Exploring the Mechanisms Critical To the Operation of Metal Face Seals through Modeling and Experiments

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Abstract

This thesis aims to explore operation mechanisms of a special type of mechanical face seals: the flexible metal-to-metal face seal (FMMFS). Unique features of the FMMFS include much more flexibility in the circumferential than in the radial direction, identical rotating and stationary seal rings, and a loading mechanism using elastomeric O-rings.

Two versions of the numerical models have been developed to evaluate seal performance under various operating conditions. Both models consider interactions among surface deformations due to thermo-mechanical twists, unsteady lubrication in the sealing band, and heat transfer in the seal pair simultaneously. Outputs include contact pressures, oil film thickness, cavitation zone, partial film density, friction coefficients, dynamic oil transport, and seal temperature distributions. In the meantime, experimental efforts have been made to measure the friction coefficients and seal temperatures during different operations. The model predictions were then compared with the experiment results through the two above-mentioned quantities. The comparisons show that the numerical simulations consistently overestimate the friction by 15%-20%. However, overall trend of friction variation with speed and even some details of the friction can be captured, indicating that the current models are able to properly predict some underlying physics of seal operations.

The numerical models were then used to evaluate scoring and leakage failures of the FMMFS through three important variables: surface temperature, contact wetness, and oil exchange. Some surface geometric features, which contribute to differences of scoring and leakage behaviors, are identified. In order to achieve higher scoring resistance and minimum leakage, the sealing surface should have the following features: (1) random or dispersed asperity distributions, (2) relatively large surface roughness, and (3) combination of concave and half-concave-half-convex radial profiles.

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Chapter 1
INTRODUCTION

This chapter intends to

- Describe the unique geometric features of the flexible metal-to-metal face seal;
- Explain the project motivations and objectives;
- Introduce the previous work and theories related to the mechanical face seal of interest.

1.1 Introduction to the flexible metal-to-metal face seal

A mechanical seal is a device used to prevent lubricant leakage and exclude the contaminant outside. Through proper lubrication, it can delay wear of mechanical components and thereby lengthen their service life. The seal that is studied in this thesis is a unique type of the mechanical face seal. The seal has a small cross section area with a relatively large diameter. The outside diameter (OD) of the seal varies from several tens millimeters to several meters depending on different applications. Because of this geometric feature, the seal is much more flexible in the circumferential direction than in the radial direction, and therefore called flexible metal-to-metal face seal (FMMFS). It is noted that although the OD of the seal changes, the dimension of the radial cross section varies little.

The FMMFS system consists of two identical elements (as shown in Fig.1.1). Each element consists of a seal ring and an elastomeric (rubber) O-ring. The seal ring, acting as the primary seal, is made up with either iron-based or nickel-based cast alloy depending upon different applications. The rubber ring, also known as a toric, acts as a secondary (static) seal between the seal and housing.

During normal operation, two flexible metal-to-metal face seals are compressed against each other and make up a dynamic sealing interface where a thin lubricant film exists. One seal (called rotor) rotates relative to the other
(called stator). Note that a pair of the FMMFS differs from common mechanical face seals in that it consists of two identical seals. In addition, loading mechanism uses rubber rings. For a given seal design, the nominal face load is set by adjusting the gap between the housings. The rubber rings are compressed between two ramps, one on the seal ring, and one on the housing of the components being sealed. As the rotating housing moves towards or away from the stationary housing, the rubber rings roll up and down these ramps. This unique loading mechanism provides a consistently uniform face load over the range of axial displacements of the assembly being sealed.

![Diagram of dynamic sealing interface](image)

**Figure 1.1 Assembly of the flexible metal-to-metal face seals**

*Macro surface geometric features*

The FMMFS has its unique surface geometric features both in circumferential and in radial directions. These unique features play significantly important roles in seal performance, such as contact pattern, oil transport, friction generation and so on.
The cross section of the seal ring in the radial direction is shown in Figure 1.2. A sealing band where actual sealing occurs consists of two parts: a transition band and a flat band. The width of the sealing band is usually a few millimeters. The region inside the sealing band is called a spherical band, which allows lubricants to feed the sealing interfaces. It also enables new faces to be continuously formed as wear takes place. This ability to form new faces considerably enhances the service life of the seal. The life of a seal is determined by the time it takes the sealing band to move from the OD to the inner diameter (ID) of the seal. The transition part connects the flat band and the spherical band with a curvature. The so-called “flat” band is not flat at all under profilometer measurement. Depending on whether the surface height at OD is greater than that at ID or not, concavity and convexity can be defined, as shown in Figure 1.2. It is noted that concavity and convexity may vary along the circumference.

![Diagram showing the radial profile of the seal cross section]

**Figure 1.2  Radial profile of the seal cross section**

There also exists variation in the circumferential direction. This variation is due to consequences of several different manufacturing processes, and is called waviness. Figure 1.3 shows the waviness measurement of an FMMFS. The waviness may vary from 2 to 20 micrometers in magnitude and normally take a second-order sinusoidal shape.
The micro geometry (or roughness) is substantially important in the lubrication operation of the FMMFS in that it determines how oil is transported within the sealing interface. To minimize leakage and wear, the metal faces making up the dynamic sealing interface are highly polished. There are three different polishing processes for the FMMFS, i.e. process 1, process 2, and process 3. The seal surface texture differs significantly for different processes. Figure 1.4 shows the roughness of the sealing surface with the process 1. The lateral dimensions are 0.2 by 0.17 millimeters. The root mean square of the surface roughness is typically on the order of 0.1 microns.
1.2 Motivation and objectives

Although the design of the FMMFS is deceptively simple, the mechanisms of its operation and its failures are complex and not clearly understood. There are several proposed theories on how the FMMFS should operate. There is some evidence to support each of these opposing ideas, but none has been proven. A fundamental understanding of the metal-to-metal interface is needed to determine which, if any of these theories, should be used as a basis for designing and improving seals for extended applications.

One theory is that the seal should operate with the flats parallel to each other to minimize contact stress and wear. In this case, the oil film thickness is on the order of the asperity heights of the surfaces (i.e. boundary lubrication), and the apparent viscosity of such thin oil films is high enough to prevent leakage.

A second theory is that (thermal and possibly mechanical) distortion causes the cross section of the seal to rotate so that the flat becomes convex. Sealing then occurs at the narrow “contact” band at the spherical-flat interface. The theory implies that only if the scratches cross the spherical-flat interface, do they contribute to leakage. There is a debate over whether the lubrication of the seal interface is hydrodynamic or boundary.

In addition, two kinds of failure modes, i.e. excessive leakage and scoring failures, are associated with the FMMFS. Excessive leakage manifests continuous leakage of an unacceptable amount of the lubricant, normally without visible signs of surface damages; while the scoring failure is signaled with sudden increase of friction and temperature, accompanied with severe damages of sealing surfaces. Note that the scoring failure is always accompanied by the leakage failure.

Figure 1.5 shows an example of failure speeds with corresponding initial axial loading for leakage and scoring failure tests. Failure speeds are defined as linear speeds in the middle of the sealing band. Measured failure speeds and loads are
normalized by 10 m/s and 433 N, respectively. Large variations of failure speeds and loads mean that performance of the FMMFS is inconsistent even though all seal pairs used in the failure tests satisfy manufacturing specifications.

![Leakage failure load and speed](image1)

![Scoring failure load and speed](image2)

**Figure 1.5 Example of seal failure conditions [18]**

Properly functioning seals are critical to the performance and life of the related components. Therefore, premature failure of the FMMFS means not only replacements of the seals themselves, but also potential damages of other components and even the whole machine. Improvements in seal design would bring tremendous economic benefits.

The extended applications of large seals keep challenging the capabilities of the current FMMFS. While the FMMFS works well in some applications, its operation is not understood well enough to successfully scale it to all applications where such a seal is needed. In order to improve the seal design, it is essential to gain some fundamental understandings of lubrication of the metal-to-metal face seals. The objective of this thesis is to develop a knowledge base of important seal design parameters through both numerical modeling and experimental study, and to explain how these design parameters affect functioning of the seal under various operating conditions. This knowledge base will then be available as a tool to effectively design robust seals for new applications.
1.3 Previous works

Numerical modeling

Several numerical models to calculate the friction in the interface and the nominal temperature distributions in the mechanical face seals, considering the viscosity dependence on oil of temperature, have been developed. Such numerical models are called thermo-hydrodynamic models. Knoll [1] developed a three-dimensional numerical simulation using a finite element method which accounted for waviness effects. Person [2] used a finite difference method considering the effects of waviness and misalignment. Tournerie [3] included the heat transfer through both a stationary and a rotating seal while Person’s model considered heat transfer through the rotating seal only. Buck [4] proposed a simplified method to estimate seal surface temperature, considering conduction, convection and seal shape, in order to avoid directly solving heat transfer equations with finite element method.

Hong [5] exclusively addressed this type of seal by developing a steady-state numerical model. He also considered interactions among surface deflections, hydrodynamic lubrication, and heat transfer, but made several justified simplifications in order to reduce computational costs. Numerical models generate final deflections, contact patterns, lubrication in the sealing band and temperature distributions of the seal pair. The numerical model was compared with experiments under various operating conditions and different lubricants for validation. Based on this model, Hong [6] further developed a scoring failure criterion with account of nominal temperature and heat flux intensity.

The lubrication and heat transfer processes are inherently unsteady due to the surface geometrical variations in the sealing interface of each seal. Therefore, in order to better and more accurately simulate seal operation in practical applications, unsteady effects should be taken into consideration in the modeling. Harp and Salant [7] developed a mathematical model to predict the transient behavior of gas or liquid lubricated hydrostatic mechanical seals. The model
includes also the squeeze film effects and outputs film thickness distributions, contact pressures, leakage rates, and heat generation rates etc..

Reviews of scoring models

One of failure modes the mechanical face seal normally experiences is scoring. Scoring failure of a lubricated surface is a complex series of events and its mechanisms do not behave in the same manner for all surfaces and operating conditions. In mixed and boundary lubrications regimes, two sliding surfaces are separated by several protective films. These protective films include micro-elastohydrodynamic (EHL) lubricating film, physical absorbed and/or chemical absorbed surface film, oxide film, and finally the layer of substrate metal, as shown Figure 1.6. When one or some of these films ceases to be effective, asperities come into contact due to relative motion and scoring tends to initiate. Scoring is a very complex process and is not well understood yet. A great deal of research has been undertaken, in an effort to gain some fundamental understandings of scoring process, in particular its initiation and propagation. Several scoring models have been proposed over the years. Dyson [8] and Bowman and Staachowiak [9] rendered excellent reviews.

![Figure 1.6 Protective films between the contacting interfaces in mixed or boundary lubrication](image)

Even though the scoring process is a very complex process, scoring failure in generally is considered to be thermal in nature. In nearly all scoring models and
theories to date the effect of the contact temperature is of paramount importance. The earliest and most famous scoring theory was given by Blok [10]. He brought about a concept of "critical temperature". His hypothesis stated that scoring occurs when the contact temperature, which is the sum of the bulk temperature and the flash temperature, reaches a critical value independent of operating conditions, lubricant and material properties of surfaces. Blok's work has been validated by some researchers [11]. However, much evidence has contradicted the Blok's postulate, showing that the critical temperature varies with different operating conditions.

Dayton [12] proposed a hydrodynamic model of scoring in 1976 based on breakdown of EHL films. This model was later extended to micro-EHL lubrication by Cheng [13]. Both models indicate that the high pressures experienced within the Hertzian contact causes the lubricant viscosity to increase. In the meantime, there is an increase in lubricant temperature due to frictional heating, which reduces the viscosity. When a critical temperature is reached, the lubricant viscosity suddenly falls, causing the EHL films to collapse. The mean hydrodynamic pressure will no longer permit a high viscosity liquid film to be formed between the interacting asperities. Scoring then initiates.

Physical adsorption provides one line of protection against wear and scoring. Under the normal operation, adsorption of the lubricant molecules on the contacting surfaces and desorption of the molecules from the surfaces are in equilibration. However, when the desorption process prevails over the adsorption process, the physical-adsorbed surface films break down, promoting scoring. Based on this theory, Lee and Cheng [14] developed the Critical Temperature-Pressure (CTP) model. The model predicted relations of scoring with hydrodynamic pressure and the contact temperature. With an increase in the surface contact temperature the adsorbate thermal excitation becomes greater, indicating higher possibility of their escape from the surface. On the contrary, increasing hydrodynamic pressure enhances striking frequencies of the neighboring molecules on the surface, improving the adsorbate concentrations.
Whether scoring failure takes place or not is determined by a competition between the hydrodynamic pressure and the contact temperature.

Another form of adsorption is chemisorption, which is an irreversible or partially irreversible process with some degree of chemical bonding between adsorbate and substrate [15]. Since the common metals, such as iron, are reactive, the reaction layer formed between the metallic surface and the lubricating medium exerts a beneficial effect on surface wear, especially when additives are used.

An oxide layer, formed by the surface with oxygen present in the atmosphere or oxygen bonded to the lubricant molecule, provides another defense line against wear and scoring. Even a small area of exposed nascent surface in a contact protected only by an oxide layer is critical to scoring [16]. The kinetics of oxide formation and removal has also been proposed by Cutiongco and Chung [17] to predict scoring. As the contact temperature increases, the concentration of the adsorbed molecules on the oxide-covered asperities becomes lower, resulting in increased oxide wear. In the meantime, the oxidation rate increases, resulting in a thicker oxide layer. When a critical temperature is reached, the oxide removal rate overruns the oxide formation rate, scoring is likely to take place.

A thin hard layer of the substrate is the final layer of protective films against scoring. When asperities come into contact in relative motion, the friction generates heat. The resulting frictional heating will increase the asperity temperatures, causing thermal stress near and/or at the asperities. At the same time increase in the asperity temperature softens the substrate layer. At a critical temperature, when the sum of the local thermal stress and the local mechanical stress exceeds the yielding stress of the substrate layer, scoring then occurs.

In summary, it has been found that any one or a combination of the following phenomena can lead to scoring, i.e., breakdown of the EHL, micro-EHL films, desorption of adsorbed surface films, removal of the oxide layers, and asperity
heating and deformation etc. Obviously the contact surface temperature plays an extremely important role in above-mentioned protective film(s) breakdown. It is noted however, this temperature is not universal and is a function of the operating conditions, surface geometrical features, and resulting oil distributions.

1.4 Thesis outline

This chapter gives the introduction to the flexible metal-to-metal face seal system. The unique geometric surface features are described in details. Related previous works on mechanical face seals and scoring models were summarized. In particular, Hong's work on this exclusive type of seal was emphasized.

Both numerical modeling and experimental efforts have been undertaken to explore the operating mechanism of the FMMFS. Chapter 2 first introduces modeling processes of a quasi-steady state model and an unsteady state model. Both models include three sub models, which are 3D contact sub model, unsteady lubrication sub model, and 3D steady state or transient heat transfer sub model. Details of the three sub models are then explained at length. Some assumptions are made and justified in order to break the coupling between them. Finally some of results from the two models, which are important for seal performance analysis, are demonstrated.

In Chapter 3 experimental study on seal operation is addressed. Firstly the experimental setup is introduced, followed by the descriptions of the test scheme. Various test data is then demonstrated and compared with the simulation results as model validation. The comparisons are through two critical measurable quantities: frictional coefficient and seal temperatures.

In the following chapter the current developed models are utilized to explore the seal leakage and scoring behaviors. One advantage of the current models is their capability of predicting dynamic oil transport, which makes leakage prediction possible. Surface geometric features which affect scoring and leakage behaviors are also identified.
Chapter 5 will summarize the current work. Some conclusions are then drawn based on the numerical simulations and the experiment work. Possible areas for future study are also proposed.
Chapter 2

NUMERICAL MODELING

This chapter aims to:

- Introduce modeling processes of the quasi-steady state model and the unsteady state model;
- Elaborate the 3D contact sub model, unsteady state lubrication sub model, and heat transfer sub models in details;
- Demonstrate some important results from both models.

2.1 Structures of numerical models

Difficulty in modeling of the flexible metal-to-metal face seal (FMMFS) rises from coupled interactions of various mechanisms. Such mechanisms include solid contact, lubrication, and heat transfer, as shown in Figure 2.1. When two seals are pressed against each other, the sealing interface will deform due to externally applied force and axial load from asperity contacts will be generated. When one seal (rotor) is sliding against the other (stator), a small amount of lubricant will be introduced into the sealing band and a very thin lubricant film will be generated between the two “contacting” surfaces. As a result, hydrodynamic forces will be generated and share some portions of the axial load by rebalancing the force. The hydrodynamic forces will further deflect the sealing surfaces. In the meanwhile, a substantial amount of frictional heating is generated from asperity contacts and viscous shearing of the lubricant films. Frictional heating leads to thermal distortions of the seal interfaces, which influences the asperity contacts and hydrodynamic force distributions within the contact band. Furthermore, increase of temperature of the lubricant film reduces not only the oil shearing contribution to the friction, but also the hydrodynamic force. The decrease in the hydrodynamic force results in more asperity contacts in the sealing interfaces to compensate for the deficient axial load.
Figure 2.1 Illustration of coupled interactions among surface deflection, lubrication, and heat transfer

Apparently there are three major mechanisms whose interactions need to be considered in the modeling. They are surface deflections, lubrication, and heat transfer. In order to solve the whole system in numerical way, it is required that three sub models be solved simultaneously and iteratively, which unfortunately will involve tremendous computation costs. In order to avoid such complexity, some assumptions are made for sub models to decouple the interactions. The following sections will focus on descriptions of each sub model at length with corresponding assumptions and their validations.

Modeling process follows the steps as below:

1. Force distribution along the circumference is determined by the macro shape of the seal (or waviness), ring stiffness, and face load. This is at the structural level and has to be resolved at first. At this level radial twist angles are determined as well.

2. Then exact clearances between two rubbing surfaces that provide the force distribution in the previous step should be determined. At this stage
some simplifications as well as further improvements are attempted. Judging from the macro geometry, load (on the order of MPa), no-leak requirement, and existence of wear all along the circumference on the sealing band, the minimum clearance at each circumferential location is in the range of asperity. Therefore, boundary lubrication is important in this study. Even though inter-asperity hydrodynamic pressure and macro hydrodynamic pressure may be important in determining the minimum clearance, there are too many uncertainties and complexities involved. In addition, one primary focus is on the oil transport and oil flow at the minimum clearance locations is less important than higher clearances since the oil flow rate is proportional to the cube of the clearance. Therefore, effects of the hydrodynamic forces which slightly affect the clearances are negligible in evaluating oil flow rate. As a result, a contact model is used to calculate the minimum clearance based on the force distribution and radial twist angle determined in step 1.

3. Oil distribution within the clearances from the previous step should then be determined. An unsteady lubrication model is applied to calculate the partial film density caused by cavitation effects.

4. Heat generation is determined from the oil distribution in step 3 as well as asperity contacts in step 2. Once all thermal boundary conditions are known, seal temperatures can be easily evaluated.

Three numerical models have been developed in attempt to explore the seal operations, namely the steady state model, the quasi-steady state model and the unsteady state model. The steady state model, which includes the contact sub model, the steady state lubrication sub model, and the steady state heat transfer sub model, has been developed and explained in details by Hong [18]. Since both rubbing surfaces are not smooth at all, contact patterns vary instantly with rotation. That will lead to constant change of oil distribution and heat generation. Therefore, lubrication and heat transfer processes are unsteady in nature. In order to simulate real physics of seal operations, unsteady effects should be
taken into consideration. An unsteady lubrication sub model and a transient heat transfer sub model were developed accordingly. Keep in mind that the lubrication process is much faster than the heat transfer process. After the thermal steady state is reached, sensitivity of the oil viscosity to the oil temperature is not highly acute. Therefore, a steady state heat transfer sub model, incorporated with the unsteady lubrication sub model and the contact sub model, could be a good approximation for such analysis. The resulting model is called the quasi-steady state model. In some transient processes, however, where temperature gradients are significantly large, oil viscosity dependence on the temperature should be accounted for. In this case, a transient thermal sub model should be applied, together with the unsteady lubrication and contact sub models. The resulting model is referred to the unsteady state model. The rest of this chapter will focus on descriptions of these two models.

**Quasi-steady state model**

The flow chart of the quasi-steady state model is shown in Figure 2.2. Note that the FMMFS has much more flexibility in circumferential direction than in radial direction. Consequently, initial waviness along the circumference, if it is not sufficiently large, can be easily conformed or even flattened out. Since the waviness is the major source of generating hydrodynamic pressures, the hydrodynamic contribution to the total axial load can be negligible. This suggests that the interaction between the contact sub model and lubrication sub model can be decoupled and the total axial load is only supported by asperity contacts.

The program starts with importing surface profiles of a seal pair with inclusion of both macro and micro geometric features. The surface profiles can be taken from measurement data. With initial thermal and mechanical (or thermo-mechanical) tilts under a certain axial load, the contact sub model calculates the surface gaps and contact pressures from asperity contacts. The surface gaps are then treated as oil film thicknesses in the lubrication sub model. The lubrication sub model solves the average Reynolds equation, predicting cavitated regions, partial film contents, and oil exchange rates within the sealing interfaces. Once
asperity contact pressures and oil distributions are known, the frictional heat flux distributions and corresponding coefficients of friction can be calculated. Due to changes in contact patterns and oil distributions, the coefficients of friction vary within one revolution. Using averaged heat flux within one revolution and heat transfer coefficients calculated somewhere else [18] as inputs, a 3D steady state heat transfer sub model estimates seal temperature distributions. Due to strong dependence of oil viscosity on the oil temperature, the lubrication sub model and friction calculation will be revisited each time when seal surface temperatures are updated.

Figure 2.2  Flow chart of the quasi-steady state model

All of above procedures are repeated until solid deflections, oil pressures, and seal/oil temperature distributions converge.
**Unsteady state model**

In this numerical model a transient heat transfer sub model is incorporated instead of the steady state one. The model is capable of simulating fully transient operations, such as acceleration and deceleration processes. Figure 2.3 shows the flow chart of the model. Starting with initial values, oil viscosity and thermo-mechanical tilts are updated and input into the contact sub model, together with surface measurements. The contact sub model calculates the asperity contact pressures and determines the surface clearances. Then the lubrication sub model is called, solving the averaged Reynolds equation with output of partial density, frictional heating, and oil exchange rates. After that, the program proceeds to the next rotation phase and the above-mentioned procedure is repeated.

It is needed to note here that there are two time steps used in this model. One is in the lubrication sub model, represented by \( \Delta T_{lb} \). The other is in the heat transfer sub model, denoted by \( \Delta T_{th} \). In cases where values of the two time steps are different, the following iteration scheme is adopted. Since the lubrication process is much faster than the heat transfer process, much larger thermal time step \( \Delta T_{th} \) can be used. The reason of doing so is to reduce the computation complexity due to the unsteady effects. Then the lubrication sub model iterates \((\Delta T_{th} / \Delta T_{lb}) \) steps. The friction heating is calculated at each time step and is averaged over the duration. The average heat flux is then input into the transient heat transfer sub model to calculate seal temperature distributions at current thermal time step. All of above procedures are looped until both hydrodynamic and thermal steady state is reached, and/or other requirements are met. As far as the transient operations or details of temperature evolution with respect to time are concerned, the smaller thermal time step is needed. In this thesis it is taken as the same value as that of the lubrication time step. In this scenario more temperature details can be explored with sacrifice of computation costs.

In the end the program outputs all the information, including surface clearances, asperity contact pressures, oil distributions, frictional heating at each
lubrication time step, and seal temperature distributions at each thermal time step.

![Flow chart of the unsteady state model](image)

Figure 2.3 Flow chart of the unsteady state model

### 2.2 Importing surface profiles

The first step of both numerical models requires input of surface profiles of the seal pair of interest. In this thesis the surface profiles are measured using an interferometer. The interferometer makes individual measurements automatically along the circumference of the seal according to a programmed script. The individual measurements are then stitched together using an algorithm developed using Matlab codes. The measured surface profiles are then used as the macro geometry for the sealing band.
Such a micro geometric parameter as roughness is also required as an input in the models. In addition, considering the roughness effects on the oil flow rate, an average Reynolds equation should be solved with addition of flow factors. Both roughness and flow factor can be obtained from measurements taken by another optical interferometer. This interferometer has higher resolutions in space and height. With magnification factor of 6.2x, the interferometer makes a measurement of a rectangle area of about 1.0x1.2 mm. A typical 3D measurement from the interferometer is shown in Figure 2.4. After form is removed, root mean square (rms) roughness $S_q$ can be calculated as follows:

$$S_q = \sqrt{\frac{1}{MN} \sum_{i=1}^{M} \sum_{j=1}^{N} z(x_i, y_j)^2}$$  \hspace{1cm} (2.1)

where $M$ and $N$ are numbers of data points in $x$-, $y$- directions, $(x_i, y_j)$ is the Cartesian coordinate of a data point, $z(x_i, y_j)$ is the surface height at $(x_i, y_j)$.

![Figure 2.4 Measurement of the seal flat band with magnification factor of 6.2X](image)

The sealing surface is assumed to be homogeneous. Pressure flow factor curves generated from the small patch can then be representative of the whole
sealing interface. In order to calculate the pressure flow factor, two rough surfaces are combined to form an equivalent surface. This equivalent surface is then running against a purely flat surface. Starting with the mean distance to roughness ratio \((h/\sigma=1.5)\), mass flow deviation is calculated due to pressure gradients with increment of \(d(h/\sigma)=0.5\), all the way to \(h/\sigma=9\). In the end, two curves are created with mass flow correction factors versus \((h/\sigma)\) in \(\theta\) and \(r\) direction respectively. The pressure flow factor curves generated from the seal surface with the polishing process 1 is shown in Figure 2.5. With increase of the mean distance, the roughness effects on the mass flow rate diminish and both curves approach unity.

![Pressure flow factor curves](image)

**Figure 2.5 Pressure flow factor curves**

### 2.3 Quasi-steady state model

#### 2.3.1 Contact sub model

*Uniformly-distributed-load assumption*

It is worthwhile to point out again that the FMMFS is much more flexible in the circumferential direction than in the radial direction. Previous study [18] showed that under uniform loadings of 1 N/mm (roughly 1 MPa), which is about a typical supporting load, for the seal with a waviness of a few microns, deflections at both
ID and OD are one order of magnitude larger than the amplitude of waviness. This means that seal surfaces are so flexible in the circumferential direction that the effects of circumferential surface variations due to the initial waviness are negligible after deformation even under relatively low loading conditions.

*Distributed-load assumption*

There recently is another type of seal in production, with larger amplitude of waviness. The larger waviness is primarily due to the manufacturing process 2. The amplitude of waviness of this type of seal is on the order of ten microns. If one of the seal pair is from the process 2, the uniformly-distributed-load assumption may be questionable. However, study showed that for a pure flat seal surface running against a computer-generated surface, which has a second-order sinusoidal waviness in the circumference and the amplitude of 20 microns, two surfaces conformed beyond the loading of 0.76 N/mm and a closed-form contact region along the circumference was established. Once conformity was achieved, increasing normal axial load would not change the shape of the circumferential load distribution. Figure 2.6(a) clearly shows the result. Mean value line shifts upward when the axial load increases. This can be seen more obviously from Figure 2.6(b). When three curves are intentionally shifted at the same mean line, they collapse into a single curve. Furthermore the curve bears the same shape as that of the initial surface waviness. This implies that the circumferential load distribution can be determined through the initial waviness of the seal.
Once uniform or sinusoidal load distribution is obtained, a 3D contact problem can be simplified by solving a 2D contact problem at each radial cross section. Based upon the previous judgment, hydrodynamic effect is negligible, which is also proved by Hong [18]. Therefore, the surface deformations are independent
of the lubrication pressure at the sealing band and the decoupling between the contact sub model and the lubrication sub model can be realized.

**Mechanical and thermal twists**

Due to circumferential flexibility, radial rigidity is crucially important in determining the surface deflections. There are two major mechanisms to twist at each radial cross section. One is the mechanical twist, due to the pressure loadings from toric rings between housings and seal ramps. The other is the thermal twist which is due to thermal expansions from temperature gradients in the seal. Figure 2.7 illustrates formation of these two twist angles. Note that contact areas tend to move radially inward with increasing twist angles. For detailed information on how to calculate both mechanical and thermal twist angles under different operating conditions, refer to Hong [18]. One thing needs to be pointed out is that the thermal twist dominates over the mechanical twist under the normal axial loadings.

![Figure 2.7](image)

**Figure 2.7** (a) Mechanical, (b) and (c) thermal twist angles
Minimum surface gap heights

For boundary lubrication, the load is supported by asperity contacts. However, real contact areas only occupy a very small fraction of nominal contact areas due to the surface roughness. Williams [19] pointed out that the real contact area was only several percent of the nominal contact area. In this sense, the averaged surface gap heights (or nominal heights) in the nominal contact area are not zero. The contact sub model adopts an equation which relates the average load with the nominal height between two rough surfaces. The equation is developed by Lee and Cheng [20] and valid for contacting surfaces with a purely longitudinal roughness. Seals from the process 1 and process 2 studied in this thesis have approximately longitudinal roughness. Therefore, Lee and Cheng’s equation was used to describe the relation between average nominal contact pressure and surface gap height. The non-dimensional form of the equation is expressed as:

\[
\frac{4P_{\text{avg}}L_0}{E'\sigma\pi} = e^{f(H)} \tag{2.2}
\]

where

\[
f(H) = 3.01002 - 3.04431H - 1.54827H^2 + 2.56951H^3 - 0.92026H^4 \tag{2.3}
\]

\(P_{\text{avg}}\): average contact pressure within the nominal contact length

\(L\): profile segment length

\(E'\): equivalent Yong’s modulus \(E' = 2\left[\frac{1-v_1^2}{E_1} + \frac{1-v_2^2}{E_2}\right]^{-1}\)

\(v\): Poisson’s ratio

\(E_1, E_2\): Yong’s modulus of surface 1 and surface 2 respectively

\(\sigma\): composite rms roughness \(\sigma = \sqrt{\sigma_1^2 + \sigma_2^2}\)

\(H\): surface gap height normalized by \(\sigma\).
2.3.2 The unsteady lubrication sub model

Fundamentals of hydrodynamic lubrication

The basic and classic theory of hydrodynamic lubrication was given by Reynolds [21] in 1886. He derived the equation from the Navier-Stokes equations for determining the pressure build-up in the lubricating film with lubrication approximation.

For thin fluid films, such as oil film between two mechanical seal surfaces, the dimension ratio of oil film thickness to the lateral length is less than $O(10^{-3})$. The reduced Reynolds number is significantly small. The inertial terms in the Navier-Stokes equations can be dropped out, reducing the momentum equations to

\begin{align*}
0 &= -\frac{\partial p}{\partial x} + \mu \frac{\partial^2 u}{\partial z^2} \\
0 &= -\frac{\partial p}{\partial y} + \mu \frac{\partial^2 v}{\partial z^2} \\
0 &= \frac{\partial p}{\partial z}
\end{align*}

(2.4)

where $x, y, z$ are coordinates. $u, v$ are linear speeds in the $x, y$ directions respectively. $\mu$ is the dynamic viscosity of the lubricant. And $p$ is the hydrodynamic pressure.

The $z$-component of Equation 2.4 asserts that to the order of approximation employed here, the pressure is invariant in the $z$ direction, i.e., across the oil film. That is, the pressure is a function of $x, y, t$ only. Integrating the rest of Equation 2.4 gives the velocity profiles as below:

\begin{align*}
u(x,y,z,t) &= \frac{1}{2\mu} \frac{\partial p}{\partial y} z^2 + cz + d
\end{align*}

(2.5)
The integration constants $a$, $b$, $c$, and $d$ are evaluated with the help of boundary conditions on velocity. The pressure distribution appearing in Equation 2.4 must be such that the equation of continuity is satisfied. By substituting Equation 2.5 into the averaged equation of continuity, we obtain the Reynolds equation, combining conservation of mass and momentum:

$$\frac{\partial}{\partial x} \left( \rho h^3 \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( \rho h^3 \frac{\partial p}{\partial y} \right) = \frac{\partial}{\partial x} \left( \rho hU \right) + \frac{\partial}{\partial y} \left( \rho hV \right) + \frac{\partial (\rho h)}{\partial t}$$

(2.6)

where $\rho$ is the oil density and $h$ is the oil film thickness.

The Reynolds equation is limited to full oil films of positive pressure and does not take into account the fact that fluids cannot sustain significant tension, and that the fluid will cavitate at pressures below the cavitation pressure, forming vapor. The classic Reynolds equation must therefore be extended to include partial film phenomena.

*Partial film phenomena*

Normal oils have very low tensile strength and therefore cannot exist at pressures below their cavitation pressure. When the pressures predicted by the Reynolds equation drops to a critical value (the cavitation pressure), as will occur for sufficiently divergent geometries, the fluid will cavitate, forming both liquid and vapor. Cavitation is the disruption of what would otherwise be a continuous liquid phase by the presence of a gas or vapor or both [22]. In the cavitation zone pressure distribution is constant at the cavitation pressure as shown in Figure 2.8.

Jakobson-Floberg Olsson (JFO) cavitation theory, as described by Elrod [23], proposes that the fluid domain can be divided into two distinct zones, a full film region where the Reynolds equation governs pressure generation, and a partial film region which is at the cavitation pressure, with only a small fraction being occupied by fluid. It is assumed that within the partial film zone, the fluid distribution can be represented by multiple streamlets of fluid, spanning the gap between two surfaces, as shown in Figure 2.8.
The Reynolds equation is only valid for cases where two rubbing surfaces are perfectly smooth. When surface roughness is excessive, its effects on the flow rate and thereby pressure generation should be taken into consideration. This can be done by applying the Patir and Cheng flow factor method [24]. Patir and Cheng proposed average flow rate equations (Eq. 2.7) for the full film region in terms of three flow factors, the average gap height and the average hydrodynamic pressure. By equating the flow rates predicted by these equations with those generated by a rough surface, the flow factors for that particular surface can be calculated, as shown in Figure 2.9.
\[
q_v = -\frac{h^3}{12\mu} \frac{\partial \bar{p}}{\partial x} + \frac{U_1 + U_2}{2} h
\]

\[
\bar{q}_x = -\phi_x \frac{h^3}{12\mu} \frac{\partial \bar{p}}{\partial x} + \frac{U_1 + U_2}{2} h + \frac{U_1 - U_2}{2} \sigma \phi_x
\]

\[
\bar{q}_y = -\phi_y \frac{h^3}{12\mu} \frac{\partial \bar{p}}{\partial y}
\]

Figure 2.9 Comparison of rough and smooth surface flow rates

\[
qv = h \left( \frac{h^2}{2} + \frac{U_1 + U_2}{2} h \right)
\]

where the over-score bar indicates statistical averaging. The shear flow factor \(\phi_s\) represents the changes in Poiseuille flow due to the pressure distribution that is generated by the sliding motion of the rough surface. If two rubbing surfaces are identical, the effects of the shear flow factors cancel out. The pressure flow factors, \(\phi_x\) and \(\phi_y\), represent the changes in the Poiseuille flow due to roughness and average pressure difference alone. The Couette flow is unchanged by roughness, if we assume that there is no inter-asperity cavitation present. The Reynolds equation for rough surfaces now takes the form:

\[
\frac{\partial}{\partial x} \left( \phi_x \frac{\rho \bar{h}^3}{12\mu} \frac{\partial \bar{p}}{\partial x} \right) + \frac{\partial}{\partial y} \left( \phi_y \frac{\rho \bar{h}^3}{12\mu} \frac{\partial \bar{p}}{\partial y} \right) = \frac{\partial}{\partial x} \left( \frac{\rho \bar{h}(U_1 + U_2)}{2} \right) + \frac{U_1 - U_2}{2} \sigma \phi_x \frac{\partial \phi_x}{\partial x}
\]

\[
+ \frac{\partial}{\partial y} \left( \frac{\rho \bar{h}(V_1 + V_2)}{2} \right) + \frac{\partial (\rho \bar{h})}{\partial t}
\]

\[
(2.8)
\]

One of unique features of the FMMFS system is that the seal pair is "identical" with the same manufacturing processes. Under this condition, the shear flow factor is close to zero and the third flow correction term on the RHS of the above equation can be dropped out. This left two pressure flow factors, which
are in sliding and traverse directions respectively, being effective in correcting flow rate change due to the roughness. Details of how to calculate the pressure flow factors can also be found in reference [25]. Pressure flow factors versus h/σ for a rough surface from the process 1 are already shown in Figure 2.4.

**Universal Reynolds equation**

Elrod [23] proposed a universal Reynolds equation, based on JFO cavitation theory, which combines the partial film and full film regions into a single set of equations based on the following assumptions:

- In the full film region:
  - The volume fraction of the oil is 1.0, and the density is that of the liquid oil.
  - The classic Reynolds equation governs the hydrodynamic pressure generation.

- In the partial film region:
  - The pressure is constant and equal to the cavitation pressure.
  - The volume fraction of the liquid oil is governed by conservation of mass.
  - The density of the partial film is given by the volume fraction of the fluid multiplied by the fluid density, that is, the mass of vapour is neglected.

The unsteady lubrication model is developed based on Payvar and Salant’s work [26], even though they presented a steady state solution and did not include flow factor effects. Further discussion of the method and numerical algorithms with both unsteady and flow factor effects were provided by Yong [27].

Based on the JFO cavitation theory, the flow field in the cylindrical coordinate is governed by the following equation
\[
\frac{1}{r} \frac{\partial}{\partial r} \left( \phi r h^5 \frac{\partial p}{\partial r} \right) + \frac{1}{r} \frac{\partial}{\partial \theta} \left( \phi \frac{h^3}{r} \frac{\partial p}{\partial \theta} \right) = 6 \mu \omega \frac{\partial h}{\partial \theta} + 12 \mu \frac{\partial h}{\partial t} \tag{2.9}
\]

in the full film zone and,

\[
6 \mu \omega \frac{\partial}{\partial \theta} \left( \frac{\rho}{\rho_c} h \right) + 12 \mu \frac{\partial}{\partial t} \left( \frac{\rho}{\rho_c} h \right) = 0 \tag{2.10}
\]

in the cavitation zone. \( \rho_c \) is the density of liquid oil film. The over-score bar is dropped for simplicity. Boundary conditions at inside and outside radius are expressed as,

\[
p = p_s, \text{ at } r = r_i \\
p = p_a, \text{ at } r = r_o \tag{2.11}
\]

where \( p_s \) is the sealed pressure and \( p_a \) is the atmospheric pressure.

Instead of solving two distinct regions separately, Equation 2.9 and 2.10 can be combined as a "universal" equation with the help of the cavitation index \( F \) and the pressure index \( \Phi \), which are defined as,

\[
\frac{p - p_c}{p_s - p_c} = F \Phi \tag{2.12}
\]

\[
\frac{\rho}{\rho_c} = 1 + (1 - F) \Phi \tag{2.13}
\]

The pressure index \( \Phi \) has a different definition in the full film zone and the partial film zone, and the cavitation index \( F \) which activates or suppresses the appropriate terms of the differential equations or boundary conditions in the way such that when \( \Phi > 0 \), \( F = 1 \) and \( \Phi < 0 \), \( F = 0 \). Note that \( \Phi \) is a dimensionless pressure in the liquid film region and that \( 1 + \Phi \) is the partial film content in the cavitation zone.

With the following dimensionless variables,
\[ \eta = \frac{r}{r_i}, \quad H = \frac{h}{h_{ref}}, \quad \gamma = \frac{6\mu \omega}{p_s - p_c \left( \frac{r_i}{h_{ref}} \right)^2} \]  

(2.14)

the resulting universal Reynolds Equation is given by,

\[
\frac{1}{\eta} \frac{\partial}{\partial \eta} \left( \phi H \frac{\partial (F\phi)}{\partial r} \right) + \frac{1}{\eta} \frac{\partial}{\partial \theta} \left( \phi \frac{H^3}{\eta} \frac{\partial (F\phi)}{\partial \theta} \right) = \gamma \frac{\partial}{\partial \theta} \left[ (1 + (1 - F)\phi)H \right] + \frac{2\gamma}{\omega} \frac{\partial}{\partial t} \left[ (1 + (1 - F)\phi)H \right]
\]

(2.15)

With boundary conditions,

\[
\phi = 1.0 \quad \text{at} \quad \eta = 1
\]

\[
\phi = \frac{p_a - p_c}{p_s - p_a} - 1 \quad \text{at} \quad \eta = \frac{r_o}{r_i}
\]

(2.16)

Equation 2.15 is then solved using the TDMA algorithm [28] with relaxation coefficients to update the values of \( \Phi \) and \( F \) at each iteration. To ensure numerical stability, small values of relaxation coefficients are required [26].

2.3.3 Frictional heat generation

Contact areas in this thesis refer to the nominal contact areas instead of actual contact areas (see Figure 2.10). It is well known that the actual contact areas only occupy a very small fraction of areas in the nominal contact regions. Therefore, the average gap height (or nominal height) in the nominal contact areas is not zero, but a certain value \( h \). Areas outside of the nominal contact zones are defined as non contact areas.
Contact areas are further divided into wet contact and dry contact depending on whether there is liquid or vapor present between the contacting surfaces. In the dry contact regions where vapor separates the two surfaces or the contact areas are surrounded by vapor, frictional heat flux only comes from asperity contact pressures. Dry frictional heat flux is then given by

\[ q_d = f_d P_{\text{con}} V \]  \hspace{1cm} (2.17)

where

- \( q_d \) : Dry frictional heat flux \([\text{W/m}^2]\)
- \( f_d \) : dry friction coefficient
- \( P_{\text{con}} \) : Asperity contact pressure [Pa]
- \( V \) : linear sliding speed [m/s]

In the wet contact regions, however, the contact areas are immersed into the liquid lubricant film. The friction is influenced by both contact pressure and viscous shearing of the lubricant. The frictional heat flux for the wet contact regions is expressed as

\[ q_w = f_s P_{\text{con}} V + \frac{\mu V^2}{h} \]  \hspace{1cm} (2.18)
where

- \( q_w \): Frictional heat in wet contact areas [W/m\(^2\)]
- \( f_b \): boundary friction coefficient
- \( \mu \): Kinetic viscosity of the lubricant, as a function of oil temperature [Pa.s]
- \( h \): Nominal surface height [\( \mu \)m]

In the no-contact areas, the only source of friction comes from viscous shearing of the lubricant. The corresponding frictional heat flux is then given by

\[
q_{n,m} = \frac{\mu V^2}{h} [W/m^2]
\]

With cavitation effects, partial film phenomena occur in some of the contact areas. In the cavitation zones, there is a mixture of the lubricant and vapor or air. Physical properties are required to be re-evaluated in these two-phase regions. The definition of the cavitation zone is shown in Figure 2.11.

**Figure 2.11 The element in the cavitation zone**

The partial film content, \( \lambda_{y,i} \), is defined as a ratio of the average density in the element (i,j) to the density of the lubricant. Apparently, \( \lambda_{y,i} = 1 \) at the areas beyond the cavitation zone, where there is only full film present between two surfaces. To another extreme, if \( \lambda_{y,i} = 0 \), it means that the lubricant vaporizes completely, and the element is filled entirely with vapor. However, if two phases are both present in the element, the physical properties of the mixture are re-
determined depending on the volume ratio of each phase. After some manipulations, the volume ratio of the oil can be approximated to the partial film content $\lambda_{ij}$. Detailed derivation is provided by Hong [18]. The physical properties of the mixture are then given by,

\begin{align}
\mu_{ij, cav} &= \lambda_{ij} \mu_{oil} + \left(1 - \lambda_{ij}\right) \mu_{vap} \approx \lambda_{ij} \mu_{oil} \\
\kappa_{ij, cav} &= \lambda_{ij} \kappa_{oil} + \left(1 - \lambda_{ij}\right) \kappa_{vap}
\end{align}

(2.20) (2.21)

where $k$ is the heat conductivity, subscripts $cav$, $oil$, and $vap$ represent cavitation zone, oil phase and vapor phase respectively.

With the cavitation effect considered, the frictional heating in the contact area is due to both dry friction and wet friction. Once the volume ratio of the mixture is determined, the frictional heat flux in that contact area can be expressed as,

\[ q_{ij, con} = \lambda_{ij} \left( f_d P_{ij, con} V + \frac{\mu_{oil} V^2}{h_{ij}} \right) + \left(1 - \lambda_{ij}\right) f_d P_{ij, con} V \]

(2.22)

While in the no-contact areas, the frictional heat flux only considers the source from the viscous shearing, and is given by,

\[ q_{ij, non-con} = \lambda_{ij} \left( \frac{\mu_{oil} V^2}{h_{ij}} \right) \]

(2.23)

Note that this frictional heat model only applies in the case where the oil lubricant is in contact with two surfaces and shear flow governs oil transport.

The lubrication sub model calculates cavitation zones and the partial film contents. With the contact pressures from the contact sub model, the frictional heat flux distribution is then determined. Knowing the distribution of the frictional heat flux in the sealing band, the coefficient of friction can be calculated for a given axial load,
where

\[
cof = \frac{\sum_{i}^{N} \sum_{j}^{M} q_{ij} A_{ij}}{N \cdot V}
\]  \hspace{1cm} (2.24)

\(q_{ij}\) : Heat flux in element \((i, j)\) \([\text{W/m}^2]\)

\(A_{ij}\) : Area of the element \([\text{m}^2]\)

\(N\) : Axial normal load \([\text{N}]\)

In general, frictional heat flux varies within one revolution due to unsteady effects of contact patterns and oil distributions. So does the coefficient of friction, as shown in Figure 2.12.

**Figure 2.12 Variation of coefficient of friction within a revolution**

### 2.3.4 Steady state heat transfer sub model

*Introduction*

In this sub model, a 3D steady state heat conduction equation in the cylindrical coordinate needs to be solved for both the rotor and the stator with specified boundary conditions. The 3D steady heat conduction equation is given by,
\[
\frac{k}{r} \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial T}{\partial r} \right) + \frac{1}{r^2} \frac{\partial^2 T}{\partial \theta^2} + \frac{\partial^2 T}{\partial z^2} = 0
\] (2.25)

It is known that circumferential temperature variations due to variations of the heat flux are negligible compared to the mean temperatures in many mechanical seals [29]. These variations are especially small with large thermal inertia and the high rotation velocity. References [30, 18] evaluated the circumferential temperature variations for the FMMFS which was subject to a periodic heat flux with amplitude of \( q_0 \), and found out that the circumferential temperature variations were one order of magnitude less than the average temperatures of the rotating surface. Therefore heat transfer process is simplified into a 2D problem for the rotor, and a 3D problem for the stator.

As mentioned in the previous section, the frictional heat flux is unsteady and varies within one revolution. However, solving the 3D steady conduction equation for every rotation step will requires tremendous computation costs. As far as the magnitude of seal temperatures are concerned, using averaged heat flux within one revolution is sufficient. When surface temperature fluctuation with respect to time is of interest, a transient heat transfer sub model with instant heat flux distribution should be employed. The method and numerical algorithm for solving 3D transient heat conduction equation will be explained in the following section.

It is also assumed that the frictional heat flux is generated in the middle of the oil film, and the temperature profiles in the oil film are linear. This assumption is valid when conduction across the oil film dominates convection along the circumference. The FMMFS operates in the boundary lubrication regime where the oil film thickness is in the range of asperity. The thermal resistance across the oil film is extremely small. This assumption has been widely used for temperature calculations of the seal system [31, 29].
Boundary conditions

There are four boundary conditions for the FMMFS system, as shown in Figure 2.13. At inside seal surface which is surrounding the lubricant oil and at outside seal surface which is normally exposed to the atmosphere, forced convection boundary conditions are applied on both sides. Hong [18] calculated heat transfer coefficients for inner surfaces using a commercial finite element package, ADINA™. Hong devised a new numerical scheme by iteratively solving a 3D oil flow model and a 2D thermal model and formulated relationship of the heat transfer coefficients with the rotation speeds. He also concluded that the heat transfer coefficients on the rotor side are substantially greater than those of the stator under a prescribed speed, due to rotation and resulting turbulent effects. The correlation of the heat transfer coefficients and rotation speeds is then integrated into the numerical model as an input. The heat transfer coefficients on the air side are based on the reference [32].

![Figure 2.13 Boundary conditions for 3D heat transfer sub model](image)
Distribution of the frictional heat flux is used as another boundary condition in the sealing interface. As mentioned in the previous section, the frictional heat flux varies within one revolution due to changing contact patterns and oil distributions. Time-averaged heat flux distribution in one revolution is employed in the heat transfer sub model.

For rest of the boundaries, including ramps where the rubber rings apply the axial load, and the taper bands where no contact occurs, adiabatic boundary conditions are imposed.

**Heat division**

From the lubrication model, the total heat flux at each element is estimated. However, individual heat flux to the stator and to the rotor (or heat division) should be known in order to solve the individual conduction equations for two seal rings. Heat division is a time-and-space-dependent variable, and is influenced by the rotation speed as well. There is an important assumption that the frictional heat is generated in the middle of the oil film and heat conduction across the oil film dominates the other heat transfer characteristics (see Figure 2.14). Under this assumption, the heat flux to the stator and to the rotor can be determined respectively when two surface temperatures are readily known. The heat division is given by,

\[
T_{oil} = \frac{1}{2} \left( T_{r,j} + T_{s,i} + \frac{q_y h_y}{2 \kappa_{oil}} \right) \tag{2.26}
\]

\[
df_{ij} = \frac{1}{2} + \frac{1}{2 h_y} \frac{\kappa_{oil} (T_{r,j} - T_{s,i})}{q_y} \tag{2.27}
\]

where the heat division factor, \( df \), is the ratio of heat flux to the stator to the total heat flux generated in the element \((i,j)\). The subscript \( r \) and \( s \) represents the rotor and the stator respectively.
Seal temperature calculations

By discretizing Eq. 2.25 and applying appropriate boundary conditions in finite difference equations, the seal temperatures can be solved using energy conservation to each control volume of an element. Details of derivation of finite difference equations and treatments of different boundary conditions are provided by Hong [18].

After obtaining the finite difference equations, the Gauss-Seidel iteration scheme is employed to solve these algebraic equations. For each iteration, seal surface temperatures for both stator and rotor are changing, which results in changes in the heat flux to the stator and the rotor (and the heat division). A relaxation coefficient is used to update the heat flux in order to ensure numerical stability. Iterations continue until temperature distributions in the stator and in the rotor converge.
2.3.5 Iteration of the quasi-steady state model

After the heat transfer sub model is completed, the program calculated the residuals of the averaged oil temperature and the twist angles. The residual of the averaged oil temperature is given by,

\[
\Delta T_{\text{oil}} = \frac{T_{\text{oil}}^{\text{new}} - T_{\text{oil}}^{\text{old}}}{T_{\text{oil}}^{\text{old}}} 
\]

where the oil temperature is the averaged value in the sealing band. \( T_{\text{oil}}^{\text{new}} \) is the updated oil temperature, and \( T_{\text{oil}}^{\text{old}} \) is the previous oil temperature.

And the residual of the twist angles is expressed as,

\[
\Delta \text{Tilt} = \frac{\sum_{i=1}^{N} (T_{\text{tilt}}^{\text{new}}_i - T_{\text{tilt}}^{\text{old}}_i)}{N}
\]

where \( T_{\text{tilt}}^{\text{new}}_i \) and \( T_{\text{tilt}}^{\text{old}}_i \) are the updated and previous twist angles at each radial cross section respectively. \( N \) is the number of circumferential nodes.

The complete model continues until the oil temperature and twist angles converge. The final outputs include local surface clearances, contact pressures, cavitation index, partial film contents, coefficients of friction, and seal temperature distributions. Some of results are demonstrated in the following section.

2.4 Unsteady state numerical model

In this model, a 3D transient heat transfer model has been developed, which was then incorporated into the afore-mentioned contact sub model and the unsteady lubrication sub model to form the unsteady numerical model. The contact sub model and the unsteady lubrication sub model have been discussed in details in previous sections; main focus in this section is then to explain the 3D transient heat transfer sub model.
2.4.1 3D transient heat transfer sub model

Instead of solving the stator and rotor separately, the transient heat transfer sub model treats the seal system as an entity and solves the transient heat conduction equation with a heat generation term. This is done by assuming a fictitious thin layer between the contacting surfaces (as shown in Figure 2.15). The primary advantage of doing so is that calculation of heat division can be avoided. However the fictitious thin layer must meet the following requirements: (1) height of the thin layer is much smaller than r- and θ- direction spacing; (2) heat capacity of the thin layer is small; (3) heating is generated within the thin layer. These requirements ensure generated heat will dissipate into the stator and rotor.

![Figure 2.15 Schematic of the fictitious thin layer](image)

Solving the stator and the rotor together requires one single reference coordinate. This is not a problem for both parts individually. However, at the grid points where relative motion is present, extra care should be paid. Figure 2.16 gives an example how rotation effect is considered in the calculation. After one phase rotation the grids in the rotor move one point forward along the rotation direction. The heat fluxes are calculated in the same manner except for the z-direction, which is given by,
\[
q_{k-1} = k \cdot dA_z \frac{T_{i-l,j,k-1} - T_{i,j,k}}{H}
\] (2.30)

where

\begin{itemize}
  \item \(q_{k-1}\) : Heat flux between the fictitious layer and the rotor surface \([\text{W/m}^2]\)
  \item \(dA_z\) : Area normal to the \(z\)-direction \([\text{m}^2]\)
  \item \(H\) : Half of thickness of the fictitious layer \([\text{m}]\)
  \item \(T_{i,j,k}\) : Temperature in the fictitious layer \([\degree\text{C}]\)
  \item \(T_{i-1,j,k}\) : Temperature on the rotor surface \([\degree\text{C}]\)
\end{itemize}

Figure 2.16 Example of one phase shift

Solving transient temperature problems in the seal system, even with a relatively coarse grid (106x120x120), will involve tremendous computation costs with traditional methods. In this thesis the Alternative Direction Implicit (ADI) method is applied, reducing computational complexity substantially. The ADI method is a derivative of the Crank-Nicolson method. It basically solves 3D or higher dimension problems by successive 1D dimension methods. The advantages of this method include unconditional stability, no limits on time steps and no large matrix inversion, and higher order accuracy (2\(^{nd}\)-order in both spatial and temporal domain).
An illustration of 3D ADI method is shown in Figure 2.17. Suppose we have a simple 3D system (mxnxl). By dividing the time step in three successive steps, we solve three 1D implicit problems instead of solving one complex 3D system directly. At the first step, we have x-direction implicit while keeping y- and z-direction explicit. In this sense, the temperatures of the neighboring points in the y- and z-direction are known and can be arranged in the RHS of the equation, and only the temperatures of the neighboring points in the x-direction are unknown and need to be solved. This is equivalent to solving 1D implicit problem in x-direction. After the step I, the temperatures in x-direction are updated. At step II similarly, we have y-direction implicit and keep x- and z-direction explicit. Then 1D implicit problem in y-direction is solved. Likewise at step III, we solve 1D implicit problem in z-direction.

Note that with the ADI method, large time steps are stable but produce large errors due to their neglect of “splitting terms”.

Figure 2.17 Illustration of the ADI method
We will take the first step as an example to demonstrate how computational cost is reduced by the ADI method, as shown in Figure 2.18. As mentioned before, at step I due to the other two directions are explicit, there is only \( m \) unknowns for each line \((j,k)\). To solve such \( m \) unknowns, a tri-diagonal linear system is established. Fortunately this special tri-diagonal matrix, which is \( m \) by \( m \), only needs \( m \) operations. Since there would be \( nxI \) such tri-diagonal linear systems at step I, the total operations needed to solve the whole system is \( mnxI \). Likewise there are also \( mnxI \) operations for step II and step III respectively. The total computational cost is therefore on the order of \( N(mxnxI) \) instead of \( N^2 \) in conventional methods.

\begin{figure}
\centering
\includegraphics[width=\textwidth]{adi_method_step_1}
\caption{Step I of the ADI method}
\end{figure}

### 2.4.2 Iteration of the unsteady state model

Two time steps used in the model lead to two loops, namely, the inner loop and the outer loop. In the inner loop the lubrication sub model is iterated with respect to time. At each rotation phase, the hydrodynamic pressure, cavitation index are calculated and then used as the old values for solving the lubrication sub model at the next rotation phase. When \((\Delta T_{th} / \Delta T_{lb})\) iterations are completed, the program proceeds to the outer loop. The outer loop calculates the seal temperature distribution at each thermal time step. Then the oil viscosity and
thermal twists are updated, and the program continues at the next thermal time step. The outer loop terminates until the hydrodynamic and thermal steady state are both achieved.

2.5 Numerical results

In this section we will illustrate some results from both numerical models. These variables are important to analyze the seal performance, such as oil leakage and scoring behaviors.

2.5.1 Results from the quasi-steady state model

All the results presented herein are for the seal pair from the process 2 and under the normalized axial load 2.2 N/mm, and the rotation speed 600 rpm, unless otherwise stated.

Surface clearances and contact pressures

The contact sub model gives the deformed surface clearances and the asperity contact pressures under a specified loading, which are shown in Figure 2.19 and Figure 2.20 respectively. As mentioned earlier, the unique feature of flexibility in the circumferential direction suggests that the deformed surface clearances are determined by the radial surface profile at each radial cross section. Figure 2.19 shows the radial surface profiles before and after deformation at four circumferential locations indicated in the figure. As one can see from the original surface profiles, minimum surface clearances occur close to OD. With the pressure loading and thermal effects, the radial surface profiles tilt, and the minimum surface clearances move radially inwards to the ID.
Figure 2.19 Initial vs. deformed radial surface clearances at four circumferential locations

Although the radial surface profiles tilt a small amount under the prescribed operating conditions, that amount is sufficient to determine where contacts take place in the sealing interface. Figure 2.20 demonstrates the contact patterns and the corresponding contact pressure distributions at a certain rotation phase. It is clearly shown in Figure 2.20 (b) and (c) that contacts occurring between the angles $\pi$ to $3\pi/2$ in the radial location $r=43.9$ mm correspond to the regions with the minimum surface clearances.
Figure 2.20 (a) 3D contact pressure distribution at the sealing band, (b) surface clearance along circumference at r=43.9 mm, (c) 2D contact pressure along circumference at r=43.9 mm.

Cavitation indices, partial film contents, and coefficients of friction

The unsteady lubrication model provides not only the cavitation zones, partial film contents, and coefficient of friction at a certain rotation phase, but also dynamic oil transport within the revolution. Such results enable analysis of oil leakage and possible scoring occurrence. Figure 2.21 (a) shows the distribution of the cavitation index at a certain rotation phase. In the cavitation zones where the mixture of the lubricant and vapor coexists, the average density is less than the lubricant density. The partial film contents corresponding to the cavitation zones are shown in Figure 2.21(b). The partial film contents are then used in calculating the frictional heat flux.
Figure 2.21 (a) cavitation index, and (b) partial film contents

As described in the friction model, the sources of the frictional heat flux come from both the asperity contacts and the oil viscous shearing with cavitation effects. The heat flux distribution in the oil film at a certain phase of the rotation is shown in Figure 2.22 (a). It is evident that most of frictional heat is generated in the contact regions where the asperity contacts will contribute significantly in either wet or dry contact zones. In the contact zones, the nominal surface clearance is small and the resulting viscous heat is also substantial. Therefore, in terms of scoring analysis, the contact zones are of most interest.

The heat flux distribution varies within the revolution due to changes in the contact patterns and oil distributions. At each phase, integrating the heat flux within the whole sealing band results in the total frictional heat at that phase. The coefficient of friction at that moment, defined as the ratio of the total frictional force to the normal load, can then be derived. The variations of the coefficient of friction within one rotation are also shown in Figure 2.22(b).
Figure 2. 22(a) heat flux distribution at a certain phase of a rotation, and (b) integrated coefficient of friction with respect to time under normalized load of 2.2 N/mm

**Seal temperatures**

The heat transfer sub model calculates the temperature distributions of both the stator and the rotor. While temperature distributions (or gradients) help to examine the surface deformations by twists, seal surface temperatures are more important to study seal performance in terms of scoring behavior. The distributions of the seal surface temperatures are shown in Figure 2.23(a). To better visualize the temperature profiles in the sealing band, 2D temperature distributions in the radial direction at three circumferential locations of \( \pi/6 \), \( \pi \), and \( 11\pi/6 \) are extracted and demonstrated in Figure 2.23 (b), (c), and (d). Again the high temperature regions correspond to the contact areas.
Figure 2.23 (a) 3D seal surface temperature distribution, (b) 2D seal surface temperature distribution at $\theta = \frac{\pi}{6}$, (c) 2D seal surface temperature distribution at $\theta = \pi$, and (d) 2D seal surface temperature distribution at $\theta = \frac{11\pi}{6}$.

2.5.2 Results from the unsteady state model

The results presented here are also for the seal from the process 2. The operating conditions are such that the axial face load is fixed at 2.2 N/mm, and the speed ramps up to the pre-defined value (1150 rpm) at an acceleration rate of 40 rpm/s.

Contact pressures

Figure 2.24 shows the contact pressure development during the acceleration process as well as the steady state process. Initially the undeformed surfaces are concave; contact takes place near the OD of the seal at low speeds. With increase in rotation speed, excessive frictional heating results in greater thermal
twist, which changes the radial surface profiles from concave to more convex. As a result, the contact moves inward to the ID of the seal.

![Contact pressure distribution](image)

Figure 2. 24 (a) contact pressure distribution at the 1st rotation; (b) contact pressure distribution at the 6th rotation; (c) contact pressure distribution at the 16th rotation; (d) contact pressure distribution at the 48th rotation.

**Oil exchange**

The unsteady state lubrication sub model provides not only cavitation information but also oil distribution and oil exchange rate at instant time. The oil exchange rate is defined as the oil flow in the radial direction, which is driven by the radial pressure gradient. The exchange rate is crucial for seal performance, especially in the contact areas. The exchange rate is very small due to blockage of surface roughness. For better visualization, only the areas, where the oil
exchange rate is significantly small (less than $1 \times 10^{-20}$ m$^3$/s in this case), are indicated by light color in Figure 2.25. At the beginning of the rotation when the speed is small, the oil hydrodynamic pressure has yet to be built up. As a result, the pressure-driven oil flows are extremely small, and a large portion of the sealing band (42.3%) has small oil exchange rate. With increase in the rotation speed, the oil pressure starts to build up, which leads to enhancement of the oil exchange rate. As shown in Figure 2.25(b), after 100 rotations, only 27.3% of the sealing band has very small oil exchange rates. The distribution of the oil exchange rate stabilizes after the thermal steady state is achieved (as shown in Figure 2.25(c)). This is because the oil viscosity is strongly dependent upon the oil temperature.

![Oil exchange rates after (a) 1 rotation, (b) 100 rotations, (c) 400 rotations](image)

**Figure 2.25 Oil exchange rates after (a) 1 rotation, (b) 100 rotations, (c) 400 rotations**

**Seal temperatures**

The advantage of the transient heat transfer sub model is the ability to calculate the temperature development during the transient processes and temperature fluctuations within one revolution. Figure 2.26 demonstrates evolution of the seal surface temperatures, starting with the room temperature. The high temperature areas also indicate the regions where asperity contacts take place. After roughly 300 rotations, the surface temperatures stabilize.
Even after the thermal steady state is reached, the frictional heating on the stator surface is changing with respect to time, due to constantly changing contact patterns. The temperature fluctuation at one point on the stator surface is illustrated in Figure 2.27.
2.6 Summary

A quasi-steady state model and an unsteady state model have been developed in attempt to explore the operation mechanism of the FMMFS. Both models consider interactions among the surface deflection, boundary lubrication in the sealing band, and heat transfer in the seal. Some assumptions which were taken to break the couplings were justified. Three sub models simulating different physical processes were explained at length.

Both models take the real surface profiles from measurements, and output the contact pressure, the surface clearance, the partial film content, the oil exchange rate, the friction coefficient, and the seal temperature. It is worthy to point out that both models have the capability to simulate dynamic oil transport process. Additionally, the unsteady state model is able to evaluate seal temperature development during the transient processes.
Chapter 3

EXPERIMENTAL STUDY AND MODEL VALIDATION

This chapter intends to:

- Introduce the experimental setup;
- Demonstrate some important experimental results;
- Validate the numerical models by comparing two important quantities: coefficient of friction and seal temperatures.

3.1 Experimental setup

3.1.1 Assembly of the FMMFS set

The FMMFS set consists of two identical elements; and each element consists of a cast-iron seal ring and an elastomeric (rubber) O-ring. These four separate parts, if installed correctly, move independently to each other. The assembly procedure of the FMMFS set is illustrated in Figure 3.1.

Firstly the seal pair and O-rings are removed from original wrapping. The housing ramps, the seal ramps and the rubber rings are then thoroughly degreased with alcohol, and wiped dry. The rubber ring is then rolled onto the ramp of the seal. With aid of an installation tool, the seal pair with the rubber rings on is inserted into two separate fixtures, namely stationary and rotary fixtures. Make sure two seals sit with faces parallel to each other. After the two seal halves are in the housings, the faces should be thoroughly cleaned with a lint free cloth and a thin film of oil (SAE 30 for example) applied to only the faces to prevent initial dry contacts. The FMMFS set is now ready to be closed together in the assembled unit. By moving the base plate, the stationary fixture moves downward and presses the seals against each other to the desired face load.
Care must be taken at all times when handling these seals. The seal rings are made of an extremely hard white iron alloy engineered specifically for wearability and corrosion resistance. As a result the rings are very brittle and need to be handled with care.

After seals are completely closed by external force, the oil pump is turned on. The lubricant oil is circulated into the cavity through the pipe in the middle of the stationary fixture. The oil flushes downward and bounces back at the bottom of the cavity. Then the oil flow exits from the annular tube between the pipe in the middle and the inside surface of the stationary fixture. Draining above level of seals ensures faces are always submerged in oil. The oil flow rate is measured by a flow meter and controlled by a variable frequency drive on the pump motor by adjusting the pump speed to match the desired flow rate. After the oil pressure

Figure 3.1 Assembly of the FMMFS set
stabilizes, the motor is turned on and drives the rotating fixture to spin at a predefined rate.

### 3.1.2 Thermocouple arrangements

Different thermocouple arrangements are made to measure the seal temperatures resulting from frictional heating. One method is to put the thermocouples on the flange at the OD of the seal ring. The advantage of this method is simplicity, and without introducing thermal effects on the contacting region. Two type J thermocouples are used, one intentionally being at the peak and the other the valley of the waviness, to check possible temperature variations in the circumferential direction. The location of the thermocouple $T_f$ is demonstrated in Figure 3.2.

![Figure 3.2 Location of thermocouple $T_f$](image)

The flange temperature $T_f$ can be used to study normal behavior of seal operation. However, surface or/and near-surface temperatures in the contacting region are always desired, especially for scoring analysis. Many different surface temperature measurement techniques have been used with some degree of success. For example, in the case where two sliding materials are different, dynamic thermocouples can use the contacting bodies or part of the bodies, as the thermocouple elements, and use the contact area as the measuring junction.
However, questions persist about the accuracy of the measurement. Tian and Kennedy [34] developed a thin film thermocouple (TFTC) to measure actual contact temperatures (flash temperature). The thermocouple devices were made from thin films of vapor-deposited copper and nickel and had extremely rapid (<1μs) response to a sudden temperature change, due to extremely small junction size and mass. In spite of capability of measuring flash temperature, fabrication of TFTC is quite complicated and expensive. Interference of surface asperities is also in question.

Using embedded thermocouple techniques (ETC) is the most common way to measure the surface temperature without interfering with the real rubbing surfaces. If ETC is fine enough and close enough to the contacting surfaces, the measurements could give a good indication of transient changes in frictional heat generation. Some doubts raise on the limited response time. However Rabin and Rittel [35] examined the time response exclusively for the solid ETCs. They found that unlike the transient response of a fluid immersed thermocouple, the transient response of a solid ETC is solely governed by the radius of the thermocouple and the thermal diffusivity of the measured metal, regardless of the thermal diffusivity of the thermocouples. Rittel [36] then measured transient temperature changes in polymers using the ETC with the response time on the order of 10 μs. The results proved feasibility of this technique on transient temperature measurements.

In this thesis, the ETC technique uses 44 gauge type T (copper constantan) thermocouples. The wire diameter of the thermocouple is 0.05 mm, and the maximum diameter with tubing around the bead is 0.28 mm. The main purpose of the tubing is to electrically isolate the thermocouple from the measured metal. The response time, from the Rabin and Rittel's model, is roughly 50 μs.

Two thermocouples are used and intentionally arranged in the way as shown in Figure 3.3. One thermocouple is located at the ID of the sealing band; while the other in the middle. As mentioned in the previous chapter, the seal ring
normally has a sinusoidal shaped waviness in the circumferential direction. In an attempt to capture possible temperature variations along circumference, the thermocouple sitting at ID is also arranged at the peak location in the circumferential direction; while the one in the middle of the sealing band is at the valley. The relative dimensions are also indicated in Figure 3.3.

To mount the fine gauge thermocouples, step holes of 0.4 and 0.8 mm diameter were machined using laser technique. Application of the laser technique tends to minimize the thermal effects on the contacting surfaces. The details of the thermocouple holes are exhibited in Figure 3.4. The thermocouple is fed through all the way to the bottom of the hole, and fixed in place with epoxy such that the junction is about 0.2 mm below the contacting surface.

![Figure 3.3](image_url)  
(a) top view of a 92 mm seal ring, (b) circumferential profiles at r=45mm, (c) thermocouple hole at the peak location indicated in (b), and (d) thermocouple hole at the valley location
3.1.3 Test procedures

Two test procedures are used in this thesis. The first procedure (or Proc I) tends to cover the wide range of operating conditions, i.e. loads and speeds. The procedure with changes of rotating speeds and loads is illustrated in Figure 3.5. There are four cycles in this procedure. For each cycle, the axial load is kept fixed and the rotating speed continuously ramps from 0.6 m/s* to 3 m/s by increment of 0.8 m/s, with each speed running for 5 minutes. After that, the cycle is repeated with the axial load increased by 0.5 N/mm until the load reaches to 2.0 N/mm. Sixteen different operating conditions form one complete test procedure. Experiments on oil temperature effects and lubricant effects (such as SAE10W and SAE50) follow this procedure.

* the linear speed base on the inner diameter of the seal ring.
The second procedure (or Proc II) is designed to investigate the torque variations during operation. The face load remains at a constant value of 2.2 N/mm. At one specific speed, the run goes through acceleration, steady operation, and deceleration processes. The acceleration/deceleration rate is 40 rpm/s. Duration of the steady operation is 5 minutes. The procedure first goes through the break-in process, i.e. 50 rpm, 75 rpm and 100 rpm. After that, speed increases by increment of 100 rpm, all the way up to 800 rpm. The schematic of the test procedure II is shown in Figure 3.6.
3.2 Experimental results

3.2.1 Consistency of test facility

Three tests were run beforehand to check consistency of the test facility. The seal pair used in all three tests is the same and well broken-in, and it is assumed that minimum wear occurred after each test. The lubricant oil is SAE 10W. The first test followed the procedure I. The second test repeated the first test with the same operating conditions. The third test used the reversed procedure of Proc I. In another word, instead of starting with the lower load and ramping up the speed, this test started with the highest load (2.0 N/mm) and highest speed (3.0 m/s), then decreased the speed by a decrement of 0.8 m/s, all the way to 0.6 m/s. After that the cycle repeated for a lower load until the face load 0.5 N/mm was reached. Figure 3.7 shows measurements of coefficient of friction (COF) for these three runs. The COF is defined as the ratio of frictional torque to thrust measurements. Each cycle, representing a load level, is separated by a red dot line. As one can see from the figure, both magnitude and variation of COF are very close under the same operating conditions for three cases. That implies that the test facility is consistent and repeatable.
3.2.2 Tests under different operating conditions

Seal operation under various conditions was investigated for different seal pairs. Figure 3.8 shows the COF measurements and corresponding temperature $T_f$ in a typical test with Proc I. The oil temperature was set to 30°C and the oil flow rate was 4 g/m. The seal used in the test is a seal from the polishing process 1 with small amplitude of waviness. No leakage was observed in the duration of the test. That suggested that even under the load 0.5 N/mm, the two seal surfaces conformed and continuous contact formed along the circumference. As shown in the figure, except for the load 0.5 N/mm, the friction coefficients for all other loads asymptote to the value of 0.1. Under the face load of 0.5 N/mm, the COF is relatively higher for all speeds. The elevated values of COF suggest that there is a greater portion of friction comes from viscous shearing besides the asperity contacts due to the large viscosity in this scenario. Temperature $T_f$
shown in Figure 3.8 (b) corresponds to the total heat flux which is input into the seal system.

![Graphs showing COF and temperature over time](image)

**Figure 3.8 Test results of (a) COF, and (b) temperature Tf with Proc I**

Significance of the oil shearing to the frictional heating generation, especially under the face load of 0.5 N/mm, is further confirmed by the case where the oil temperature is elevated to 100°C. The test result of COF is exhibited in Figure 3.9. The mean value at each operating condition is also indicated in the figure as a red cross. As one can see, the friction coefficient is at the same level over the entire range of operating conditions. The explanation for this is that the oil viscosity strongly depends upon the oil temperature, and viscosity variation is relatively small in that temperature ranges of interest. Therefore, oil viscous shearing contributes a similar amount to the total frictional heating. That would lead to the similar values of the friction coefficient in the boundary lubrication regime.
3.2.3 Lubricant effects on seal application

Different lubricants were also tested to examine their applicability on the FMMFS. The major difference of lubricants lies in the viscosity and application scenario. Typical characteristics of three lubricants (i.e. SAE 10W, SAE 50 and Optigear) are listed in Table 3.1. Comparisons of COF measurements using these three lubricants are also shown in Figure 3.10. The discrepancy in COF, especially under face load 0.5 N/mm, resulting from the difference of viscosity is clearly demonstrated in Figure 3.10 (a) and (b). With increase in oil temperature at higher operating conditions, the difference becomes smaller and smaller due to convergence of viscosity. However, it is noted that Optigear oil, which performs superiorly in gear applications, behaves remarkably worse than other lubricants. The reason probably lies in the fact that the Optigear oil does not favor the applications in which the contact pressures are relatively small.
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</tr>
<tr>
<td>cSt@100°C (ASTM D445)</td>
<td>6.1</td>
<td>18.0</td>
<td>19.8</td>
</tr>
<tr>
<td>Viscosity index (ASTM D2270)</td>
<td>97</td>
<td>96</td>
<td>98</td>
</tr>
</tbody>
</table>

Table 3.1 Typical characteristics of SAE 10W, SAE 50, and Optigear

Figure 3.10 COF measurements for (a) SAE 10W, (b) SAE 50, (c) Optigear
3.2.4 Transient test

There were observed fluctuations of the frictional torque measurements, which are believed to be due to the inherent variations of contact patterns and oil distributions. In order to examine the torque variation within revolution, higher resolution load cell (specifically torque sensor) and higher sampling rate were required in the new test. The test procedure II was employed. Twenty tests in total have been conducted, with combinations of seals from both the process 1 and the process 2. The detail test arrangements are displayed in Table 3.2.

<table>
<thead>
<tr>
<th>Test No.</th>
<th>Seal Pair</th>
<th>Thermocouple Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Process 1 vs. Process 1*</td>
<td>Type J</td>
</tr>
<tr>
<td>5</td>
<td></td>
<td>Type T</td>
</tr>
<tr>
<td>6</td>
<td></td>
<td></td>
</tr>
<tr>
<td>7</td>
<td></td>
<td>Type J</td>
</tr>
<tr>
<td>8</td>
<td></td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>Process 1 vs. Process 2</td>
<td>Type T</td>
</tr>
<tr>
<td>10</td>
<td></td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>Process 2 vs. Process 1</td>
<td>-</td>
</tr>
<tr>
<td>12</td>
<td></td>
<td>Type J</td>
</tr>
<tr>
<td>13</td>
<td></td>
<td></td>
</tr>
<tr>
<td>14</td>
<td></td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>Process 2 vs. Process 2</td>
<td>-</td>
</tr>
<tr>
<td>16</td>
<td></td>
<td></td>
</tr>
<tr>
<td>17</td>
<td></td>
<td></td>
</tr>
<tr>
<td>18</td>
<td>Process 1 vs. Process 2</td>
<td>Type T</td>
</tr>
<tr>
<td>19</td>
<td>Process 2 vs. Process 1</td>
<td>-</td>
</tr>
<tr>
<td>20</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

* The first seal is the stator and the latter is the rotor.

Table 3.2 Seal arrangements of the unsteady operation test
Coefficient of friction

Figure 3.11 (a) shows the torque measurements for test #4 in a typical test cycle. Within the duration of the entire run, the average values of the torque readings over the speeds are more or less similar; while amplitude and patterns of torque variations differ from speed to speed. Figure 3.11 (b) and (c) display the torque variations within two revolutions for 50 rpm and 100 rpm respectively. As one can see, the frictional torque pattern is complex and periodic; and the pattern differs with the rotation speed as well. The variance and complexity of the pattern are assumed to be caused by the oil distributions.

Figure 3.11 (a) Torque measurements in an unsteady operation test with Pro II, (b) torque variation at 50 rpm, (c) torque variation at 100 rpm

Overall, the FMMFS operates in the boundary lubrication regime under the normal conditions. This is also validated by the unsteady operation test. As we already show in the Figure 3.11(a), the variation of frictional torque with speed changes little within the duration of the entire test. For a better view of this
validation, a standard Strubeck curve was generated for all the tests with different seal groups. It is assumed that the width of the contact band for all the cases is roughly 1 mm, so the averaged contact pressure is explicitly represented by the face load. For instance, if the face load is 2.2 N/mm, as in this case, the average contact pressure is 2.2 MPa. Keep in mind also that the viscosity is a strong function of the oil temperature. Since we are unable to directly get the oil temperature, the averaged temperatures sensed by the two thermocouples in the sub-surface are used as an approximation for the oil temperature. The resulting relation between the COF and the Sommerfeld number is provided by Figure 3.12. The values of the friction coefficients cluster around 0.1 with a wide span of the Sommerfeld number. It is noted that the values shown in the figure are the mean values after the steady state is reached.

![Figure 3.12 Strubeck curve for the entire test](image)

**Surface temperatures**

The test tends to measure the surface temperature variations during one revolution. However, due to limitations on resolution of hardware of the data acquisition system, details of temperature variation are not available at this time. Still, the steady state temperature is able to be monitored. Figure 3.13 shows the
temperature recordings during one test cycle. The locations of the temperature sensors are also indicated in the figure. This test reveals that the temperature at the middle of the flat band is always higher than that at the ID of the seal, even though the temperature sensor at the middle is sitting in the valley location with respect to the circumferential direction. And the discrepancy between two thermocouple readings becomes larger with the rotation speed. This behavior can be explained using the second operating theory stated in Chapter 1. That is, the flat band of the FMMFS would become convex due to the thermo-mechanical twists. When the speed is low, the thermal deflection is not sufficient to alter the concavity of the radial profile; contact takes place close to the OD of the seal. Both thermocouples are far away from the concentrated heat source, giving the similar temperature readings. With increase in the rotation speed, the heat flux input increases, leading to larger thermal twists. The thermal twists change the radial profile from concavity to convexity gradually, moving the contact points radially inwards. However the contact area never moves beyond or very close to the ID of the seal since the temperature close the ID is always lower.

![Figure 3.13 Temperature measurements in a transient test](image)
3.3 Comparison between experiments and simulations

The numerical models have to be validated before being put into real applications. The experimental results, for this purpose, serve as a benchmark for the model validation. Two measurable quantities, namely friction coefficient and seal temperature, are used for comparison. As mentioned before, the friction coefficient is dependent upon the asperity contact and partial film contents. By matching the overall trend and detail variations of the friction coefficient, the contact and lubrication sub models can be justified. And comparison of the seal temperatures can justify the heat transfer sub model. Surely in order to validate the contact model, a separate effort has to be done. In the following sections, the detail comparisons between the experimental and the simulation results from the quasi-steady state model will be addressed.

3.3.1 Sensitivity study

There are always challenges to simulate the "real" physics of seal operation based on some simplifications made in the modeling. The following sections will introduce some influence factors and explain their effects on simulation results. All the results, if otherwise stated, are obtained from the Test #2 and under the working conditions of the face load 2.2 N/mm and the rotation speed 100 RPM.

Boundary and dry friction coefficients

Both the boundary and dry friction coefficients are empirical values, which are referred from the experiments and literature respectively. The friction coefficients have direct effects on the friction prediction and the resulting seal temperatures. The frictional torque increases with increasing coefficients in general, even though the increasing temperatures offset the oil shearing a bit. The seal surface temperatures increase accordingly. Figure 3.14 shows the results.
Surface roughness (Rq and flow factor)

The roughness (rms) is a very crucial parameter in the numerical model and has substantial effects on results of the contact pressure, the cavitation, thereby the surface temperatures. In the simulation, the roughness is taken as a constant value over the entire sealing band. Even though it is a reasonable argument due to the assumption of the seal surface being homogeneous, the roughness has deviations which may be up to 20%. Roughness effects on the frictional torque are illustrated in Figure 3.15. The breakdown of the total torque is also shown in the figure. With increasing roughness, reduction of the total frictional torque is mainly because of decrease of the dry friction. As roughness increases, the nominal surface clearance increases to balance the axial load since the contact pressure is a function of the dimensionless variable h/σ. Higher surface clearance will result in less oil flow blockage, therefore less dry areas. In the meantime, cavitation effects decrease with increasing roughness [37]. Less dry areas also means more "wet" areas, so the torque due to the boundary friction increases a little. Viscous shearing is determined by the competition between the "more" wet area, which promotes more viscous shearing, and greater absolute surface clearance, which reduces the viscous heat generation. In this case of interest, the latter takes the lead, causing slight decrease in the viscous torque. In general, the total frictional torque decreases with increasing roughness.
Figure 3.15 Roughness effects on the total frictional torque and its three contributors

As mentioned in Chapter 2, the rough surfaces will also affect the oil flow rate. For this reason, a flow factor method is introduced to derive the average Reynolds equation. The details of how to get the flow factor curves for each seal group were explained at length in Section 2.2. It is noted, however, that the pressure flow factor curves, generated from a so-called small representative patch, may vary a great deal at different locations of the sealing surface. Furthermore, the flow factor is determined by the equivalent surface or combination of the two rubbing surfaces. Because of the relative movement, generation of the flow factor curve should in fact a dynamic process. Figure 3.16 displays the two pressure flow factor curves obtained from the different locations of the sealing band, and corresponding frictional torque calculations based upon these two flow factor curves. The roughness is in the circumferential direction because of specific manufacturing processes. This unique micro geometric feature will impose blockage effect on the oil transport in the radial direction, in particular at low surface clearance. When more blockages occur due to smaller oil flow rate correction, more dry area results, causing higher frictional heat generation.
Viscosity

It is well known that viscosity of lubricant has significant impacts on viscous heating generation and cavitation. The preliminary study [27] showed that due to the air boundary condition at the OD of the seal, air can easily access to the lubricant oil and transport much faster than the oil in fact. Once the air is entrapped in the oil, it is hard to escape. As a consequence, air entrainment may decrease the viscosity of lubricant tremendously even in the “flooded” areas. The reduced viscosity will decrease the frictional torque significantly. Figure 3.17 demonstrates the effects of different viscosity coefficients on the total frictional torque.
3.3.2 Values of the baseline parameters

Uncertainty of the baseline parameters, such as heat transfer coefficients, the roughness, the pressure flow factor, and the oil viscosity etc., will bring deviations from reality. To make a valid comparison, values of the baseline parameters should be properly selected. Firstly the heat transfer coefficients (HTCs) on the oil side were calculated using ADINA™. At the rotation speed of 900 RPM, The HTCs are 900 W/(m²°C) and 6600 W/(m²°C) for the stator and the rotor respectively. On the air side, the HTC of the rotor is taken as 38 W/(m²°C); and that of the stator is 10 W/(m²°C). The HTCs at the other speeds are correlated with the rotation speeds as a power function with the exponent of 0.85.

Values of 0.09 and 0.4 are used as the boundary friction coefficient, \( f_b \), and the dry friction coefficient, \( f_d \), respectively. The number of \( f_b \) is an asymptotic value of boundary lubrication tests; while the value of \( f_d \) is obtained based on the literature.
As explained in the previous chapter, the equivalent roughness, $\sigma_q$, is calculated from the small patches of two contacting surfaces. In this study there are four patches measured for each seal at four different circumferential locations. Therefore, sixteen equivalent surfaces in total can be formed with 16 possible values of roughness. Then the average value is taken as the input to the model. The pressure flow factor curves are also generated from the equivalent surfaces. There are also 16 pressure flow factor curves; and a most representative one is employed in the model.

It is known that the oil viscosity (SAE 30 for instance) depends highly on oil temperature. The relation between kinematic viscosity and the oil temperature follows the Vogel equation. Take SAE 30 as an example, the oil viscosity as a function of the oil temperature is expressed as,

$$ v = 2.894 \times 10^{-5} \exp\left(\frac{1432.29}{132.94 + T}\right) $$  \hspace{1cm} (3.1)

where $v$ is the kinematical viscosity and $T$ is the oil temperature.

### 3.3.3 Coefficient of friction

Thirteen cases have been simulated and results of friction coefficient are compared with the experiment data. Figure 3.18 shows the overall comparisons between the simulations and experiments under the same operating conditions. The friction coefficient is the averaged value after the steady state is reached. Note that in order to validate comparison, uncertainties from both simulation results and experimental measurements should be considered. The factors which affect the results of COF have been explained in the previous sections. As regard with uncertainty due to the measurement instrument, refer to Appendix A. As one can see from the figures, except for the low speeds, where break-in and wear may take place in some cases, calculated friction coefficients agree well with the experiment measurements in general.
Figure 3. 18 COF comparisons between simulation and experiment under the same operating conditions

The comparison is also presented on COF variations with the rotation speed for individual tests, as shown in Figure 3.19. Four typical tests, each representing one combination of seal groups, are listed. From baseline point of view, the numerical simulation generally overestimates the COF in terms of magnitude for all the seal groups. However, the simulation results demonstrate the fair similarity across the speeds. And the differences between simulation and experiment are consistent except for few cases, such as at some low speeds where the seal from the process 2 running against each other. By investigating those cases, it is not hard to find that such inconsistency is due to the break-in process, in which moderate wear may take place and it is signaled by spiking of the torque and temperature readings.
Figure 3.19 Comparison of COF with speeds

As we already shown in the experimental results, there observed variations of the frictional torque measurements within one revolution. The variations are considerably complicated, and show periodic patterns (as shown in 3.11). These complex periodic patterns are also predicted by the numerical model. Figure 3.20 shows the comparison of detailed friction variations within two revolutions for four different seal groups under different speed levels. Because of differences in the magnitude of COF, all the simulation results are shifted down on purpose by a certain value for better comparisons. One can see from the figures that the patterns of the frictional torque vary a great deal from case to case. The discrepancy is resulted from various surface geometrical features, thereby resulting different contact patterns and oil distributions. As shown in the figures, remarkable similarity has been achieved, which implies that the current model is able to predict some physics of seal operation. Despite of some success, there
are still some cases in which the similarity is not achieved, suggesting that future efforts are needed for further model improvements.

![Graphs showing COF comparison](image)

**Figure 3.19** Comparison of detailed friction variations within two revolutions for different seal groups

### 3.3.4 Seal sub-surface temperature

Within the manufacturing specifications, the width of the flat band of the seal ring ranges from 2.5 to 3.0 mm. However, the model always takes the width of the flat band as 3.0 mm. The possible discrepancy leads to hardly match the thermocouple location precisely in the radial direction. In addition, the laser technique, which is used to drill holes for holding thermocouples, has difficulty in controlling the hole depth. Both challenges impose some difficulty in precisely locating the thermocouples. To consider the possible range of the thermocouple locations, the temperature was calculated corresponding to the range. Figure 3.20(a) demonstrates schematically the possible locations of the thermocouple
T1 and the corresponding temperature range calculated under the different operating conditions are shown as shaded areas in Figure 3.20(b).

![Diagram](image)

**Figure 3.20** (a) Possible locations of the Thermocouple T1, (b) corresponding temperature range calculated by the numerical model

With consideration of the temperature range, both temperature T1 and T2 are compared with the experiment measurement on the case basis. Since there are no thermocouple arrangements on the seal with the process 2, only two combinations of the seals, with the seal from the process 1 as the stator, are compared. Figure 3.21 exhibits the temperature comparisons with rotation speed for the seal pair from the process 1 as well as the seal pair with one from the process 1 and the other from the process 2. The temperature range is also indicated in the figure. As expected, both temperatures are overestimated by the numerical model. That is primarily because of higher frictional torque prediction, causing higher heat flux to the seal system. Despite of overestimation of temperature magnitude at each individual speed, the overall trend agrees very well between the simulation and the experiment.
Figure 3. 21 Comparisons of temperature T1 and T2 (a) for the seal pair from the process 1; (b) for the seal pair with one from the process 1 and the other from the process 2

The overestimation of the temperatures is the consequence of the friction overestimation can be better explained by the relation of temperatures and the total frictional heating. By putting all the experiment data and simulation results together, it is obvious that for both temperatures T1 and T2, under the same total frictional heat, simulated temperatures agree very well with the experiment. The
results are shown in Figure 3.22. This suggests that the current thermal model is valid.

Figure 3.22 Relation of temperature and the total frictional heating for (a) temperature T1, and (b) temperature T2
3.4 Summary

In this chapter, the vertical small seal simulator was introduced. The major components of the VS$^3$ were described as well. Two tests have been conducted in individual efforts. One test attempted to study the seal performance in the steady state under the various operating conditions; and the other aimed to investigate the unsteady behaviors of the seal operation. The steady state study verified that the FMMFS operates in the boundary lubrication regime under the normal operating conditions. Furthermore, oil shearing makes a significant contribution to the total friction heating, especially at lower face loading. While the unsteady study revealed that the FMMFS presents a complex and periodic friction variation within one revolution. The detailed friction pattern differs with the surface geometrical features as well as the rotation speed. The variance of the friction pattern is due to the various contact patterns and oil distributions.

The results from the experiments were then used as a benchmark to compare with the simulation results from the numerical model developed in the previous chapter. The comparisons have been undertaken through two important quantities: friction coefficient and seal temperature. In general speaking, the current numerical model is able to predict the seal behaviors with some success. Even though the current model overestimates the COF by roughly 20%, the overestimation is consistent over the tests and remarkable similarity in detailed friction variation has been achieved in some cases.
Chapter 4
APPLICATIONS

This chapter aims to:

- Explain different scoring behaviors qualitatively and identify surface features contributing to such differences;
- Predict possible leakage failure and provide feasible explanations;

4.1 Scoring failure

4.1.1 Scoring failure test procedure

A design of experiment (DOE) test has been conducted in the sponsor's effort for scoring failure analysis. The procedures of the scoring failure tests are as follows. First, the seal pair is inserted into the fixtures. Then two seals are drawn together by moving down the flat plate and the face load is maintained as a constant value of 2.2 N/mm. The test firstly runs through the break in process, i.e. from 50 rpm all the way to 100 rpm, in steps of 25 rpm. Then starting from 100 rpm, the test continues to run all the way to the maximum speed of 2100 rpm or until failure, which ever occurs first, in steps of 50 rpm. At each speed, the test runs both in forward and reverse sequences. The detailed test sequence for a certain speed is explained as follows. At first, the test is accelerated from 0 rpm to a predetermined speed at a predetermined acceleration rate (40 rpm/s in this case). After the predetermined speed is reached, the test runs forward at this speed for 240 seconds. Then the deceleration process triggers, reducing the speed to 0 rpm at the predetermined deceleration rate (40 rpm/s also). The test then dwells for 60 seconds. After that, the test starts to speed up at the same acceleration rate, in the reverse rotation. Once the predefined speed is reached, the test keeps running for another 240 seconds. Then the test experiences the same deceleration and dwell processes as those in the forward rotation. Finally the test ramps up to another speed and the above sequences repeat. Note that
the acceleration and deceleration time should not be included in the forward, reverse nor the dwell time. The test sequence for a predetermined speed is illustrated schematically in Figure 4.1.

![Diagram of test sequence](image)

**Stage 1:** 0 rpm  
**Stage 1 - 2:** Accelerate to 50 rpm (or any speed) at 40 rpm/sec  
**Stage 2 - 3:** Forward Time, usually 240 sec at 50 rpm  
**Stage 3 - 4:** Decelerate to 0 rpm at 40 rpm/sec  
**Stage 4 - 5:** Dwell Time, usually 60 sec 0 rpm  
**Stage 5 - 6:** Accelerate to -50 rpm (or any speed) at 40 rpm/sec  
**Stage 6 - 7:** Reverse Time, usually 240 sec at -50 rpm  
**Stage 7 - 8:** Decelerate to 0 rpm at 40 rpm/sec  
**Stage 8 - 9:** Dwell Time, usually 60 sec 0 rpm

**Figure 4.1 Representation of the test sequence for a predetermined speed during a scoring failure test**

**4.1.2 Analysis of scoring failure**

Four representative seal pairs were selected for scoring failure analysis. The surface conditions and failure speeds of the selected seal pairs are listed in Table 4.1. It is seen that the failure speed spreads a large range from 900 RPM to 1850 RPM.
Table 4.1 Selected seal pairs for the scoring failure analysis

Surface temperature

It is well known that scoring is a very complicated phenomenon and is affected by lots of factors. Among these factors, the surface temperature is still believed to be the most critical variable to break down the protective films and initiate scoring. Therefore, the surface temperature is still the focus of interest for scoring analysis. Table 4.1 lists measured flange temperature as well as calculated max surface temperature right before scoring takes place for the four representative seal pairs. It is obvious that the seal pair in case 3 survives through the highest speed and highest surface temperature; while the seal pair in case 4 fails miserably at premature speed of 900 RPM and low surface temperature as well. It is observed that the temperatures for different cases span a considerably wide range (~60°C), which means that the surface temperature solely can not be a “universal” criterion to determine whether the scoring occurs or not. There must be some other factors involved to affect such temperature. Those factors include contact wetness and oil exchange in the contact area.

Contact wetness

There is no doubt that the scoring always initiates at the contact spots which experience intense frictional heating and high temperatures. Since the scoring is a transient process, an instant spike of temperature of some contact spots may not be catastrophic if their surrounding conditions are favorable. As we know, due to the cavitation effects, mixture of oil and vapor/air is present in the cavitation zones. Since the oil can serve as an effective coolant in the sealing
band, the greater amount of oil is certainly beneficial to relieve the heat intensity of the contact regions. Furthermore, oil molecules have better chance to adsorb on the surface, re-forming the protective film if more oil is around. Therefore, how much oil within the contact regions will somehow impose a limitation on how high of the surface temperature the seal can tolerate. In another word, the more oil content is available around the contact asperities, the higher temperature the asperities can tolerate. This concept is represented by the contact wetness, which is defined as the average partial oil amount per unit area within the contact region.

Figure 4.2 illustrates the contact wetness for the four cases. For each case, three specific instances within one revolution are presented. At a certain instance, distribution of the oil amount is only shown in the contact regions. The oil partial density is then integrated along the circumference at each radial location. And the average value is plotted out versus the radius, reflecting the oil amount in the contact region. The radial range of contact is also indicated in the figure.

Among the four cases, the case 3 (Figure 4.2(c)), in which the seal pair failed at 1850 RPM, has the least cavitation effects and largest average partial density, especially in the contact areas. The larger amount of the oil content around the asperities helps the seal pair to operate through the higher speed and survive the higher surface temperature. The second best behaved seal pair in terms of contact wetness is the case 4 (as shown in Figure 4.2(d)). The average partial density is about 0.7~0.8 within the contact areas. As pointed out in the previous section, this seal pair failed at the lowest speed and the lowest surface temperature. Therefore, there must be another factor(s) to trigger it to fail at a much premature stage. Both case 1 and case 2 have relatively small partial density within the contact regions. And the case 1 has the smaller contact area than the case 2 over one revolution.

The primary reason for the large discrepancies of the contact wetness for the above four cases are due to the cavitation effects. We already know that the
cavitation effects increase with increasing waviness, increasing oil viscosity, increasing sliding speed, increasing asperity radius, decreasing roughness, and decreasing pressure relative to the cavitation pressure. Since the oil viscosity is dependent upon the oil temperature, and the oil temperature is affected by the total frictional heating, which is approximately proportional to the sliding speed, decrease in the oil viscosity and increase of the sliding speed can be somehow offset. Otherwise is the same, waviness and surface roughness are the most critical features to influence cavitation phenomenon. As mentioned in Chapter 2, the rubbing surfaces conform under the normal face load, the initial waviness does not sustain after deformation. Therefore, the roughness is the only major contributor to the difference of the cavitation effects. Both case 1 and case 2 have smaller roughness. Their equivalent rms roughness is 0.062µm and 0.075µm respectively. While case 3 and case 4 have equivalent rms roughness 0.14µm and 0.12µm.

(a) Case 1
(b) Case 2

(c) Case 3
Contact wetness within one revolution for four cases

Oil exchange

Oil will experience thermal degradation, especially in the high-temperature areas. Replacement of the degraded oil by the fresh oil is, therefore, essential to ensure normal operation of the seals. Considering oil supply at the inner surface of the seal as well as longitudinal roughness directionality, oil transport in the radial direction is of interest. Fresh oil transport in and degraded oil out of the contact areas is represented by oil exchange rate. Obviously the larger the exchange rate, the higher temperature the seals can tolerate.

Figure 4.3 shows oil exchange rate for the four cases. Similar to representation of the contact wetness, three rotation instances within one revolution after the steady state are presented for each case to reflect variations. At each instance, distribution of the oil exchange rate is shown only in the contact areas. Integrating along the circumference at each radial location gives the magnitude of the oil exchange rate.

Due to the small magnitude, the distribution of the oil exchange rate is plotted in log scale. There is no oil exchange at all in some spots; log values for those
spots are unavailable. Therefore, for visualization purpose in the 3D image the oil exchange rate is set to a very small value ($10^{-20}$) in the spots with no oil exchange. In addition, to differentiate between the contact and no-contact areas, the oil exchange rate in the no-contact areas is set to $10^{-23}$. However, the true values are used in integration (in the 2D plot). Since the graph is plotted using the log scale, the locations where there are no oil exchange are indicated by broken segments of the line. Another point which needs to be stressed is that even though just few spots within the contact areas have considerable oil exchange at some instance, the exchanged oil can be carried over to the other contact locations by sliding motion in the circumferential direction.

Two extremes have been observed, i.e. case 3 and case 4. As one can see in Figure 4.3(c), the oil exchange is continuous across boundaries of the contact areas. Furthermore, the contact area is rather small and oil exchange within that area is active. Even discontinuity exists at some instance during the entire revolution, from the dynamic process’s prospect, most contact asperities can be accessible by the fresh oil “pumped” from the inner cavity. Effective oil transport capability, especially in/out of the contact areas, is one of reasons to explain why the seal pair in this case failed at the highest speed and the highest temperature. In contrast, oil exchange rate is the worst in case 4. There is always a cutoff from the oil supply to the contact region. Although there is some oil exchange between the contact region and the areas close to the OD (as shown in the middle graph of Figure 4.3(d)), the exchanged oil is not “fresh”. As the degraded oil keeps operating, lubrication may only remain functioning for a short period of time before the protective films are broken down, causing scoring failure. The case 1 and case 2 lie in between. There are sometimes oil transport in or/and out of the contact area; while sometimes there is no oil exchange at all. Comparatively, the case 1 has the better oil transport performance than the case 2 since there are more times in the case 1 that the fresh oil is supplied into the contact region. In general, oil exchange rate can support the evidence on the different scoring behaviors.
Figure captions:
(a) Contact chemical exchange rate for Case 1 at different rotations.
(b) Contact chemical exchange rate for Case 2 at different rotations.
4.1.5 Micro surface geometric features

Variations of surface temperature, contact wetness and oil exchange rate may result from difference of the micro surface geometric features of the seal pairs. Even though they all have similar macro geometries, the micro structures differ greatly. Figure 4.4 shows the surface topography of equivalent seal
surfaces for the four cases. The lateral dimensions of each image are about 0.4 by 0.35 millimeters. It is clearly seen that the seals in both case 3 and case 4 have higher roughness than those in case 1 and case 2. As we already know, the extent of the cavitation effects is inversely proportional to the surface roughness. This is why we have higher contact wetness in the case 3 and case 4. The surface roughness also determines actual clearances between the two rubbing surfaces. The greater the roughness, the larger the surface clearance. The surface clearance is extremely important to the oil flow since it is proportional to the cube of the clearance. Another noticeable difference is asperity distributions caused by the different manufacturing processes. As one can see that the seal surfaces in the case 2 and case 4, formed by the process 2, have obvious ridges and troughs and asperities align with the circumferential direction. In contrast, asperity distributions for surfaces in the case 1 and case 3 with the process 1, are more random and dispersed. "Line-up" asperities help oil transport in the circumferential direction, but also function as a barrier to the radial flow. For the case 2 and case 4, it is more likely that contact concentrates on the asperities on top of one or few ridges and those contacts serve as blockage to the radial flow. Concentration of the contact also tends to increase the surface temperature, unfavorable for the scoring failure. On the other hand, the more “random” distributions of the asperities in the case 1 and case 3 lead to dispersed contact patterns, thereby facilitating the radial flow. It is no wonder that the seals in the case 1 and case 3 have better oil exchange performance.
4.1.6 Thermal instability

The current quasi-steady state numerical model has been proved to be able to qualitatively explain wide variations of the scoring failure speeds under the thermal steady state operation. Since the scoring failure is a transient thermal process, unsteady effects on the temperature development should be taken into consideration to properly predict initiation and propagation of the scoring failure. Therefore, the fully unsteady state should be employed to represent this concept due to the thermal instability.

The following assumptions are taken to simulate the scoring failure process. Firstly the critical surface temperature, which is the driving factor to break down the protective films, can be determined by the contact wetness and oil exchange rate. Once the critical temperature is exceeded locally, the local boundary friction coefficient increases immediately from 0.09 to 0.4 to reflect the surface film break-down. The protective film break-down leads to increasing frictional heat generation locally. The heat is then propagated with aid of surface sliding to the neighboring asperities, causing their temperatures to jump. When the critical
surface temperature covers a certain percentage of the surface area (which needs to be quantified in the future effort), the whole surface will experience a catastrophic damage, causing the seal to score.

Take one case as an example. The critical temperature is set to 210 °C on purpose such that there are only 0.23% of spots in the entire sealing band whose temperatures initially exceed this critical temperature. The demonstration of the initiation and propagation process of thermal instability during one revolution is illustrated in Figure 4.5. At the very beginning, only few spots have temperatures slightly higher than the critical temperature (Figure 4.5 (1)). To illustrate how local temperatures evolve with time, two temperatures are traced within one revolution. One is denoted as Ta with its initial temperature lower than the critical temperature. The other, denoted as Tb, is 120° behind the Ta (relative to the sliding direction). Its initial temperature is higher than the critical temperature. As shown in Figure 4.5(2), Tb quickly shoots up over 250°C when its local protective film is broken. This is due to local increase of heat generation and the elevated temperatures in the neighboring areas. Ta remains low since it is still far away from either of two high temperature regions. It is noted that the areas where the front of the high temperature regions sweep, are also brought up to the high temperature regions. Ta has a sudden jump when it is hit by one of the high temperature regions, as shown in Figure 4.5(3). Both temperatures are “stabilized” before two high temperature regions join (Figure 4.5(4)). When Tb is stricken by the front of the high temperature region behind it, it has a second jump (Figure 4.5(5)). Eventually the whole contact area experiences the elevated temperatures (Figure 4.6(6)). It is worthy to note that the whole process may not be completed because the high temperature regions may be sufficient enough at some stage of the revolution to break down the surface films, causing fatal damage on the surface.
Surface temperature distribution at t = 0.002 s

(1)

Surface temperature distribution at t = 0.003 s

(2)

Surface temperature distribution at t = 0.023 s

(3)

Surface temperature distribution at t = 0.036 s

(4)
4.2 Leakage failure

Model prediction

The unsteady lubrication sub model has capability of calculating dynamic oil transport, enabling prediction of oil leakage. Two seal pairs, namely LK1 and LK2, are selected for leakage failure analysis. Based on experimental observations, under the face load 2.2 N/mm, the seal pair LK1 leaked at a very premature speed, while the LK2 survived through a high speed. Numerical simulation has been conducted for these two seal pairs accordingly under the same operating conditions such that the face load is 2.2 N/mm and the rotation speed is 100 RPM. Oil flow out of the OD of the seal is then calculated and treated as oil leakage. It is noted that simulation overestimates the oil leaks since the model only allows outflow and disables inflow at the OD of the seal. By integrating the oil flow rate at the OD of the seal along the circumference, instant oil leaks result. Figure 4.6 shows the instant oil leaks within one revolution. Since the total amount of oil leakage is more of interest, the instant oil flow rate is then integrated with respect to time. This leads to the total amount of oil leaked after a period of time. Figure 4.7 shows the total amount of leaks after one revolution. It is clearly seen that the oil flow out of the seal pair LK1 is almost one order of
magnitude greater than that of the counterpart by the end of one revolution. This result agrees well with the experimental observations.

**Figure 4.6** Instant oil leaks through the OD of the seal within one revolution

**Figure 4.7** Integrated oil leaks within one revolution
Macro surface geometric features

Macro surface geometric features are investigated in attempt to understand the difference of oil leakage. 3D surface profiles of the surface pairs LK1 and LK2 are shown in Figure 4.8. Difference of the radial profiles in terms of concavity and convexity is clear. Refer to the Chapter 1 for the definitions of concavity and convexity. Generally speaking, concavity leads to contacts close to the OD, thereby preventing leakage. And convexity results in contacts at the ID, forming a “barrier” there and preventing oil flow into the sealing band. However, once oil passes that barrier, it has chance to escape from the sealing band. As one can see in Figure 4.8(a), the seal LK1 has combination of convex and concave radial profiles. And the concavity and convexity vary along the circumference. This unique geometric feature facilitates the radial oil flow in the following way. Firstly the oil supply at the inner cavity of the seal is “pumped” into the sealing band at the locations with concave radial profiles. Portion of this amount of oil is then driven to other circumferential locations by combination effects of pressure flow and sliding motion. If those locations happen to be convex, open channels to the OD of the seal form. As a consequence, possible oil leaks can take place. The seal pair LK2, however, is purely concave. In this case, the oil can easily access to the sealing band, but hardly pass the barrier line formed at (or close to) the OD of the seal. Without sufficient thermal twists which tend to change the radial profile from concave to convex, no leakage will happen.

Figure 4.8 Macro equivalent surface geometries of (a) LK1, (b) LK2
4.3 Summary

Through comparisons of four seal pairs with different surface features, it has been shown that the surface temperature, contact wetness, as well as oil exchange rate in the contact areas are three critical parameters for scoring failure analysis. Qualitative explanations on the different scoring behaviors can be offered via these three parameters. The underlying micro surface geometrical features provide the physical explanations on the difference of seal performance. Large surface roughness tends to improve the wetness by reducing the cavitation effects as well as enhance the radial oil flow relatively. More importantly, "random" asperity distribution leads to more disperse contacts, which not only reduce the surface temperature, but also facilitate the oil transport into the contact areas more easily.

Numerical prediction of oil leakage also agrees well with the experimental observations. Macro geometric features of concavity and convexity lead to different oil transport behaviors. Both purely concave and purely convex surface profiles tend to prevent oil leakage. Switches between concave and convex make it possible for oil to leak.

For the sake of better scoring resistance, better oil transport, especially in the radial direction, is desired. However, excessive radial oil flow may lead to leakage failure. A tradeoff is required for proper operation of the FMMFS without scoring and leakage failures. In order to achieve the tradeoff and better performance, the seals should have the following surface features. Firstly, the sealing surface should have features such that the asperity distribution is more disperse and random. Secondly, the surface roughness should not be too small. Finally, the sealing surface consists of both concave and half-concave-half-convex radial profiles. The half-concave-half-convex means the largest surface height in the radial profile locates neither at ID nor at OD, but somewhere between.
Chapter 5

SUMMARY AND CONCLUSIONS

5.1 Summary

The metal face seal of interest in this thesis has unique features such as more flexibility in the circumferential direction than in the radial direction, identical rotating and stationary seals, and a loading mechanism using elastomeric rings. Despite of its wide range of applications with some success, few research efforts into its performance have been made in the past.

Extensive work from both numerical modeling and experiment has been conducted, in attempt to gain some fundamental knowledge of operating mechanisms of the FMMFS. Two versions of numerical models have been developed on this purpose. The quasi-steady state model composed of a 3D contact, an unsteady lubrication, and a steady state heat transfer sub model, can evaluate seal performance such as contact pressure, surface clearance, hydrodynamic pressure, partial film content, frictional heating, and surface temperature etc. The unsteady state model is the result of replacing the steady state heat transfer sub model in the quasi-steady state model with a transient heat transfer sub model. Besides the outputs from the quasi-steady state model, the unsteady state model can predict seal behaviors such as friction and temperature evolution, during transient operation (acceleration/deceleration for example). The unsteady state model is also able to simulate a possible thermal instability process. Both models are capable of calculating dynamic oil distributions and oil transport within the entire sealing band. However, considering dependence of the lubricant viscosity on the oil temperature, the unsteady state model can provide more realistic and accurate predictions, especially during the transient processes.

A great deal of experimental efforts have also been performed to investigate operation of the FMMFS. Friction due to the relative motion and seal temperature
are two parameters of most interest. A wide range of operation conditions, i.e. face load, rotating speed, and lubricant temperature, was tested with different surface geometric features. The test also applied different types of lubricants to see their effects on the seal applicability. With the help of the high-resolution load cell, fluctuations of the frictional torque were monitored during operation under various operating conditions. A DOE test was also carried out in a separate effort to study effects of the seal characteristics, such as surface waviness and surface roughness, on scoring failure. A by-product of this test is that oil leaks, if present, can be captured by a leak detection system. The experiments also serve as a benchmark to validate the developed numerical models by comparing two critical parameters, i.e. friction coefficient and seal temperatures.

The numerical models are then utilized to examine two major applications: scoring failure and leakage failure. Physical analysis was given qualitatively to explain the great variation of scoring failure and leakage failure conditions for the FMMFS with different surface geometric features. A thermal instability process, which may leads to the scoring failure, is also simulated using the unsteady state model.

5.2 Conclusions

Two versions of numerical models are available. Their usage is dependent upon different applications. The quasi-steady state model is the primary model to examine the operation and performance of the FMMFS. As far as the transient operation is concerned, the unsteady state model can be applied. Both numerical models can take input of the real seal surface profiles and output important seal operation-related parameters such as contact pressures, deformed surface clearance, dynamic oil distribution, coefficient of friction, and seal temperature distribution. Heat transfer coefficients, on the convection boundary conditions, were obtained as a function of the rotating speed from a commercial finite element package.
Evidence from both experiments and simulations suggests that the FMMFS operates in the boundary lubrication regime over a wide range of operating conditions. Furthermore, under the typical loading conditions, wavy surfaces will conform during operation. As a result, the hydrodynamic effects due to the circumferential waviness are negligible. The experimental results also reveal that there exists friction variation within one revolution. The variation is complex and periodic and due to change of the contact patterns and oil distributions. The variation pattern differs with surface geometric feature as well as with the rotation speed.

Comparisons between the experimental measurements and the simulation results show that the friction coefficient is slightly overestimated by the numerical simulation. Temperature overestimation is the consequence of the friction overestimation. Generally speaking, overall trend of friction coefficient and seal temperature with rotation speed is captured. Even some detailed friction variation has been achieved, suggesting that the numerical models capture governing physics from distinct features of the FMMFS.

From the scoring failure analysis, the micro geometric features which affect the scoring failure are found using the results from the quasi-steady state model. The surface roughness can impact extent of the cavitation effects as well as the radial oil flow by changing the oil film thickness. The larger the roughness is, the less the extent of the cavitation effects and the greater the radial oil flow rate. In the meantime, asperity distributions also impose considerable effects on contact patterns and oil transport capability. Differences of the asperity distributions are caused by the manufacturing process. The surfaces from the process 1 tend to have more “random” asperity distributions, leading to dispersed contacts in the radial direction and enhanced radial oil flow. On the contrary, the surfaces with the process 2 tend to not only form concentrated contact line circumferentially, but also block the radial oil flow by presence of the ridges.
The surface geometric features which affect the leakage failure are also found. Both purely concave and purely convex surfaces prevent oil transport in the radial direction, causing zero or minimum leakage. Switches from the concave to the convex or vice versa are necessary to transport the oil out of the sealing band.

There is a conflict between scoring and leakage from seal design point of view. On the one hand, effective oil transport in the radial direction is desired for delaying the scoring failure. On the other hand, excessive radial oil flow may leads to leakage failure. The following surface features are proposed to achieve the tradeoff between mature scoring failure and minimum leakage. That is, the sealing surface should have dispersed asperity distribution with relatively large surface roughness. In addition, the sealing surface should also include both concave and half-concave-half-convex radial profiles.

5.3 Suggestions for future work

The first and most critical step of the modeling process is to determine the circumferential load distribution which is determined by the initial waviness. The current models assume the second order sinusoidal load distribution. Since the circumferential variation of the FMMFS is irregular and may take more than the second order waviness, the higher order load distribution should be more appropriate and pursued.

The current numerical models consistently overestimate the friction coefficient. Furthermore, even though remarkable consistency with the experimental results is achieved for the detailed friction variation in some cases, some cases have significant differences. Therefore, the friction model needs to be further investigated and improved. At the same time, effects of air entrainment in the sealing system should be looked into. Because the boundary condition at the OD of the seal is atmosphere, the abundant air can easily access into the bulk lubricant. Due to the fact that the air transports much faster than the oil and does not easily escape once it is trapped in the oil, the viscosity of the bulk lubricant
will reduce to the degree which is dependent upon how much air is present. The viscosity of the bulk lubricant has strong effects on friction generation and the air entrainment will also affect cavitation and oil transport significantly.

Even though the current numerical models can qualitatively provide the physical explanations on different scoring behaviors among various seals with distinct surface geometric features, a more quantitative method should be pursued to relate the contact wetness and oil exchange rate with the surface temperature. The corresponding scoring failure criterion should be developed based on the "varying" critical surface temperature.

The unsteady lubrication sub model can show a difference in oil leakage among different seals. However, there is no standard or criterion for the leakage failure. So future efforts can be made through mutual works of simulations and experiments to set up a leakage criterion for this unique type of seal.
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Appendix A

UNCERTAINTY ANALYSIS OF FRICTION COEFFICIENT MEASUREMENT

Because of the dependence of friction coefficient on the material surface, environment, and measuring equipment, accurate and repeatable friction coefficient measurement remains challenging to date [38]. During the test process, the instantaneous coefficient of friction ($\mu$) is defined as the ratio of the measured friction force ($F_f$) and the measured normal force ($F_n$) as shown in Equation (A.1),

$$\mu = \frac{F_f}{F_n}$$  \hspace{1cm} (A.1)

The load cell located on the top of upper fixture directly measures the normal force and torque ($T$) exerted on the interfaces. The torque is contributed by the tangential friction force. The friction force is then readily derived as,

$$F_f = \frac{T}{R}$$  \hspace{1cm} (A.2)

Where $R$ is the characteristic radius, i.e. inner radius in this case. Later on, the friction force is discussed instead of torque. Misalignments between the force measurement axes and the directions normal and tangent to the reciprocating or rotating surface constitute one of the most significant contributions to friction coefficient measurement errors. The misalignment geometry is shown in Figure A.1.
The intent of the instrument designer is to measure forces along the normal and tangential directions. However, manufacturing tolerances inevitably produce some misalignment between axes of the force measurement system and the normal and tangential axes. As shown in Figure A.1, the transducer axis Y that intended to measure the normal force is misaligned from the normal direction by an angle of $\beta$; the transducer axis X which is intended to measure the friction force is misaligned from the tangential direction by an angle of $\alpha$. The normal and tangential force could be projected onto the measurement axes, resulting in the measured normal and tangential force $(F_x, F_y)$, defined as,

\[ F_x = F_f \cos \alpha + F_n \sin \alpha = F_n \left( \sin \alpha + \mu \cos \alpha \right) \]

\[ F_y = F_n \cos \beta - F_f \sin \beta = F_n \left( \cos \beta - \mu \sin \beta \right) \]  \hspace{1cm} (A.3)

The coefficient of friction can then be derived in terms of the measured forces and misalignment angles,

\[ \mu = \frac{F_x \cos \beta - F_y \sin \alpha}{F_y \cos \alpha + F_x \sin \beta} \]  \hspace{1cm} (A.4)
If the misalignment angles are neglected, the friction coefficient is directly calculated from Equation (A.1) as the ratio of the measured normal force and the measured friction force, i.e. $\mu' = \frac{F_y}{F_x}$. An error associated with neglecting the misalignment angles is consequently defined by.

$$e = \frac{\mu - \mu'}{\mu} = \frac{\mu \cos \beta - \mu^2 \sin \beta - \mu \cos \alpha - \sin \alpha}{\mu \cos \beta - \mu^2 \sin \beta}$$  \hspace{1cm} (A.5)

Assuming the transducer axes are perpendicular to each other (i.e. $\alpha = \beta$), Equation (A.5) can be reduced to

$$e = \frac{\mu^2 \sin \alpha + \sin \alpha}{\mu^2 \sin \alpha - \mu \cos \alpha}$$  \hspace{1cm} (A.6)

It is reasonably assumed that the misalignment angle $\alpha$ is sufficiently small. The above equation can be further simplified as

$$e = \frac{2\alpha (1 + \mu^2)}{\mu (\alpha^2 + 2\mu \alpha - 2)}$$  \hspace{1cm} (A.7)

In order to evaluate the measurement uncertainty for $\mu$, the law of propagation uncertainty is applied to determine the combined standard uncertainty, which represents one standard deviation of the friction coefficient measurement result. The combined standard uncertainty is a function of the standard uncertainty of each input measurement and the associated sensitivity coefficients. Based on the definition of $\mu$ in Equation (A.4), the expression for the square of the combined standard uncertainty for the friction coefficient measurement result is given by,

$$u_c^2(\mu) = \left( \frac{\partial \mu}{\partial F_x} \right)^2 u^2(F_x) + \left( \frac{\partial \mu}{\partial F_y} \right)^2 u^2(F_y) + \left( \frac{\partial \mu}{\partial \alpha} \right)^2 u^2(\alpha) + \left( \frac{\partial \mu}{\partial \beta} \right)^2 u^2(\beta)$$  \hspace{1cm} (A.8)
Note that the correlation between the separate input variables, i.e. zero covariance, is assumed here. The partial derivatives of the friction coefficient with respect to the input variables are given by Equations (A.9)-(A.12), respectively,

\[
\frac{\partial \mu}{\partial F_x} = \frac{\cos \beta}{F_x \cos \alpha + F_x \sin \beta} - \frac{\sin \beta (F_x \cos \beta - F_y \sin \alpha)}{(F_x \cos \alpha + F_x \sin \beta)^2} \tag{A.9}
\]

\[
\frac{\partial \mu}{\partial F_y} = \frac{-\sin \alpha}{F_y \cos \alpha + F_y \sin \beta} - \frac{\cos \alpha (F_x \cos \beta - F_y \sin \alpha)}{(F_y \cos \alpha + F_y \sin \beta)^2} \tag{A.10}
\]

\[
\frac{\partial \mu}{\partial \alpha} = \frac{-F_y \cos \alpha}{F_y \cos \alpha + F_y \sin \beta} + \frac{F_y \sin \alpha (F_x \cos \beta - F_y \sin \alpha)}{(F_y \cos \alpha + F_y \sin \beta)^2} \tag{A.11}
\]

\[
\frac{\partial \mu}{\partial \beta} = \frac{-F_x \sin \beta}{F_x \cos \alpha + F_x \sin \beta} - \frac{F_x \cos \beta (F_x \cos \beta - F_y \sin \alpha)}{(F_x \cos \alpha + F_x \sin \beta)^2} \tag{A.12}
\]

The force transducers normally generate voltage signals. The voltage signals is then converted to the desired force values in the conditioner by multiplying a calibration constant C. Consequently, the process of calibration is to determine such constants. Obviously the uncertainty in the measured calibration constant is dependent upon the uncertainty in both the measured voltages and the forces applied during the calibration sequence. The uncertainty in the average measured voltages is obtained using Eq. (A.13) and (A.14), where \(s^2(V_x)\) and \(s^2(V_y)\) are the variances in the recorded voltages by the force transducers in the X and Y directions respectively, \(n\) is the number of the samples, and \(\bar{V}_x\) and \(\bar{V}_y\) are the mean voltage values in the X and Y directions respectively,

\[
u^2(\bar{V}_x) = \frac{s^2(V_x)}{n} = \frac{1}{n-1} \sum_{i=1}^{n} (V_{x,i} - \bar{V}_x)^2 \tag{A.13}
\]

\[
\nu^2(\bar{V}_y) = \frac{s^2(V_y)}{n} = \frac{1}{n-1} \sum_{i=1}^{n} (V_{y,i} - \bar{V}_y)^2 \tag{A.14}
\]
As previously discussed, the average X and Y-direction force values are then determined by multiplying the average voltages with calibration constants,

\[ F_x = C_x \bar{V}_x \]  
\[ F_y = C_y \bar{V}_y \]  

(A.15)  

(A.16)

The standard uncertainty in the mean values of \( F_x \) and \( F_y \) are given by,

\[ u^2(F_x) = \left( \frac{\partial F_x}{\partial C_x} \right)^2 u^2(C_x) + \left( \frac{\partial F_x}{\partial V_x} \right)^2 u^2(\bar{V}_x) = (\bar{V}_x)^2 u^2(C_x) + (C_x)^2 u^2(\bar{V}_x) \]  

\[ u^2(F_y) = \left( \frac{\partial F_y}{\partial C_y} \right)^2 u^2(C_y) + \left( \frac{\partial F_y}{\partial V_y} \right)^2 u^2(\bar{V}_y) = (\bar{V}_y)^2 u^2(C_y) + (C_y)^2 u^2(\bar{V}_y) \]  

(A.17)  

(A.18)

The calibration constant uncertainty can be determined in the following way. A known load is applied on the transducer of interest and the corresponding voltage is recorded. The calibration constant \( C \) is then taken as the slope of the applied force versus recorded voltage. By repeating above procedure, a pool of \( C \)s is collected. Because there are also uncertainties both in load and voltage, a Monte Carlo simulation is used to generate constant curve. The mean and standard deviation of the constants are then obtained from the constant curve. The uncertainties in average measured forces are then determined by Equation (A.17) and (A.18).

The angular misalignment for our test setup is determined by measuring the horizontal force by the application of a known vertical force. The mean and standard deviation of angle of misalignment were found from the sample distribution of multiple repetitions. Furthermore, the transducer axes were perpendicular, such that \( u(\alpha) = u(\beta) \).

The combined standard uncertainty of the friction coefficient can then be determined using Equation (A.8), where \( F_x \) and \( F_y \) are replaced by \( \bar{F}_x \) and \( \bar{F}_y \), respectively.
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