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METHODS TO EFFECTIVELY EVALUATE MODERN JOURNAL BEARING PERFORMANCE AND ACHIEVE HIGH RELIABILITY

Patrick J. Smith
Air Products Fellow
Air Products & Chemicals
Allentown, PA, USA

Robert E. Benton, Jr.
Rotoflow Expander Technology Manager
Air Products & Chemicals
Allentown, PA, USA

Victor Obeid
Manager
Rotor Bearing Technology & Software (RBTS), Inc.
Phoenixville, PA, USA

John K. Whalen
Consultant
Houston, TX 77062, USA
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Patrick Smith is an Air Products Fellow in the Operation Excellence Technical Team at Air Products & Chemicals. He is based in Allentown, PA, and he is the Machinery Technology Manager and the Machinery Functional Lead for Global Operations. As Machinery Technology Manager he provides a balanced approach to the assessment and mitigation of technical risk for rotating machinery. As Machinery Functional Lead for Global Operations he provides solutions for difficult rotating machinery problems that have broad applications or have a significant impact on safety, reliability or performance. Patrick started his career with Ingersoll-Rand in the Pump Division in 1982 after graduating from Villanova University with a B.S. Degree in Mechanical Engineering. He joined Air Products & Chemicals, Inc. in 1986 working as a rotating machinery specialist. He went on to get a Masters in Mechanical Engineering from Villanova University in 1990. He has published dozens of articles on rotating machinery. He is also a registered professional engineer in the state of Pennsylvania and has been a member of the Turbomachinery Advisory Committee since 2016.

Robert Benton graduated in 1989 from Carnegie-Mellon University with a B.S. in Mechanical Engineering. He has spent the last 30 years in Air Products, designing and commissioning cryogenic expanders and compressors for use in air separation and hydrocarbon service. Focuses have included equipment for oxygen enriched, flammable and toxic services, rotordynamics and finite element analysis. From 1989 to 1995, he was responsible for Engineering, Testing, and Project Execution for various expander projects. In 1999, Mr. Benton became the lead engineer in the mechanical engineering group within Rotoflow, An Air Products Business (formerly the CryoMachinery department of Air Products) and in 2003 assumed the role of Head of the Engineering Group. In 2012 Mr. Benton entered into the role of Global Expander Technology Manager. He is an active member of API-617 and API-614 task forces.

John Whalen spent seven years (1981-1987) at Dresser-Rand Steam Turbines as a Product Engineer in the Large Turbine Engineering Department and then as an Analytical Engineer in the Rotordynamics Group of the Advanced Engineering and Development Department. In 1988, John accepted a position with Centritech Corporation, as the Assistant Chief Engineer. In 1989, he was promoted to Manager of Engineering. In 1991, he left Centritech to help start TCE/Turbo Components & Engineering, Inc. At TCE, he was responsible for the Engineering Department and engineering for the product lines, which include babbitted journal and thrust bearings, labyrinth seals and related engineering services. John was president and primary owner of TCE when it was acquired by John Crane in 2011, John retired from John Crane in 2015. In August of 2018 John agreed to support Gulf Coast Bearing & Seal (GCBS) in a consulting role as their Chief Engineer. John received his BSME (1981) from Rochester Institute of Technology. John is a registered Professional Engineer in the State of Texas and holds an Emeritus position on the Turbomachinery Symposium Advisory Committee.

Victor K. Obeid has over 30 years of experience in the fields of rotor dynamics, fluid-film and rolling-element bearings, machinery vibration, failure analysis and troubleshooting. He is a pioneer in the development and application of PC based state-of-the-art computer aided design software for predicting the dynamics of complex *rotor-bearing systems. A former Staff Engineer at the Franklin Institute Research Laboratories and a technical leader at RBTS, he directs government and industry sponsored projects involving design, analysis and trouble-shooting of rotating machinery systems and their components. He has been instrumental teaching and training in the fields of bearings and rotor dynamics, and their application to common as well as unique equipment design, operation, and failure analysis. He taught seminars and training sessions worldwide at rotating equipment OEM, end users, packagers, government agencies, and open seminars to machinery engineers. Mr. Obeid holds a Bachelor's degree from Drexel University and Master of Science degree from Penn State University, both in Mechanical Engineering.*

ABSTRACT

Centrifugal compressors and steam turbines commonly use hydrodynamic journal bearings. These bearings can be fixed geometry (sleeve) or tilting pad type. Most end users expect bearings to operate trouble free from major turnaround to major turnaround. Compressor manufactures continue to push the design limits of their equipment, reduce costs and optimize machine performance. However, this can increase the risk of bearing issues in new machines and in operating machines as they age. As discussed in this tutorial, industry standards offer limited guidance on how to design or apply hydrodynamic bearings to prevent wear or damage. So, how can end users evaluate bearing design performance to achieve high reliability? The objective of this tutorial is to answer this question.

INTRODUCTION

This tutorial focuses on babbitt lined hydrodynamic tilting pad journal bearings used in centrifugal compressors, gearboxes, and steam turbines. The purpose of this tutorial is to provide guidance on how to identify and avoid potential bearing issues with these types of bearings on new machines or when retrofitting a new bearing design into an existing machine. The primary audience is the specifier or end user. What end users care about most in the design of hydrodynamic bearings is that the minimum oil film thickness is enough to prevent babbitt surface wear during all operating conditions, the bearing temperature is not high enough to cause the formation of deposits on the bearing surface, and the bearing provides the dynamic characteristics required to avoid unacceptable rotor vibration levels.

The body of the tutorial is divided into seven (7) sections as follows:

- Examples of failed bearings
- Bearing terminology and design features
- Bearing requirements for industry standards
- Bearing analysis
- Case studies
- Rotordynamic considerations
- Miscellaneous topics

The tutorial begins with *examples of failed bearings* from operating machines. These examples are used to illustrate the typical problems encountered in the field and they are also used as case studies presented later in the tutorial.

The next part of the tutorial introduces *bearing terminology and design features* including:

- Journal surface speed
- Unit loading
- Diametral (assembled bearing) clearance
- Preload
- Pivot offset
- Bearing pad material
- Pad flexibility
- Enhanced features such as hot oil carryover blockers, back of pad cooling and directed lube.

This is followed by a discussion of the *bearing requirements and industry standards*. The failed bearings are then evaluated against the criteria in these leading industry standards.

Bearing failures are typically characterized by the actual damage observed, such as babbitt fatigue or babbitt wiping. However, this doesn't get to the true underlying causes, which include:

- Design
- Installation
- Operational
- Maintenance
- Quality

A design deficiency is caused by poor selection of bearing features for the application. In basic terms this means that the bearing design leads to insufficient minimum oil film thickness, excessive dynamic loads and/or high metal temperatures when operating within design conditions. Installation issues include problems such as excessive clamping (crush), improper mounting or excessive misalignment. Operational issues include running the machine at loads and/or speeds beyond the design limitations and/or supplying lubricating oil outside of design temperature and pressure limits. Maintenance issues include things such as operating the machine with degraded oil, poor oil supply piping cleanliness, oil filter degradation and problems with excessive corrosion. Quality issues includes bearings and parts that are outside of design specifications. Bearing requirements and industry standards are shown to give guidance regarding the bearing design for the application while assuming the other considerations above are properly addressed. This is followed by the topic of *bearing analysis*, which demonstrates how end users can evaluate a bearing design to ensure the bearing is properly applied and the design is able to achieve high reliability.

The key outputs from the bearing analysis are discussed and include:

- Minimum oil film thickness
- Side leakage
- Oil supply to bearing
- Frictional (power) losses
- Maximum pad temperature
- Peak oil film pressure

Case studies are then presented. As discussed, the example failed bearings are used as case studies and each is analyzed and evaluated. In these case studies, parametric studies of speed, load, clearance, preload and various oil conditions are presented to show the impact and sensitivity to small changes the various factors have on bearing performance. The results of these studies are used to provide the end user with guidance on how to evaluate a bearing design for an application to achieve high reliability. Another case study on a new machine with a suspect bearing will be used to illustrate this.

The next part of the tutorial covers *rotordynamic considerations*. Examples of how the dynamic bearing characteristics can impact the rotordynamic behavior and lead to rotor vibration problems are presented. Journal bearing dynamic characteristics, which include stiffness and damping properties, have a strong influence on the dynamic behavior of the rotor, which to an end user shows up as high vibration and rotor instability issues. In this tutorial, a parametric study from one of the case studies is also performed to show the influence parameters such as bearing clearance and preload have on the rotordynamic behavior of a rotor. This helps explain how turbomachinery design engineers balance bearing static performance (minimum oil film thickness and bearing temperature) and rotordynamic performance (vibration and stability).

Miscellaneous topics including protection systems, condition monitoring techniques, operational controls and basic maintenance practices for ensuring good bearing performance over the life of the machine will then be presented. This includes a discussion on bearing temperature limits, vibration limits, oil requirements and oil analysis. The case studies illustrate how these factors can be used to protect, monitor, control and maintain good bearing performance over the life of the machine.

EXAMPLES OF FAILED BEARINGS

All three bearing examples presented below come from integrally geared centrifugal compressors from different manufacturers. Integrally geared compressors use helical type gears which impart significant loads to the pinion bearings due to the gear reaction forces. The bearing load is determined from the rotor weight and the variable gear reaction forces, which are a function of the gas power, rotative speed and gear geometry. Unlike an inline machine where the rotor weight dominates the bearing load; the gear reaction forces dominate the bearing load in an integrally geared compressor.

Figure 1 is a photograph of the lower half of a tilting pad journal bearing showing deposits forming on the babbitted surface. This bearing utilizes babbitted bronze pads and is a five-pad design oriented so the full operational load is between pads (LBP). The oil is introduced between pads with a single oil inlet at each location and has end seals to contain oil in the bearing shell; this is referred to as flooded operation.

Figure 1: Bearing A

Figure 2 is a photograph of the upper and lower halves of a tilting pad journal bearing showing babbitt wiping and deposit formation. This bearing utilizes babbitted steel pads and is also a five-pad bearing design oriented so the full operational load is between pads (LBP). Here, there are three spray nozzles that introduce fresh cool oil between the pads. This bearing operates evacuated; that is the oil exits the bearing through the large oil drain holes in the lower half of the aluminum end plates; and as such, the interior of the bearing shell is not flooded with oil. This reduces oil churning between pads and allows for cooler oil to be introduced to the pad leading edges.

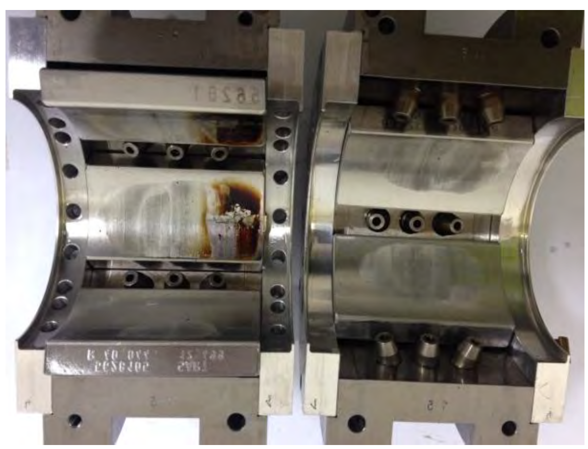

Figure 2: Bearing B

Figure 3 is a photograph of the lower half of a five-pad tilting pad journal bearing. Severe deposit formation and babbitt damage is noted on the loaded pad. This bearing is installed such that the full load is on a single pad, referred to as a load on pad (LOP) configuration. This bearing has end seals that contain the oil in the shell and as such runs flooded. Oil is introduced though orifice holes drilled in the outer shell in between each pad.

Typically, when a bearing is designed to run flooded, oil is introduced through single orifices between pads and has end seals that have a designed tight clearance to the journal, to ensure the bearing shell runs flooded with oil. Bearings that run evacuated (i.e. not flooded), typically have a way to introduce oil along the axial length of the pad leading edge via spray nozzles, spray bars or a groove at the leading edge of the pad. Evacuated bearings typically have end plates (not seals) that have generous clearance to the journal to encourage the oil to leave the bearing. By doing this the oil does not heat up by churning and the cool inlet oil can be introduced to the pad leading edge without having to mix with the oil trapped in the shell. The benefit of running flooded is that one can be assured that each pad will get all the oil it needs, and no part of the bearing will be starved of oil.

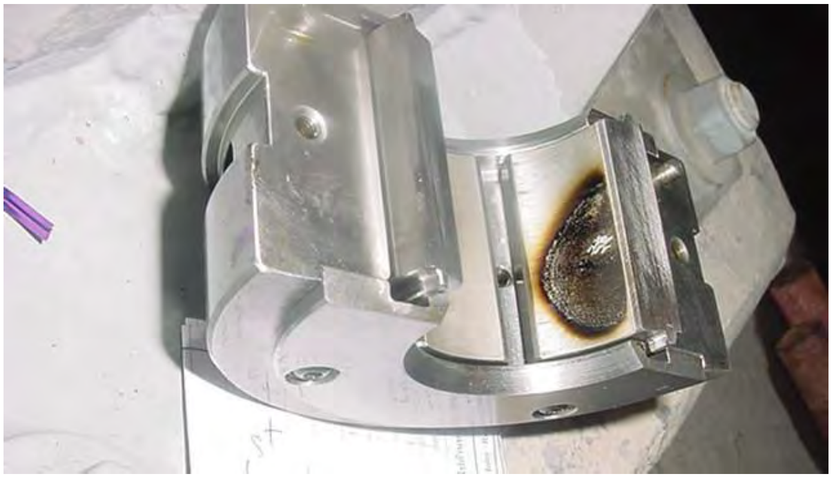

Figure 3: Bearing C

BEARING TERMINOLOGY AND DESIGN FEATURES

Oftentimes bearing surface journal speed and unit loading are used as operational limits for high speed bearings. These parameters will be discussed first, and it will be explained why these simple limits can be ineffective if used by themselves.

Journal speed is determined by the following formula:

$$U = \pi/60 * (N * D) / 12 \tag{1}$$

Where;
- U = journal surface speed, ft/s
- N = journal rotational speed, RPM
- D = journal diameter, inches

At one time, journal speeds were limited to less than 200 ft/s (61 m/s). Today, bearings have been successfully designed with journal speeds up to 450 ft/s (137 m/s).

Bearing unit or projected load is determined by dividing the bearing applied load by the bearing projected area. The general equation is:

$$W_U = W / (L * D) \tag{2}$$

Where;
- W_U = unit load, psi
- W = bearing load, lbs
- L = bearing axial length, in
- D = journal diameter, in

At one time, unit load was limited to less than 250 psi (17 bar or 1.7MPa). Today, bearings have been successfully designed with unit loads up to as high as 900 psi (62 bar or 6.2 MPa).

In general, journal speed and unit loading must be evaluated together and not separately as bearings can generally tolerate a higher unit load at lower journal speeds. Figure 4 shows an example of how the operational limits on some bearings has been extended using newer technologies.

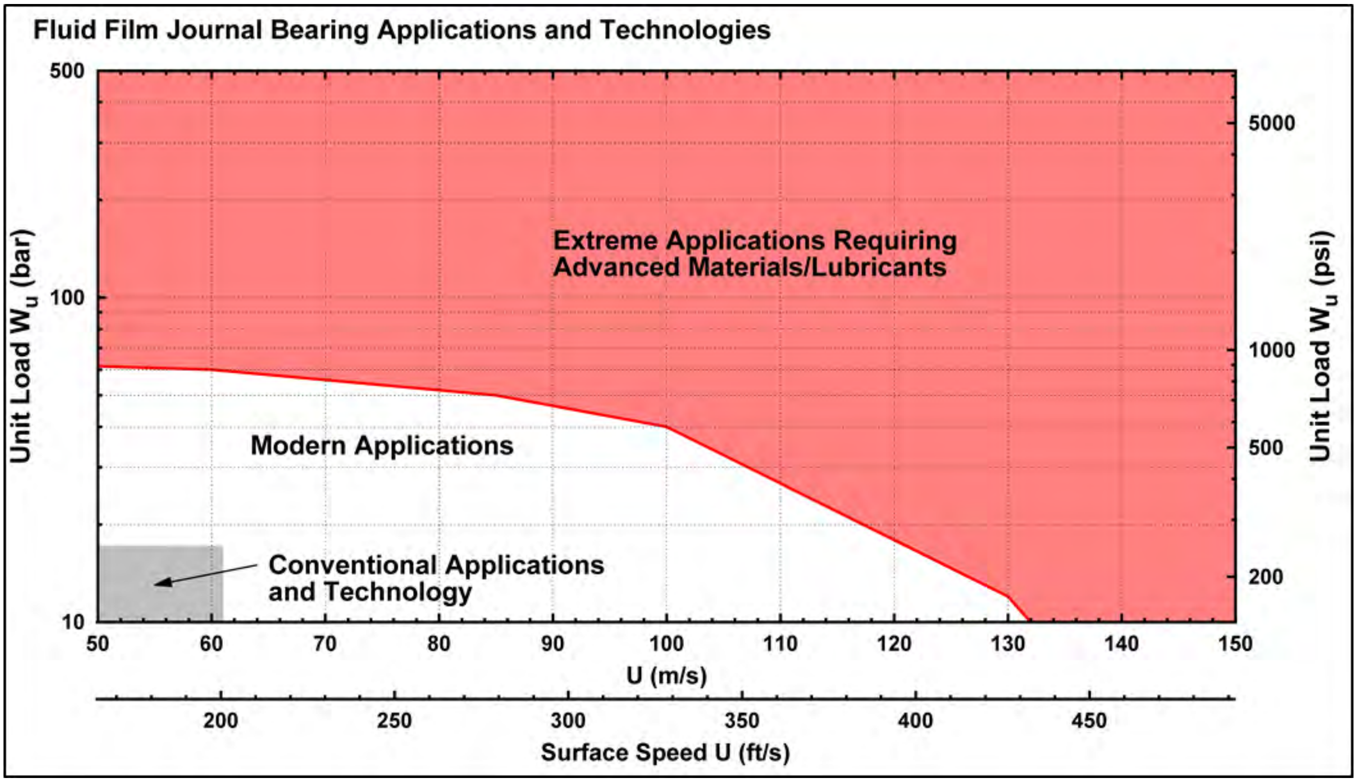

Figure 4: Journal Bearing Operational Limits (Courtesy of BRG Machinery Consulting)

However, other factors such as bearing clearance, preload, conduction, pad flexibility (thermal and mechanical), spray blockers, enhanced pad cooling and oil conditions also impact bearing performance and applying these simple speed and load limits can be ineffective in preventing bearing issues or result in a more conservative design than is needed.

Bearing clearance and preload have significant effects on the bearing characteristics. In a tilting pad bearing, two clearances of interest are the pad clearance, C_p and the assembled (or diametral, or “set”) clearance, C_b. The diametral pad clearance is equal to the machined pad diameter minus the journal diameter. Said another way, it is the unassembled bearing clearance (see Figure 5). The assembled bearing clearance is the actual installed bearing clearance. Bearing preload is defined as:

$$m = 1 - (C_b / C_p) \tag{3}$$

Where;

m = preload
$C_b = D_b - D_j$ = assembled diametral clearance
$C_p = D_p - D_j$ = pad diametral clearance
D_j = journal diameter ($2R_j$)
D_p = pad bore diameter ($2R_p$)
D_b = bearing bore diameter ($2R_b$)
O_p = center of pad bore
O_b = center of journal and bearing bore

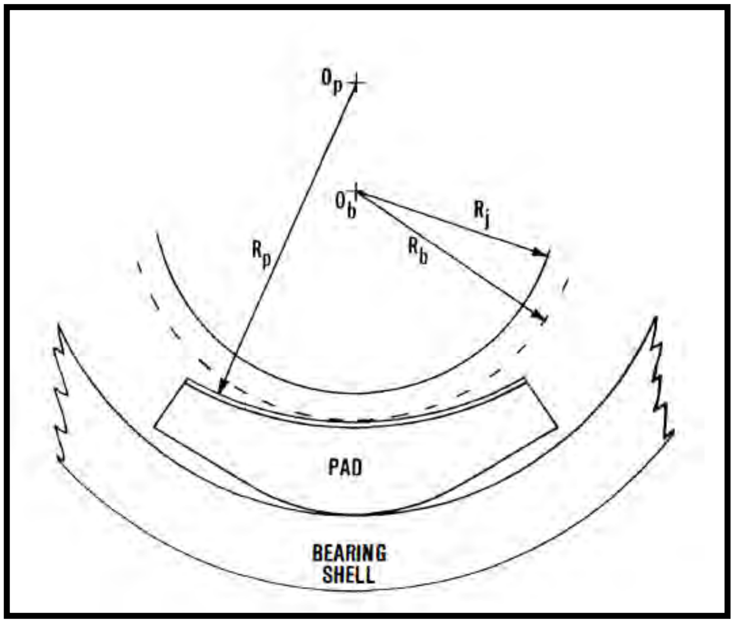

Figure 5: Bearing Bores (ref. Salamone 1984)

As the name implies, tilting pad bearings have pads that tilt. As such they must have a pivot point to tilt about. Figure 5 shows a pad in a shell where the pivot is at the interface between the pad and the shell. In this configuration the pad outer surface has a curvature less than the shell ID, this is referred to as a rocker back pivot design. With a rocker back pivot, the pad can tilt circumferentially but not axially; if everything is aligned correctly axial tilting is normally not needed. However internal alignment can change over time, or even as machines warm up and load up, leading to edge loading of the pad. Spherically seated pads (ball & socket pivot design) can pivot in both the circumferential and axial directions. Therefore, this design can handle some shaft/bearing misalignment and reduce the chance of pad edge loading. Edge loading can cause higher local loads since the load is spread across a smaller area. Pivot selection and design is a whole topic in itself; Nicholas and Wygant (1995) provide a good discussion on various pivot designs.

Centrally pivoted pads tilt at the center of the pad arc. Offset pivots can handle higher loads in the normal direction of rotation. Figure 6 shows the differences between center and offset pivot pads. Figure 7 shows a typical pressure profile in a 4 pad LBP tilting pad bearing. The pad is like a beam and with the offset pivot, the pressure profile across the pad will cause the pad to open more at the leading edge. This will result in a larger converging clearance, which will increase load capacity of the bearing. It will allow more oil to flow into the bearing and result in a thicker oil film and cooler pad temperatures. So, for the same load, a bearing with offset pivot pads should run cooler than a bearing with center pivot pads. One disadvantage of offset pivot pads is that these pads have less load carrying capability in the reverse direction of rotation. Other disadvantages include a larger oil flow requirement and a stiffer bearing, which may cause rotordynamic issues.

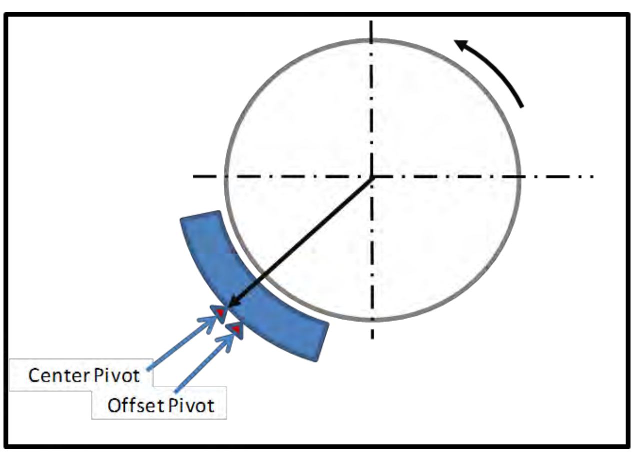

Figure 6: Offset pivot

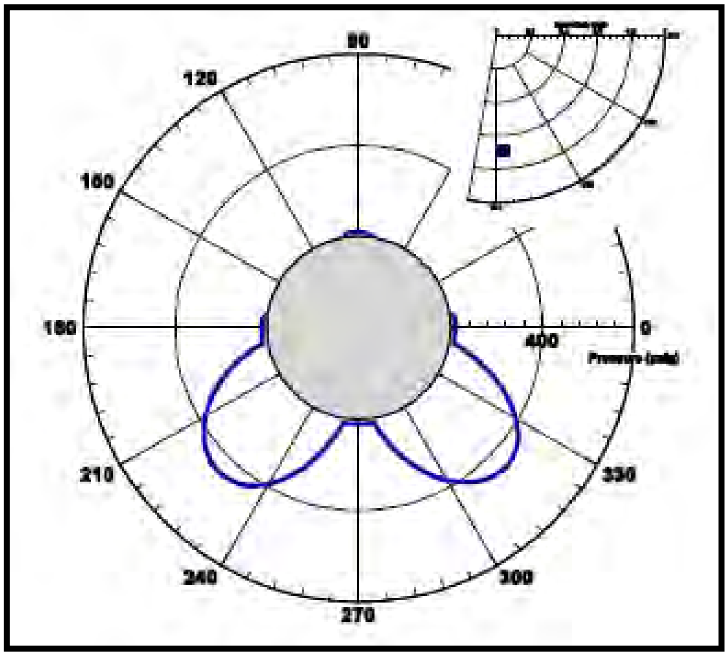

Figure7: Typical Tilting Pad Bearing Oil Pressure Profile (Courtesy of BRG Machinery Consulting)

Steel pads with a babbitt lining are the most common pad materials for tilting pad bearings. Chromium copper has a higher thermal conductivity than steel (450% more heat conductive). Therefore, heat generated in bearings with babbitted chromium copper pads is conducted away faster than bearings with steel pads. Typically, this results in cooler running bearings.

Pads can change dimensionally in service due to applied pressure profiles and thermal gradients. The pressures generated in the hydrodynamic film can "unwrap" the pad bore resulting in increased preload. Likewise, the thermal gradients can reduce bearing clearances, since the pad thermally expands and "grows" in, closing clearances (and the journal can also expand reducing clearance). The thermal gradients also can influence the operating pad shape. As such any analysis should account for all these effects.

In tilting pad bearings, some of the hot oil exiting the trailing edge of a pad mixes with cool inlet oil, and this oil mixture feeds the leading edge of the next pad. San Andres and Abdollahi (2018) go into detail on this mixing and the various models to account for it during a bearing analysis. Hot oil carryover blockers installed between pads can reduce pad to pad hot oil carryover, which would reduce the overall supply oil temperature at the inlet of the pads. This can result in a cooler running bearing. Recent research (Coghlan and Childs 2015) has cast some doubt on the true effectiveness of hot oil carryover blockers. The hot oil leaving a pad trailing edge is very thin (1-2 mils, for instance) and the blocker designs utilized do not get close enough to the journal to effectively scrape away the hot oil before it is introduced to the leading edge of the next pad.

With bypass cooling, a small amount of oil is directed behind the pads to aid in convective heat removal. This is usually utilized with chromium copper pads taking advantage of the increased conductivity. This requires a small amount of additional oil supply, but also results in a cooler running bearing. Nicholas (2003) has a good description of bypass cooling and spray bar blocker design.

BEARING REQUIREMENTS FROM INDUSTRY STANDARDS

The following five industry standards are referenced in Table 1. These standards cover compressors, gears and steam turbines.

- API-617 Eighth Edition (Centrifugal Compressors)
- API-613 Fifth Edition (Gearbox)
- API-612 Seventh Edition (Steam Turbines)
- API-672 Fourth Edition (Integrally Geared Compressors)
- AGMA 6011-J14 (High Speed Helical Gear Units)

Table 1: Radial Tilting Pad Bearing Requirements in Various Industry Standards

STANDARD	MAXIMUM DESIGN METAL TEMP	MINIMUM OIL FILM THICKNESS	MAXIMUM UNIT LOAD	MAXIMUM JOURNAL SPEED	MAXIMUM OIL TEMPERATURE RISE
	°F/°C	in/mm	psi/MPa	ft/s / m/s	°F/°C
API-617 (1)	212/100				
API-613		0.001/.025	500/3.4		
API-612 (1)					50/28
API-672 (1)					50/28
AGMA 6011-J14 (2)	239/115	0.0008/.020	609/4.2	410/125	
(1) Based on a maximum oil supply temperature of 120°F (48.89°C) (2) Limits are based on an average diametral clearance ratio of 2 mils/inch (2 mm/m) of journal diameter					

As shown, the bearing requirements vary considerably from standard to standard, both in actual limits and what criteria are specified. Other than the AGMA standard, the other references have minimal criteria, and there is no consistency from standard to standard. This makes it difficult for an end user to evaluate the risk of a bearing design for a given application.

Comparison of Failed Bearings to Industry Standards

The following is some of the basic operational data for the example bearings.

- Bearing A
  - Journal surface speed = 287 ft/s (87.5 m/s)
  - Unit load = 273 psi (1.88 MPa)
  - Clearance ratio = 2.66 mils/inch of journal diameter

- Example B
  - Journal surface speed = 309 ft/s (94.2 m/s)
  - Unit load = 330 psi (2.28 MPa)
  - Clearance ratio = 2.20 mils/inch of journal diameter

- Example C
  - Journal surface speed = 287 ft/s (87.5 m/s)
  - Unit load = 222 psi (1.53 MPa)
  - Clearance ratio = 3.30 mils/inch of journal diameter

In each case, the journal surface speed, unit loading, and clearance ratio are well within the industry norms. The oil used for the bearings in all these examples is a standard ISO VG 46 turbine oil with no EP additives.

BEARING ANALYSIS

The bearings in this tutorial have been analyzed with the computer program THPAD. This is a thermoelastic hydrodynamic (TEHD) program available through membership in the ROMAC industrial research program at the University of Virginia. *"This program calculates the operating eccentricity and dynamic characteristics, including a variety of thermal and deformation effects. The algorithm solves Reynolds equation for the circumferential pressure distribution based on an assumed axial profile. Viscosity and temperature distributions are determined by the solution of a bulk flow energy equation, also allowing circumferential and cross-film variation. Models to account for conduction in the bearing shell, in the shaft, and in the cavitated film region and to account for heating of the sump oil surrounding the bearing are included. Deformations of the pads and of the pivots can be considered and pivot stiffness is included explicitly in the dynamic coefficient reduction. The convergence of the solution to the operating eccentricity is enhanced by the use of a novel non-dimensionalization."* (Branagan, et al. 1988).

The output to be evaluated includes several parameters. Static output (pressure, film thickness, temperature, oil flows, etc.) will be discussed first while dynamic output (stiffness and damping terms) will be addressed later.

Minimum Oil Film Thickness

A minimum oil film thickness is required to ensure reliable bearing operation. API-613 specifies a minimum oil film thickness of .001 inches (.025 mm) for all specified operating conditions. AGMA 6011-J14 specifies a slightly lower minimum oil film thickness of .0008 inches (.020 mm). Based on more current operating experience, the minimum oil film thickness specified by AGMA seems reasonable. Minimum oil film thicknesses less than this may be acceptable but require a careful review of other parameters such as the frictional losses and operating temperatures.

For a given bearing design, the factors which mostly affect the minimum oil film thickness are load, speed and oil viscosity. There are three things that are needed for any journal bearing to generate hydrodynamic pressure or load capacity: a convergent wedge, surface motion and a viscous lubricant. It is important to understand that oil is not pumped into a bearing clearance; it is dragged in by shaft rotation. As a result, shaft speed is a primary factor in the minimum oil film thickness. All things being equal, higher speeds result in thicker oil films. However, higher speeds also increase friction which increase oil film temperature and reduces film thickness. So, one factor affects the other and the overall effect on minimum oil film thickness can only be determined by going through a comprehensive bearing analysis.

Side Leakage and Oil Supply

Oil flows onto each pad at the leading edge. Some of the oil leaves the trailing edge, enters the oil sump between the pads, mixes with cool supply oil and then enters the leading edge of the next pad. Some of the oil leaves axially from the pads and then flows out of the bearing through the end seals and drain holes (if supplied). This side leakage must be made up by the oil supply. As such the primary purpose of calculating the side leakage is to ensure that there is enough oil supply to the bearing and the bearing is not operated in an oil starved condition. So, it is important to ensure that there is enough oil supply to prevent the bearing from being starved of oil. Higher oil film temperatures will result, and dynamic coefficients will be impacted, if a bearing is operated in a starved condition.

The oil supplied to the bearing is typically fed into a circumferential groove on the outside diameter of the bearing shell which feed orifices between each of the pads. The total amount of oil supplied to the bearing is a function of the number of orifices, orifice diameter, oil supply pressure at the bearing, and end seal clearance.

There are two types of side leakage: pad side leakage and end seal side leakage. In an evacuated bearing, the bearing cavity is non-pressurized, and the oil flow is controlled at the inlet to the bearing by orifices. For bearings running in an evacuated condition each pad must be analyzed to ensure it receives enough oil to develop a full film. In a common configuration, the oil seal clearances are open, and all the oil exits the pad axially. In the flooded configuration the bearing cavity is typically slightly pressurized by using tight end seal clearances. In this case, the oil flow is controlled by an equilibrium between oil flowing into the bearing shell through orifices and oil flow out of the end seals. Per Nicholas (1994), "*The advantage of this setup is that the hot oil produced by the lower pad can exit directly through the end seal clearance. The disadvantage is that all the oil exits along the shaft making oil slingers and oil baffles necessary to prevent oil leaks*."

Frictional Losses

Bearing frictional power loss is caused by viscous shearing of the oil film which produces a resistive torque on the shaft and consumes mechanical power. From Newton's law of viscosity, the shear stress caused by the relative movement of two plates separated by a fluid can be determined from the following equation:

$$\tau = \mu * dv/dt \quad (4)$$

Where;

- τ = shear stress, psi
- μ = oil viscosity, reyns ($lb_f \cdot s/in^2$)
- v = journal velocity, in/s
- t = minimum oil film thickness, in

Load can be a factor, but in general, speed has a much larger impact on the power loss than load.

Petrov's Law can be used to provide some further insight into the affect that bearing clearance and other parameters have on power loss. Petrov developed the following equation for the frictional losses in an unloaded (centered) journal bearing:

$$P = 2.61 x 10^{-6} * \mu D^3 L N^2 / C_d \quad (5)$$

Where;

- P = power, HP
- μ = oil viscosity, reyns
- D = bearing diameter. in

L = bearing length, in
N = shaft speed, RPM
C_d = diametral bearing clearance, in

So, smaller clearances should result in higher frictional power losses. However, higher friction also results in higher temperatures which lower oil viscosity and power loss. So, again, one factor affects the other and the power loss can only be determined by going through a comprehensive bearing analysis.

Maximum Pad Temperature
The babbitt material typically determines the temperature limit for a bearing. Although the melting temperature of babbitt is around 455°F (235°C), babbitt begins to soften at around 275°F (135°C) to 300°F (149°C). API 617 specifies maximum bearing metal temperature of 212° F (100°C) as design criteria. The maximum pad temperature in a tilting pad bearing is usually on the trailing edge of the loaded pad(s). However, the other aspect that needs to be considered is the formation of deposits on the pad surface which is related to the pad temperature and can negatively impact the bearing performance. Pad temperature, oil supply temperature and the oil characteristics are key contributors.

Peak Oil Film Pressure
The hydrodynamic oil film pressure profile is an output of the bearing analysis. It is not unusual for the peak hydrodynamic pressure to be three to five times the specific unit load. This peak pressure can be compared with the babbitt strength to evaluate bearing limits. Babbitt strength is a function of the babbitt thickness and operating temperature. Babbitt loses strength at elevated temperature. Babbitt fatigue strength generally varies inversely with the thickness of the babbitt layer once the babbitt thickness is below 0.020 inches (.51 mm). Thinner babbitt thicknesses have a higher fatigue strength due to the backing material strength. In addition to the static loads, rotor vibrations cause dynamic loads which result in peak film pressure fluctuations and can lead to babbitt fatigue damage. So, there needs to be some margin to the temperature versus babbitt strength curve to account for this.

As discussed by Whalen, et al. (2012), *"The location of the temperature sensors is also important; it is not necessary to know the point in the bearing of maximum temperature, but it is important to know the temperature at the most distressed location. The most distressed location is determined by analysis of the temperature coupled with the applied hydrodynamic pressure. Because of this understanding API 670 specifies very good locations for temperature sensor placement."* This is discussed in more detail later in the tutorial.

A Note on Bearing Codes
There is neither a standard industry code nor a standard methodology for analyzing bearing performance. There are commercially available codes, company internally developed codes and codes developed by universities and design institutes that machinery and bearing manufacturers use to model bearing performance. There are differences on how the bearing problem is solved. Some codes assume an adiabatic model, and some use a non-adiabatic model with variable viscosity and temperature. Some codes consider pad deformation and others don't. Some codes include a model for turbulence.

The purpose of this tutorial is not to evaluate the different codes. But as with any code, the user needs to understand the limitations of the code. Kocur et al (2007) presented results where they asked several users of various bearing codes to evaluate the same bearing, they presented a large spread in the results. Five different codes were used by 16 different users and the maximum temperatures (from 4 of the codes) varied from 173°F to 221°F; for the same bearing operating under the same conditions.

The code used for this tutorial is based on a non-adiabatic model (i.e., it includes conduction) and includes both mechanical and thermal deformations of the pads. THPAD was selected due to its ubiquitous use and familiarity.

CASE STUDIES

The three failed bearings presented earlier as examples will each be analyzed. Then a fourth analysis will be performed on a machine in the design stage.

Case Study A
This case study pertains to an integrally geared centrifugal compressor driven by a 5,000 HP, 1,794 RPM induction motor. The gearbox consists of a bullgear and three rotors. The stage one/two rotor consists of a pinion with overhung impellers mounted at both ends while the stage three rotor consists of a pinion with a single overhung impeller. These rotors are mounted at the horizontal split line. The stage four/five rotor is in the top of the gearbox cover and consists of a pinion with overhung impellers at both ends.

The gearbox utilizes tilting pad journal bearings for all three pinions with "X" and "Y" non-contacting proximity type shaft vibration probes adjacent to each bearing. The pinions are fitted with thrust collars which transmit pinion axial thrust to the bullgear. The bullgear rotor is fitted with a sleeve type journal bearing on the drive end and a combined sleeve type journal bearing and tapered land type thrust bearing on the non-drive end. The pinion bearings are all fitted with temperature probes mounted just below the babbitt surface on the loaded pads. The compressor control system includes high vibration alarms, high vibration shutdown protection and high bearing temperature alarms. The gearbox arrangement for stages 1 to 3 is shown in Figure 8; the stage four/five rotor is omitted for clarity.

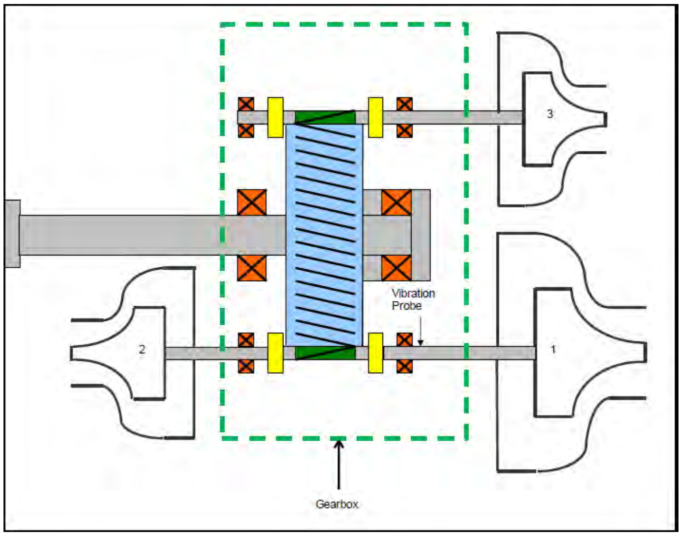

Figure 8: Compressor Arrangement for Case Study A

This compressor ran for twelve years in continuous service without any bearing problems. The second stage bearing typically operated between 220°F (104°C) and 230°F (110°C). However, when the bearing was replaced with a new spare, the bearing temperature increased to approximately 250°F (121°C). The measured bearing clearance of .0052 inches (.132 mm) was 0.001" (.025 mm) tighter than the minimum design clearance due to an oversized bearing journal. The rotor was not replaced, but the bearing was eventually replaced with a modified bearing with a larger assembled clearance and preload and the bearing temperature fell to approximately 170°F (77°C) to 180°F (82°C). A comprehensive bearing analysis was performed to understand the bearing behavior and ensure this change corrected the underlying problem.

The second stage bearing is a five pad, tilting pad type with spherical seats and a center pivot design. The bearing geometry and operating conditions are shown in the table below (note "nominal" clearance is the clearance at the midpoint of the clearance range):

Table 2: Case Study A Design Data

BEARING GEOMETRY							
Journal Diameter in/mm	**Bearing Axial Length in/mm**	**Diametral (Assembled) Nominal Design Clearance in/mm**	**Clearance Ratio mil/inch / (micron/mm)**	**Preload**	**Pad Angle degrees**	**Pivot Angle degrees**	**Pad Orientation (Under Operating Load)**
2.7559/70	1.9291/49	0.0073/0.186	2.66/2.66	0.286	60	30	Load on Pad

BEARING OPERATING PARAMETERS						
Pinion Speed RPM	**Journal Load lbf/kN**	**Journal Speed ft/s / m/s**	**Bearing Unit Load psi/MPa**	**Lubricant**	**Oil Supply Temperature Deg F/Deg C**	**Calculated Oil Flow to Bearing GPM/lpm**
23849	1454/6.5	287/87.5	273/1.9	ISO VG 46	113/45	11.1/42

The bearing journal speed and load are well within industry experience and so bearing application would not be considered aggressive. Because the second stage bearing appeared to show sensitivity to diametral clearance, bearing analyses were performed at various diametral clearances while maintaining the design pad clearance.

As mentioned, the measured assembled bearing clearance was tighter than the design limits. A smaller diametral clearance results in a higher preload when using the same pads because the pad clearance doesn't change. The smaller diametral clearance impacts the bearing performance, including the minimum oil film thickness and the maximum pad temperature. The results of the bearing analyses presented below assume a constant pad clearance, which results in varying preload for different diametral clearances.

Figure 9 is a plot of the calculated minimum oil film thickness as a function of diametral clearance for both the original bearing design (including below design clearances) as well as the modified bearing (with the design clearance range). As shown, reduced diametral clearances generally results in smaller minimum oil film thicknesses. To meet a minimum oil film thickness of .0008 inches (.020 mm), the minimum diametral clearance based on the original design bearing is around .004 inches (.102 mm). Also note that the modified bearing with the higher diametral clearance has a higher calculated minimum oil film thickness.

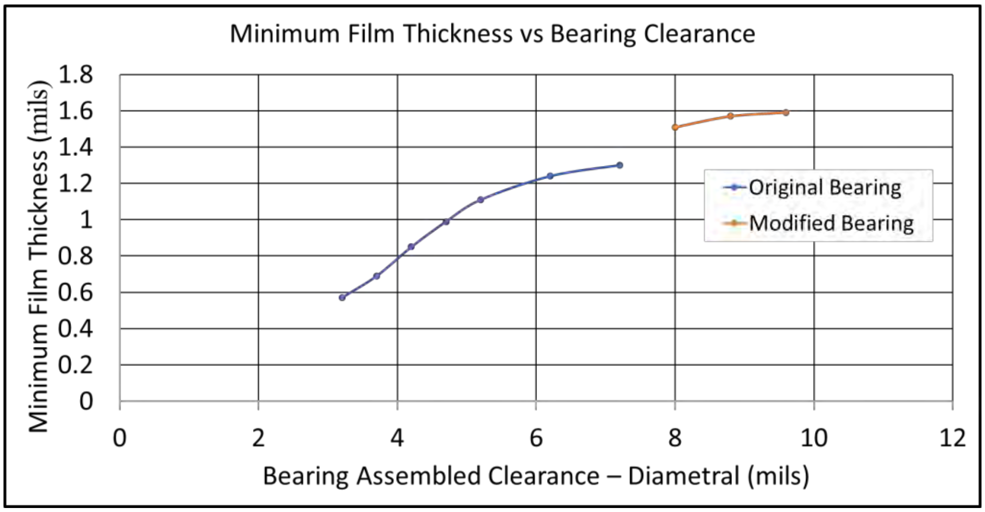

Figure 9: Calculated Oil Film Thickness Versus Clearance for Case Study A

Figure 10 is a plot of the calculated maximum pad temperature versus diametral clearance. As shown, the diametral clearance has a moderate impact on the maximum pad temperature until the diametral clearance is below .0052 inches (.132 mm) and then it is much more sensitive to small changes in diametral clearance. Recall that the bearing clearance measured in the field was .0052 inches (.132

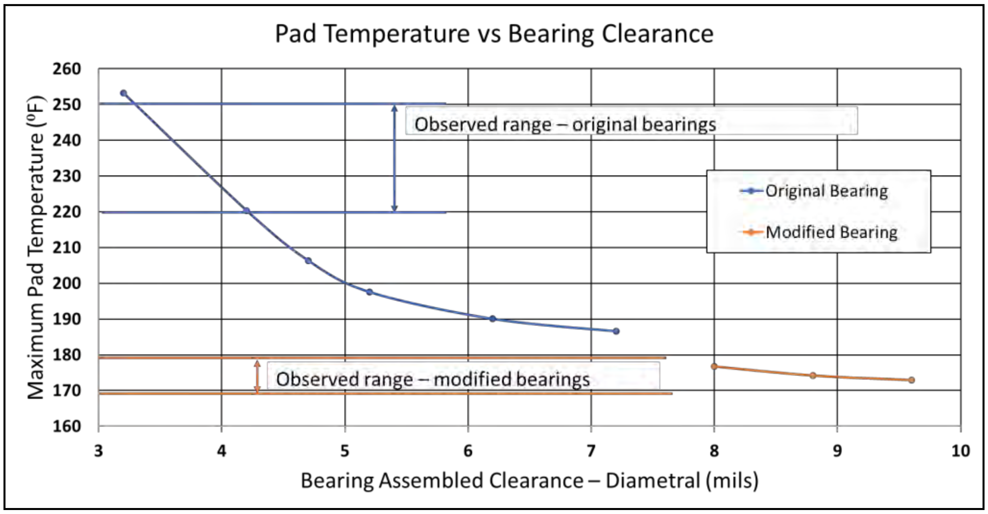

Figure 10: Calculated Maximum Pad Temperature Versus Bearing Assembled Clearance for Case Study A

mm). However, there is always some variability in field measurements, and it is possible the actual clearance was a little tighter. In addition, the clearance could tighten up a bit due to over clamping of the bearing, misalignment, etc. The measured clearance is right at the knee in the calculated clearance versus temperature curve and is thus not as tolerant to small changes that could quickly cause higher temperatures. When reviewing the results of a bearing analysis this should be a consideration.

Figure 11 is a plot of the calculated side leakage as a function of diametral clearance. This allows us to see at what oil flow the bearing starts to run starved. Note that this is an extreme value since it assumes all the oil supplied is available to go into the pad leading edges, and this is an ideal assumption. Typically, higher oil flows are needed to ensure every pad gets the oil it needs. Note that the modified bearings with the higher diametral clearance has a higher calculated side leakage than the original bearing design. The higher side leakage results in more hot oil leaving the bearing and more cool oil entering each pad. This results in a cooler running bearing.

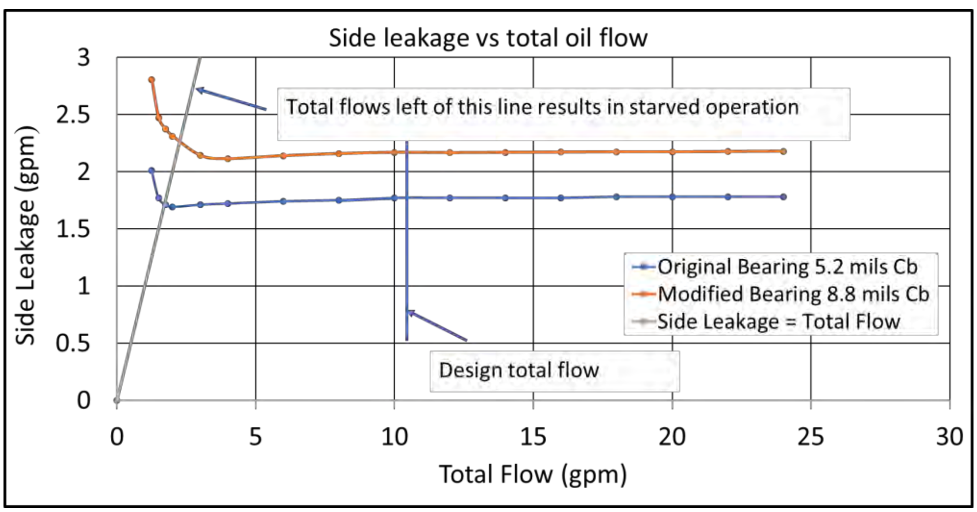

Figure 11: Calculated Side Leakage Versus Total Flow for Case Study A

Figure 12 is a plot of maximum pad temperature as a function of oil flow. The hydraulics in an integrally geared compressor gearbox are more complex than that of an inline compressor and it is possible that the oil pressure at the supply to the bearing is lower than the measured value at the supply to the gearbox. This would result in a lower oil flow, and as shown, also has a minor impact on maximum pad temperature until the flow is about half of the design value and then it has a much larger impact.

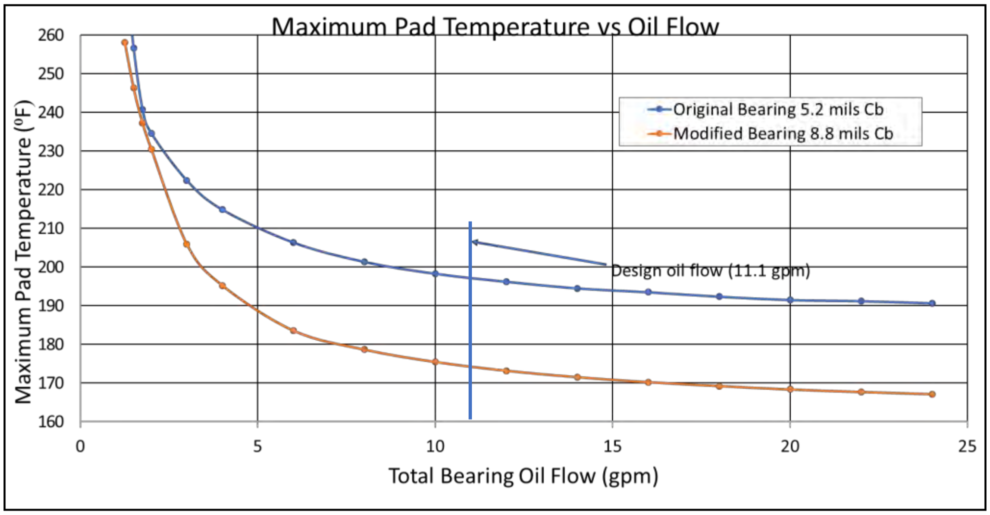

Figure 12: Maximum Pad Temperature Versus Calculated Oil Flow for Case Study A

Figure 13 is a plot of the calculated power loss as a function of the diametral clearance. As shown, power loss varies inversely with the clearance. Although not shown on the plot, the calculated power loss is not affected much by load in this example.

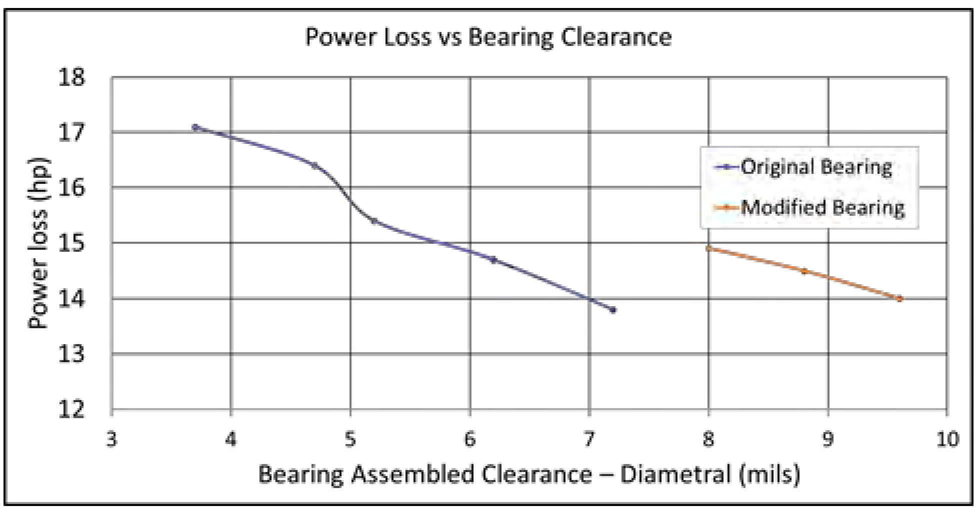

Figure 13: Calculated Power Loss Versus Diametral Clearance for Case Study A

The bearing analysis for this case was based on bronze pads and includes mechanical and thermal deformations. Deformations have a lesser impact on bearing performance at moderate bearing temperatures and loads, but as the temperate and/or load increases, pad deformation can have a significant impact due to the change in the bearing's operating geometry. As the bearing heats up in operation, the bearing clearance will close due to thermal growth of the shaft and the pad. The effect is more significant at higher bearing temperatures. Also, if the loaded pad is thought of as a curved beam that is simply supported at the pivot point, applying the generated hydrodynamic pressure profile across the pad will tend to flatten the pad, or "unwrap" it, opening the leading edge. This will increase the pad clearance, and therefore will result in an increase in preload.

Pad material can also be a factor. Some of the differences include:

- Bronze has a higher thermal conductivity than steel, which results in better heat transfer. However, for moderate bearing temperatures heat transfer in the bearing is dominated by the convection term as most of the heat is carried away by the flowing lubricant. The higher thermal conductivity of the bronze pads only results in a reduction in bearing temperature of a few degrees, if any. At higher bearing temperatures, conduction can be more significant and should always be included in the analysis.

- Bronze has a higher coefficient of thermal expansion than steel. At moderate bearing temperatures, the impact is small. However, as the temperatures increase, this reduction in bearing clearance is more significant. For this reason, the clearance may need to be adjusted when using or changing from steel pads to bronze pads.
- Bronze has a much lower modulus of elasticity. For a given pad geometry and applied pressure, a bronze pad will deflect more than a steel pad. A higher deflection means that the pad will open more, allowing more oil flow into the bearing and resulting in lower temperatures. However, the effect is generally minor except at higher loads.

Figure 14 is a plot of maximum pad temperature as a function of bearing clearance for both bronze pads and steel pads. When running with larger clearances, the bronze pads are cooler than the steel thanks to the extra heat conduction. At clearances less than .0047 inch (.119 mm), the bronze pads run hotter because the increased thermal expansion (closing the operating clearance) overtakes the benefit of the extra conduction. Comparing bronze to steel:

- Thermal expansion coefficient: Bronze is 65% larger than steel
- Thermal conductivity: Bronze is 30% higher than steel
- Modulus of elasticity: Bronze is half that of steel

In addition to the above analyses, bearing performance was also reviewed as a function of speed at the measured bearing clearance of .0052 inches (.132 mm). As shown in Figure 15, the calculated maximum pad temperature is just below 200°F (93°C) at the design journal speed. Since this is a motor driven compressor the speed will be constant and is a known value.

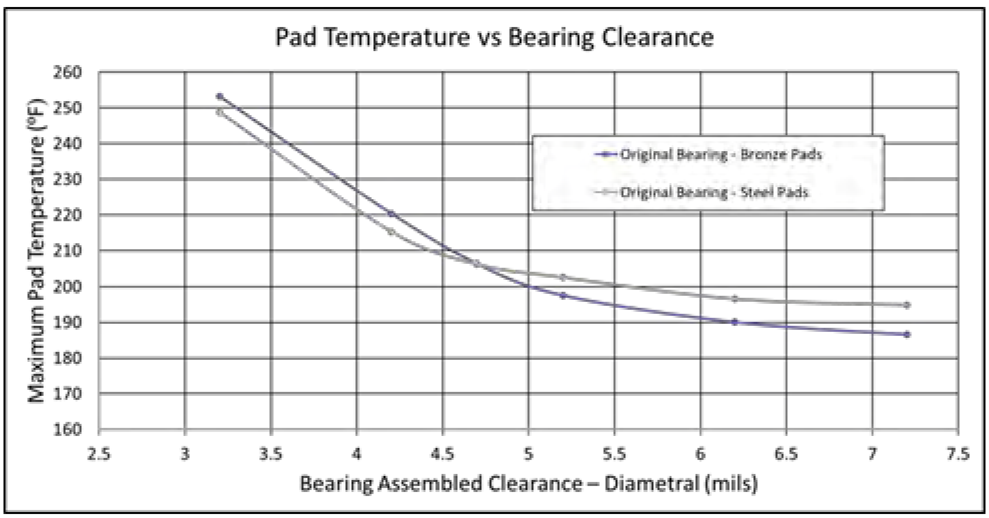

Figure 14: Calculated Maximum Pad Temperature Versus Bearing Clearance Comparing Bronze and Steel Pads, Case Study A

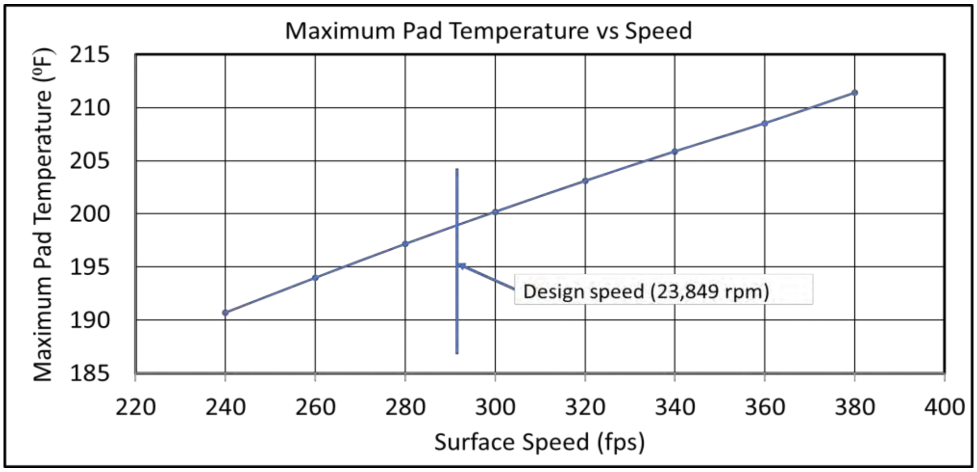

Figure 15: Calculated Maximum Pad Temperature as a Function of Journal Speed for Case Study A

Finally, bearing performance was also reviewed as a function of load at the measured bearing clearance of .0052 inches (.132 mm). As shown, the calculated maximum pad temperature is just below 200°F (93°C) at the design load. See Figure 16. With integrally geared compressors the load is a function of the gas being compressed and the process conditions, and in some cases can vary.

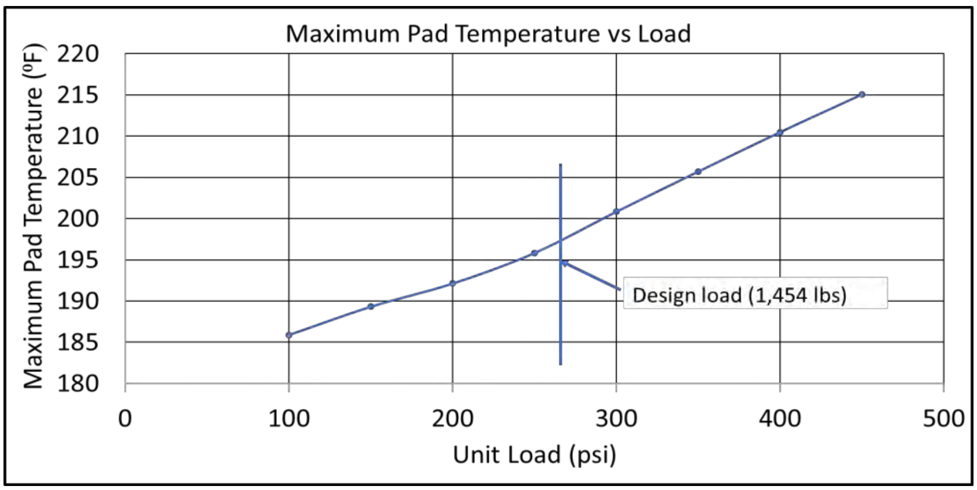

Figure 16: Calculated Maximum Pad Temperature as a Function of Load for Case Study A

There is a higher probability of oil varnish deposit formation at higher bearing temperatures. An operating limit of 212°F (100°C) seems like an appropriate criterion and provides some margin to the temperature where deposit formation started to occur for this case study. However, as shown, various factors can cause the bearing to operate a little hotter than the calculated maximum pad temperature. Therefore, a lower design value seems appropriate. The analytical design values in this case study meet these criteria, but as shown in Figures 15 and 16, it is close due to a tight bearing clearance. And in this case, there wasn't enough margin resulting in a hotter running bearing.

Figure 17 is a plot of the maximum oil film pressure versus bearing clearance. Notice how the peak oil film pressure increases at a faster rate when the clearance is below .004 inches (.102 mm) to .0045 inches (.114 mm). Figure 18 is a plot of babbitt strength versus babbitt temperature. The babbitt strength plot shows how a tight bearing will run hot but also run with a higher peak film pressure – bringing it closer to the "line." At even more extreme cases (higher pressures and temperatures associated with very tight clearances) the point can go above the "line."

What is also important to note is the loss in babbitt strength in going from a moderate temperature of 180°F to 250°F. The babbitt strength is reduced by 43%. For highly loaded bearings this could be a significant factor. The specifics of the bearing babbitt limits need to be discussed with the bearing supplier.

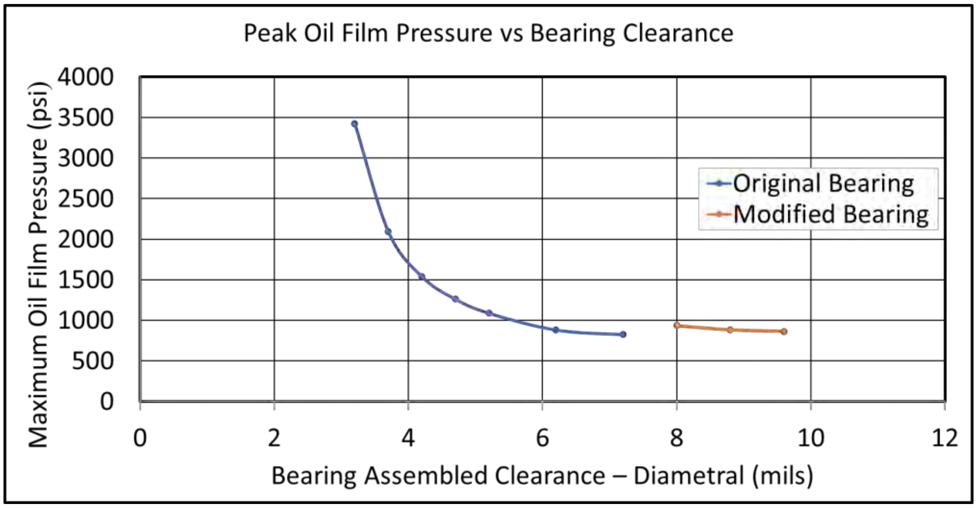

Figure 17: Maximum Oil Film Pressure Versus Calculated Bearing Clearance for Case Study A

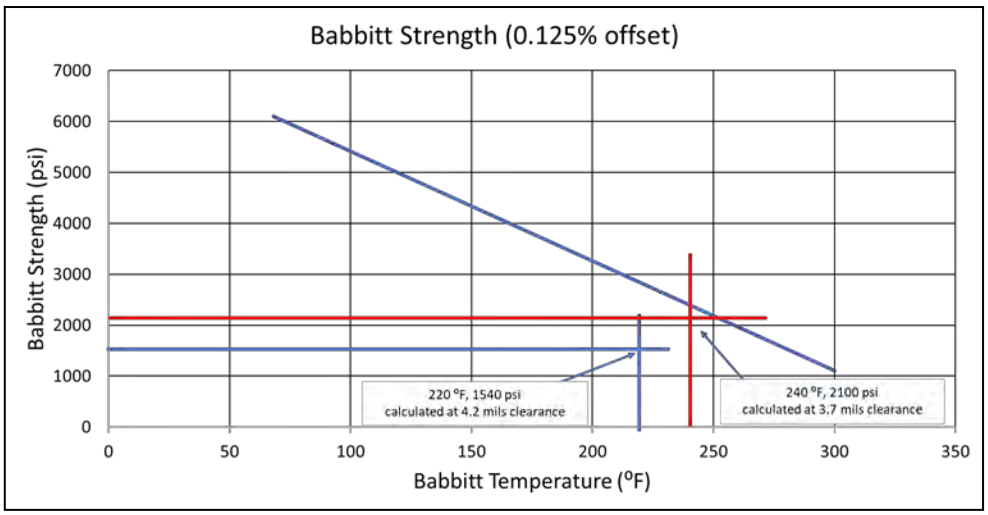

Figure 18: Calculated Babbitt Strength Versus Babbitt Temperature for Case Study A

Conclusion for Case Study A

What can be learned from this case study is the sensitivity of bearing performance with diametral (assembled) clearance. Had the bearing clearance been within the design tolerances, the bearing performance likely would have been fine. The other aspect is the impact that thermal and mechanical deformations have on bearing performance. If these are not accounted for in a high-speed bearing such as this one, the analysis may yield optimistic results.

The journal speed and unit load criteria in AGMA 6011-J14 are based on minimum clearance ratio of 2 mils/inch of journal diameter. This might sound like a reasonable criterion but based on the observations and analysis for this bearing, it is close to the point where high bearing temperatures were observed. When reviewing bearing performance, it is important to consider how fault tolerant the design is. If there is not much margin, a slightly tight clearance, slightly excessive clamping, hotter oil or other factors could result in a hot running bearing or a bearing more prone to failure. Just increasing bearing clearances changes the stiffness and damping of the bearing, which impact the rotordynamics. This is discussed later in this tutorial.

The other take-away from this case study is that care must be taken when using or switching to bronze pads. Old reasoning was that this would result in a cooler running bearing. Although not shown, the benefits of using bronze pads for this example were insignificant. The difference in mechanical deformation with steel and bronze pads was insignificant because of the thick pads. And, although bronze has a better thermal conductivity, this was offset somewhat because the thermal growth of bronze is greater than steel and this resulted in a slightly reduced operating bearing clearance.

Case Study B

This case study pertains to a five stage integrally geared centrifugal compressor driven by an 8,949 KW, 1,500 RPM synchronous motor. The gearbox consists of a bullgear and three rotors. The pinion one, stage one/two rotor and the pinion two, stage three/four rotor consist of pinions with overhung impellers mounted at both ends. These rotors are mounted at the horizontal split line. The pinion three, stage five rotor is in the top of the gearbox cover and consists of a pinion with a single overhung impeller and a free end on the other side.

The gearbox utilizes tilting pad journal bearings for all three pinions with “X” and “Y” non-contacting proximity type shaft vibration probes adjacent to each bearing. The pinions are fitted with thrust collars which transmit pinion axial thrust to the bullgear. The pinion bearings are all fitted with temperature probes. The temperature probe tips are positioned in the loaded pads just below the babbitt surface. The compressor control system includes high vibration alarms, high vibration shutdown protection and high bearing temperature alarms.

During commissioning there were high temperature issues with the pinion two, stage three/four bearings. The fourth stage bearing temperature would increase to around 266°F (130°C) during start-up and then drop to about 248°F (120°C) during steady state operation with an oil supply temperature of around 118°F (48°C). Higher pad temperatures were seen at start-up with colder oil supply temperatures. The fourth stage bearing was inspected and there was no visible damage or distress. Based on experience with a similar compressor, the oil supply temperature was increased to 131°F (55°C). At this oil supply temperature, the steady state third stage bearing temperature was around 235°F (113°C) and the fourth stage temperature was around 214°F (101°C).

The third stage bearing temperature was deemed acceptable for short term operation but was unacceptable for long term operation. The bearing design and operational data were reviewed with the compressor manufacturer and the following were concluded:

- The third and fourth stage bearing clearances are in the range where they are supposed to be for moderate bearing temperatures.
- Based on assembly records, the bearing clamping is not excessive.
- Based on assembly records the bearing orientation is correct.
- Various bearing modifications to reduce the temperature were evaluated and the following changes were recommended:
    - Machine a chamfer on the trailing edge of all the pads to increase the oil flow into the pads oil film. This functionally changes the bearing to an offset pivot design.
    - Increase the diameter of the oil nozzles in the bearing from .078 inches (2.0 mm) to .094 inches (2.4 mm) to increase the oil flow into the bearing. This would increase oil flow about 45%.
    - Machine the pinion journal diameter in the bearing area to increase the bearing clearance.

The third stage bearings were inspected a couple years later, and no visible damage or distress was observed. See Figure 19.

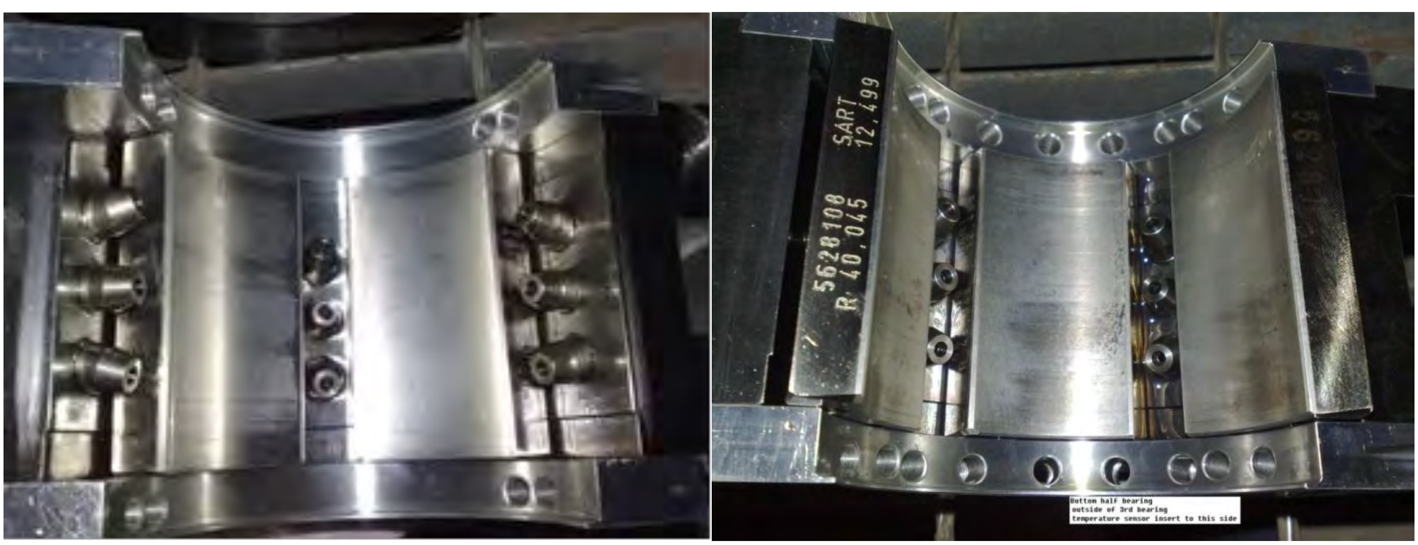

Figure 19: As Found Third Stage Bearing for Case Study B

While the first two recommendations were implemented, it was decided not to machine the bearing journal. After restarting, the third stage bearing temperature was reduced to around 158°F (70°C) at an oil supply temperature of 131°F (55°C). At cooler oil supply temperatures, the bearing temperature would rise rapidly. It was decided not to inspect or modify the fourth stage bearing. However, during the start-up after the outage, the fourth stage bearing temperature climbed to about 275°F (135°C) before dropping and eventually stabilizing at around 212°F (100°C). The third and fourth stage pinion vibrations were unchanged and were stable at around 1.0 mils (.025 mm) p-p (third stage) and 0.4 mil (.010 mm) p-p (fourth stage).

Several years later the fourth stage vibration increased to about 0.5 and 0.6 mils p-p after a compressor shutdown and then slowly increased to about 0.5 and 0.85 mils p-p. A few months later, the compressor tripped due to a sudden spike in fourth stage vibration. The bearings were removed and inspected. There was some bearing babbitt damage on the fourth stage bearing pads and the pads were replaced. See Figure 20.

Figure 20: Fourth Stage Bearing Damage for Case Study B

The bearing and journal dimensions were also checked. The diametral clearance was under the design value due to an oversized bearing journal. The as found diametral clearance was .006 inches (.152 mm). Nothing could be done at the time to address the tight clearance. During the outage, the oil was replaced. The compressor was restarted and the fourth stage vibration stabilized at about 0.5 and 0.6 mils p-p. But the vibration slowly trended up to 0.8 and 0.9 mils p-p before tripping again on high fourth stage vibration one month later. The bearing temperature after the bearing repair was 212°F (100°C) and increased to about 235°F (113°C) when the compressor tripped the second time. Again, babbitt damage was found. See Figure 21.

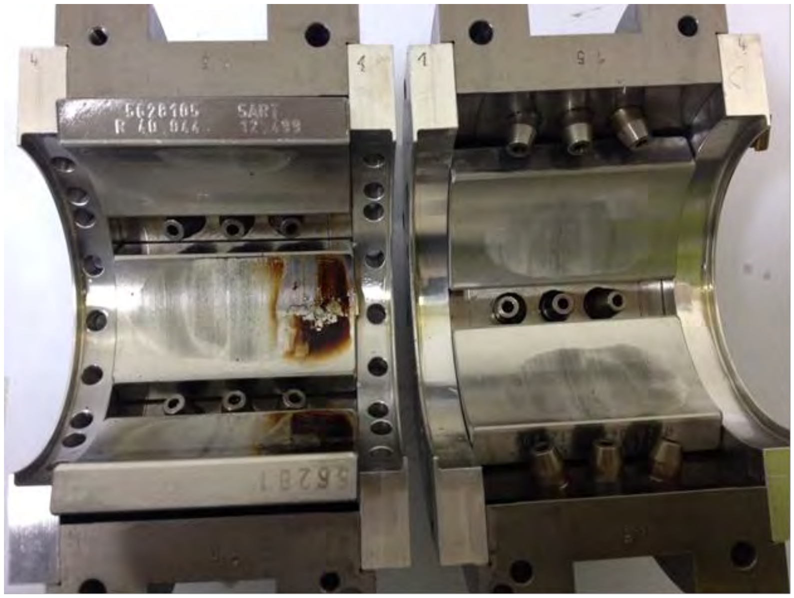

Figure 21: Fourth Stage Bearing Damage for Case Study B

The fourth stage bearing pads were replaced with the same modifications as was done with the third stage bearing. The bearing clearance was still tight due to the oversized bearing journal. The compressor was restarted with an oil supply temperature of 124°F (51°C). The fourth stage bearing temperature quickly climbed to 284°F (140°C) at which time the oil supply temperature was increased to 131°F (55°C), and the bearing temperature dropped to around 194°F (90°C).

The sensitivity of bearing temperature to oil supply temperature has been observed on other similar machines. This was a concern because this can make the bearings less “fault tolerant,” meaning that small problems like a slightly tight bearing clearance, a little misalignment, a little varnish, etc. that most bearings could tolerate could be a problem for these bearings. To understand this, a comprehensive bearing analysis was performed.

The fourth stage bearing was analyzed. It is a 5 pad, tilting pad type bearing with non-aligning, cylindrical, center pivot pads. The bearing geometry and operating conditions are shown in the table below. Note that the calculated oil flow is based on the original 0.078-inch (2.0 mm) nozzles and not the .094 inches (2.4 mm) nozzles that was implemented later.

Table 3: Case Study B Design Data

BEARING GEOMETRY							
Journal Diameter inches/mm	**Bearing Axial Length inches/mm**	**Diametral (Assembled) Minimum Design Clearance inches/mm**	**Clearance Ratio mil/inch**	**Preload**	**Pad Angle degrees**	**Pivot Angle degrees**	**Pad Orientation (Under Operating Load)**
3.1423/79.8	2.7953/71	0.0071/0.181	2.2	0.31	52	26	Load between Pad

BEARING OPERATING PARAMETERS						
Pinion Speed RPM	**Journal Load Lbf/kN**	**Journal Speed ft/s / m/s**	**Bearing Unit Load Psi/MPa**	**Lubricant**	**Oil Supply Temperature Deg F/Deg C**	**Calculated Oil Flow to Bearing GPM/lpm**
22562	2898/12.9	309/94	330/2.3	ISO VG 46	113/45	10.6/40

A plot of minimum oil film thickness as a function of diametral clearance is shown in Figure 22. Even at diametral clearances much less than design, the calculated minimum oil film thickness is greater than .0008 inches (.020 mm) for both the original bearing and the modified bearing with the offset pivot.

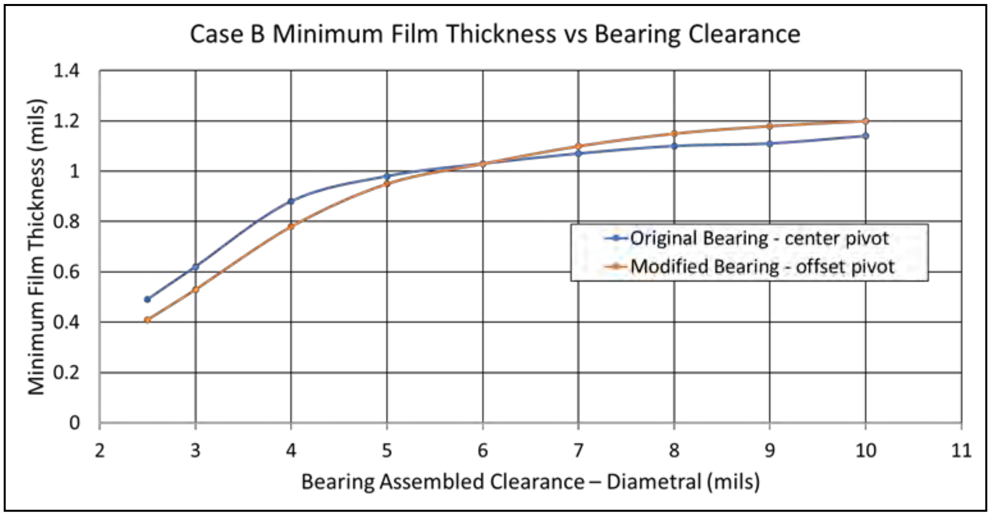

Figure 22: Minimum oil film thickness Versus Diametral Clearance for Case Study B

A plot of maximum pad temperature as a function of diametral clearance is shown in Figure 23. Based on the original bearing design, the calculated maximum pad temperature is above 212°F (100°C) even at the design clearance of .0071 inches to .0083 inches (.181 mm to .211 mm) and is within the observed temperature range in the field. The analysis clearly shows the improvement in changing to an offset pivot design, a 20°F (11°C) drop in temperature in the design clearance range.

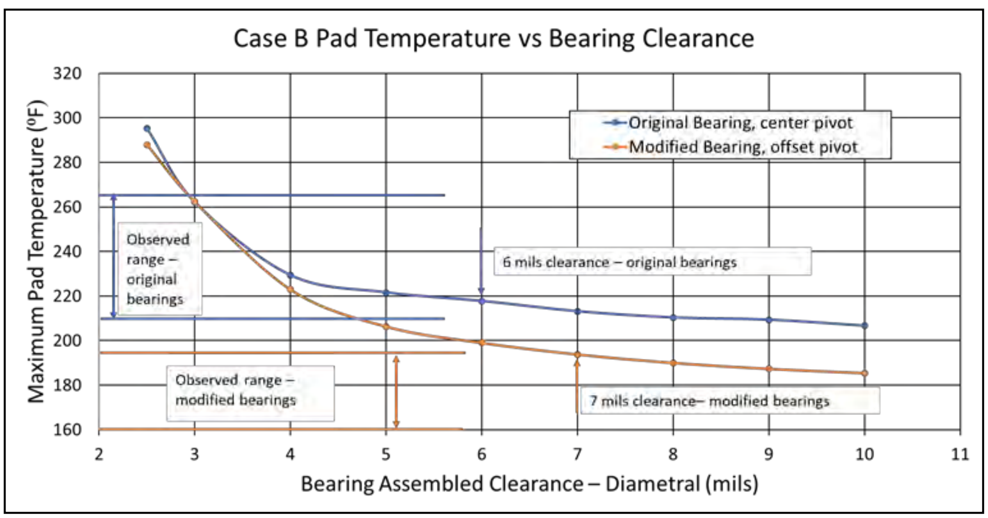

Figure 23: Maximum Pad Temperature Versus Diametral Clearance for Study B

The offset pivot has a large impact on lowering the bearing temperature and largely explains why the field modifications were successful. To evaluate the potential benefit of increasing oil flow, a plot of oil flow versus maximum pad temperature was generated. See Figure 24. As shown, the impact of increased oil flow was predicted to have a very modest impact (about 5ºF).

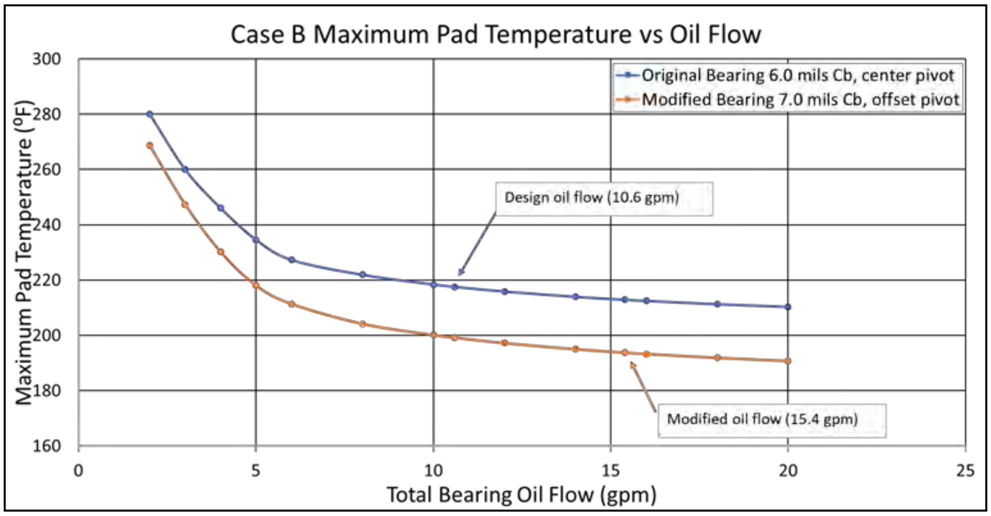

Figure 24: Maximum Pad Temperature Versus Oil Flow for Study B

The following permanent changes were incorporated to lower the bearing temperature.

- Increase the diametral clearance to a minimum of 0.0070 inches (.178 mm)
- Incorporate a circumferential groove on the pads adjacent to loaded pads. The circumferential groove directs hot oil out of the unloaded bearing pads, which brings in additional cooler oil and results in cooler oil to the loaded pad. Note that this will likely not be effective if a hot oil blocker is present.
- Change from a centrally pivoted pad to 55% offset pivot pad by changing the pivot point.

To understand the sensitivity of oil supply temperature, bearing analyses were also run at various oil supply temperatures. See Figure 25. As shown, the analyses showed lower bearing temperatures at lower supply temperatures. So, what can explain the drop in bearing temperature when the oil supply temperature was increased?

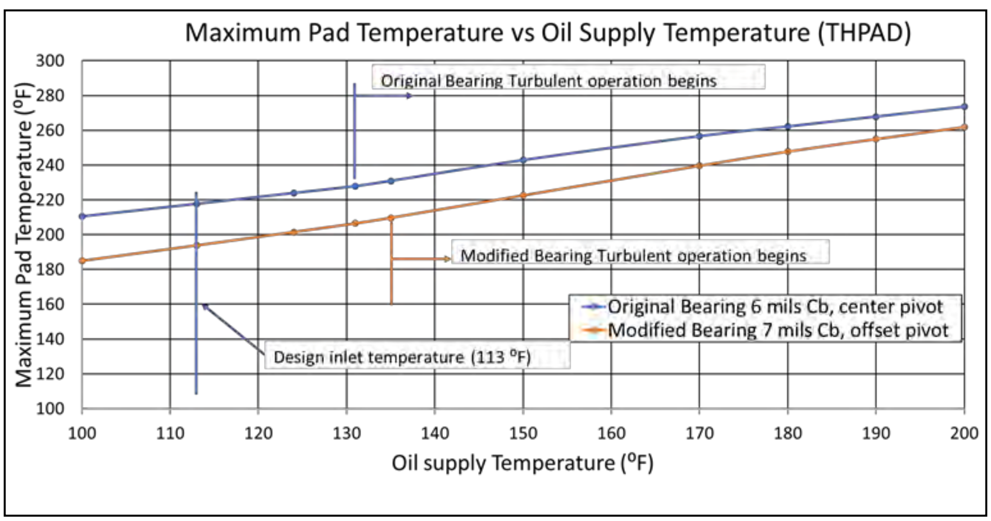

Figure 25: Maximum Pad Temperature Versus Oil Supply Temperature for Study B

One explanation is raising the oil supply temperature brings about the onset of turbulent flow in the bearing. Although the bearing code (THPAD) does not have a sophisticated turbulent model, it did predict the possible onset of turbulent flow at about 131°F (55°C). This results in enhanced heat transfer leading to a corresponding drop in bearing temperature. This is discussed by He, et al (2016). They explain the significant cooling effects attributed to turbulent flow. They concluded that this is more common with large bearing sizes, high shaft speed and low lubricant viscosity. They concluded that turbulent effect should be included in the bearing modeling, especially high-speed applications. However, not all bearing codes incorporate a turbulent model, and when they do the models vary in complexity, so when reviewing bearing analyses this should be discussed as it could influence the accuracy of the results and the conclusions.

Plots of journal speed and unit loading against maximum pad temperature were calculated based on design oil flow. See Figures 26 and 27. As shown, the original bearing design resulted in high temperatures. The modified bearing has some margin to a calculated maximum pad temperature of 200°F (93°C). This does show why it may be important to go beyond the simple journal speed and unit load limits when evaluating bearing designs.

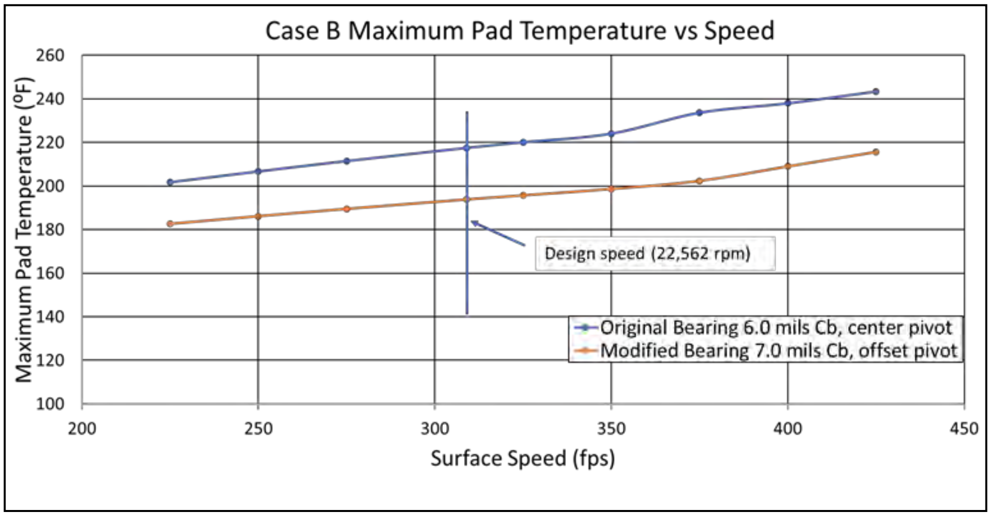

Figure 26: Maximum Pad Temperature Versus Journal Surface Speed for Study B

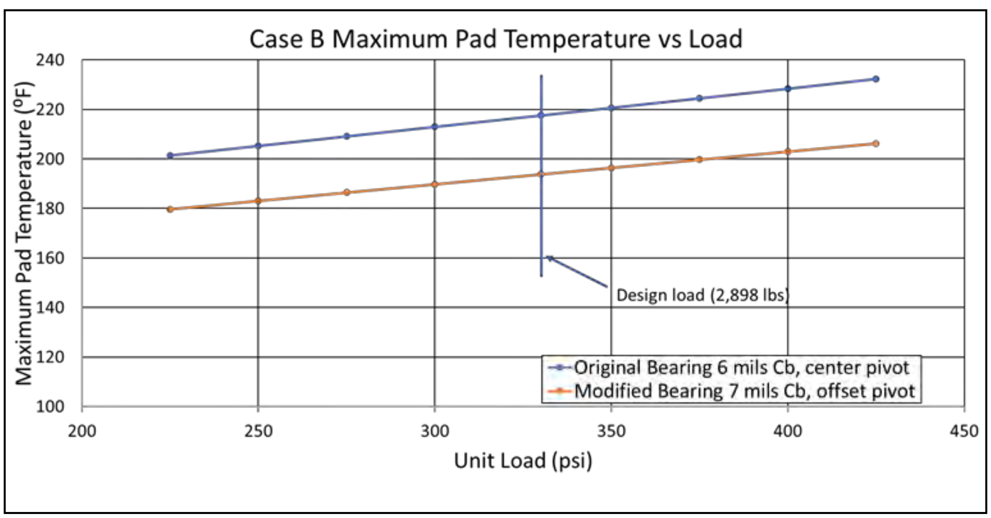

Figure 27: Maximum Pad Temperature Versus Unit Load Case Study B

As with Case Study A, a plot of babbitt strength versus babbitt temperature is shown in Figure 28. The original and modified bearing designs are shown on the plot and as shown; both are below the line. This explains why there weren't chronic problems with babbitt damage even with the original bearing design.

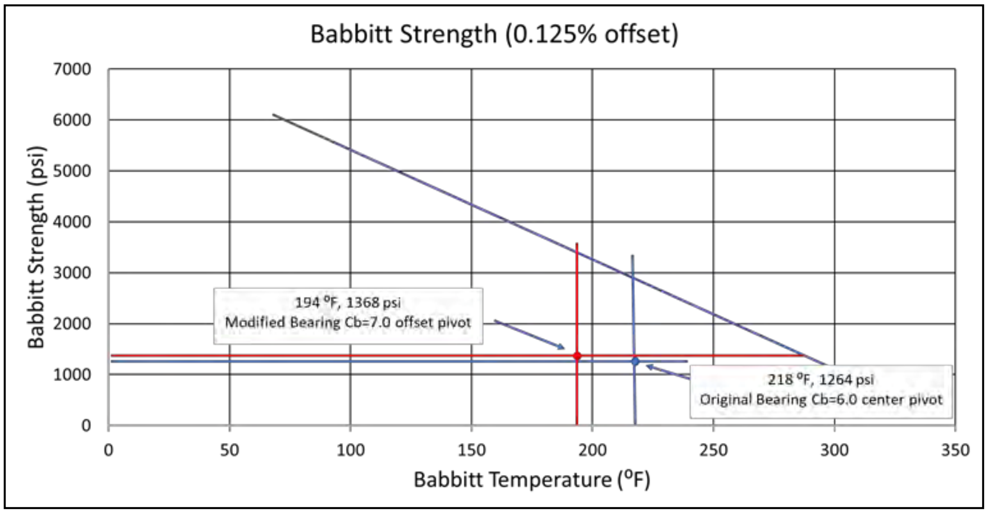

Figure 28: Babbitt Strength Versus Babbitt Temperature for Case Study B

Effect of Turbulence

Turbulence is normally only a concern with very large bearings (16 inches and larger) running at 3600 rpm. Their film thicknesses and surface velocities are such that turbulence is generated in the oil film due to local high Reynolds numbers. At 3600 rpm, a 16-inch journal's surface velocity is 250 ft/s while the three bearings we are analyzing in this tutorial all operate over 285 ft/s; indeed, bearing B operates at 309 ft/s. So, with today's high-speed machinery, even though they have small bearings, they have surface velocities high enough to generate turbulence. The formula for Reynolds Number in a bearing is:

$$R_e = vh/\upsilon \tag{6}$$

Where:

v=fluid velocity, in/s
h=film thickness, in
υ=kinematic viscosity, in^2/s

The Reynolds number varies linearly with the oil film velocity and film thickness and is inversely proportional to viscosity. As such higher speeds, thicker films and decreasing viscosity all work to increase the Reynolds number. Since viscosity decreases when temperatures go up then a high inlet oil temperature will work to increase the Reynolds number. Large Reynolds numbers can indicate the presence of turbulence. One of the biggest questions with the analysis is at what Reynolds number does the film go turbulent. This can be influenced by the bearing geometry and surface conditions and as such there are no universal values.

Turbulent oil films cause an increase in power loss and a decrease in pad temperatures. This can be modelled with increased viscosity (to match power loss) and increased oil film conductivity (to match bearing temperatures). THPAD has a turbulence model that relies heavily on empirical data. Testing of the code against published test data shows pretty good correlation for power loss and poor correlation with pad temperatures when turbulence is present. A newer code, MAXBRG (He, et al 2003), has much more sophisticated turbulence models that use theoretical approaches rather than the empirical ones used by THPAD. This results in more accurate temperature predictions.

Since THPAD could not predict the observed drop in pad temperature with an increased oil supply temperature MAXBRG was utilized. MAXBRG requires much more input, has more switches and more coefficients to enter and runs a lot longer (hours per set of cases versus seconds for THPAD). As such MAXBRG is typically not utilized for day to day analyses, but it comes in handy for those analyses that require a deeper look.

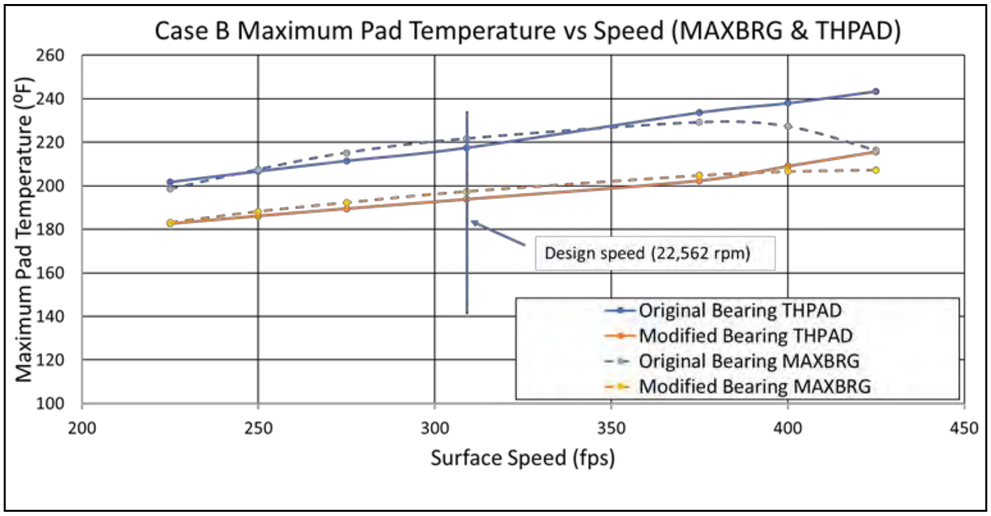

Figure 29 Pad Temperature Prediction versus Surface Speed, THPAD and MAXBRG (Case Study B)

Figure 29 is Figure 26 repeated but with additional data plotted. This allows comparison of THPAD to MAXBRG and the effect of turbulence. Note how the two sets of data agree very closely but diverge at the higher operating speeds. This is due to the initiation of turbulence in the oil film. MAXBRG predicts a reduction, or leveling off, in temperature as speed increases while THPAD continues to predict a rise in temperature as the speed increases. Turbulence initiates at different locations in the bearing depending on film thickness. As such it typically starts in the unloaded pads and then moves to the loaded pads as the film thickness and speed increases.

MAXBRG also assumes there is a conversion period between fully developed laminar flow and fully developed turbulent flow, this is referred to as transitional flow. As an example, at the lower speed (225 ft/s) the unloaded three pads are in transition while the two loaded pads are still laminar. As the speed increases, for instance at 275 ft/s, the entire bearing is in transition, and then at the extreme speed (425 ft/s) the very top pad is fully turbulent while the other pads are still in transition.

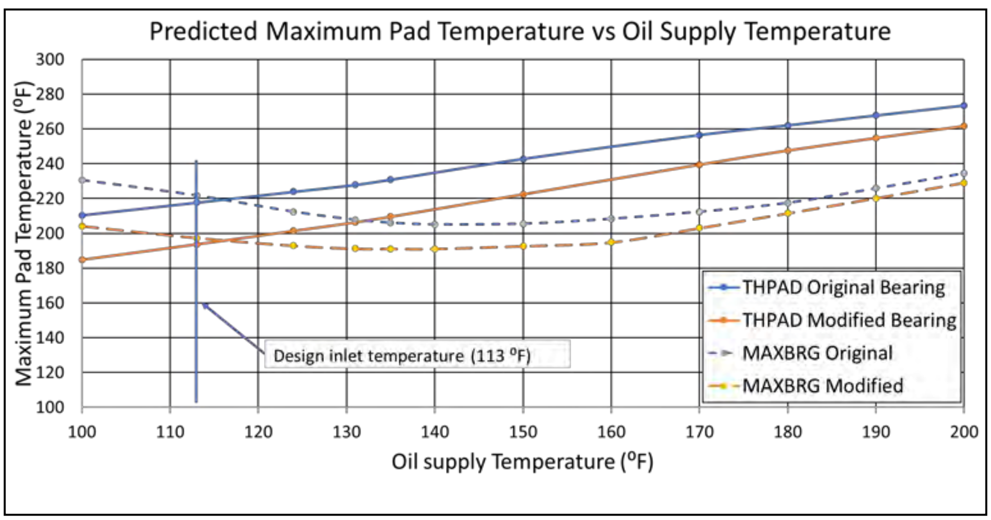

Figure 30 Predicted Pad Temperature Change as Oil Supply Temperature Changes, THPAD and MAXBRG (Case Study B)

For Figure 30 additional data was plotted onto Figure 25 to illustrate the differences between THPAD and MAXBRG. Here there are significant differences again attributed to turbulence. Note that in the “normal” oil supply temperature range of 110-120 ºF the two programs agree closely. As the oil supply temperature increases beyond those normal values the programs diverge with THPAD predicted a steady increase in pad temperatures while MAXBRG predicts a decrease in pad temperature with increasing oil supply temperature, to a minimum at about 140 - 145ºF oil supply temperature. As with Figure 29 this predicted decrease in pad temperature is attributed to turbulence as the warmer oil supply decreases viscosity and therefore increases the Reynolds Number. For instance, at the 113 ºF oil supply temperature MAXBRG predicts the modified bearing will have all 5 pads in transitional flow while at 140 ºF it predicts the top three pads are fully turbulent while the bottom two pads are still in transition.

Note that more research needs to be performed to understand turbulence in tilting pad journal bearings; especially the onset of the transitional and then the fully turbulent flow. For the comparisons presented above there were assumptions made about those Reynold Numbers, and they may not be accurate. The intent was to show the very real possibility of decreasing pad temperature as the oil supply temperature is increased, in these high-speed applications.

Conclusions for Case Study B

After the modifications were made to the third and fourth stage bearings, the bearing temperatures dropped to around 160°F – 190°F (71°C - 89°C). The improvement is likely due to the offset pivot design and increased clearance. This doesn't work in all cases. It depends on the bearing design and the loading. But it is important to understand the implications of this when evaluating a new design or troubleshooting a problem. While there might be times when there are advantages to pushing bearing limits, these need to be evaluated against the risks of problems and having good mitigation and fallback strategies in case the bearing doesn't perform as predicted.

The other lesson is the consideration for including turbulent flow when modelling high speed bearings. While there is certainly a lot of research that still needs to be done in this area it can possibly explain the drop in pad temperature with an increase in oil supply temperature.

Case Study C

This case study pertains to a three pinion, integral gear centrifugal compressor, driven by a 3500 HP, 3000 RPM induction motor. There are two pinions in the lower half of the gearbox with impellers mounted on both ends of both pinions. These rotors are used to compress dry air. There is a single pinion in the gear case cover with impellers mounted on both ends of the pinion. This rotor is used to compress nitrogen. There are thrust collars on all the pinions that transmit pinion axial load to the bullgear.

The gearbox utilizes tilting pad journal bearings for all three pinions with single non-contacting proximity type shaft vibration probes adjacent to each bearing. There are no bearing temperature probes.

The nitrogen section of this compressor was operated at reduced flow and power levels for several years without any problems. However, about a year after the flow and power were increased, the pinion radial vibrations increased. The compressor was shutdown to investigate, and heavy deposits were discovered on both bearings. The shaft journals were cleaned and both pinion bearings were replaced. The compressor ran for approximately one more year and the problem repeated itself.

At the time this problem occurred, a comprehensive bearing analysis was not performed. It was decided to change from the original OEM bearing design to a higher tech bearing that incorporated chromium copper pads, spherical seats, offset pivots, hot oil carryover blockers, bypass cooling and directed lubrication. A picture of this bearing is shown in Figure 31. Note that this bearing design is run in an evacuated condition (no end seals) as discussed in Nicholas (2005).

Figure 31: Replacement Bearing

There were no bearing problems after these bearings were retrofitted. To understand the initial problem, a bearing analysis of the original bearing design was performed. However, not all the bearing information is available. What is known is that this bearing is a 5 pad, tilting pad type with non-aligning, cylindrical, center pivot pads. The nominal bearing journal is 1.875 inches (47.6 mm), the length to diameter ratio is one and the diametral clearance is 0.005 inches (.127 mm) to 0.008 inches (.203 mm). The pinion speed is 35,060 RPM and calculated load at design conditions is 780 lbf (3.48 kN). To perform the analyses, assumptions of the preload, pad angle and pivot angle were made. All the data used for the analyses is summarized in Table 4.

Table 4: Study 3 Design Data – Original Bearing

BEARING GEOMETRY							
Journal Diameter inches/mm	**Bearing Axial Length** inches/mm	**Diametral (Assembled) Design Clearance Range** inches/mm	**Clearance Ratio** mil/inch	**Preload**	**Pad Angle** degrees	**Pivot Angle** degrees	**Pad Orientation (Under Operating Load)**
1.875/47.6	1.875/47.6	0.005-.008/0.13-0.20	2.7-4.3	0.41	58	29	Load on Pad

BEARING OPERATING PARAMETERS						
Pinion Speed RPM	**Journal Load** Lbf/kN	**Journal Speed** ft/s / m/s	**Bearing Unit Load** Psi/MPa	**Lubricant**	**Oil Supply Temperature** Deg F/Deg C	**Calculated Oil Flow to Bearing** GPM/lpm
35060	780/3.5	287/87.5	222/1.5	ISO VG 46	113/45	11.2/42

A plot of minimum oil film thickness against diametral clearance is shown in Figure 32. As shown, the film thickness is above 0.001 inches (0.025 mm) down to a diametral clearance of about 0.0043 inches (0.109 mm). This is below the minimum design clearance of 0.005 inches (0.127 mm). Also note that both the original bearing and the upgrade bearing have similar minimum oil film thickness values through the range. For simplicity .0062 inches clearance will be used for comparison purposes.

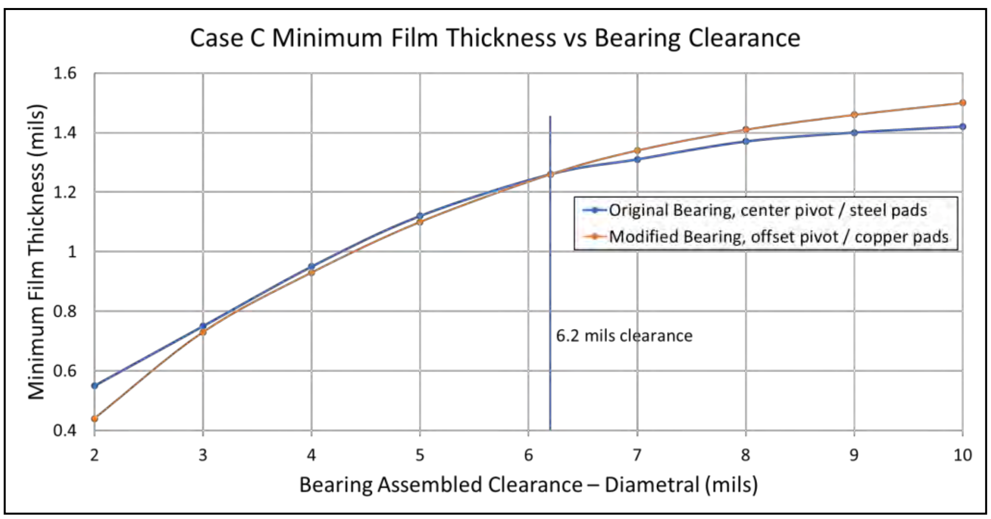

Figure 32: Calculated Minimum oil film thickness Versus Diametral Clearance for Study C

A plot of maximum pad temperature versus radial diametral clearance is shown in Figure 33. As shown, the maximum pad temperature is quite low at the design clearance and doesn't exceed 200°F (93°C) until the diametral clearance is about 0.004 inches (0.102 mm). The bearing clearance was checked in the field and was in accordance with the design. There was no evidence of a tight bearing clearance or excessive clamping. And, there were no problems with any of the other bearings. Oil testing did not find any issues. And so, it seemed unlikely to be an issue with the oil.

The upgrade demonstrates a significant calculated decrease in pad operating temperature of about 35°F (19°C). Of course, with no temperature instrumentation it is not possible to verify these values.

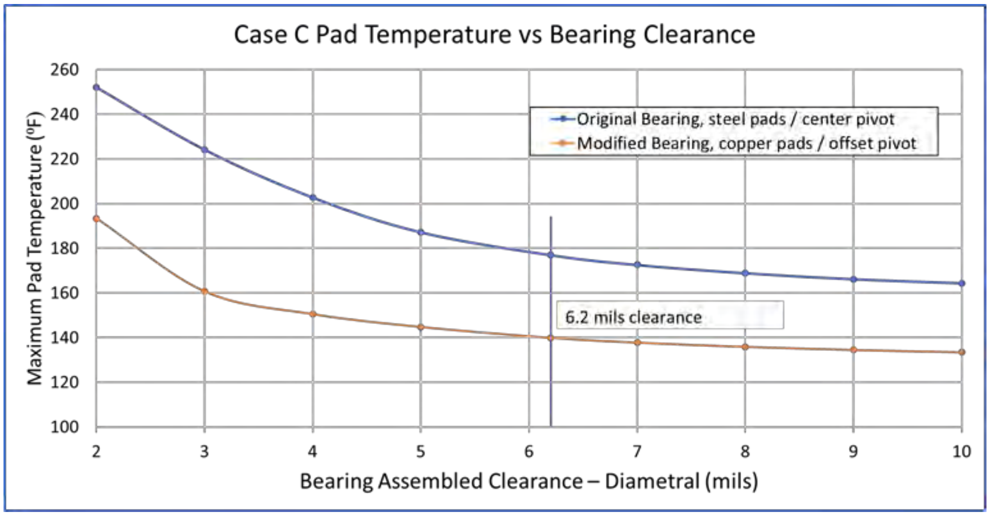

Figure 33: Calculated Maximum Pad Temperature Versus Diametral Clearance for Study C

One possibility is that there was an issue with oil supply to these bearings. The hydraulics within the gearbox are complicated and it is possible that there is an issue with either getting oil flow to the bearing or something is preventing the oil from draining properly from the bearing, also resulting in lower oil flow. Although these bearings are very common for this supplier and have been used successfully in many other machines at higher speeds, the problem bearings in this machine are in the gearbox cover, furthest from the oil supply. Based on inspections at site, there were no issues with excessive clamping or misalignment. So, based on experience and inspections, there didn't appear to be any issues with the bearing selection or application. This gives some credibility to an issue with oil flow. So, the bearing analysis was rerun assuming a lower oil flow.

A plot of oil flow versus maximum pad temperature is shown in Figure 34. As shown, the maximum pad temperature doesn't exceed 200°F (93°C) until the oil flow is about 2.5 GPM. Also, as shown the upgraded bearing doesn't predict operation above 200°F until the flow is below 1.25 gpm, making it less susceptible to reductions in oil flow.

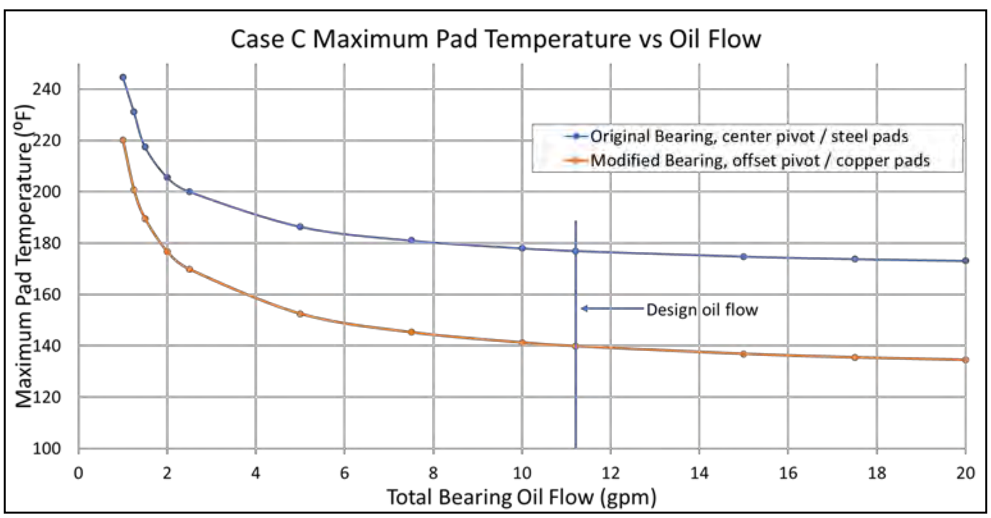

Figure 34: Calculated Maximum Pad Temperature Versus Oil Flow for Case Study C

Conclusions for Case Study C

There have been no bearing issues since the new bearings were installed. The improvement is likely due to higher capability of the new bearing design. The interesting take-away from this example is the improvement that is available with a bearing design that utilizes several significant design upgrades. The copper pad upgrade alone can significantly reduce the temperature of a hot running bearing. Likewise, the use offset pivots has been proven to also significantly reduce operating temperatures. Adding in bypass cooling, directed lubrication and hot oil carryover blockers can further enhance the bearing design.

New Bearing Application

When a compressor was being considered for a new project, the bearing data was reviewed, and a concern was raised about a relatively high journal speed. This bearing is very similar to the bearing discussed in Case Study C. The bearing information is below.

Table 5: Bearing Data for New Application

BEARING GEOMETRY							
Journal Diameter inches/mm	**Bearing Axial Length inches/mm**	**Diametral (Assembled) Nominal Design Clearance inches/mm**	**Clearance Ratio mil/inch**	**Preload**	**Pad Angle degrees**	**Pivot Angle degrees**	**Pad Orientation (Under Operating Load)**
1.875/47.6	1.875/47.6	0.0051/0.130	2.7	0.045	58	29	Load on Pad
BEARING OPERATING PARAMETERS							
Pinion Speed RPM	**Journal Load Lbf/kN**	**Journal Speed ft/s / m/s**	**Bearing Unit Load Psi/MPa**	**Lubricant**	**Oil Supply Temperature Deg F/Deg C**	**Calculated Oil Flow to Bearing GPM/lpm**	
40691	544/2.4	333/102	155/1.1	ISO VG 46	110/43.3	11.2/42.4	

The data about the bearing tolerances was not available, except that this is a low preload bearing. A plot of diametral clearance versus maximum pad temperature and minimum oil film thickness is shown in Figure 35 and is based on a constant preload of 0.045. As shown, the bearing temperature does not exceed 200°F (93°C) until the clearance drops below 0.0045 inches (0.110 mm) and the minimum oil film thickness is not less than 0.0008 inches (.0254 mm) until the diametral clearance is below 0.0033 inches (.084 mm). At 0.8 mils (0.0254 mm) minimum film thickness the predicted pad temperature is 225°F (107°C) so operation with clearances below 0.0035 inches should be avoided. Although the journal speed is a bit elevated, the load is very low, which results in good bearing performance. Also, there is a lot of internal history with this bearing design in similar applications. Based on this, the bearing selection was deemed acceptable.

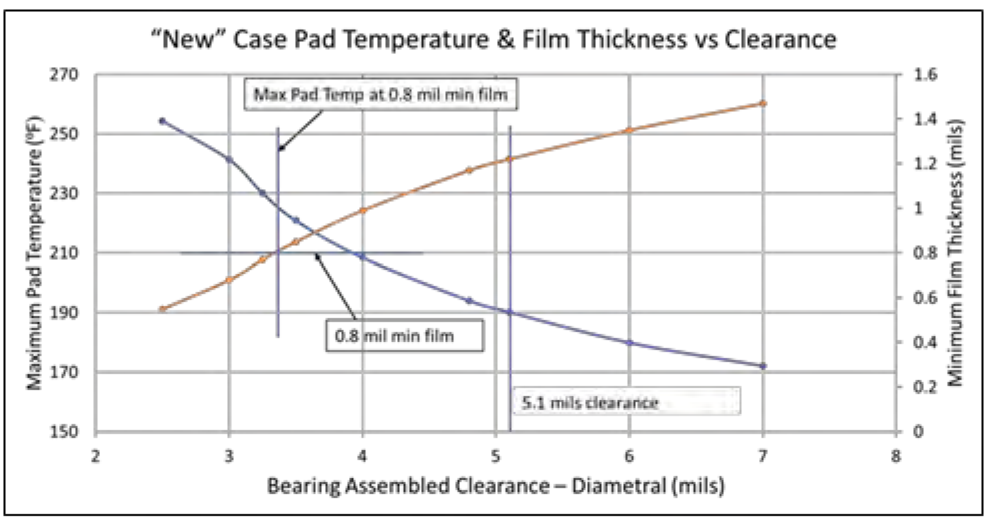

Figure 35: Maximum Pad Temperature and Minimum Oil Film Thickness Versus Diametral Clearance for New Application

ROTORDYNAMIC CONSIDERATIONS

Journal bearings have both static and dynamic characteristics. Static aspects include bearing load capacity, side leakage, frictional power loss, oil film temperature and pad temperature. Dynamic characteristics include stiffness and damping properties.

Making some minor bearing changes to address a performance issue can also impact the rotordynamic behavior. This should get reviewed to avoid a potential vibration issue. To illustrate this, the rotor in Case Study 1, Bearing A will be used as an example. Rotordynamic analyses were performed based on the original bearing design clearance and the modified bearing clearance to understand the effect on the rotordynamic behavior and to determine if there were any rotordynamic issues with the modified bearing design.

Rotordynamic Analysis

From API-617, if a rotor-bearing system is modeled as a simple mass-spring-damper system, the governing equation of motion for this system can be written as

$$m\ddot{x} + c\dot{x} + kx = F(t) \tag{7}$$

Where:
m = mass of the block.
c = viscous damping coefficient.
k = stiffness of the elastic element.
x = displacement of the block.
F(t) = force applied to the block (time-dependent function).

While this simple model does not reflect the actual rotordynamic behavior of a compressor rotor/bearing system, it can be used to illustrate the basic concepts. In this example, the forced response is counteracted by the rotor mass and the support's stiffness and damping characteristics. Fluid film bearings provide stiffness and damping and so changing a bearing's stiffness and damping properties will affect the rotordynamic behavior. Table 6 shows the differences in the key bearing design parameters and the stiffness and damping coefficients for both the original design and the modified design. Note that when the diametral clearance was changed, the pad clearance, was also changed to maintain a similar preload.

Table 6: Bearing A Rotordynamic Data

CASE	CLEARANCE	Pad Radial Clearance	Assembled Radial Clearance	PRELOAD	Oil Pressure	STIFFNESS		DAMPING	
		inches	inches		psig	KXX lbf/in	KYY lbf/in	DXX lbf-sec/in	DYY lbf-sec/in
Original	Design	0.0037	0.0026	0.291	26	8.71E+05	1.39E+06	6.10E+02	6.65E+02
Original	Maximum	0.0037	0.0036	0.018	26	6.46E+05	1.37E+06	3.18E+02	6.17E+02
Modified	Minimum	0.0063	0.0040	0.364	26	6.12E+05	1.10E+06	2.74E+02	4.09E+02
Modified	Nominal	0.0063	0.0044	0.304	26	5.60E+05	1.07E+06	2.48E+02	3.77E+02
Modified	Maximum	0.0063	0.0048	0.237	26	5.21E+05	1.04E+06	2.34E+02	3.72E+02

The purpose of performing a rotordynamic analysis is to ensure that there are no undesirable rotor/bearing performance issues. The key results from the lateral rotordynamic analysis include a damped eigenvalue analysis and a damped unbalance response analysis.

Stability Analysis

The damped critical speed analysis uses bearing damping and stiffness coefficients to predict the rotor/bearing system natural frequencies and associated mode shapes of vibration. The dynamic coefficients vary with speed and so the damped critical speed analysis is performed based on the rotor operating at the design speed. The output of the damped critical speed analysis includes all the calculated damped natural frequencies within a speed range of 0% to 125% of the maximum operating speed, and the associated mode shapes and stability values.

A resonance condition exists when the frequency of a harmonic (periodic) forcing function coincides with a natural frequency of the rotor/bearing system. Potential excitation sources include rotor unbalance, oil film instability, internal rubs, dynamic seal effects, loose parts, etc. These sources can excite subsynchronous, synchronous and supersynchronous resonant frequencies. When this happens, the forced vibrations resulting from the given exciting mechanism dynamically amplify the levels of vibration at the resonant frequency. The level of stability or damping refers to the unit's resistance to these excitations. So, the greater the damping, the less the dynamic amplification of the system vibration.

A variety of parameters can be used to evaluate the level of stability, or damping. An example that is commonly used to evaluate stability is the critical damping ratio. The critical damping coefficient Cc is the damping value required to completely suppress any free vibration of the system. The critical damping ratio is the ratio of the actual damping coefficient divided by the critical damping coefficient, or C/ Cc. If the critical damping ratio is negative the system is said to be unstable. A ratio of zero indicates that there is no damping, and the amplification is infinity. If the critical damping ratio is greater than zero, there is some damping. A critical damping ratio of 1 indicates that the system is critically damped and there is no amplification when operating near or at this resonant frequency. Ratios between 0 and 1 indicate the degree of damping. Figure 36 shows how decreasing damping affects the amplification of the vibration.

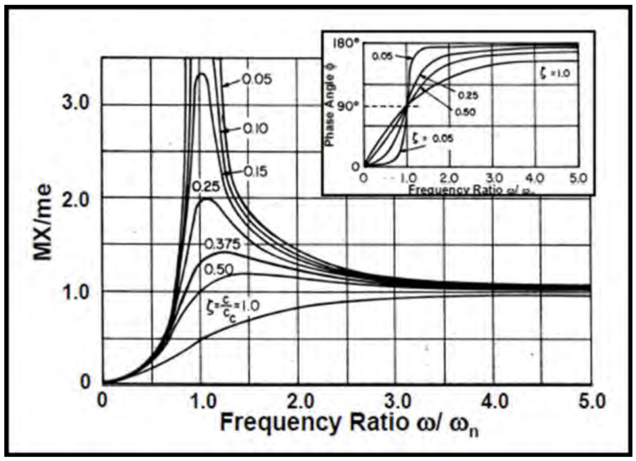

Figure 36: Critical Damping Ratio (Courtesy of Franklin Institute Research Laboratories)

Another stability parameter is called the log dec. The logarithmic decrement (log dec) is the natural logarithm of the ratio of any two successive amplitude peaks in a free harmonic vibration. The log decrement is a measure of how quickly the free vibrations experienced by the rotor system decay. When the log decrement is positive, the system is stable. Conversely, when the log decrement is negative, the system is unstable. According to level 1 screening criterion in API-617, if the log dec is less than 0.1, further stability analyses shall be performed. However, there is no industry standard of acceptance criteria on the minimum log dec. Even API-684 states that "…interpretation of results and suitability of design to be mutually agreed upon by the vendor and purchaser."

The results of the damped natural frequency analysis for both the original bearing clearance case as well as the modified bearing clearance case are shown below in Tables 7 and 8. These are all forward modes.

Table 7: Results of Damped Natural Frequency Analysis for Original Diametral Bearing Clearance

MODE #	ORIGINAL DESIGN BEARING CLEARANCES					
	Natural Frequency CPM	Critical Damping Ratio	Log Dec	Amplification Factor	Separation Margin	API Required SM
1	7288	0.0604	0.38	1.3	69%	110%
2	9531	0.1013	0.64	0.8	60%	41%
3	10395	0.0703	0.44	1.1	56%	63%
4	12386	0.0851	0.53	0.9	48%	47%
5	54281	0.0558	0.35	1.4	128%	0%
6	67580	0.1577	0.99	0.5	183%	0%

Table 8: Results of Damped Natural Frequency Analysis for Modified Diametral Bearing Clearance

MODE #	MODIFIED BEARING CLEARANCES					
	Natural Frequency CPM	Critical Damping Ratio	Log Dec	Amplification Factor	Separation Margin	API Required SM
1	6615	0.0654	0.41	7.6	72%	14%
2	8535	0.0727	0.46	6.9	64%	14%
3	9724	0.0561	0.35	8.9	59%	15%
4	11554	0.0851	0.53	5.9	52%	13%
5	20336	0.4072	2.56	1.2	15%	0%
6	27023	0.4501	2.83	1.1	13%	0%
7	36812	0.2838	1.78	1.8	54%	0%
8	51447	0.3998	2.51	1.3	116%	0%

What is interesting is that what may be viewed as a relatively minor increase in bearing clearance had a moderate impact on the calculated natural frequencies and the stability. Based on the original design case, there are no natural frequencies near the operating speed. With the modified clearance case there are 2 modes (modes 5 and 6) which are close to the operating speed. Fortunately, these modes are very well damped. While there are no stability issues for either case, the magnitude changes show the value in considering this analysis anytime a change is made to something in the rotor/bearing system.

A mode shape is the deflected shape of a rotor calculated at the particular natural frequency. Each resonant frequency will have a different mode shape. Understanding the mode shapes is valuable for a couple of reasons:

1. The location(s) of the greatest displacement(s) are places that are the most sensitive to excitation forces for that particular mode. So, when performing the damped unbalance response analysis, unbalance should be located at these locations(s).
2. Knowing the mode shapes permits an estimation of rotor displacements at close fit areas such as bearing and seal locations, and at vibration probe locations.

The mode shape for mode 3 for the modified bearing clearance case is shown in Figure 37. This mode can be excited by adding unbalance at the locations of greatest displacement, locations 1 and 39 as shown.

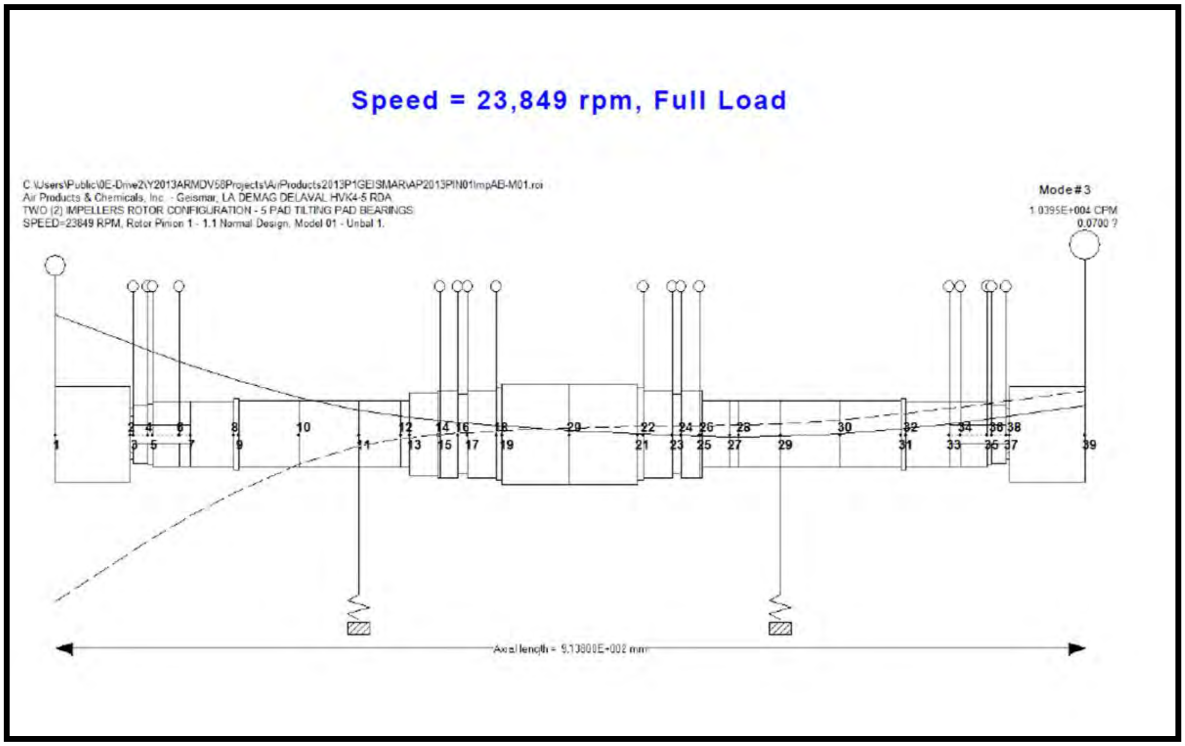

Figure 37: Mode #3 for Modified Bearing Clearance Case

Unbalance Response Analysis

The damped unbalance response analysis is the principal tool used by the rotordynamicist to evaluate relevant lateral rotordynamics characteristics including lateral critical speeds and associated amplification factors. The result of an unbalance response analysis is a calculation of the rotor response (vibration) to a set of applied unbalances as a function of speed. The location, phase and distribution of the unbalances are based on the mode that is being excited. It is important that all critical speeds below the maximum operating speed and the critical speed immediately above the maximum operating speed are analyzed. API also defines how to determine the magnitude of the unbalances that are applied. For this case study, an unbalance of 33.6 gram-mm was added at both impeller locations. These unbalances were in phase with one another.

The results of the unbalance response analysis are displayed as a plot of vibration versus shaft rotational speed. This is commonly called a Bode plot. These plots are generated for several axial shaft locations which typically include bearing locations, shaft displacement probe areas, and mid-span. For this case study, unbalance response plots were generated for the following locations:

- Station 1: second stage impeller
- Station 11: second stage bearing
- Station 20: rotor mid-span
- Station 29: first stage bearing
- Station 39: first stage impeller

The vibration peaks are the resonant speeds and these peaks are reviewed for the magnitude of the response and the proximity to the operating speed. The magnitude is evaluated based on the amplification factor (AF). The method of calculating the amplification factor is shown on the Bode plot in Figure 38. This also introduces the term called the separation margin (SM), which is the margin of the peak vibration to the operating speed. API defines critical speeds as those resonant speeds with an amplification factor greater than 2.5. If the amplification factor is less than 2.5, no separation margin is required. If the amplification factor for any critical speed below the operating speed is greater than 2.5, the minimum separation margin shall be 16% or the value from the equation below, whichever is less.

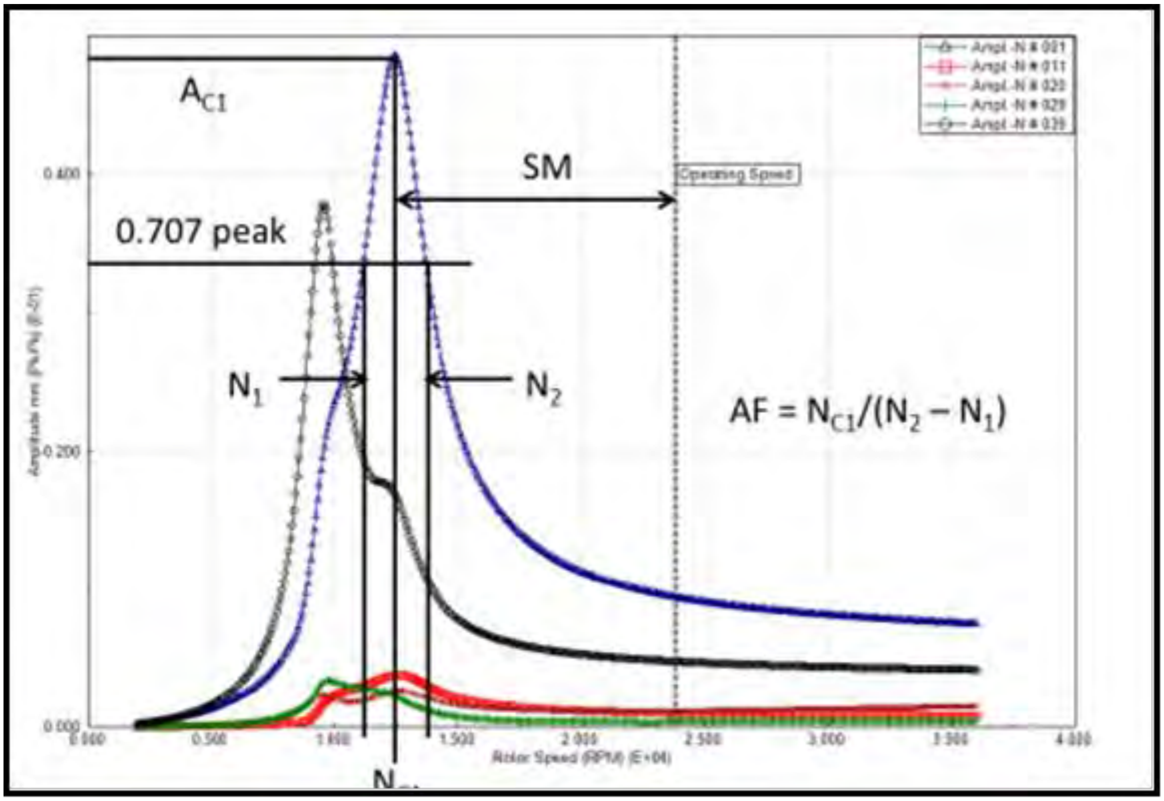

Figure 38: Bode Plot

$$SM = 17\left(1 - \frac{1}{AF - 1.5}\right) \tag{8}$$

If the amplification factor for any critical speed above the operating speed is greater than 2.5, the minimum separation margin shall be 26% or the value from the equation below, whichever is less.

$$SM = 10 + 17\left(1 - \frac{1}{AF - 1.5}\right) \tag{9}$$

For this case study the unbalance response plots for both the original bearing clearance case as well as the modified bearing clearance case are shown in Figures 39 and 40. As shown, for both cases the separation margins are acceptable. However, the unbalance response at similar natural frequencies is slightly higher in the modified bearing clearance case. This again shows the value in considering a rotordynamic analysis anytime something in the rotor bearing system is changed.

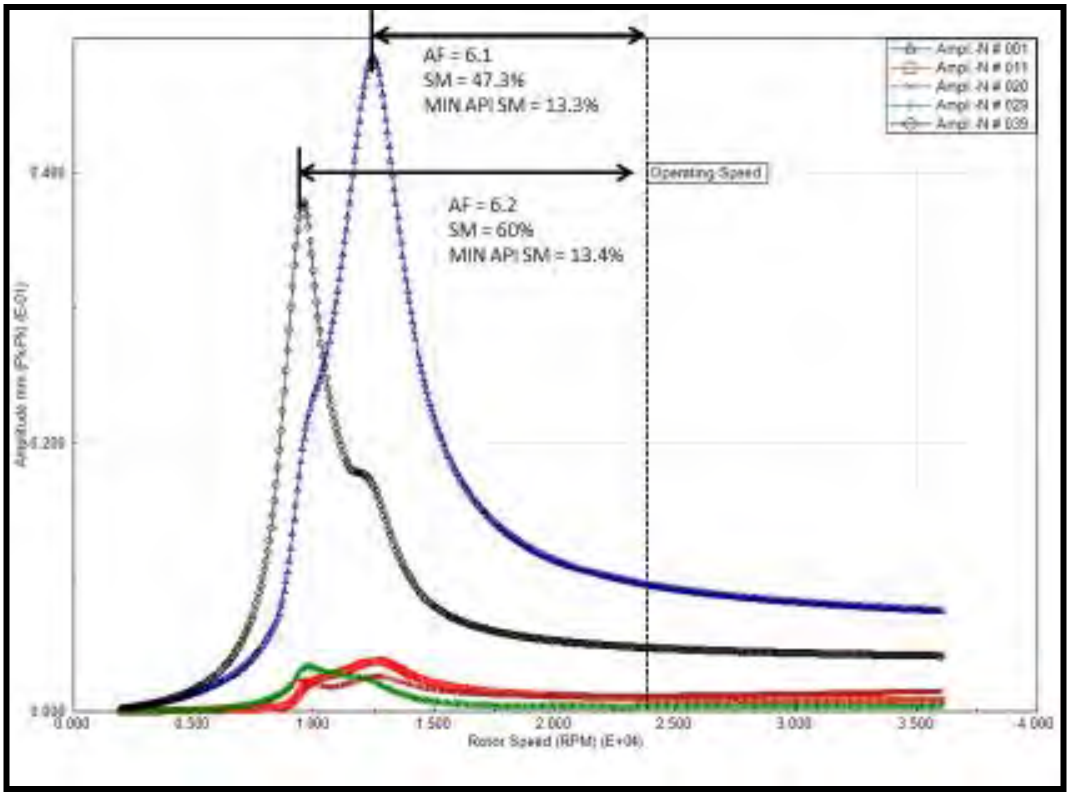

Figure 39: Unbalance Response Plot; Original Bearing Clearance Case

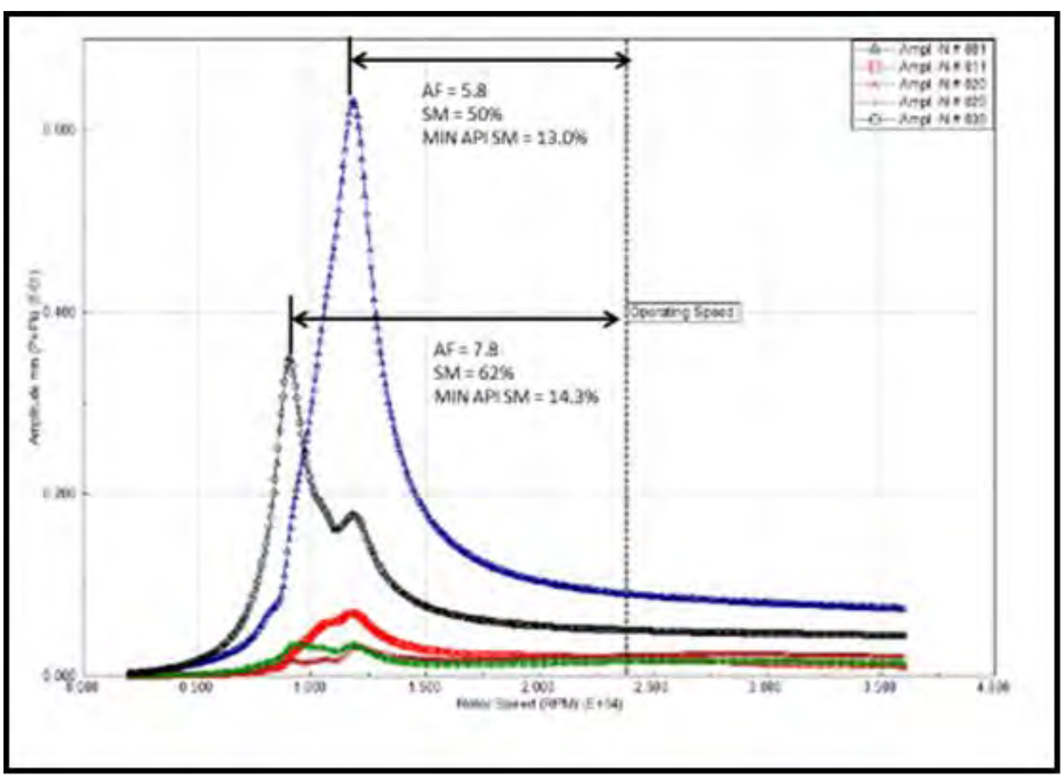

Figure 40: Unbalance Response Plot; Modified Bearing Clearance Case

Rotordynamic Discussion Conclusions

The lateral rotordynamics for both the original bearing clearance case as well as the modified bearing clearance case are acceptable per API-617 standards. However, the analyses do show that the rotordynamic behavior changes even when making what appears to be a minor bearing clearance change. And for this reason, a lateral rotordynamic analysis should be considered whenever something is changed in the rotor bearing/system.

MISCELLANEOUS TOPICS

There are a few important miscellaneous topics addressed covering monitoring, operation and maintenance aspects.

Condition Monitoring

The purpose of a machinery condition monitoring system is to identify machine degradation early so that action can be taken before it progresses to a point of machine failure. Failures usually occur at the bearings because they are designed to fail before the shaft does. Condition monitoring can also be a very powerful tool in preventing sudden, catastrophic bearing failures. Sometimes the signs of progressive damage are very subtle and difficult to detect. In these cases, a bearing may appear to suddenly fail without warning.

The instruments used to monitor bearing condition include bearing temperature sensors and vibration probes. When reviewing bearing temperature probes, it is important to understand where the probe tip is located. A sensor tip located near the bearing surface will show a higher temperature reading than one located on the back of a bearing pad, and it will also react much quicker to changes in temperature. API-670 is a good reference on the design and installation of bearing temperature sensors. Figures 41 and 42 shows the ideal sensor placement locations for tilting pad bearings.

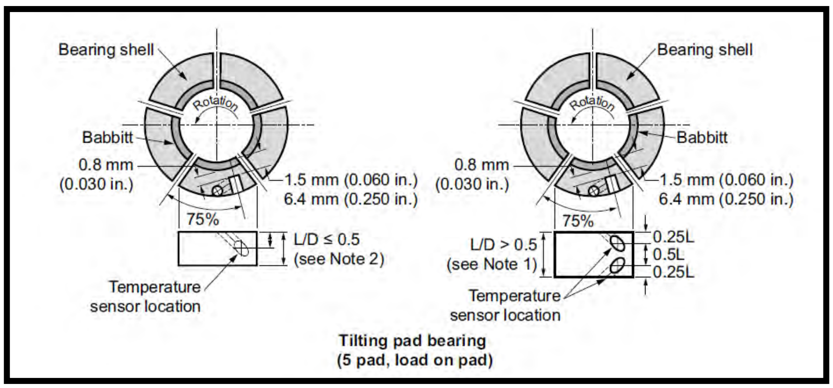

Figure 41: Bearing Temperature Sensor Location for Load on Pad Design (Courtesy of API-670)

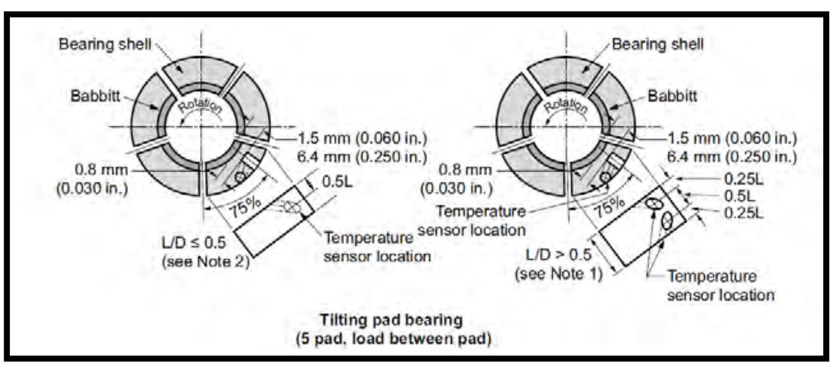

Figure 42: Bearing Temperature Sensor Location for Load Between Pad Design (Courtesy of API-670)

When the rotor vibrates, the resulting force is transferred through bearings to support structures and then to ground. In general, the best place to measure vibration is either on the shaft near the bearings, or on the bearing housing or other structural part which responds to the rotor vibration. Vibration can be measured radially in usually in the two planes adjacent to the bearing, and in the axial plane with respect to the shaft.

API-670 also provides good guidance on the design and installation of vibration probes. Ideally two radially orientated probes are provided per bearing. With two probes per bearing located 90 degrees apart it is possible to observe the motion of the rotor in the bearing; both dynamically (vibration) and statically (rotor position). Rotor position analysis can provide extra useful information for diagnosis. However, there are many cases where only one probe is used. This still works but relying on a single instrument for monitoring and protection is a compromise in cost versus reliability.

Bearing temperature and vibration can change depending on operating conditions such as load, speed, oil temperature and pressure, ambient conditions, cooling water temperature, etc. The ideal condition monitoring systems uses statistical analysis to predict the bearing temperature and vibration behavior based on these other parameters, and then use adaptive control limits based on the predicted values. This allows for tighter controls than constant level limits. And it allows for much earlier detection on a progressive issue than the protection limits and/or a manual review.

Regarding bearing temperatures, experience has shown a tendency for the formation of deposits on the bearing surfaces when the bearing temperature, as measured by a probe close to the bearing surface, is above 221°F (105°C). This can also vary either higher or lower depending on the oil quality, amount of water in the oil and other conditions. This is an oil issue and not a bearing limitation. However, the formation of deposits often results in an increase in vibration and/or further increases in bearing temperature, which can lead to a trip.

Protection Systems
The purpose of a machinery protection system is to ensure that a machine is operated within its normal parameters, provide detection of an imminent problem, and to trip a machine if there is a serious problem. A machinery protection system consists of the instruments that measure the key parameters, and a control system that displays, alarms and/or trips the machine if a parameter exceeds predefined limits.

When looking at a protection system there are two things to consider. First are the set points for alarms and trips, and second is the protection logic. API-617 provides good guidance on vibration limits but defers to the machinery supplier for the bearing temperature limits. Common bearing temperature alarm and trip set points for the type of machines and bearings discussed in this tutorial are 230°F (110°C) and 248°F (120°C).

In general, when bearing damage starts to occur, vibration is a leading indicator and bearing temperature is a lagging indicator. Although this is not always the case, vibration protection is still the primary indicator and bearing temperatures tend to occur more slowly.

As discussed by Smith (2017), most protective systems focus on steady state operation. This makes sense since this is where the machine will run most of the time. However, transient cases often don’t get the same level of attention during either the design or operational phase. Many, if not most catastrophic compressor failures occur when operating in transient operation. Some recommendations to protect bearings include:

- Start-up vibration protection per API-670. This should also include no time delay on a trip signal, bad quality as a vote to trip and the set point and duration for the trip multipliers should be minimized based on previous starts.
- “x” and “y” vibration probes should be considered. If “”x” and “y” probes are supplied, the trip logic for start-up should be nonvoting for start-up, meaning that if the vibration reaches the trip set point on either probe, the machine will trip.
- Bearing temperature protection. Since bearing temperatures tends to change slowly, either an automated or manual trip can be used for protection.

Basic Maintenance Practices
If properly designed and applied, hydrodynamic radial bearings should not wear for the life of the machine because these bearings are not in contact with the rotating shaft. However, if the bearings are operated at conditions during steady state or transient operation outside of the design limitations, the bearings can be damaged. This should get picked up by the condition monitoring and/or protection system.

However, oil condition is critical to the reliability of bearings and oil analysis is an essential part of a good maintenance system and can also provide early indication of a bearing problem. Whalen, et al. (2012) discuss oil analysis in detail and can be referenced by the reader. The one area that is newer to the industry and deserves special attention is varnish detection. As mentioned throughout this tutorial, deposits such as varnish cause bearing damage and/or rotor instability leading to an unplanned trip. Early detection and action are essential because once started, varnish is more difficult to address.

As discussed by Whalen, et al. (2012), "*Varnish detection and mitigation has become an industry upon itself with the introduction of Group II base oils. Many of the routine tests used to monitor oil quality cannot detect varnish problems until it is too late. It has been a source of many bearing failures*." "*If varnish is discovered there are ways of treating the problem through special types of filtration. Ultimately, the source of the heat generation should be located to eliminate the problem. The following tests have shown promise in the early detection and the onset of varnishing.*

- *The potential for varnish build-up often correlates with a decrease in an oil's antioxidant levels. Monitoring these levels have shown to provide a source of early detection. The monitoring of oxidation inhibitors (Amines & Phenols) can be accomplished with the use of FTIR and/or RULER. To perform these tests properly, it is important to provide a base line sample before monitoring depletion rates. Typical depletion limits are 25 to 50% of the baseline before a partial change out or a complete oil change out is required. The need for frequent change outs could be a key indicator there is an internal heat source causing the accelerated depletion of the antioxidants.*
- *QSA – If the presence of varnish already exists within the system, a well-known test developed by Analysts Inc. can be used to detect the severity of the varnish or sludge that exists. Analysts Inc developed a color patch test (QSA) that correlates the contrast of a sample patch to a numerical severity of the varnish or sludge that plagues the system. It also provides a patch weight that can be monitored and trended as well.*

Before applying these practices, it is imperative that the physical act of obtaining a representative sample by a trained individual is accomplished. If this is ignored, much of the data gathered will be misleading in performing a final diagnosis. To determine the best location for sampling a bearing, the method of oil application must be understood. (e.g. In force fed systems it is best practice to set up a primary and secondary sample point. The primary sample point should be located on the common oil return line before it empties into the oil reservoir. The secondary sample point should be located on the dedicated return line of each bearing to further narrow down wear and contamination related problems).

Much of the analysis data stated above is geared around building a primary testing slate for routine testing. Secondary test methods may be required to further troubleshoot negative sample results."

CONCLUSIONS

The cornerstones of tilting pad bearing reliability, which has been the focus of this tutorial, include design/application, condition monitoring, protection strategy and maintenance practices. When reviewing a bearing design for an application, the old method of applying simple limits to journal speed and unit load can be ineffective in preventing bearing issues or result in a more conservative design than is needed. Based on the discussion in this tutorial, the following criteria should be considered when reviewing a bearing for an application:

- Require a calculated minimum oil film thickness of .0008 inches (.020 mm) under all design operating conditions. Minimum oil film thickness less than this requires review and approval by the end user.
- Require a maximum measured bearing operating temperature of 212°C (100°C) in the shop and in the field under all process and oil operating conditions for tilting pad babbitt bearings. This is an acceptance criterion, not the alarm and trip temperatures, which would be higher. If the bearing temperature is not monitored, the calculated bearing maximum pad temperature shall not exceed 200°F (93°C) without the approval of the end user.
- The rotordynamic report shall be provided by the manufacturer for review.

To detect progressive bearing issues and protect the bearings, the end user should consider:

- A robust condition monitoring system that can provide early detection of bearing wear or damage.
- A robust protection system that includes appropriate protections during transient conditions.
- A robust oil analysis program to detect early signs of oil degradation.

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