Reducing Power Losses in Tilting Pad Bearings

Michael Blumenfeld, Brian R. Fabijonas, Wei Tian, Howard Hawkins, and Robert Rodzwic. 24th Pprime Workshop: Futuroscope, October 2–3, 2025

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Reducing Power Losses in Tilting Pad Bearings
Réduction des pertes de puissance dans les butées hydrodynamiques à patins oscillants

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Mots clés: fluides non newtoniens, pertes de puissance, butées hydrodynamiques à patins oscillants, réduction de la viscosité.

Bearing losses are significant in the power generation sector where small changes in efficiency can scale rapidly. We describe a set of tests of a thrust bearing lubricated by a non-Newtonian fluid. In this fluid, the viscosity depends on the local shear rate in such a way that a thicker oil is guaranteed at low shear and a thinner oil at high shear which in turn reduces power loss in the bearing. Modelled data will be compared to experimental determinations of power loss and bearing pad temperature measured on the test rig. Finally, novel approaches to turbine lubricant design will be proposed that may enable step-change improvements in efficiency while still maintaining the durability required for reliable operation.

Les pertes dans les butées sont significatives dans le secteur de la production d'énergie où de petites améliorations d'efficacité peuvent rapidement prendre de l'ampleur. Nous décrivons une série de tests sur une butée lubrifiée par un fluide non-newtonien. Dans ce fluide, la viscosité dépend du taux de cisaillement local de telle sorte qu'un grand film fluide est garanti à faible cisaillement et un film plus mince à fort cisaillement, ce qui réduit ainsi les pertes de puissance dans la butée. Les données modélisées seront comparées aux déterminations expérimentales de perte de puissance et de température des patins de butée mesurées sur le banc d'essai. Enfin, des approches novatrices de conception de lubrifiant pour turbines seront proposées, pouvant permettre des améliorations substantielles de l'efficacité tout en maintenant la durabilité nécessaire à un fonctionnement fiable.

1 Introduction

Global energy demand is forecast to grow in the next 25 years. Whereas some sources of energy will rise slowly or even plateau, the demand for electricity is anticipated to have a steep slope. See Figure 1. One very common generator of electricity is gas turbines. Mid-sized turbines are currently experiencing a boom for use as generators of electricity in data centers [1]. The need to reduce power loss in these turbines is paramount. Every kilowatt of power loss preserved translates to kilovolts of electricity. Consequently, industry has been looking at turbines critically.

A key place to look for efficiency gains is where the turbines use lubrication, of which there are many: bearings (thrust and journal), gearboxes (accessory, turning gears), torque converters, lubrication oil pumps, just to name a few. Each of these elements has power loss. Improving efficiency in each of these can have dramatic effects on the turbine. As an example, ExxonMobil recently demonstrated that changing the lubrication oil for four journal bearings in a generator-compressor train from ISO VG 32 to ISO VG 18 resulted in a 15% efficiency gain for each of the generator bearings and a 16% gain for each of the compressor bearings. This net was a reduction

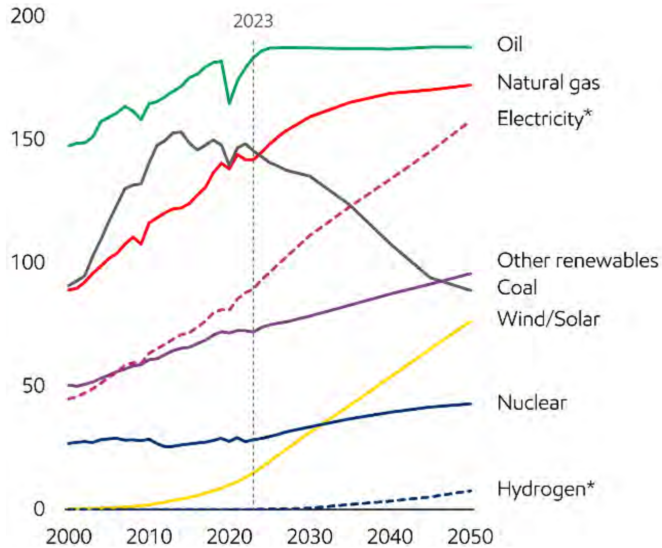

Fig 1- Anticipated global energy demand. Note that electricity and hydrogen are secondary sources (dashed lines) derived from primary sources (solid lines) shown. [2]

of 136 kW in power loss for the full system [3]. The caveat, of course, is that most turbines cannot survive using ISO VG 18 oil at all operating conditions; in particular, thicker oil is needed at startup.

This paper explores the following question: is it possible to blend an oil that will have the benefits of a thicker oil for low shaft speeds and thinner oil at higher speeds? The answer is yes and lies in the realm of shear-dependent non-Newtonian oils (SDNNO). We describe a suite of bearing performance tests carried out on a test rig which uses tilting-pad thrust bearings with two Newtonian oils and one SDNNO and compare the measured power loss and pad temperatures.

2 Shear Dependent Non-Newtonian Oils

Non-Newtonian engine oils have become integral to modern automotive lubrication systems due to their ability to adapt viscosity in response to changing mechanical and thermal conditions. Unlike Newtonian fluids, whose viscosity remains constant regardless of applied stress, non-Newtonian oils exhibit variable viscosity under shear forces—internal forces generated when layers of fluid slide past one another at different velocities. This behaviour is particularly advantageous in engines, where oil must flow easily at startup temperatures to reduce wear, yet maintain sufficient thickness under high-speed, high-temperature conditions to ensure effective lubrication. The inclusion of viscosity modifiers and synthetic base oils enables these adaptive properties, allowing the oil to thin under high shear (reducing friction) and thicken when needed (enhancing protection).

Figure 2 shows the viscosity of such a fluid as a function of shear for a 5W-50 engine oil fit to the Carreau-Yasuda equation for shear-induced temporary viscosity loss. The change in viscosity is inversely proportional to the temperature of the oil. The graph demonstrates that such an oil is possible based on existing technology.

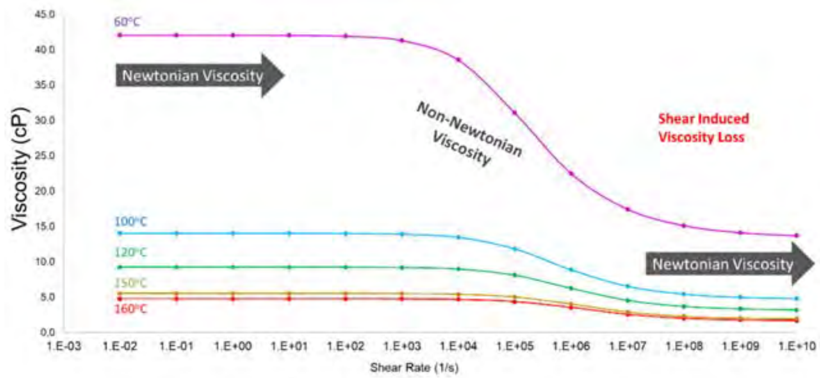

Fig 2: Viscosity of SDNNO as a function of shear. Note that the viscosity drop zone lies between shear rates of 1E3 and 1E8 (1/s). Graph courtesy of ExxonMobil.

The widespread adoption of non-Newtonian engine oils began in earnest during the 1980s, driven by advancements in engine design and increasing demands for fuel efficiency and durability. As automotive technology evolved, so did the need for lubricants capable of performing across a broader range of operating conditions. Multi-grade oils, which often exhibit non-Newtonian behaviour, became standard due to their ability to maintain optimal viscosity across temperature extremes. This shift marked a significant improvement over earlier monograde oils, offering enhanced engine protection, reduced wear, and improved performance. Today, non-Newtonian oils are a cornerstone of engine lubrication, supporting the longevity and efficiency of internal combustion engines in a wide array of vehicles.

However, non-Newtonian oils have not generally been used in power generation applications such as gas and steam turbines. Most turbines operate at constant high speeds and temperatures for extended periods, where consistent long-term viscosity performance is critical to prevent wear and maintain long-term operational reliability. Non-Newtonian oils have historically been disadvantaged in these areas due to lower thermal stability, potential permanent mechanical shear of polymers as well as cost and complexity considerations. However, given recent advancements in polymer science and fluid engineering, coupled with the increasing emphasis on maximizing turbine efficiency, the potential application of non-Newtonian fluids in the power generation sector warrants renewed investigation.

3 Numerical Simulations

To optimize oil formulation for power generation applications, numerical simulations of bearing performance were conducted to better understand the operating conditions the oil experiences in the test setup. A thermo-elasto-hydrodynamic (TEHD) simulation program, THRUST [4], was used for this purpose. Like most bearing simulation tools currently available for modeling tilting pad thrust bearings, THRUST is limited to Newtonian fluid behavior and does not account for non-Newtonian effects such as shear thinning.

Despite this limitation, the simulation provides valuable insights into the range of temperatures, pressures, and shear rates the oil is subjected to—information that is critical for guiding formulation design. Figure 3 presents a representative set of simulation results, illustrating the range of shear rates encountered by the oil in the

Kingsbury test rig. The minimum (orange) and maximum (blue) shear rates within the bearing are estimated based on the corresponding oil film thicknesses and the linear speed of the bearing.

To achieve energy efficiency benefits, the oil must exhibit shear-thinning behavior within this shear rate range. The desired rheological profile is achieved by carefully selecting and blending base oils with appropriate viscosity modifiers.

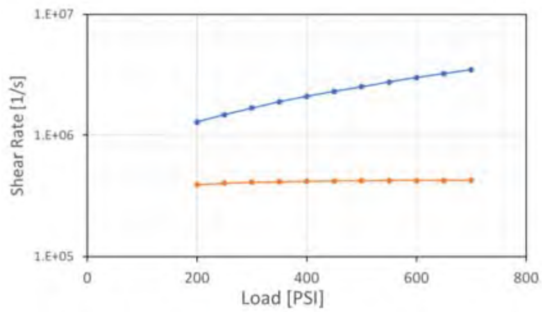

Fig 3: Computed shear rates for the test bearing on the Kingsbury, Inc test rig at anticipated maximum and minimum viscosities. The shear rates fall in the viscosity drop zone for the SDNNO being investigated (see Fig 2).

4 Physical Tests

4.1 Test Rig Description

The test rig used in this study located at Kingsbury, Inc is a vertically oriented machine show in Fig 4 where the bearings sit in a pot of oil. The shaft is driven by a variable-speed, 50 hp electric motor that is coupled to a vertical shaft that has a nonintegral runner. The runner diameter is 10.625 in (269.875 mm) and has a total thickness of 1.751 in (44.48 mm). A plain journal bearing is used at the top end of the runner for radial positioning. A tilting-pad thrust bearing sits on each side of the collar. The lower bearing consists of six tilting pads sitting on hydraulic jacks. These jacks control the load on the upper bearing and act as an equalization mechanism for the lower bearing. The upper bearing is a traditional flooded design with the pads sitting on leveling plates that rest inside a base ring. The shaft rotates in the range of 500-3500 rpm, and bearing loads of up to 2400 psi can be applied. Except for the fact that both thrust bearings are loaded, this configuration is typical of hydroelectric turbines.

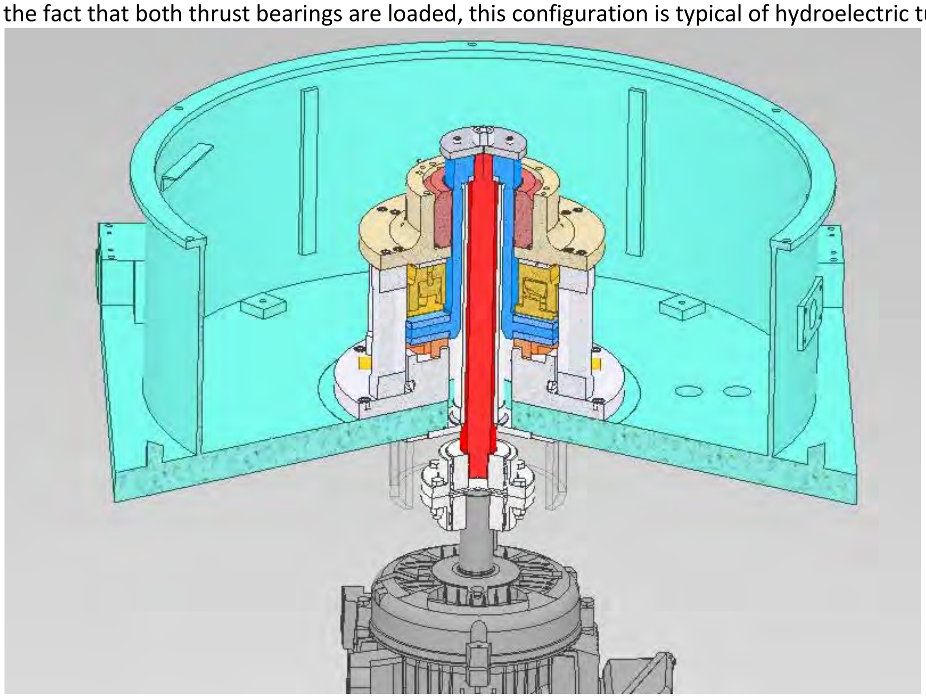

Fig 4: Schematic of the test rig at Kingsbury, Inc used for these tests.

LabVIEW software is utilized monitor the operation of the test rig and control the motor speed. Thermocouples with a range of 24°F - 1400°F (0°C -760°C) and accuracy of 0.4% of reading are used to collect the temperature data, while an Omega PX303 sensor with 0 – 10,000 psi range and 0.25% accuracy is used to measure the pressure in the loading mechanism. The test rig is instrumented with a motor power meter with 65 hp capacity and 0.5% accuracy. Transient and steady-state data logs are stored where the transient log stores the data every second by averaging 120 samples per second and the steady-state log stores 30 second averages of the transient data.

Thermocouples for the babbitted pads are placed at the 75/75 location (75% of the pad arc as measured from the leading edge and 75% of the pad width as measured from the inner radius) just below the surface of the babbitt in the steel body. This type of measurement fails for PEEK-lined shoes because PEEK is a thermo-insulator. Instead, the shoes were instrumented with "bleed holes" machined through the PEEK pad surfaces into a cavity below the hole at the 80/60 location. The idea was to allow hot oil at the surface of the pad to flow, or bleed, into the hole, and that a thermocouple mounted there would measure the oil temperature. There was no separate exit path for the oil. See Gokaltun et al. [5] for further discussion about instrumenting PEEK shoes.

4.2 Test Series

Ten tests were carried out in this campaign, nine of which were steady-state tests. They are described in Tab 1. In all tests, the shoes in the lower bearing were a flooded chrome copper design with a 65% offset. The upper bearing was changed between tests as indicated. Tests A-C were carried out at an earlier time and serve as baseline performance for 50% and 60% offset babbitted and PEEK-lined thrust bearings using oil HO (a commercially available premium Newtonian ISO VG 32 hydraulic oil). Tests for Series A & B were only conducted at 140°F pot temperature only. A series of steady state tests (D-F) were carried out using the SDNNO. Test G was an extended duration test, where the bearing was run continuously for 6-7 hours per day at a speed of 3000 rpm and a bearing load of 500 psi for a total of 276 hours of run time. An oil sample was collected before and after the extended duration test to measure if extended testing resulted in permanent viscosity change. Test H was carried out to compare with Test F (before and after extended duration test). Finally, tests I-J were conducted using a commercially available Newtonian ISO VG 32 turbine oil that serves as a fair comparison for the SDNNO to double check the results.

Test	Oil	Steady-State	Oil Temperature		Test (upper) bearing shoe style		
			140°F	158°F	50% babbitted	60% babbitted	50% PEEK-lined
A	HO	X	X		X		
B	HO	X	X				X
C	HO	X	X	X		X	
D	SDNNO	X	X	X		X	
E	SDNNO	X	X	X	X		
F	SDNNO	X	X	X			X
G	SDNNO		X				X
H	SDNNO	X	X	X			X
I	TO	X	X	X			X
J	TO	X	X	X	X		

Tab 1 – List of tests performed in this study. TO is a commercially available Newtonian ISO VG 32 turbine oil and HO is a commercially available premium Newtonian ISO VG 32 hydraulic oil.

4.3 Test Results

We investigate power loss and pad temperatures as measured on the test rig. We do this by looking at contour maps of these variables as a function of collar speed and bearing load. The colors in Figs. 5-12 are identical: green is a test with the HO, red is a test with the SDNNO, and blue is a test with the TO.

For the 50% offset babbitted bearing, we compare tests A, E, and J in Figs 5 and 6. Note that the test conducted with the SDNNO (Test Series E in red) has a lower power loss than those conducted with other oils. Similarly, the pad temperatures for Series E are lower as well.

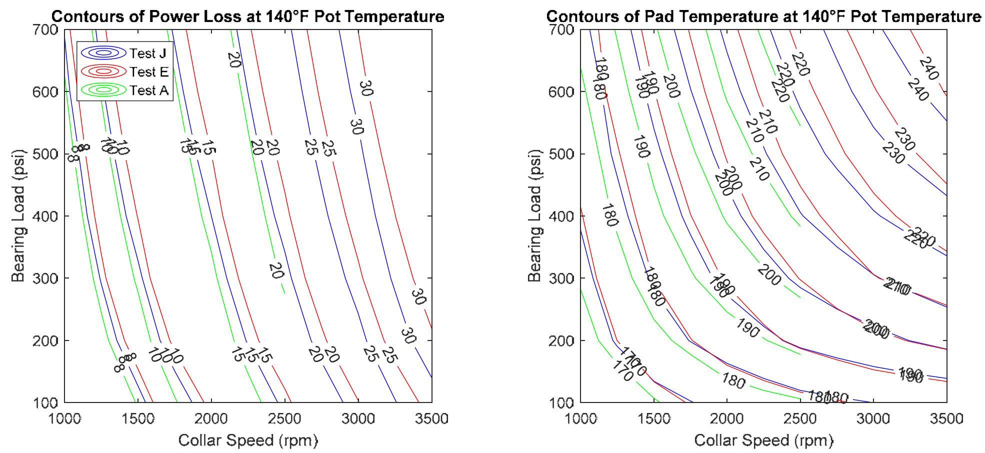

Fig 5: Comparison of bearing performance in tests A, E, and J at 140°F pot temperature.

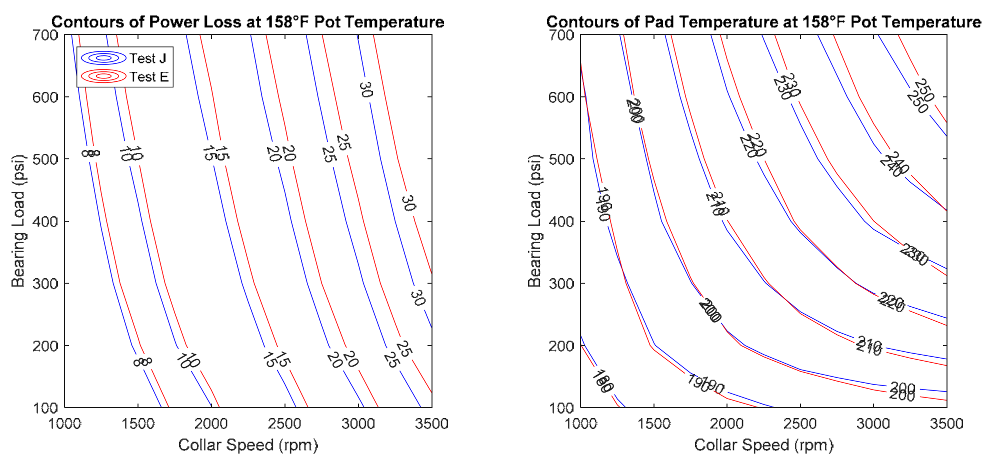

Fig 6: Comparison of bearing performance in tests E and J at 158°F pot temperature.

For the 60% offset babbitted bearing, we compare tests C and D in Figs 7 and 8.

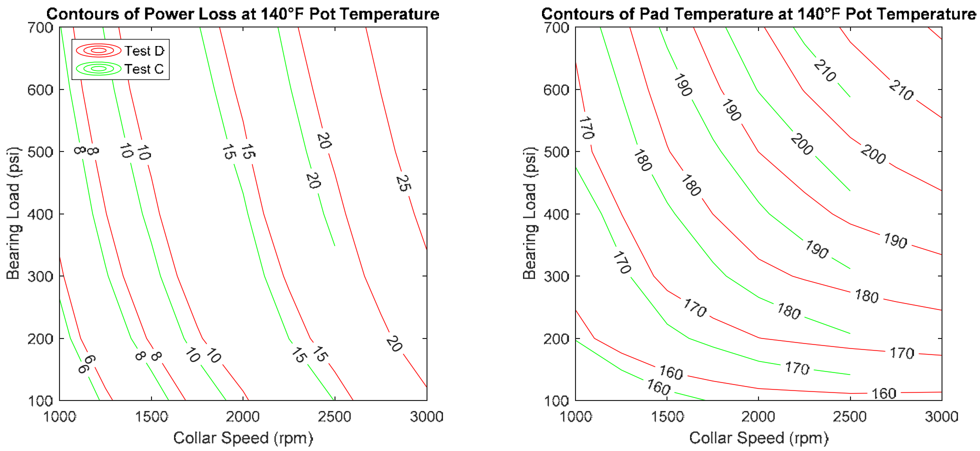

Fig 7: Comparison of bearing performance in tests C and D at 140°F pot temperature.

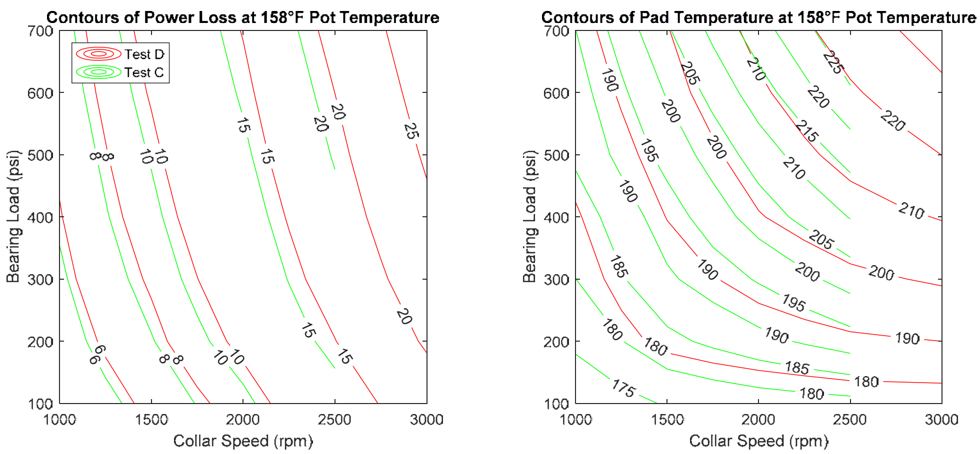

Fig 8: Comparison of bearing performance in tests C and D at 158°F pot temperature.

For the 50% offset PEEK-lined bearing, we compare tests B, F, and I in Figs 9 and 10.

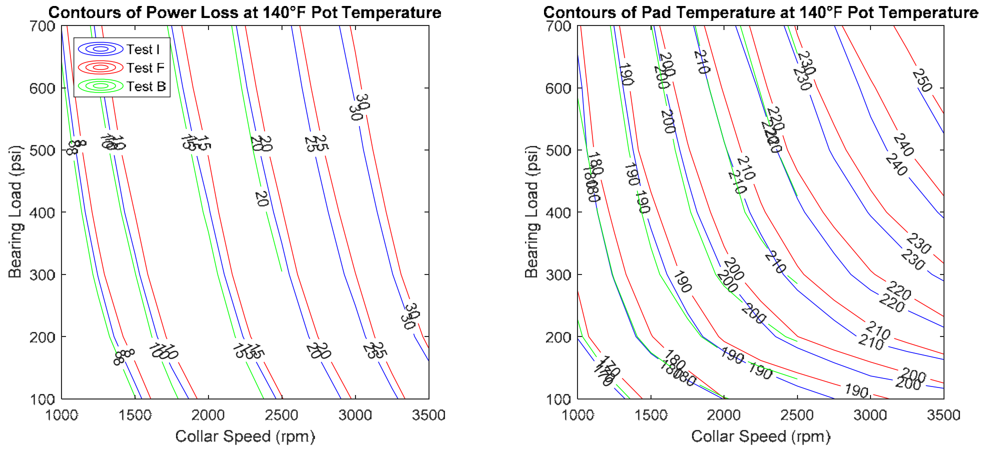

Fig 9: Comparison of bearing performance in tests B, F, and H at 140°F pot temperature.

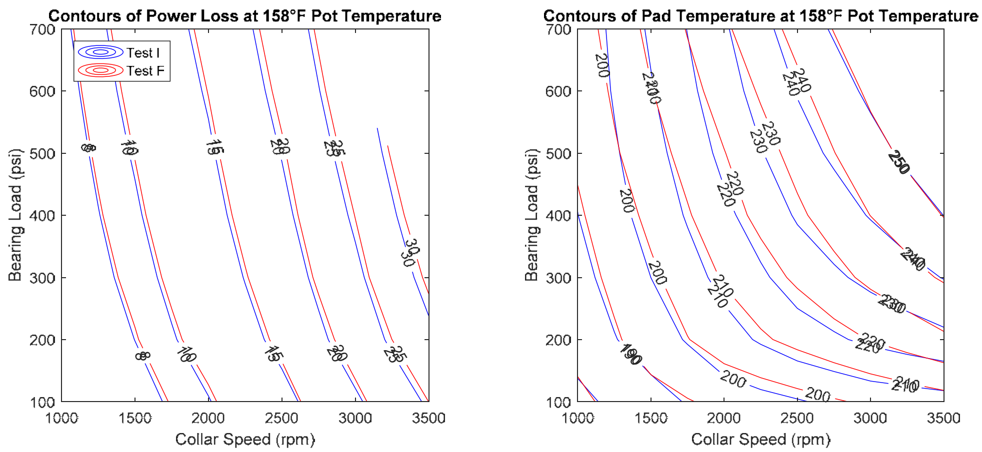

Fig 10: Comparison of bearing performance in tests F and H at 158°F pot temperature.

Our final comparison in Figs 11 & 12 is that of Series F and H which were conducted with the same bearings and oil before and after the extended duration test, respectively. There is no discernible difference in power loss between these two tests, and pad temperatures are within statistical uncertainty. This suggests that the extended duration test (Series G) had no effect on oil viscosity. This agrees with tests of the oil samples.

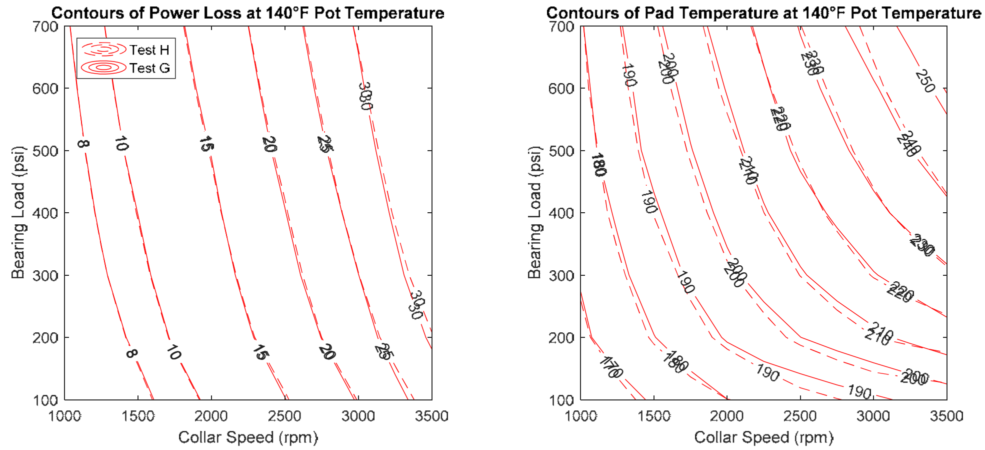

Fig 11: Comparison of test Series G and H at 140°F pot temperature before and after the extended duration test. There is no discernable difference in the output of the tests.

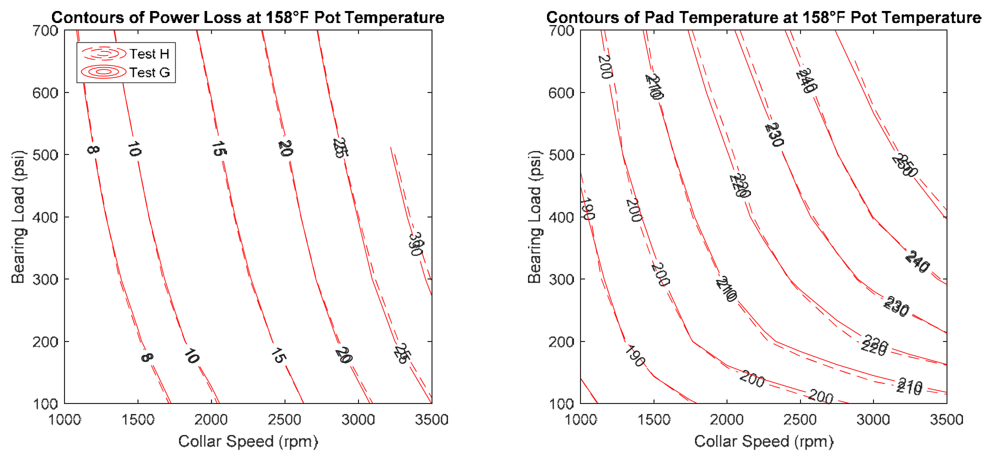

Fig 12: Comparison of test Series G and H at 158°F pot temperature before and after the extended duration test. There is no discernable difference in the output of the tests.

5 Discussion

{The powerpoint for the Low Loss test rig makes the statement "It's the dynamic viscosity (density), not kinematic viscosity (velocity) that determines power loss". I this is a good place to elaborate on this point. That this statement is true for a Newtonian fluid, and that what we did here is show that we can circumvent this statement using a SDNNO. Viscosity in the contact zone determines the power loss, not the bulk viscosity.

Also, we should include here the "novel approaches to lubricant design" as promised in the abstract.} Why this hasn't been done before, what needs to be overcome, advances in polymer design,

6 References

[1] Data Centers fueling gas turbine sales. Gas Turbine World. https://gasturbineworld.com/data-centers-fueling-gas-turbine-sales/; US Department of Energy. https://www.energy.gov/articles/doe-releases-new-report-evaluating-increase-electricity-demand-data-centers

[2] ExxonMobil. (2024). *Global outlook: Our view to 2050*. https://corporate.exxonmobil.com/sustainability-and-reports/global-outlook

[3] ExxonMobil. (2021). Mobil SHC 918 EE Tech Deck [PDF]. Toronto STLE. https://torontostle.com/wp-content/uploads/2021/03/Mobil-SHC-918-EE-tech-deck-final.pdf

[4] Brockett TS, Barrett LE & Allaire PE. (1996) Thermoelastohydrodynamic analysis of fixed geometry thrust bearings including runner deformation. Trib. Trans. 39, pp. 555–562.

[5] Gokaltun S, Fabijonas BR & Rodzwic, RC. (2021) Condition monitoring of PEEK bearings using temperature measurements. Proceedings of the 20th Tribo-Pprime Workshop, Futuroscope, France. 12pp.