HIGH PRESSURE PERFORMANCE OF FOIL JOURNAL BEARINGS IN VARIOUS
GASES

by

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List of Abbreviations

A	Area
D	Bearing diameter
$\mathcal{D}_j$	Bearing load capacity coefficient
g	Acceleration due to gravity
H	Characteristic film thickness
h	Film thickness
L_a	Characteristic axial length
L_c	Characteristic circumferential length
P_{ref}	Reference Pressure
p	Pressure
Re	Reynolds Number
Re_m	Modified Reynolds Number
S'	Modified Sommerfeld Number
Ta	Taylor Number
t	time
U	Characteristic Circumferential Surface Velocity
U_s	Characteristic Horizontal Surface Velocity
u	Horizontal velocity component
V	Characteristic vertical velocity
v	Vertical velocity component
W	Characteristic axial velocity
W_{preload}	Component of bearing load caused by preload

W_s	Characteristic axial velocity
W_t	Total Bearing Load
W_{vert}	Component of bearing load caused by vertical loading
w	Axial velocity component
X	Body force in the x-direction
Y	Body Force in the y-direction
Z	Body force in the z-direction
Θ	Characteristic time
μ	Dynamic viscosity
ν	Kinematic viscosity
ρ	Gas density
Ω	Shaft rotational speed

Subscripts

amb	Property measured at ambient chamber conditions
char	Characteristic value
cont	Quantity referenced by the continuity equation
L	Quantity measured during lift-off
max	Maximum value
ref	Reference value
x,y,z	Components measured in the x, y, and z directions

Superscripts

*	Non-dimensional quantity
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High Pressure Performance of Foil Journal Bearings in Various Gases

Abstract

by

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Foil bearings are being considered for use in high ambient pressure applications which use lubricants other than air. However, experimental performance data under these conditions is limited, which hinders the modeling efforts of system designers. To collect a more comprehensive data set in these conditions, a new test facility is developed which measures bearing torque in helium, nitrogen, and carbon dioxide, at pressures up to 4.83 MPa and speeds up to 42,000 RPM. Bearing torque is found to increase with both pressure and test gas molecular weight, and this effect is amplified as speed increases. The effects of pressure and test gas molecular weight, and speed are analyzed in terms of the kinematic viscosity and modified Reynolds number. Finally, laminar inertial effects, transition to turbulence, and flow instability are evaluated as possible causes of the observed increase in torque using the modified Reynolds number, Reynolds number, and Taylor number, respectively.

Chapter 1

Introduction

1.1 Foil Gas Bearings. Foil gas bearings are a type of gas lubricated, self acting hydrodynamic bearing. Traditional rigid hydrodynamic bearings consist of a journal rotating inside of a rigid bearing sleeve with a small radial clearance (Fig. 1). This clearance is filled by a lubricant, typically oil although various gases have been used [1]. As the journal rotates, a lubricant film develops which separates the journal and bearing surfaces and supports the load of the journal, eliminating solid sliding contact and dramatically reducing wear on the journal and bearing surfaces.

Figure 1 – Rigid Hydrodynamic Bearing

Foil bearings improve upon rigid hydrodynamic bearings by using a compliant top-foil to trap the fluid film. This compliant surface enables foil bearings to accommodate thermal and centrifugal shaft growth which would cause a rigid bearing to seize, and increases tolerance to shaft misalignment. Furthermore, foil bearings use

ambient gas as their lubricant, eliminating the need for oil systems. Elimination of the oil system reduces weight, maintenance costs, and risks associated with oil leakage and degradation. In closed-loop, unmaintainable systems, such as the closed Brayton cycle (CBC) power generation system proposed for human and robotic space exploration, eliminating the oil system provides the necessary benefit of maintaining a contaminate-free working fluid [2][3].

As with all self acting hydrodynamic bearings, foil bearings require solid lubrication during periods of rubbing contact which occur during start-up and shut-down, when low operating speeds prevent a full hydrodynamic film from forming and supporting the entire shaft load. In order to take advantage of the high temperature capabilities of foil bearings, solid lubricants must exhibit good friction and wear characteristics above 500° C, where traditional solid lubricants, such as polytetrafluoroethylene, cannot be used [4]. The development of high temperature solid lubricants in recent years, such as the NASA developed PS300 series, has dramatically improved wear characteristics and performance of foil bearings over a temperature range of 25 – 700 °C, allowing them to be considered for a variety of high temperature applications [5].

The present work considers bump-foil type foil bearings, which consist of a rigid bearing sleeve, at least one layer of corrugated foils, known as bump-foils, stiffener foils, and a top foil. The top foil is a smooth metallic sheet which fits around the journal and is supported by stiffener foils which are, in turn, supported by the bump foils (Fig. 2).

Figure 2 – Bump-Foil Type Foil Journal Bearing

The use of stiffener foils and multiple bump foils allows stiffness to be varied axially, radially, and circumferentially, allowing for the gas film entrapment to be controlled more precisely, enabling bearings to be optimized for specific applications [6]. Bearings that tailor stiffness in the axial, circumferential, and radial directions show dramatic improvements in load capacity over prior designs, as well improvements in stiffness and damping characteristics. These bearings have been termed by DellaCorte and Valco as Generation III bearings and represent the current state of the art of foil bearing design [6].

The advancements in journal coatings and bearing load capacity over the past two decades have made foil bearings candidates for more demanding, higher temperature turbomachinery applications such as microturbines [7] and turbochargers [8][9]. These foil bearing applications use air at atmospheric pressure as the bearing lubricant. However, some potential applications require foil bearings to operate at high pressures, in

gases other than air. One such application is the closed Brayton cycle, a leading candidate for nuclear dynamic power conversion systems [2][3][10]. This closed Brayton cycle uses a nuclear fission heat source and high pressure recirculating gas as the working fluid. Since the working fluid is contained in a closed loop and must remain free of contamination, oil-free lubrication is required. CBC's use high pressure gas (i.e. CO₂ or He-Xe mixtures) as the working fluid and, therefore, foil bearing performance models must be updated to include the effects of gas properties and elevated ambient pressure.

1.2 Prior Experimental Work. In order to update current foil bearing performance models to include the effects of elevated ambient pressure and varying gas properties, reliable experimental data is needed. However, until recently, only one report on foil bearing performance in lubricants other than air at elevated ambient pressure had been published [11]. These tests were performed at a constant speed of 52,000 RPM while load was varied from 0 – 4 lbs. Test gases included air, argon, and krypton, at a maximum pressure of 100 psig. This study shows that bearing power loss increases substantially with both ambient pressure and test gas molecular weight (Fig 3). However, this test was done at constant speed, preventing the effect of speed on high pressure bearing performance from being determined. In addition, the bearings used in this study were designed using first generation bearing technology and the results may not be applicable to more advanced third generation bearing designs.

Figure 3 – Experimental Results of Dobler and Miller [11]

More recently, a study was published in air at pressures up to 820 kPa [3]¹. These tests were conducted using various applied loads (4.45 – 35.0 N) while running up to 24,000 RPM. Results reported in these tests show an increase in bearing power loss with increasing ambient pressure as well (Fig 4). However, testing a single gas does not permit determination of the effects of changing gas properties on bearing performance.

Figure 4 – Experimental Results of Howard et al [3].

¹ This study was conducted on a previous design iteration of the High Pressure Rig at the NASA Glenn Research Center. A more recent design iteration of this test rig was used to conduct the current research.

While these two studies show that bearing power loss (friction) increases with increasing pressure, there has yet to be a published data set which shows the effect of journal speed, gas properties, and ambient pressure in the operating ranges of potential high pressure foil bearing applications. This limited range of available data prevents system designers from accurately predicting high pressure foil bearing performance. Therefore, there is a need to develop a new test rig capable of acquiring a foil bearing performance data set which includes the effects of gas properties, elevated ambient pressure, and varying speed in the operating range of foil bearing applications. This data set can then be used to update current analytical and numerical performance models.

1.3 Pressure Effects – Reynolds Equation. Hydrodynamic lubrication with compressible fluids is typically analyzed using the Reynolds equation, which appears below and is derived from the continuity equation and Navier-Stokes equations in Appendix A.

Reynolds Equation

$$\frac{\partial}{\partial x} \left(\frac{\rho h^3}{12\mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left(\frac{\rho h^3}{12\mu} \frac{\partial p}{\partial z} \right) = \frac{\partial}{\partial x} \left[\frac{\rho h U_s}{2} \right] + \frac{\partial}{\partial z} \left[\frac{\rho h W_s}{2} \right]$$

This equation can be solved numerically to obtain the pressure distribution, velocity field, and film thickness as a function of the x and z coordinates.

Solving this equation for compliant foil bearings is complicated by the need to couple the above equation with a model of the top-foil deflection. Carpino and Talmage have developed an algorithm which solves this system of coupled partial differential equations in cylindrical coordinates, assuming an isothermal gas film [12]. This algorithm is run with the test conditions listed in Table 1 to show the trend in the friction

coefficient as a function of pressure and modified Reynolds number, as predicted by the Reynolds equation. This numerical solution of the Reynolds equation predicts that the coefficient of friction remains nearly constant, regardless of increases in ambient pressure and corresponding increases in modified Reynolds number (Fig 5 & 6).

Diameter (mm)	51.0
Length (mm)	25.4
Load (N)	44.3
Speed (krpm)	30
Pressure Range (MPa)	.10 – 5.1
Re _m Range	.02 – 1.3

Table 1 – Operating Conditions for Numerical Solution of the Reynolds Equation

Figure 5 – Reynolds Equation Prediction of Apparent Friction Coefficient versus Pressure

Figure 6 – Reynolds Equation Prediction of Apparent Friction Coefficient versus Modified Reynolds Number

This result contradicts the previously published experimental results of Dobler & Miller [11] and Howard et al [3] (Fig. 3 & 4), as well as the results of the current work, which all suggest that friction coefficient increases with pressure. Therefore, the assumptions used in the derivation of the Reynolds equation are examined more closely.

The validity of the assumptions used to reduce the Navier-Stokes equations to the Reynolds equation are discussed in detail in Appendix A, and summarized herein. In hydrodynamic lubrication, the neglect of inertial terms in the Navier-Stokes equations is justified by showing that the modified Reynolds number, defined below, is small. However, as pressure increases, density increases, causing kinematic viscosity (dynamic viscosity divided by density) to decrease (Fig. 7). Furthermore, for a given pressure and temperature, high molecular weight gases have lower kinematic viscosities than low molecular weight gases. Therefore, the modified Reynolds number can approach and

exceed unity, especially when bearings operate at high pressures and speeds, and use high molecular weight gas as the bearing lubricant (Table 2). Under these conditions, the inertial terms in the Navier-Stokes equation may be non-negligible.

Figure 7 – Effects of Pressure on the Kinematic Viscosity of Various Molecular Weight Gases

Modified Reynolds Number Definition

$$Re_m \equiv \left(\frac{\rho_{amb} U H^2}{\mu_{amb} L_c} \right)$$

Ω (krpm)	ρ_{amb} (Kg / m ³)	H (μ m)	Rem
5	0.2	5.08	2.87E-05
5	200	5.08	2.87E-02
20	0.2	5.08	1.15E-04
20	200	5.08	1.15E-01
40	0.2	5.08	2.29E-04
40	200	5.08	2.29E-01

5	0.2	12.7	1.79E-04
5	200	12.7	1.79E-01
20	0.2	12.7	7.17E-04
20	200	12.7	7.17E-01
40	0.2	12.7	1.43E-03
40	200	12.7	1.43E+00
5	0.2	50.8	2.87E-03
5	200	50.8	2.87E+00
20	0.2	50.8	1.15E-02
20	200	50.8	1.15E+01
40	0.2	50.8	2.29E-02
40	200	50.8	2.29E+01

Table 2 – Modified Reynolds Number at Various Operating Conditions

(Viscosity assumed to be 15×10^{-6} kg/m-s

Low Density corresponds to helium at atmospheric pressure and 80° F

High density corresponds to krypton at 50 ATM and 80° F)

Retaining the inertial terms in the Navier-Stokes equations explicitly identifies the kinematic viscosity as the gas property of interest and the modified Reynolds number as the dimensionless parameter of interest when evaluating high pressure foil bearing performance (Appendix A). These observations are used to guide the data analysis in the current work.

Other assumptions of Reynolds equation analysis include isothermal gas film, stable flow, and laminar flow. Assumption of an isothermal gas film leads to the conclusion that kinematic viscosity is the only gas property which affects bearing torque. This result can be checked and used to comment on the validity of the isothermal approximation. If the flow between the journal and top-foil may be approximated as Couette flow then flow instability is expected when the Taylor number, defined below, exceeds 41.3 [13] and turbulence is expected when the Reynolds number, defined below, exceeds 1500 [13]. When the flow between concentric cylinders becomes either unstable or turbulent, increases in torque are expected, and since both Taylor number and Reynolds number increase with ambient pressure (ambient density), both instability and

turbulence will be investigated as possible causes of increasing torque with ambient pressure.

Taylor Number

$$Ta \equiv \frac{\rho_{amb} UH}{\mu_{amb}} \sqrt{\frac{2H}{D}}$$

Reynolds Number

$$Re \equiv \frac{\rho_{amb} UH}{\mu_{amb}}$$

Chapter 2

Methodology

2.1 Test Equipment. A new test rig was designed and built to obtain a more comprehensive foil bearing performance database which includes the effects of ambient pressure, speed, and gas properties, upon which analytical and numerical models can be based. The test rig consists of a motor driven journal mounted inside a pressure vessel rated to 4.83 MPa (700 psig) (Fig 8). The motor spins the journal which supports a foil bearing and an annular weight. The rotational speed of the motor and the bearing friction force are measured, allowing bearing power loss to be calculated. This rig, referred to herein as the High Pressure Rig, is capable of measuring foil journal bearing performance at speeds up to 42 krpm and pressures up to 4.83 MPa (700 psi), using several inert test gases including: air, nitrogen, argon, helium, xenon, carbon dioxide, and krypton, and various selected mixtures. The current data set was taken using helium, nitrogen, and carbon dioxide as test gases, having molecular weights of 4, 28, and 44 g/mol respectively.

Figure 8 – High Pressure Rig Diagram

Figure 9 – High Pressure Rig Detail View

Pressure Vessel / Gas Handling

The pressure containment system consists a pressure vessel, certified to handle 4.83 MPa (700 psig) from -29 °C to 38 °C (-20 °F to 100 °F), which can be opened at a bolted joint (Fig 8). One half of the pressure vessel is bolted to a table, while the other is mounted to one meter long linear bearing and rail system for operational convenience. Pipe flanges are secured to the pressure vessel using grade 8 bolts and commercially available one-thousand psi gaskets provide sealing. Instrumentation and power enter the pressure vessel penetrations via commercially available multi-wire feedthroughs and sealing assemblies.

Test gases are fed into the pressure vessel from bottles adjacent to the test rig. Pressure is regulated by a bottle-top regulator, while two sets of ball valves and a fill valve, and two pressure gauges, provide multiple checks to prevent chamber over pressurization. Chamber pressure is monitored with electronic transducers and an analog mechanical pressure gauge.

Spindle and Drive

Mounted inside the pressure vessel is a commercially available, one kilowatt, variable frequency spindle, which is designed for use in high speed machining applications and, therefore can support large radial loads with limited deflection while running at speeds as high as 42,000 rpm (Fig. 9). The spindle end has a collet into which a test journal is secured. The rotating end of the spindle has a portion painted flat black to allow an optic probe to measure the time in between successive light-to-dark transitions, which is used to calculate spindle rotational speed. The controller for the

spindle is a commercially available, 1500 Hz Variable Frequency Drive, which can be used to create preset velocity profiles or changed manually during testing.

Test Articles

The journals currently used on the High Pressure Rig are specially designed to minimize imbalance, runout, and overhung mass. The original journal design caused large amplitude vibrations which jeopardized spindle bearing health and prevented the High Pressure Rig from reaching its intended maximum speed of 42 krpm. A full description of this rotordynamic problem and the journal redesign which solved it can be found in Appendix B.

The newly designed High Pressure Rig journals fit directly into the spindle collet and maintain concentricity with the axis of rotation to within 13 μm (.0005 in) (Fig. 10 and 11). The journal is trim balanced, in-place, using a single plane influence coefficient technique to reduce vibration levels and corresponding dynamic loading. The journals are made of a light weight, high strength, titanium alloy in order to reduce overhung mass and maintain high stiffness. The journals are coated with PS-304 [5], a high temperature solid lubricant used to reduce friction during periods of solid rubbing contact which occur during start-up and shut down. The journal and PS-304 coating are then ground to a nominal diameter of 3.493 cm (1.375 in) to improve surface finish.

Figure 10 – Titanium Journal Diagram before Plasma Spray and Grinding

Figure 11 – Titanium Journal Picture after Plasma Spray and Grinding

The High Pressure Rig is designed to accommodate bearings with a nominal diameter of 34.9 mm (1.375 in) and an axial length of 28.6 mm (1.125 in), a size chosen for compatibility with other foil bearing test facilities and readily available hardware. The current work is performed with commercially available Generation III bump foil bearings, which use multiple split bump foils and a stiffener foil to tailor stiffness radially, circumferentially, and axially, increasing load capacity and enhancing bearing performance (Fig. 2 & 12)[6]. Typical testing is done using bearings with uncoated top

foils, in a partially run-in condition, as described in DellaCorte et al [14]. The estimated load capacity coefficient, $\mathcal{D}_j$, of a bearing running in this condition is .45 lbs/in³-krpm [6][14].

Figure 12 – Third Generation Bump-Foil Type Foil Journal Bearing Picture

Loading

The bearing is mounted inside of an annular weight which is used to apply a radial load (3.14 lbs) onto the bearing. The bearing / annular weight assembly is then fit onto the journal. A radially extending rod (torque arm) is oriented vertically downward from the annular weight and connects to a load cell for accurate torque measurement (Fig. 9). Manufacturing imperfections cause the annular weight to be slightly asymmetric and therefore slightly imbalanced. This imbalance results in a tare torque (0.8 N-mm) which is taken into account when measuring frictional torque.

Instrumentation

Measurements made on the High Pressure Rig include frictional torque, motor speed, chamber pressure, and chamber temperature. The torque and speed measurements

are used to calculate bearing power loss, while chamber pressure and temperature measurements are used for gas property calculations and pressure vessel health monitoring.

Attached to the spindle mounting plate is a load cell, for torque measurement, and an optic probe, used for speed measurement. The load cell is a commercially available linear variable displacement transducer. As the spindle turns, solid-solid friction or viscous shearing of the gas film produces a frictional moment on the bearing and annular weight assembly. The load cell, located $8.4 \text{ cm} \pm .1 \text{ cm}$ ($3.37 \text{ in} \pm .04$) from the bearing center, measures the force required to constrain the bearing and load assembly rotationally, which can be used to calculate frictional torque. The error in the torque measurement is dominated by the uncertainty in the distance from the load cell center to the bearing center (1.3%).

The optic probe is placed under the rotating portion of the spindle and detects the light-to-dark transition on the painted spindle shaft, creating a square-wave voltage whose period is used to calculate the angular speed of the spindle accurate to within .5%, resulting in a total power loss error of (1.4%).

The pressure measurement is taken using a low pressure (1.0 MPa maximum) and a high pressure (6.5 MPa maximum) transducer, both with an accuracy of .04% of the full scale pressure. The use of the low pressure transducer improves accuracy from 2.6 kPa (.38 psi) for high pressure measurement, to .4 kPa to (.06 psi) for low pressure measurements.

Housed in the pressure vessel, is a thermocouple, which measures chamber temperature to an accuracy of .08%. It may be argued that gas properties should be calculated at the average gas film temperature, not the chamber temperature, and that this would result in large errors due to the lack of an accurate gas film temperature measurement. However, using average gas film temperature deviates from the convention established for oil and gas lubricated bearings in which the lubricant inlet temperature is used for property calculation [1][15][16].

During journal balancing and motor vibration tests, two piezoelectric accelerometers are placed on the spindle mounting bracket, accelerations are measured and analyzed in both the time and frequency domains, for dynamic balancing and health monitoring.

Data Acquisition

The signals recorded on the High Pressure Rig include motor speed as measured by the optic probe, motor speed as measured by the variable frequency drive, torque, pressure, temperature, and vibratory acceleration amplitude (during vibration testing). These signals are collected with commercially available 12-bit and 14-bit data acquisition cards. The data is then processed and logged using commercially available data acquisition software. Transient data output files include raw time trace data, while steady data output files include time averaged signals and their associated errors.

2.2 Test Preparation. Before high pressure testing begins, the journal must be balanced and the PS-304 journal coating must be conditioned. Dynamic loads must be minimized through balancing because calculation of the total bearing load assumes a negligible dynamic load. In addition, balancing the journal prolongs the life of the

spindle bearings by lowering bearing forces. The solid lubricant coating must be conditioned so that the surface finish does not change over time, ensuring that power loss measurements are repeatable.

Balance / Vibration

Single-plane balancing is performed using a tracking filter to isolate the synchronous vibration amplitude and phase, as measured by accelerometers mounted on the spindle shaft. The amplitude and phase of the synchronous vibration of the journal alone are compared to those measured with a trial weight, and this comparison is used to estimate the magnitude and circumferential location of a correction weight. This process is then repeated until the journal vibrations reach acceptably low levels (<1.0 g peak) through the entire operating range. The typical magnitude of imbalance after balancing is ~ 3 mg-m.

After journals are balanced, broad-spectrum vibration tests begin. These vibration tests establish baseline vibration levels for the motor which can be used to monitor journal balance, monitor spindle bearing health, and identify resonant frequencies of the spindle and journal assembly. These resonant frequencies are then avoided during steady data collection. Figure 13 shows the broadband vibration amplitude before and after balancing.

Figure 13 – Broadband Vibration Amplitude before and after Balancing

Solid Lubricant Conditioning

Conditioning of the solid lubricant improves the surface finish and generates a thin oxide layer that provides additional solid lubrication [14], which improves bearing performance. The typical PS304 conditioning process involves subjecting the journal and bearing to numerous start-stop cycles at high temperature (540 °C)[14]. Unconditioned journals are subjected to start-stop cycles during testing begin to improve their surface finish and generate this oxide layer over time, resulting in improved performance and unrepeatable torque measurements

Conditioning of bearings and journals in this study is not done to optimize bearing performance, but simply to bring bearing power loss at constant speed to a repeatable value, ensuring consistent results. This can be accomplished without conditioning the bearing at high temperature. Bearing and journal combinations are subject to 500 to 1000 start-stop cycles at room temperature with a 1.4 kg (3.14 lbs) radial load. Each start-stop

cycle consists of an acceleration, speed plateau, and deceleration phase (Fig. 14). At speeds below the bearing lift-off speed there is solid-to-solid rubbing contact and power loss increases rapidly with increasing speed. As the journal approaches the lift off speed, the load is partially supported by a fluid film and power loss begins to decrease with increasing speed (Fig. 14). At speeds above the lift-off speed the journal may operate safely at steady-state, and power loss repeatability can be tested.

Figure 14 – Individual Start-Stop Cycle from Solid Lubricant Conditioning

At the start of the conditioning process, power loss during constant speed operation is high and erratic. However, after several hundred start-stop cycles the power loss during the speed plateau decays to a lower, steady value, indicating that journal and bearing surfaces are no longer undergoing significant changes, and that the measurement of power loss has become repeatable (Fig. 15). This conditioning is repeated whenever bearing characteristics are observed to have changed significantly during testing, due to solid-solid rubbing that may occur while operating near bearing load capacity.

Figure 15 – Solid Lubricant Conditioning Power Loss Time Trace

2.3 Characterization Testing. Most traditional, oil lubricated, hydrodynamic journal bearings have an inner diameter which is slightly larger than the journal that rotates inside of it, resulting in a small clearance which can be quantified and compared between bearings. Foil bearings, on the other hand, have bump foils which preload the top foils against the journal. This preload represents the minimum force that must be generated by the hydrodynamic film before dry sliding friction gives way to full film hydrodynamic lubrication, and must be taken into account when determining the total load supported by the bearing. Characterization testing is performed prior to data collection in order to measure bearing preload and dry-sliding friction coefficient. This data is used to ensure that trends in data are the result of changing test parameters, not changing bearing characteristics and to appropriately account for total bearing load when comparing data collected from different journal and bearing pairs.

Preload and friction coefficient are measured by rotating the spindle manually and measuring the frictional torque for various applied radial loads. Friction force is plotted

against applied load (Fig. 16) and the least squares fit is used to calculate the friction coefficient (slope) and the preload (x-intercept), as described in Reference [14].

Figure 16 – Preload and Friction Coefficient Data

Another common parameter used to define bearings is the load capacity coefficient [6]. The High Pressure Rig does not currently have the capability of measuring the load capacity coefficient, which requires load to be varied during operation. A full discussion of this limitation can be found in Appendix C.

2.4 High Pressure Alternate Gas Testing. High pressure testing provides the data necessary to evaluate bearing performance as a function of pressure, gas properties, and speed. This data includes frictional torque, spindle rotational speed, chamber pressure, and chamber temperature measured with various gases as the bearing lubricant. Testing is done in helium, a low molecular weight gas, carbon dioxide, a high molecular weight gas, and nitrogen, a moderate molecular weight gas and surrogate for air.

Performance Testing

During high speed operation the total bearing load (applied radial load plus preload) is supported by a hydrodynamic film. In this flow regime there is no solid-solid rubbing contact and bearings may be run in this condition at steady-state without risking damage to the bearing or journal. Data collected while running in this regime is referred to herein as steady testing, and involves time averaging data at constant speed. During low speed operation the hydrodynamic film cannot support the entire bearing load, so solid-solid rubbing contact occurs. In this flow regime, steady testing cannot be done without damaging the journal and top-foil surfaces. Testing in this regime requires an alternate testing method, referred to herein as transient testing, which involves measuring torque and speed during a single start-stop cycle.

The two speeds which can be used to mark the transition between these two regimes are the lift-off speed and the speed at which the power loss reaches a minimum. The lift-off speed is the speed at which the top-foil and journal surfaces are no longer in contact and can be estimated using the “Rule of Thumb” method described in DellaCorte and Valco [6]. A very conservative estimate of the lift-off speed (8.5 krpm) was used in the current work in order to minimize the risk of damage to the test articles. The speed at which the power loss reaches a minimum defines the low end of the desirable operating range for gas foil bearings as described in DellaCorte et al [17]. In order to further minimize the risk of damage to the test articles, steady testing is stopped when either the power loss minimum or the estimated lift off speed is reached, and transient data is used to complete the low speed portion of the power loss versus speed curve. Transient data

can be used to find important features of the power loss curve, such as the power loss extrema, but is not the focus of the current work and is not used in the data analysis.

During steady testing the journal is accelerated to maximum speed, and that speed is maintained until the torque and power loss reach a steady value (< 2 seconds). Once the torque and power loss reach a steady value, torque and speed data are acquired, time averaged, and recorded. The speed is then decremented by 1 krpm, and the process is repeated until the power loss minimum (Fig. 17) is identified or the spindle speed approaches the estimated lift-off speed of the bearing. During transient testing the journal is accelerated to a speed well above the estimated lift off speed, allowed to plateau briefly, and decelerated (Fig. 18). The combination of these two test methods completes the power loss versus speed curve as seen in Figure 19.

Figure 17 – Time Trace from Steady-State Test

Figure 18 – Time Trace of Transient Test

Figure 19 – Identification of the Power Loss Minimum with Steady-State and Transient Data

High Pressure Testing

To begin each high pressure data set, a steady test and a transient test are run in ambient air. This data set is used as a control to ensure that bearing characteristics are not changing over time. Next, in order to decrease the concentration of air in the pressure vessel, the chamber is pressurized, with the test gas, to 210 kPa (30 psia) then vented to 100 kPa (15psia). This process is repeated two more times so that, at 15 psia, there is a 7:1 mixture of test gas to air. The chamber is then pressurized to the maximum pressure allowed by the chamber relief valve (4.49 - 4.83 MPa), resulting in a gas-air mixture which is ~.3% air. A steady test and transient test are run at maximum pressure, and then the chamber is vented to the next lowest pressure set point, and the process is repeated until the chamber pressure is reduced to atmospheric pressure. The chamber is then flushed with compressed air and another data set is taken in air at ambient pressure, to verify that bearing performance has not changed.

2.5 Analysis Method – High pressure foil bearing research in alternate gases aims to generate an empirical database that can be used to accurately predict bearing torque given bearing dimensions, bearing characterization parameters (i.e. preload), gas properties, ambient pressure, load, and speed. The current work takes a first step towards this goal by examining the combined effects of gas properties, ambient pressure, and speed. First, bearing power loss is analyzed as a function of speed and pressure for each gas. Next, the effects of molecular weight, viscosity, and pressure are lumped together using the kinematic viscosity. Then the effects of molecular weight, viscosity, pressure, and speed are combined using the modified Reynolds number. Finally, the modified Reynolds number, Reynolds number and Taylor number are used to evaluate whether

laminar inertial effects, transition to turbulence, or instability of the primary flow can be linked with trends in bearing performance.

Dobler and Miller [11] took the first step toward understanding high pressure bearing performance by reporting an increase in power loss with both pressure and molecular weight. Quantitative comparisons to the Dobler and Miller data are difficult due to the lack of published characterization parameters and difference in bearing technology. However, current bearing power loss data will be compared qualitatively to those published by Dobler and Miller by examining trends in power loss with varying molecular weight and pressure. In addition, the raw measurements of power loss as a function of pressure can, themselves, be valuable guides for modeling efforts for Closed Brayton Cycle designers.

The next step is to lump the effects of varying pressure, molecular weight, and absolute viscosity, into a single varying parameter, the kinematic viscosity, as suggested by the scaling of the Navier-Stokes Equations (Appendix A). Evaluating bearing performance as a function of kinematic viscosity allows bearing performance in different gases to be compared directly. Absolute viscosity is a very weak function of pressure and gas density is proportional to pressure and molecular weight, assuming an ideal gas². Pressure and molecular weight are inversely proportional to kinematic viscosity and bearing torque is expected to decrease with increasing kinematic viscosity, for set operating conditions.

The effect of speed on the apparent friction coefficient will then be lumped with the effects of kinematic viscosity using the modified Reynolds number as suggested by

² Carbon dioxide, at the high pressure range of the tests conducted, does not behave ideally, but the effects of increasing pressure and molecular weight are qualitatively similar.

the scaling of the Navier-Stokes equations (Appendix A). The modified Reynolds number can then be evaluated for its ability to adequately account for the combined effects of kinematic viscosity and speed. The modified Reynolds number can also be used to examine the possibility of laminar inertial effects altering the flow field and accounting for trends observed trends in torque. Laminar inertial effects are expected to become significant as the modified Reynolds number approaches and exceeds unity, which occurs at high speeds and pressures in high molecular weight gases (Appendix A).

The Reynolds number can be used to evaluate the feasibility of turbulence affecting the apparent coefficient of friction. The onset of turbulence causes an increase in the apparent coefficient of friction and is expected to occur as the Reynolds number approaches 1500 [13] for Couette flow.

Finally, the Taylor number is used to evaluate the feasibility of primary flow instability affecting the apparent friction coefficient. Instability of the primary flow causes an increase in apparent friction coefficient and is expected to occur as the Taylor number approaches 41.3 [13] for Couette flow between concentric rotating cylinders.

Calculation of the modified Reynolds number, Reynolds number, and Taylor number is complicated by the inability to directly measure the film thickness. A characteristic film thickness is approximated using the equation below, which is derived from the scaling of load capacity and pressure, and is described more completely in Appendix D.

Film Thickness Scaling

$$H \sim \sqrt{\frac{\mu_{amb} U D^2 L_a}{W_t}}$$

Chapter 3

Results

3.1 Raw Data. Prior to testing, the journal is balanced, conditioned, and characterized according to the procedures described above. Measurements of preload and friction coefficient as well as test conditions for each gas appear in Table 2. Bearing performance data in helium and nitrogen data were taken using the newly designed titanium alloy journal, while carbon dioxide data was taken using the original nickel based super-alloy journal described in Appendix B, limiting the maximum speed to 22 krpm and resulting in a different preload for carbon dioxide tests. When comparing carbon dioxide torque data with helium and nitrogen torque data, it is necessary to normalize the friction force by the total bearing load, which is equivalent to comparing the apparent friction coefficient.

Test Gas	Load (N)	Preload (N)	Total Load (N)	Max Pressure (psi)	Max Rotational Speed (krpm)	Friction Coefficient
Helium	13.97	11.73	25.70	645	42	.33
Nitrogen	13.97	11.73	25.70	660	42	.33
Carbon Dioxide	13.97	4.76	18.73	700	22	.31

Table 3 – Bearing Characterization Data and Test Conditions

Another speed limitation, due to windage losses in the rotor-stator gap of the motor, limited the top motor speed during high pressure nitrogen testing to 32 krpm above 450 psi and 41 krpm between 300 psi and 450 psi. The effect of windage losses in rotor-stator systems decreases with decreasing molecular weight, allowing helium testing to span the entire operating range of speed and pressure. Performance testing in carbon

dioxide was limited in speed by the rotordynamic issues discussed in Appendix B, so windage losses were not an issue.

Results of the current study are compared to those of Dobler and Miller by examining pressure contours of power loss versus speed (Fig. 20-22). Power loss is shown to increase with pressure most substantially in carbon dioxide, less substantially in nitrogen, and almost unnoticeably in helium. This supports the results of the Dobler and Miller study which show an increase in power loss with both increasing ambient pressure and increasing molecular weight (Fig. 3) [11]. Furthermore, at low speeds no gas appears to show a significant increase in power loss with pressure.

Figure 20 – Power Loss versus Speed in Helium

Figure 21 – Power Loss versus Speed in Nitrogen

Figure 22 – Power Loss versus Speed in Carbon Dioxide
(Speed limited by rotordynamic issues described in Appendix B)

The effects of speed can be seen more easily by examining speed contours of power loss versus pressure (Fig. 23 - 25). In carbon dioxide, power loss begins to increase with pressure at approximately 15 krpm, while in nitrogen power loss does not begin to increase with pressure until the speed reaches approximately 19 krpm, and in helium power loss does not appear to increase with pressure for any of the speeds tested (maximum test speed of 42 krpm). Figures 20 – 25 suggest that increasing the molecular weight of the test gas both amplifies pressure effects on power loss and decreases the speed at which these pressure effects are seen.

Figure 23 – Power Loss versus Pressure in Helium

Figure 24 – Power Loss versus Pressure in Nitrogen

Figure 25 – Power Loss versus Pressure in Carbon Dioxide
(Speed limited by rotordynamic issues described in Appendix B)

This result can be seen most readily by comparing the apparent friction coefficient (friction force divided by total load) versus pressure at various speeds for each gas (Fig. 26-28). At 10 krpm no gas shows the variation in friction coefficient with pressure, at 17 krpm friction coefficient increases with pressure in carbon dioxide, but not in any other gas, and at 22 krpm friction coefficient increases with pressure most substantially in carbon dioxide, less substantially in nitrogen, with no variation of friction coefficient with pressure seen in helium.

Figure 26 – Apparent Friction Coefficient versus Pressure at 10 krpm for All Test Gases

Figure 27 – Apparent Friction Coefficient versus Pressure at 17 krpm for All Test Gases

Figure 28 – Apparent Friction Coefficient versus Pressure at 22 krpm for All Test Gases

3.2 Kinematic Viscosity Analysis. In order for bearing performance data taken in helium, nitrogen, and carbon dioxide to be effectively applicable to other gases, it is necessary to identify the gas property most relevant to bearing performance. As a first approximation, the gas film is assumed to be isothermal and, therefore, the scaling of the governing equations identifies the kinematic viscosity as the relevant gas property (Appendix A). Kinematic viscosity is inversely proportional to both molecular weight and pressure, so torque and power loss are expected to decrease with increasing kinematic viscosity, based on the conclusions of Dobler and Miller and this data set.

A necessary but not sufficient condition for justifying the assumption of an isothermal gas film is that the friction coefficient is a function of kinematic viscosity only for given operating conditions. In other words, the plots of friction coefficient versus kinematic viscosity must be independent of test gas. This result is observed experimentally, as seen in Figures 29-32. This data also shows the effects of speed more completely, by extending the range of kinematic viscosity beyond values achievable by testing in a single gas for the available operating conditions.

Figure 29 – Apparent Friction Coefficient versus Kinematic Viscosity at 10 krpm for All Test Gases

Figure 30 – Apparent Friction Coefficient versus Kinematic Viscosity at 17 krpm for All Test Gases

Figure 31 – Apparent Friction Coefficient versus Kinematic Viscosity at 22 krpm for All Test Gases

Figure 32 – Apparent Friction Coefficient versus Kinematic Viscosity for All Test Gases and All Speeds

This data confirms that the magnitude of the friction coefficient is a decreasing function of kinematic viscosity for given operating conditions, as expected. The above data also shows that the coefficient of friction remains constant for kinematic viscosities above a critical value. This critical value is independent of test gas and is an increasing function of speed.

At a given pressure, increasing molecular weight decreases kinematic viscosity, assuming an ideal gas. Therefore, bearings operating in high molecular weight gases have higher friction coefficients and reach the critical value of kinematic viscosity at lower speeds than bearings operating in low molecular weight gases. Over the pressure and speed ranges tested, bearings operating in helium do not show a significant increase in friction with increasing pressure because the kinematic viscosity of helium remains above the critical value.

3.3 Non-Dimensional Analysis. The apparent friction coefficient of a foil bearing has been shown to be a function of both kinematic viscosity and speed, for a given load. The scaling of the Navier-Stokes equations identifies the modified Reynolds number as the dimensionless parameter which takes into account the combined effects of kinematic viscosity and speed (Appendix A). The modified Reynolds number accounts for the combined effects of speed and kinematic reasonably well as seen by the agreement of data collected from all three gases throughout the entire range of speed and pressure (Fig. 33). Although there is significant scatter present in the data, it appears that the apparent friction coefficient trends with the modified Reynolds number. Furthermore, friction coefficient is not a function of the modified Reynolds number until the modified Reynolds number approaches unity, which agrees with the results predicted

by the Navier-Stokes equations (Appendix A). This suggests that laminar inertial effects are a feasible cause of the increasing trend of friction coefficient with increasing pressure, speed, and molecular weight.

Figure 33 – Apparent Friction Coefficient versus Modified Reynolds Number for All Test Gases and All Speeds

The effect of transition to turbulent flow on the apparent friction coefficient is evaluated by analyzing the apparent friction coefficient with the Reynolds number as the independent variable (Fig. 34). Figure 34 shows an increase in apparent friction coefficient as the Reynolds number approaches 1500, the value predicted for turbulent transition in Couette flow [13]. This suggests that a transition from laminar to turbulent flow could be responsible for the increase in friction coefficient that is seen when

bearings operate at high pressures and high speeds, using high molecular weight gas lubricants.

Figure 34 – Apparent Friction Coefficient versus Reynolds Number for All Test Gases and All Speeds

The possible effect of instability of the primary flow on the apparent friction coefficient is evaluated by analyzing the apparent friction coefficient with the Taylor number as the independent variable (Fig. 35). Figure 35 shows that the apparent friction coefficient begins to increase with Taylor number as the Taylor number approaches 7, a value significantly lower than the predicted value for the onset of instability, 41.3. For the tested geometry, the effects of inertia on the flow field and/or the transition to turbulence appear to precede the onset of primary flow instability, suggesting that flow instability is not a likely cause of the increase in friction coefficient seen during testing.

Figure 35 – Apparent Friction Coefficient versus Taylor Number for All Test Gases and All Speeds

Chapter 4

Discussion

4.1 Discussion of results. A data set has been collected which takes into account the combined effects of ambient pressure, gas properties, and speed. This data set extends the range of available experimental data into operating ranges that are associated with high pressure foil bearing applications such as the Closed Brayton Cycle, which allows high pressure foil bearing performance models to be revised and updated.

Prior experimental data has shown that bearing friction force increases substantially with both increasing ambient pressure and increasing molecular weight. This increase in friction is not predicted by the Reynolds equation, so the validity of the simplifying assumptions of the Reynolds equation are examined more rigorously.

It is shown that the assumption of a small modified Reynolds number and neglect of inertial terms in the Navier-Stokes equations is invalid at high pressures and speeds. Retaining the inertial terms and assuming an isothermal gas film identifies the kinematic viscosity as the significant gas property and the modified Reynolds number as the significant dimensionless parameter predicted by the Navier-Stokes equations to effect bearing torque. Analysis of bearing friction with respect to these two parameters reveals that, for given operating conditions, bearing friction trends with kinematic viscosity and, for a given loading condition, friction trends with modified Reynolds number. Increases in friction coefficient occur as the modified Reynolds number approaches and exceeds unity, which supports the hypothesis that laminar inertial effects cause the observed increase in friction coefficient.

Although the isothermal gas film approximation used in this analysis is not proven to be valid, the trending of friction coefficient with kinematic viscosity satisfies a necessary but not sufficient condition for validation of this assumption, and thus supports its use.

The transition to turbulent flow is examined as a possible cause of the increase in bearing friction by assuming Couette flow and evaluating changes in bearing friction with respect to the Reynolds number. Increases in friction are consistent with predicted transitions to turbulence, which supports the hypothesis that transition to turbulence causes the observed increase in friction coefficient.

The effect of instability of the primary flow is evaluated by once again assuming Couette flow and evaluating frictional force with respect to the Taylor number. The apparent friction coefficient begins to increase at Taylor numbers below those predicted for the onset of instability. Taylor instability is not a likely cause of the observed increasing trend of friction coefficient with pressure, molecular weight, and rotational speed. However, for other bearing geometries and loading conditions, it is possible for flow instability to precede the onset of turbulence and/or laminar inertial effects, and could therefore result in increased bearing friction at lower speeds and pressures than would otherwise be predicted. It is cautioned that the numerical values of the Reynolds number, modified Reynolds number, and Taylor number are determined using an estimated film thickness based on the Reynolds equation itself.

4.2 Concluding remarks. High pressure foil bearing research in alternate gases aims to generate an empirical database that can be used to accurately predict bearing torque given

bearing dimensions, bearing characterization parameters (i.e. preload), gas properties, ambient pressure, applied load, and speed. The current study takes a first step toward this goal by evaluating bearing performance with respect to the modified Reynolds number, which lumps the effects of molecular weight, ambient pressure, viscosity, and speed into a single parameter. The modified Reynolds number, Reynolds number, and Taylor number are then used to evaluate the feasibility of laminar inertial effects, transition to turbulence, and primary flow instability as the mechanism of increasing torque. This data set suggests that laminar inertial effects, the onset of turbulence, or some combination of the two, causes the increase in frictional torque seen as pressure and test gas molecular weight are increased. This result can be used to evaluate bearing performance models to ensure that they take into account the combined effects of speed, ambient pressure, and molecular weight appropriately.

Although the mechanism which causes the torque to increase with increasing speed, ambient pressure, and molecular weight has not been definitively identified, it is clear that bearing efficiency decreases and the thermal load on the bearing increases quickly as the Reynolds number and/or modified Reynolds number exceed their critical values and bearing torque increases. The effect of bearing geometry, applied load, and characterization parameters on this critical value remains undetermined as well, but it is clear that the design of systems which incorporate foil bearings operating at high pressures and speeds, with high molecular weight lubricants must identify this critical value to guarantee that bearings are operating in regions of acceptably low torque and power loss, preventing thermal runaway and bearing failure.

Appendix A

Derivation of the Governing Equations

The Navier-Stokes equations form the basis of the science of fluid mechanics. Typically, hydrodynamic lubrication is analyzed by scaling the Navier-Stokes equations, assuming or showing that body forces, inertial forces, and unsteady effects can be neglected and reducing these equations to the familiar Reynolds equation. The following derivation of the Reynolds equation from the continuity equation and Navier-Stokes equations for a slider bearing in Cartesian coordinates is included in order to identify scaling parameters that will aid in the data analysis and to explicitly identify the assumptions that go into the Reynolds equation in order to evaluate their validity in high pressure applications.

The continuity equation and full Navier-Stokes equations appear below for an isotropic Newtonian fluid [13]:

Continuity Equation

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} = 0$$

Navier-Stokes Equations

$$\rho \frac{Du}{Dt} = X - \frac{\partial p}{\partial x} + \frac{\partial}{\partial x} \left[\mu \left(2 \frac{\partial u}{\partial x} - \frac{2}{3} \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) \right) \right] + \frac{\partial}{\partial y} \left[\mu \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right) \right] + \frac{\partial}{\partial z} \left[\mu \left(\frac{\partial w}{\partial x} + \frac{\partial u}{\partial z} \right) \right]$$

$$\rho \frac{Dv}{Dt} = Y - \frac{\partial p}{\partial y} + \frac{\partial}{\partial y} \left[\mu \left(2 \frac{\partial v}{\partial y} - \frac{2}{3} \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) \right) \right] + \frac{\partial}{\partial z} \left[\mu \left(\frac{\partial v}{\partial z} + \frac{\partial w}{\partial y} \right) \right] + \frac{\partial}{\partial x} \left[\mu \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right) \right]$$

$$\rho \frac{Dw}{Dt} = Z - \frac{\partial p}{\partial z} + \frac{\partial}{\partial z} \left[\mu \left(2 \frac{\partial w}{\partial z} - \frac{2}{3} \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) \right) \right] + \frac{\partial}{\partial x} \left[\mu \left(\frac{\partial w}{\partial x} + \frac{\partial u}{\partial z} \right) \right] + \frac{\partial}{\partial y} \left[\mu \left(\frac{\partial v}{\partial z} + \frac{\partial w}{\partial y} \right) \right]$$

The following non-dimensional scaling parameters are then substituted into the continuity equation:

Scaling Parameters

$$x^* = \frac{x}{L_c}, y^* = \frac{y}{H}, z^* = \frac{z}{L_a}, P^* = \frac{P}{P_{ref}}, \rho^* = \frac{\rho}{\rho_{amb}}, u^* = \frac{u}{U}, v^* = \frac{v}{V_{ref}}, w^* = \frac{w}{W_{ref}},$$

$$t^* = \frac{t}{\theta}, X^* = \frac{X}{\rho g_x}, Y^* = \frac{Y}{\rho g_y}, Z^* = \frac{Z}{\rho g_z}, \mu^* = \frac{\mu}{\mu_{amb}}$$

This scaling leads to the following non-dimensional form of the continuity equation:

Dimensionless Continuity

$$\left(\frac{L_c}{U \theta_{cont}} \right) \frac{\partial \rho^*}{\partial t^*} + \frac{\partial(\rho^* u^*)}{\partial x^*} + \left(\frac{V_{ref} L_c}{UH} \right) \frac{\partial(\rho^* v^*)}{\partial y^*} + \left(\frac{W_{ref} L_c}{UL_a} \right) \frac{\partial(\rho^* w^*)}{\partial z^*} = 0$$

This non-dimensional equation is used to find the characteristic time scale for the decay of the transient term, θ_{Cont} , and to find an appropriate scaling for the characteristic y and z velocities, V_{ref} and W_{ref} , respectively. The maximum value of the time constant for High Pressure Rig bearing geometry and operating speeds, is approximately one-hundredth of a second. Therefore, transient effects, in the continuity equation, become negligible after a few hundredths of a second and can be neglected when analyzing steady-state bearing performance. The steady-state non-dimensional continuity equation appears below.

Dimensionless Continuity Equation

$$\frac{\partial(\rho^* u^*)}{\partial x^*} + \frac{\partial(\rho^* v^*)}{\partial y^*} + \frac{\partial(\rho^* w^*)}{\partial z^*} = 0$$

Where the characteristic y and z velocities have been defined as:

Definition of Characteristic Velocities in the Y and Z directions

$$V_{ref} \equiv U \left(\frac{H}{L_c} \right), W_{ref} \equiv U \left(\frac{L_a}{L_c} \right)$$

Using the above scaling of vertical and axial velocities gives the following form of the dimensionless x-momentum equation.

Dimensionless X-Momentum Equation

$$\begin{aligned} \rho^* \left[\left(\frac{\rho_{amb} H^2}{\theta \mu_{amb}} \right) \frac{\partial u^*}{\partial t^*} + \left(\frac{\rho_{amb} U H^2}{\mu_{amb} L_c} \right) \left(u^* \frac{\partial u^*}{\partial x^*} + v^* \frac{\partial u^*}{\partial y^*} + w^* \frac{\partial u^*}{\partial z^*} \right) \right] = & \left(\frac{\rho_{amb} g_x H^2}{\mu_{amb} U} \right) X^* - \left(\frac{P_{ref} H^2}{L_c \mu_{amb} U} \right) \frac{\partial p^*}{\partial x^*} \\ + \frac{\partial}{\partial x^*} \left[\left(\frac{H^2}{L_c^2} \right) \mu^* \left(2 \frac{\partial u^*}{\partial x^*} - \left(\frac{2}{3} \right) \left(\frac{\partial u^*}{\partial x^*} + \frac{\partial v^*}{\partial y^*} + \frac{\partial w^*}{\partial z^*} \right) \right) \right] + \frac{\partial}{\partial y^*} \left[\mu^* \frac{\partial u^*}{\partial y^*} + \left(\frac{H^2}{L_c^2} \right) \mu^* \frac{\partial v^*}{\partial x^*} \right] + \frac{\partial}{\partial z^*} \left[\left(\frac{H^2}{L_c^2} \right) \mu^* \frac{\partial w^*}{\partial x^*} + \left(\frac{H^2}{L_a^2} \right) \mu^* \frac{\partial u^*}{\partial z^*} \right] \end{aligned}$$

Reduction of this equation to the Reynolds equation requires the assumption that the coefficients of the transient term, inertial terms, body force term, and the film thickness ratios $(H/L_c)^2$ and $(H/L_a)^2$ are very small. Since the horizontal component of the gravity field is zero, the assumption of negligible body force is valid. The film thickness ratios are both indeed much less than 1 ($\sim 1 \times 10^{-6}$), which justifies the neglect of terms with these ratios as their coefficients. Table 4 shows the maximum value of the time constant for each of the three momentum equations over the density range of the High Pressure Rig, using various estimated film thicknesses.

ρ_{amb} (Kg / m ³)	H (μ m)	Θ_{max} (μ s)
0.2	5.08	.34
200	5.08	340
0.2	12.7	2.2
200	12.7	2200
0.2	50.8	34
200	50.8	34000

Table 4 – Time Constant for Transient Terms in the Navier-Stokes Equations
(Viscosity 15×10^{-6} kg / m-s)

Therefore, transient effects become negligible after a few tenths of a second and are neglected in the steady-state analysis.

Inertial terms are usually neglected in the Navier-Stokes equations by showing that the modified Reynolds number, defined below, is much less than unity. Table 5 shows the modified Reynolds number for High Pressure Rig operating conditions and various estimated film thicknesses [18]:

Modified Reynolds Number Definition

$$Re_m \equiv \left(\frac{\rho_{amb} U H^2}{\mu_{amb} L_c} \right)$$

Ω (krpm)	ρ_{amb} (Kg / m ³)	H (μ m)	Rem
5	0.2	5.08	2.87E-05
5	200	5.08	2.87E-02
20	0.2	5.08	1.15E-04
20	200	5.08	1.15E-01
40	0.2	5.08	2.29E-04
40	200	5.08	2.29E-01
5	0.2	12.7	1.79E-04
5	200	12.7	1.79E-01
20	0.2	12.7	7.17E-04
20	200	12.7	7.17E-01
40	0.2	12.7	1.43E-03
40	200	12.7	1.43E+00
5	0.2	50.8	2.87E-03
5	200	50.8	2.87E+00
20	0.2	50.8	1.15E-02
20	200	50.8	1.15E+01
40	0.2	50.8	2.29E-02
40	200	50.8	2.29E+01

Table 5 – Modified Reynolds Number at Various Operating Conditions
(Viscosity 15×10^{-6} kg / m-s)

This table shows that for testing at low speeds in low density gases (low pressure and/or molecular weight) the low modified Reynolds number approximation is certainly valid. However, during high speed testing in high density gas (high pressure and/or molecular weight) the low Reynolds number approximation may break down, depending

on the actual magnitude of the film thickness. Therefore, since the Reynolds equation assumes small modified Reynolds number, it may fail to accurately predict the performance of foil bearings operating at high speeds and pressures. The break down of this assumption provides useful information for high pressure foil bearing performance analysis and will be discussed more completely following the conclusion of the derivation of the Reynolds equation.

In order to complete the derivation of the governing equations used in current foil bearing models, the small modified Reynolds number assumption is made, resulting in the following simplified form of the x-momentum equation:

Dimensionless X-Momentum Equation

$$\frac{\partial p^*}{\partial x^*} = \frac{\partial}{\partial y^*} \left[\mu^* \frac{\partial u^*}{\partial y^*} \right]$$

Where the reference pressure, P_{ref} , is defined as follows:

Reference Pressure Definition

$$P_{ref} \equiv \left(\frac{\mu_{amb} U L_c}{H^2} \right)$$

This pressure scaling is then substituted into the Y and Z-momentum equations, giving the following non-dimensional results:

Dimensionless Y-Momentum Equation

$$\begin{aligned} \rho^* \left[\left(\frac{\rho_{amb} H^4}{L_c^2 \theta \mu_{amb}} \right) \frac{\partial v^*}{\partial t^*} + \left(\frac{\rho_{amb} U H^4}{\mu_{amb} L_c^3} \right) \left(u^* \frac{\partial v^*}{\partial x^*} + v^* \frac{\partial v^*}{\partial y^*} + w^* \frac{\partial v^*}{\partial z^*} \right) \right] &= \left(\frac{\rho_{amb} g_y H^3}{L_c \mu_{amb} U} \right) Y^* - \frac{\partial p^*}{\partial y^*} \\ + \frac{\partial}{\partial y^*} \left[\left(\frac{H^2}{L_c^2} \right) \mu^* \left(2 \frac{\partial v^*}{\partial y^*} - \left(\frac{2}{3} \right) \left(\frac{\partial u^*}{\partial x^*} + \frac{\partial v^*}{\partial y^*} + \frac{\partial w^*}{\partial z^*} \right) \right) \right] &+ \frac{\partial}{\partial z^*} \left[\left(\frac{H^4}{L_c^2 L_u^2} \right) \mu^* \frac{\partial v^*}{\partial z^*} + \left(\frac{H^2}{L_c^2} \right) \mu^* \frac{\partial w^*}{\partial y^*} \right] + \frac{\partial}{\partial x^*} \left[\left(\frac{H^2}{L_c^2} \right) \mu^* \frac{\partial u^*}{\partial y^*} + \left(\frac{H^4}{L_u^4} \right) \mu^* \frac{\partial v^*}{\partial x^*} \right] \end{aligned}$$

Dimensionless Z-momentum Equation

$$\rho^* \left[\left(\frac{\rho_{amb} H^2}{\theta \mu_{amb}} \right) \frac{\partial w^*}{\partial t^*} + \left(\frac{\rho_{amb} U H^2}{\mu_{amb} L_a} \right) \left(u^* \frac{\partial u^*}{\partial x^*} + v^* \frac{\partial u^*}{\partial y^*} + w^* \frac{\partial u^*}{\partial z^*} \right) \right] = \left(\frac{\rho_{amb} g_z H^2 L_c}{\mu_{amb} U L_a} \right) Z^* - \left(\frac{L_c^2}{L_a^2} \right) \frac{\partial p^*}{\partial z^*} + \frac{\partial}{\partial z^*} \left[\left(\frac{H^2}{L_a^2} \right) \mu^* \left(2 \frac{\partial w^*}{\partial z^*} - \left(\frac{2}{3} \right) \left(\frac{\partial u^*}{\partial x^*} + \frac{\partial v^*}{\partial y^*} + \frac{\partial w^*}{\partial z^*} \right) \right) \right] + \frac{\partial}{\partial x^*} \left[\left(\frac{H^2}{L_c^2} \right) \mu^* \frac{\partial w^*}{\partial x^*} + \left(\frac{H^2}{L_a^2} \right) \mu^* \frac{\partial u^*}{\partial z^*} \right] + \frac{\partial}{\partial y^*} \left[\left(\frac{H^2}{L_a^2} \right) \mu^* \frac{\partial v^*}{\partial z^*} + \mu^* \frac{\partial w^*}{\partial y^*} \right]$$

Although the gravitational component in the y-direction is not zero, the maximum value of the coefficient in front of the body force term is much less than 1 ($\sim 1 \times 10^{-5}$), and therefore, the body force in the y-direction will be neglected. Arguments similar to those used to simplify the x-momentum equation can be used to further simplify the y and z-momentum equations to the forms shown below:

Dimensionless Y-Momentum Equation

$$\frac{\partial p^*}{\partial y^*} = 0$$

Dimensionless Z-momentum Equation

$$\left(\frac{L_c^2}{L_a^2} \right) \frac{\partial p^*}{\partial z^*} = \frac{\partial}{\partial y^*} \left[\mu^* \frac{\partial w^*}{\partial y^*} \right]$$

The Y-momentum equation reveals that the pressure does not vary in the y-direction, and therefore the x and z-momentum equations can be integrated and solved for their respective velocity components. Integrating the dimensional form of the continuity equation across the film thickness, and substituting the dimensional x and z velocity components gives the Reynolds equation in Cartesian coordinates. The form of the Reynolds equation shown below assumes a slider in motion with a stationary bearing surface:

Reynolds Equation

$$\frac{\partial}{\partial x} \left(\frac{\rho h^3}{12\mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left(\frac{\rho h^3}{12\mu} \frac{\partial p}{\partial z} \right) = \frac{\partial}{\partial x} \left[\frac{\rho h U_s}{2} \right] + \frac{\partial}{\partial z} \left[\frac{\rho h W_s}{2} \right]$$

This equation assumes a small modified Reynolds number, an assumption that may break down at high speed and high gas density as discussed above. Therefore, it is useful to analyze the most simplified version of the Navier-Stokes equation that does not require the small modified Reynolds number assumption:

Dimensionless X-Momentum Equation

$$\left(\frac{\rho_{amb} U H^2}{\mu_{amb} L_c} \right) \rho^* \left(u^* \frac{\partial u^*}{\partial x^*} + v^* \frac{\partial u^*}{\partial y^*} + w^* \frac{\partial u^*}{\partial z^*} \right) = \left(\frac{P_{ref} H^2}{L_c \mu_{amb} U} \right) \frac{\partial p^*}{\partial x^*} + \frac{\partial}{\partial y^*} \left[\mu^* \frac{\partial u^*}{\partial y^*} \right]$$

Dimensionless Y-Momentum Equation

$$\frac{\partial p^*}{\partial y^*} = 0$$

Dimensionless Z-momentum Equation

$$\left(\frac{\rho_{amb} U H^2}{\mu_{amb} L_a} \right) \rho^* \left(u^* \frac{\partial u^*}{\partial x^*} + v^* \frac{\partial u^*}{\partial y^*} + w^* \frac{\partial u^*}{\partial z^*} \right) = \left(\frac{L_c^2}{L_a^2} \right) \left(\frac{P_{ref} H^2}{L_c \mu_{amb} U} \right) \frac{\partial p^*}{\partial z^*} + \frac{\partial}{\partial y^*} \left[\mu^* \frac{\partial w^*}{\partial y^*} \right]$$

This form of the Navier-Stokes equations retains the inertial terms neglected by the Reynolds equation. Retaining these terms identifies kinematic viscosity (μ_{amb}/ρ_{amb}) as the only gas property relevant to bearing performance. If gas properties other than the kinematic viscosity affect bearing performance (i.e. specific heat, thermal conductivity), it suggests that the isothermal gas film approximation may be invalid, and that accurate calculation of the bearing torque may require coupling of the Navier-Stokes equation and the energy equation. Furthermore, this form of the equation identifies the modified Reynolds number as a relevant non-dimensional parameter. Bearing torque is expected to be predicted accurately by the Reynolds equation as long as the modified Reynolds number remains small and is expected to deviate from Reynolds equation predictions as

the modified Reynolds number approaches and exceeds unity. Identification of the kinematic viscosity and modified Reynolds number as parameters relevant bearing torque guides the analysis of high pressure foil bearing performance data.

Appendix B

Rotordynamic Considerations

The original journals used on the High Pressure Rig were duplicates of journals tested on other foil journal bearing rigs. These journals were not designed for the collet fastening arrangement used on the High Pressure Rig; therefore an adapter and tie-bolt were necessary to accommodate the journals. A drawing of the original journal and adapter assembly appear in (Fig 36). When the original journal design was run on the High Pressure Rig spindle, large amplitude vibrations were seen at and above 23,000 RPM (383 Hz). Analysis of waterfall plots of accelerometer data revealed that the large vibration occurred at the synchronous frequency (Fig 37), suggesting that imbalance, despite being minimized, was exciting a rotordynamic mode of the shaft and journal assembly. The effect of overhung journal mass on the resonant frequencies was a key design consideration, and was further investigated through a series of experiments described below.

Figure 36 –Nickel-Based Super Alloy Journal Diagram

Figure 37 – Waterfall Plot for Nickel-Based Super Alloy Journal Vibrations

First, fixtures of different overhung masses were run in the collet, and their vibration levels were recorded as a function of speed (Fig. 38). This data shows that the resonant frequencies varied substantially with overhung mass, suggesting that the overhung mass of the original journal was too great, and a new journal would have to be designed. A finite element analysis was then used to determine the resonant frequencies of a new journal design. However, it was first necessary to know the mode shape being excited by the journal imbalance. A preliminary model assumed that the collet was infinitely stiff, and that the imbalance was exciting a bending mode of the journal itself. This assumption resulted in predicted resonant frequencies much higher than those measured experimentally, so the model was amended to include collet of finite stiffness, thus allowing rigid body modes of the journal to be excited. The updated model

produced results that agreed reasonably well with experimental values for a collet stiffness of 8.1×10^7 N/m (4.5×10^5 lbf/in). Predicted and measured resonant frequencies can be seen in Table 6.

Figure 38 – Vibration Levels for Various Overhung Masses

Collet Fixture	Predicted Bending Mode Resonant Frequency (Hz)	Predicted Rigid Body Resonant Frequency (Hz)	Actual Resonance Frequency (Hz)
Adapter Only	1208	611	583
Adapter w/ Additional Mass	967	527	533
Journal w/Adapter	964	334	383
Titanium Journal	N/A	800	>700 Hz (maximum drive frequency)

Table 6 – Predicted and Measured Resonance Frequencies for Nickel-Based Super Alloy Journal
(Collet Stiffness used in Rigid Body Calculation was 4.5×10^5 lbs/in)

The estimated collet stiffness was then used in another finite element model to predict the resonance frequencies of the new journal design. The new journal design

improved on the original design by eliminating the need for the adapter and tie-bolt and removing material from the journal interior, thus reducing overhung mass (Fig. 10). Since the High Pressure Rig does not have high temperature capability, a titanium alloy replaced the high temperature nickel based super alloy used in the original journal design, resulting in further mass reduction. The new design reduced the mass from 320 g to 90 g (.7 lbs. to .2 lbs) and the reduced the length from center of mass to the end of the collet from 2.8 cm to 1.9 cm (1.1 in. to .7 in.). The predicted resonance frequency for the new design was 800 Hz, which lies outside of the operating range of the High Pressure Rig, compared to 334 Hz predicted for the original journal design. The new journal was then fabricated, balanced and tested, resulting in vibration levels lower than those of the spindle rotating alone throughout most of the operating range (Fig. 39). Therefore, the new titanium journal has provided the rotordynamic solution necessary to restore the originally specified high speed capability of the High Pressure Rig (Fig. 40).

Figure 39 – Vibration Levels of Titanium Journal

Figure 40 – Evolution of High Pressure Rig Operational Envelope

Appendix C

Load Capacity Coefficient Limitation

A common measure of bearing performance is the Load Capacity Coefficient, described by DellaCorte and Valco (Ref. 6). This coefficient is a measure of bearing load capacity at a given speed and is defined as:

$$\mathcal{D}_j \equiv \frac{W_t}{L_a D^2 \Omega_L}$$

Accurate measurement of the load capacity of foil bearings, as described in Radil & DellaCorte [14], involves running the bearing at high speed, and increasing load until the bearing can no longer support the load while running at steady state. The maximum load which the bearing can support at steady state (including preload) is the load capacity, W_t , and the speed at which it can support this load is called the lift off speed, Ω_L . The High Pressure Rig does not currently have the capability to vary load during a test, so the load capacity coefficient is estimated based on the qualitative guidelines established in Dellacorte and Valco and Radil and Dellacorte [6][14].

Measurement of the load capacity coefficient on the High Pressure Rig was attempted by decreasing the speed for a constant load (Fig. 41). The spindle was run well above an estimated lift-off speed with a known radial load and speed was reduced, incrementally, until torque and power loss can no longer maintain a steady value [14] (Fig 42). The inability of the bearing to produce steady torque and power loss shows that the hydrodynamic film is breaking down and that torque and power loss runaway are soon to follow. Therefore, the lowest speed at which power loss maintained a steady

value was used to estimate Ω_L (lift off speed), and the radial load and preload were used to calculate W_t . These measurements, as well as bearing dimensions were then used to estimate the load capacity coefficient. The results obtained using this method do not agree with published results or results of similar test rigs with variable load capability, and therefore were not used to calculate the load capacity coefficient.

Figure 41 – Torque and Speed Time Trace from Load Capacity Testing

Figure 42 – Load Capacity Test Data
5.5 Krpm data reaches a steady value while 4 krpm data becomes unstable.
4.5 krpm is taken to be the lift off speed.

The following scenario is offered merely as a possible explanation of the load capacity coefficient discrepancy. For the sake of this research it is enough to know that the discrepancy exists and that load capacity coefficient measured on the High Pressure Rig does not conform to the conventions established in prior work, and therefore, will not be used.

During high-speed, variable-load, load capacity testing, increased friction causes rapid power loss increase and, therefore, rapid heat transfer into the journal. This increase in heat transfer causes the journal temperature to increase, the journal to expand, which leads to more rubbing contact, resulting in rapid thermal runaway. During low speed, constant load, load capacity tests, increased friction results in much smaller power loss increases, and much less heating of the journal, which could allow the bearing to support the journal load at steady state despite being in a state of solid-solid contact,

resulting in inaccurate measurements of lift-off speed. However, this hypothesis has not been rigorously tested.

Appendix D

Film Thickness Scaling

Traditional hydrodynamic lubrication takes place in a flow regime in which the effects of inertia can be neglected. However, experimental results show that as the density of the gaseous lubricant increases so does bearing torque. This suggests that inertial effects are substantial and must be taken into account when analyzing high pressure foil bearing performance. Inertia could potentially play a roll in bearing performance through laminar inertial effects altering the velocity field, transition from laminar to turbulent flow, or through primary flow instability. All three of these possibilities are investigated using different dimensionless parameters, describe below, each of which require knowledge of the film thickness.

Modified Reynolds Number – Inertial Flow

$$\text{Re}_m \equiv \frac{\rho U H^2}{\mu_{amb} D}$$

Reynolds Number - Turbulent Transition

$$\text{Re} \equiv \frac{\rho_{amb} U H}{\mu_{amb}}$$

Taylor Number – Instability

$$Ta \equiv \frac{\rho_{amb} U H}{\mu_{amb}} \sqrt{\frac{2H}{D}}$$

Film thickness is not measured on the High Pressure Rig, so in order to calculate the Reynolds number, Taylor number, and modified Reynolds number it is necessary to estimate a characteristic film thickness from scaling arguments, as shown below:

Load Capacity Scaling

$$W_t = (W_{vert} + W_{preload}) \sim P_{ref} A_{char}$$

Reynolds Equation Pressure Scaling

$$P_{ref} \sim \frac{\mu_{amb} UD}{H^2}$$

Bearing Projected Area

$$A_{char} \sim (L_a)(D)$$

Which gives the following scaling for H:

Film Thickness Scaling

$$H \sim \sqrt{\frac{\mu_{amb} UD^2 L_a}{W_t}}$$

This is an expression for a characteristic film thickness, H, based on test conditions and bearing geometry. This approximation assumes that the circumferentially uniform preload and the vertical applied load can be added directly. It further assumes that the viscous pressure scaling, obtained for modified Reynolds numbers much less than unity, is appropriate although the modified Reynolds number approaches and exceeds unity for the conditions tested.

A comparison of this approximation of the film thickness to experimental results published by Ruscitto et al [18] and computational results predicted by the Carpino algorithm [12] are shown in Figures 43 and 44 respectively:

Figure 43 – Comparison of Scaled Characteristic Film Thickness to Measured Minimum Film Thickness
(Measured Values Taken from Ruscitto et al [18])

Figure 44 – Comparison of Scaled Characteristic Film Thickness to Calculated Minimum Film Thickness
(Calculated Using Carpino and Talmage Algorithm [12])

Since both the experimental and computational methods report minimum film thickness, it is not surprising that the estimate of the characteristic film thickness overestimates measured and computed values. Also, since no bearing preload is reported by Ruscitto, the preload assumed to be negligible compared to the applied load. However as the applied load becomes vanishingly small, this assumption is no longer valid, which explains the large deviations of the scaled film thickness from the measured film thickness as the applied load approaches zero. The film thickness scaling overestimates the measured and computed minimum film thickness by approximately a factor of 3, while adequately capturing the trend with both speed and load, suggesting that it can be used to estimate, roughly, the non-dimensional parameters of interest.

The resulting forms of the Taylor number, Reynolds Number, and modified Reynolds number based on this approximation of the film thickness are shown below:

Reynolds Number - Turbulent Transition

$$\text{Re} \equiv \rho_{amb} \left[\frac{\Omega^3 D^5 L_a}{8 \mu_{amb} W_t} \right]^{1/2}$$

Taylor Number – Stability Transition

$$\text{Ta} \equiv \text{Re} \left[\frac{2 \Omega \mu_{amb} D L_a}{W_t} \right]^{1/4}$$

Modified Reynolds Number – Inertial Flow

$$\text{Re}_m \equiv \frac{\rho \Omega^2 D^3 L_a}{4 W_t}$$

Appendix E:

Discussion of Error

Torque measurement is subject to error from the linear variable displacement transducer (LVDT), uncertainty in the measurement of the distance from the bearing center to the center of the LVDT, and repeatability of the torque measurement. The repeatability error is 30% for a 90% confidence interval, which is much larger than any of the previously mentioned sources of error.

The error in the calculation of the kinematic viscosity is introduced by uncertainty in the temperature, uncertainty in the pressure, and uncertainty in the tabulated values of viscosity and density obtained from the National Institute of Standards and Technology (NIST). An average temperature of 303 K was used for property calculation of all gases and all test conditions, resulting in a maximum error of 7 K from the measured value. Using Sutherland's formula for the viscosity of an ideal gas, this produces approximately 2% error in the calculation of viscosity. Using the ideal gas law, the error in density resulting from the error in temperature is found to be approximately 2% as well. The pressure used in property calculations is the averaged value of the pressure for the individual data point and therefore reflects only the error introduced by the transducers which is negligible by comparison. The uncertainty in the tabulated property values from NIST is approximately 2% for the viscosity and .05% for density for the conditions tested. This results in a total error of 3.5% in kinematic viscosity.

The calculation of the Modified Reynolds number, Reynolds number, and Taylor number all depend on the estimate of the film thickness, which introduces the most error. For example, the bearing diameter was chosen as the characteristic length scale for the

current work, however it can be argued that the bearing circumference is an equally valid choice. In addition, the total load, calculated by adding the preload to the applied radial load, was used as the characteristic load in the scaling of the film thickness. Since the preload acts uniformly around the circumference and the applied radial load acts on a specific circumferential location, it may not be appropriate sum the two directly when calculating total load. Therefore, differing treatment of the film thickness scaling parameters can result in significant changes in the values of modified Reynolds number, Reynolds number, and Taylor number. However, consistency in the choice of these scaling parameters between tests ensures relative accuracy between test conditions. This relative accuracy allows for qualitative comparisons between tests to be made.

Furthermore, the non-dimensional analysis used to determine the possible effects of laminar inertial effects, instability effects, and turbulence effects are not presented as a precise quantitative analysis, but rather as a more general feasibility analysis of these three mechanisms. Direct measurement of the film thickness or different selection of scaling parameters will change the absolute value of the modified Reynolds number, Reynolds number, and Taylor number, but unless deviations from the scaled values exceed an order of magnitude the conclusions of this feasibility assessment will remain the same.

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