis was used to determine how the frictional heat generated affects the mating rings. The simulations were performed with ANSYS 8.1 finite element software. The element chosen to simulate the mating rings was Solid 65. Solid 65 had the ability to include non-linear material models. Due to symmetry a three dimensional model representing 1/16 (one sixteenth) of the cylindrical model of the mating ring was made for analysis.

Meshing

The mesh generation for the three dimensional models for both rings were done using ANSYS preprocessor. Mesh refinement is critical to the accuracy of the results gained from simulation, and with this in mind different mesh sizes were simulated. A mesh sensitivity analysis was done by refining the mesh size until the answers for the simulations remain the same. The results were compared for deviations as this was necessary to evaluate if the simulations were converging.

The simulations were run with the mesh size that gave the best and most consistent result

The boundary condition for the ANSYS simulations were as follows

a) The mesh drawing was 1/16 (22.5 degrees) of the entire primary and mating ring assembly

b) Symmetry condition were used at the inner boundaries where the rings were cut, therefore the boundary condition for these sections were insulated
c) The mesh drawing was 1/16 (22.5 degrees) of the entire primary and mating ring assembly

d) Symmetry condition were used at the inner boundaries where the rings were cut, therefore the boundary condition for these sections were insulated

e) At the inner diameter for the primary ring and the mating ring insulation boundary conditions were used

f) For the outer diameter of both rings their respective heat transfer convection coefficient was calculated using equations 1, 2 and 3.

g) At the outer diameter the heat transfer convection coefficient was used as the boundary condition

h) The heat generation value obtain from using equation 4 was used for the boundary condition at the contact interface
Figure 30: ANSYS plot of the fin ring with the test boundary conditions

Figure 31: ANSYS plot of conventional ring with the test boundary conditions
**Figure 32:** ANSYS plot of fin ring with the test boundary conditions

**Figure 33:** ANSYS plot of conventional ring with the test boundary conditions
Solution

The ANSYS (F.E.A.) plots shown in figures 30 to 33 showed different positions of the temperature profiles for the fin and conventional mating ring. These ANSYS simulations were done using the tests boundary conditions. The simulations showed the surface temperatures for the fin mating ring ranging between 45 to 46.6 °C and 56 to 59.6 °C for the conventional ring. The surface temperature for the conventional ring was higher than the fin ring.

4.2 Test Results

Figure 34: Pump in an Industrial plant (courtesy of senior design team 2001-02)

The mechanical seal using the different mating rings were tested with the specifications shown in the table below.
**Table 3: Equipment specification and test parameters for dynamic, cyclic and static tests**

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Circulating pump</strong></td>
<td>Model AMT 551011, maximum flow rate 2.85 gpm (0.000180 m$^3$/s) maximum head 40 psi (275800 pa), power rating 0.5 hp (0.373 Kw).</td>
</tr>
<tr>
<td><strong>Variable Speed Drive</strong></td>
<td>Yaskawa, 3-phase AC variable speed drive. Volts 230-460, maximum 2 hp (1.492 Kw), maximum speed 3600 rpm</td>
</tr>
<tr>
<td><strong>Reservoir</strong></td>
<td>Volume, 5.28 (0.020 m$^3$) gallons of water with the cooling coils.</td>
</tr>
<tr>
<td><strong>Valves and Fittings</strong></td>
<td>All rated above 18,000 psi (124.11 Mpa)</td>
</tr>
<tr>
<td><strong>Piping</strong></td>
<td>½ inch (0.0126 m) diameter stainless steel tubing rated @ 18000 psi (124.11 Mpa) and ¾ inch (0.0190 m) pvc sch 40 pipe rated @ 480 psi (3.31 Mpa) @ 73 degrees Fahrenheit (22.9 °C)</td>
</tr>
<tr>
<td><strong>Gland Specification</strong></td>
<td>Seal chamber pressure = 10 psi (68950 pa)</td>
</tr>
<tr>
<td></td>
<td>Spring load = 42.4 lbs (19.272 Kg)</td>
</tr>
<tr>
<td></td>
<td>Rotational speed = 3600 rpm</td>
</tr>
<tr>
<td></td>
<td>Coolant flow rate, water @ 1 gpm (0.0000633 m$^3$/s)</td>
</tr>
<tr>
<td></td>
<td>Heat generated at interface at steady state,</td>
</tr>
<tr>
<td></td>
<td>Conventional mating ring</td>
</tr>
<tr>
<td></td>
<td>Fin mating ring</td>
</tr>
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</table>
Static Test

Table 4: Static test parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coolant Flow Rate</td>
<td>0 gpm (0 m³/s)</td>
</tr>
<tr>
<td>Seal Chamber Pressure</td>
<td>20 psi (137900 pa)</td>
</tr>
<tr>
<td>Rotational Speed</td>
<td>0 rpm</td>
</tr>
<tr>
<td>Temperature</td>
<td>Ambient</td>
</tr>
<tr>
<td>Test Time</td>
<td>4 Hours</td>
</tr>
<tr>
<td>Leaks Recorded</td>
<td>0</td>
</tr>
</tbody>
</table>

These parameters were used to conduct the static test for both conventional and fin mating ring seal arrangement. The test rig was inspected periodically and after four hours there was no sign of leakage. This showed that both design were capable of sealing properly.

Cyclic Test Results

The cyclic test results were not done in accordance with API standard because no hot water was available to change the coolant temperature during the test from steady state temperature to 80 degrees Fahrenheit. However the test was run using the remaining test criteria and the results are plotted below.
Figure 35: Fin ring cyclic test

Figure 36: Fin ring surface temperatures
Figure 37: Conventional ring cyclic test

Figure 38: Conventional ring surface temperature
Dynamic Test results

The test results were sampled using this equation \( y^{0.5} = a + \frac{b}{x^{0.5}} \)

This was necessary as the data had a lot of noise associated with it. The data was first run through the “Table curve 2D V4” program to find an equation that sampled the data with the least degree of error. The equation chosen was the one that gave average values for the test results over the test period. For the equation used, \( a = 6.5219721, b = 0.87000762 \) for the fin ring. The conventional ring results were sampled using the same equation with \( a = 48.916404, b = 19.18969 \). The % error associated with sampling are 0.00312 and 0.00214 respectively.

The static tests on both rings showed that the mechanical seal was sealed properly as there were no leaks after the prescribed test period. Also the cyclic
test was successful. There was no leak for the test period.

**Figure 40:** Coolant in for conventional ring

Figure 40 represents the average temperature profile for the coolant in (flush) for the conventional ring test.

**Figure 41:** Coolant in for fin ring

Figure 41 represents the average temperature profile for the coolant in (flush) for the fin mating ring test.
Figure 42: Surface temperature at radius 1.2” conventional ring

Figure 42 shows the average temperature profile for the conventional ring at a radius of 1.20 inch.

Figure 43: Surface temperature at radius 1.2” fin ring

Figure 43 showed the average temperature profile for the fin ring at a radius of 1.20 inch.
Figure 44: Surface temperature at radius 1.25” conventional ring

Figure 44 represent the temperature profile for the conventional ring at a radius of 1.25 inch.

Figure 45: Surface temperature at radius 1.25” fin ring

Figure 45 represent the temperature profile for the fin ring at a radius 1.25 inch
**Figure 46:** Surface temperature at radius 1.305" conventional ring

Figure 46 shows the average temperature profile for the conventional ring at a radius of 1.305 inch.

**Figure 47:** Surface temperature at radius 1.305" fin ring

Figure 47 shows the average temperature profile for the fin ring at a radius 1.305 inch.
4.3 Calculation of the Coefficient of Friction

The value of the coefficient of friction is an important value as it was used to calculate the heat generated by rubbing at the contact interface. The equation used to calculate this dimensionless value was

\[ f = \frac{MC_p \Delta T}{P_m VA_f} \]

where \( M \) is the mass flow rate of the coolant (flush) \( C_p \) is the specific heat capacity, \( \Delta T \) is the average temperature change, \( P_m \) is the contact pressure, \( V \) is the rotational velocity and \( A_f \) is the face area. Comparing figures 48 and 49 and evaluating the average temperature difference for the coolant in and coolant out.

**Figure 48:** Coolant in temperature for the flush

The coefficient of friction for the fin mating ring was 0.0810 and the heat generated at the interface using equation 4 \( P_m VA_f f \) was 1056.65 Watts.

Comparing figures 40 and 50, the temperature difference between the coolants
Figure 49: Coolant out for the flush was greater than the value for the fin ring. For the conventional ring the coefficient of friction was 0.18 and the heat generated at the interface 2377.9 Watts. The friction value was higher as the temperature difference between coolant in and coolant out was greater.

Figure 50: Coolant out conventional ring
4.4 Analysis of the Results

The temperature gradient in the circumferential direction for the rings was expected to be much less that those for the radial an axial direction. For a mating ring, temperatures at the same radius for the entire circumference are expected to be close. The closer these temperatures are the better the performance of the ring. Analyzing figures 51 and 52 the surface temperatures at a radius of 1.305 (0.0331 m) inch was measured and the difference in the average surface temperatures was approximately 0.5 °C.

Figure 51: Surf temperature at radius 1.305”

Figure 52: Surf temperature at radius 1.305”
This proves that the temperature gradient in the circumferential direction is small.

Comparing figures 42 and 43, it was obvious that the surface temperature for the fin mating ring is approximately (five) 5 °C less than those for the conventional ring even though the average coolant in temperature is higher for the fin ring. For radius 1.25 inch which is represented by figures 44 and 45 the fin ring temperature was 5.5 °C less than the conventional ring and the same trend was obvious when figures 46 and 47 were compared. The coolant used to cool the mechanical seal was water and its inlet temperature depended on the ambient temperature. The average temperature for the flush coolant temperature for the fin ring test was 38 °C while it was 34 °C for the conventional ring. The average maximum temperature measured by the fin ring was 44.5 °C with a temperature difference between this value and the flush inlet temperature of 6.5 °C. However, for the conventional ring the temperature difference between the average maximum temperature and flush temperature was 15 °C.

Temperature measurements were taken by the thermocouples for circumferential and radial measurement. Figures 51 and 52 represent temperature measurements in the circumferential direction while figures 42 to 47 show measurements taken in the radial direction. The temperature difference in the circumferential direction was 0.5 °C at an angle of 90 degrees. For the radial direction with reference to the fin ring data, the temperature difference was 2.5 °C and 2 °C for the conventional ring. The radial distance was 0.1 inch (0.00254 m) and the circumferential distance was 1.33 inch (0.03378 m).
The face area at the contact interface for both the fin mating ring and the conventional mating ring are the same, along with the material properties, sliding speed, coolant flow rate and test parameters. However, the coefficient of friction for the conventional seal is higher than that for the fin seal. This is a direct result of temperature effect due to the interface lubrication. As the surface temperature get higher the lubricant at the contact interface in localize hot spots flash into vapor. This causes the lubricant to lose its effectiveness. The resulting hot spots expand into high spots. The load is supported by both the fluid and the contact surfaces as explained earlier by equation 9 and 10. As higher temperature causes more of the lubricant to flash into vapor the asperities at the contact interface support more loads. This however, increases the coefficient of friction for the mechanical seal and accelerates the wear rate.

4.5 Comparison and Discussion

The results for the cyclic tests for both rings showed the temperature for the fin ring to be lower. Comparing the data associated with the conventional ring and the fin ring. The conventional ring data tended to be noisier even though the operating conditions for both seals were identical.

For the Fin mating ring the temperature at the interface surface was solved analytical using 1-D conduction.

\[ Q = \frac{-kA\Delta T}{\Delta x} \]

Where \( Q \) is the heat generated, \( k \) is the thermal conductivity, \( \Delta T \) is the temperature change and \( \Delta x \) is the change in the x-direction. The average
surface temperature at the interface for the test was 49 °C. For the conventional mating ring the average test interface surface temperature was 54 °C. These values when compared with those from the ANSYS plots in figures 34 and 35, showed good correlation as the fin ring surface temperature is 46.6 °C and the conventional ring surface temperature 59.6 °C. Hence the results for the simulations surface temperatures are in good agreement with the test results. Therefore the model simulations and the tests validate each other.

The surface temperatures measured for the fin mating ring were 5 °C less than the conventional mating ring surface temperatures even though the coolant temperature was four degrees higher for the fin ring. The fin ring had lower surface temperatures even when the coolant (flush) was at a higher temperature. This showed that this ring had a better heat transfer characteristic than the conventional ring. The lower the surface temperatures for a mating ring the better the thermal management and the lower the chance for thermo-elastic instabilities that affect the contact surface.

A increase in the surface temperature can result in an increase in the number of hot spots on the contact surface. As the surface temperature increases the coolant temperature which lubricates the contact surface temperature rises. The coolant at high temperature flash into vapor and its lubrication effectiveness may be reduced.

Seals undergo mostly adhesive wear and if their surface temperature is kept as low as possible then this would facilitate the existence of the transfer film. This happens as the softer face material adheres to the harder face material after
being in contact over a period of time, thus forming a film on its surface. This transfer film reduces the coefficient of friction for the mechanical seal. However, in areas where hot spottings are likely to occurs, this film will break down. Therefore it is imperative that mechanical seal face temperatures are kept as low as possible as this will facilitate proper interface lubrication, reduce thermo-elastic instabilities, reduce the coefficient of friction and reduce the wear rate.

The wear rate for a mechanical seal can be predicted using equation 11. Equation 11 is base on the contact pressure, rotational speed and the material properties. Another factor is the interface lubrication. If hot spots form on the surface, then surface wear become non-uniform and the seal will leak. The wear coefficient $K$ remains a constant as long as the mechanism of lubrication remains constant (Peterson and Winer 1980). Since the mechanism of lubrication in mechanical seals is sensitive to temperature then the value of $K$ would vary gradually and systematically. $K = \frac{VH}{WL}$ with $V$ being the wear volume, $H$ is the material hardness, $L$ is the sliding distance and $W$ is the imposed load.

The approximate wear rate found analytically was 7.33e-6m/s. However, since the material properties are assumed to be constant the temperature recorded does not appear directly in the equation. Hence the same wear rate is recorded for both rings. However, it is expected that the fin mating ring would have less wear and a longer life due to lower temperature at the interface.
CHAPTER 5: CONCLUSION

Thermo-elastic instabilities influence seal wear and its performance. The lower the surface temperature at the contact interface, the lower the effect of thermo-elastic instabilities. One of the main reasons for waviness and distortion of the rings shape is the temperature effect. The higher the interface temperature of a mechanical seal the more likely that a non-uniform shape would develop at the interface. Distortion in the flatness of the rings would cause hot spots to develop at areas where they are touching off. These hot spots lead to high spots which basically supports the majority of the contact force. The end result is that at these areas more wear will take place that in the adjoining areas and leads to a non-uniform surface. Whenever there was deviation from the machine flat surface on the rings, it causes an increase in the leakage from the mechanical seal. The hot spots temperature rose as they expand more. This causes the coolant between the contact interface in that region to flash into vapor. Whenever the coolant flashes into vapor due to high surface temperature, it reduces the lubrication effectiveness of the coolant. Therefore the coefficient of friction would increase and causes a subsequent rise in the wear rate.

The experimental tests for both the fin mating ring and the conventional ring were carried out according to the API STANDARD 682. The results from the ANSYS simulations and experimental tests for the surface temperatures for both rings were closed. This showed that the approximations for the boundary conditions chosen for the ANSYS simulations for the fin mating ring and the conventional mating ring mechanical seals along with the estimated heat
generated at the interface were good. The simulations and the tests showed that the surface temperature for the fin mating ring was lower than the conventional ring.

A new design utilizing a fin mating ring is introduced in this thesis. The design was built and tested in the laboratory. The results of a series of experiments revealed that the fin mating ring had a lower contact surface temperature than the conventional ring.
CHAPTER 6: FUTURE RESEARCH

Experimental testing and measuring the depth of the rings before and after the test in order to calculate the wear volume present a more accurate assessment of the wear rate of a mechanical seal. This way, the temperature effect on wear can be accounted for in the wear rate. Therefore future research could be done on the conventional mating ring and the fin mating ring with the thickness of the ring being measured before and after the tests. The data gathered from these test should be used to set up a numerical equation which gives a more accurate prediction of the wear rate than the existing one.
REFERENCES


PPc, Mechanical Seal Handbook, Power Packing Company, Baton Rouge LA.


Somanchi A.K., A Novel Mechanical Seal Design with Superior Thermal Characteristics, MS Thesis, Department of Mechanical Engineering, Louisiana State University, 2004
APPENDIX A OPERATING PROCEDURE FOR MECHANICAL SEAL TEST RIG

Picture of the Test Rig

The mobile mechanical seal test rig in Ceba 1201 can be used to test

1) Conventional mechanical seals

2) Heat exchanger rings

3) Mechanical seals of different sizes

4) Facilitate quick changeover of different seals for testing

5) Test operation for over 100 hours
These operations can be facilitated if the test rig is used properly.

The following is a list of instructions and procedures for using the test rig.

**Starting Up**

1) Check to see if water is in the reservoir by checking the sight glass attached to it. Make sure the water level indicated by the sight glass is above the centerline for the level switch.

2) Check the oil level in the bearing housing. This is done by inspecting the sight glass (oil level glass) attached to the housing. Make sure that some oil is seen within the sight glass itself, as this would indicate the level in the housing. Failure to do so might result in bearing failure.

3) Check the valve below the level switch. Make sure that it is adjusted properly (i.e. 5-10% open) so that water can flow through it and at the same time reduce the suction on the switch.

4) Check the mechanical seal to make sure it is seated properly as improper seating can cause leakage and accelerate the failure of the seal.

5) Make sure the gages and instrumentation are working properly.

6) Adjust the valves on the coolant inlet, recycle and coolant discharge lines to partially open so that flow through both stuffing box and gland can be initiated as soon as circulating pump is turn on.

7) Make sure the pump suction line valve is in the off position before turning on the pump.
8) Check the air tank to see if air is inside of it. Adjust air to the system as the test desire by using the air pressure valve control.

9) Make sure there is power to the computer and thermocouple reader before you start running a test.

10) Peruse the variable speed drive to make sure it in the off position.

11) Finally, connect the power cable for the pump and the variable speed drive to a power source. Make sure the pump run for approximately two minutes before starting the variable speed drive.

**Operational Procedures**

During operation there are certain key aspects that you should pay attention to.

1) Make sure the flow rate of the coolant is the required flow rate for the test being conducted. (By manipulating the valves and reading the flow meter you can adjust the flow)

2) Adjust the pressure from the air tank to the system to the desired pressure. (i.e. system employs a pressure pad so that you can increase or decrease the pressure depending on the test parameters)

3) Set the speed of rotation to the desired range by using the read out counter for the variable speed drive.

4) Check and make sure that all the thermocouples and the leads to the thermocouple reader are attached properly.

5) Make sure the surge protector for the computer and the thermocouple reader is on.
6) Program the computer so that it can collect store and record data for analysis.

7) Check the air pressure gages and the flow meter to make sure you are getting the required flow and pressure rates.

8) If test is required for over four hours make sure that pipe water is piped thru the heat exchanger.

9) Check system during operation that there are no leaks as this would lead to coolant and pressure loss.

**Shutting Down Operation**

1) By using the read out counter turn of the variable speed drive and let the pump run for about two minutes after.

2) Remove power source to both pump and variable speed drive

3) Turn off the air pressure source.

4) Check your results on the computer.

5) If the heat exchanger is used, remember to turn of the water supply and remove the hoses.

6) Do a check on all the relevant equipment, making sure they are okay.
APPENDIX B TESTING PROCEDURE TO EVALUATE THE FRICTION COEFFICIENT OF A CONVENTIONAL RING

To calculate the friction coefficient for the conventional ring, some modification of the existing rig was done. The problem was that we could not measure the temperature of the coolant after work was done. To do this the stuffing box was modified and two thermocouples were placed in it. The position of these thermocouples was critical as it must no touch the rotating ring as the temperature it would measure would be false.

Fig 1 stuffing box with thermocouples exposed
Figure 2 and 3 show how the stuffing box was modified. Holes were threaded into the stuffing box 180 degrees apart. A threaded bolt that fit these holes was bored through the center. Afterward, thermocouples were place through these holes and epoxy was used to seal it so there are no leaks.

The stuffing box was assembled with the rest of the rig components. The thermocouples were attached to the webdaq thermocouple reader so that data can be gathered for analysis.
The test will be run for two hours so that the seal faces in question can reach steady state conditions. The coolant flow rate was 1gpm at 10 psi. The rotational speed was 3600rpm. Afterward analysis will be done to ascertain the friction coefficient of the seal.

Analysis

Fig 5 primary ring layout with nose dimensions (diameter)
Spring force = 43.4lbf (19.73 Kg)
Nose area = 0.468 square inches (0.00031 m$^2$)
Gage pressure of coolant between seal faces = 10 psi (68950 pa)
Net pressure on seal nose = (10 psi [1-0.5] + 43.4/0.468)
Net pressure = 97.6psi (672952 pa)

Since

\[ E_{in} + E_g - E_{out} = E_{st} \]

With \( E_{in} = \) energy in
\( E_g = \) energy generated
\( E_{out} = \) energy out
\( E_{st} = \) energy stored and energy generated = \( \rho C_p \frac{\partial T}{\partial t} \)

At steady state the energy input is equal to the energy generated

With the following assumptions

1) Flow through the stuffing box is the 1 gpm (0.0000633 m$^3$/s) flow of the coolant through the gland.
2) Greatest conduction is in the Z-direction
3) In the \( \Phi \) and \( r \) direction convection with the coolant dominates
4) At steady state energy in = energy out
5) Using cylindrical coordinates with \( \Phi = \) circumferential, \( r = \) radial and \( Z = \) axial

The energy generated at steady state \( MC_p \frac{\partial T}{\partial t} = PVA_t f \)
With $A_f$ the area of the mating ring face, $f = \text{friction coefficient}$ and $V = \text{coolant velocity}$

Therefore $f = \frac{MC_p \delta T}{PVA_f}$
APPENDIX C CALCULATING THE COEFFICIENT OF FRICTION

Heat generated = $P_m V A_f f$

$P_m V = 3302\, kPa\, m/s$

$A_f = 0.004m^2$

Mass flow rate ($M$) = $A V$ with $A$ the area of the coolant port, $V$ the coolant velocity and $\rho$ the density of the coolant. $M = 0.0633\, kg/s$

$C_p = 4.23\, kJ/kg.k$

$\Delta T = 4\, ^\circ C$

$f = \frac{M C_p \Delta T}{P_m V A_f}$

$f = 0.0810$ for fin mating ring

For the conventional ring

$\Delta T = 9.8\, \text{degrees Celsius}$

with $f = 0.180$
VITA

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