Technical Article

Wind Main Bearing Grease Evaluation

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Abstract

This study presents a comprehensive evaluation of six commercially available greases either commonly used or emerging in wind turbine main bearing applications. The selected greases, which vary in formulation and industry adoption, were subjected to a series of fluid characterization and tribological tests to assess their performance under relevant operating conditions. Key metrics of performance included stability, friction coefficient, and wear behavior. Results revealed that each grease exhibited distinct performance trade-offs, with no single formulation outperforming across all criteria. Additionally, the use of diamond-like carbon (DLC) coatings on bearing surfaces did not adversely affect the friction or wear characteristics of most tested greases, suggesting compatibility with surface-engineered components. These findings provide practical insights for grease selection in modern wind turbines.

Wind Turbine Mainshaft Bearings

Wind turbine mainshaft bearings are designed to carry dynamic loads associated with constantly changing wind speeds and directions in a wide variety of environmental conditions. Proper lubrication is critical to optimizing mainshaft bearing performance. The relative motion between the rolling elements and the race during operation entrains the lubricant to generate an oil film that separates the steel surfaces. The oil film thickness must be large enough to prevent contact between the microscopic peaks, or asperities on the two interacting surfaces. A common metric to evaluate the oil film of a given lubricant in a rolling contact is lambda ratio. The lambda ratio of a lubricated contact is the ratio of the lubricant film thickness to the composite surface roughness of the contacting surfaces (Figure 1).

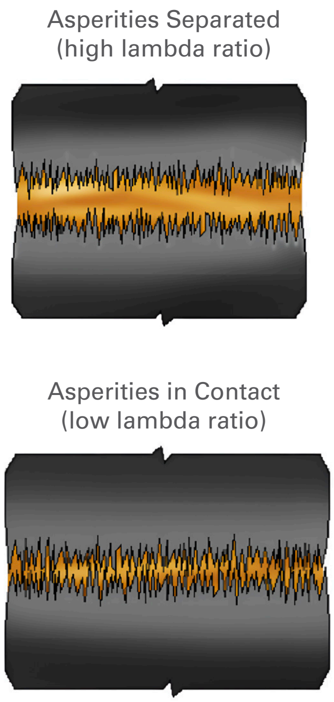

Figure 1. Lambda Ratio Example

$$\text{Lambda Ratio, } \lambda = \frac{\text{Film Thickness}}{\text{Composite Surface Roughness}}$$

A lambda ratio of 1.0 suggests that the oil film present in a contact is at the minimum thickness to span the mean peaks and valleys of the two mating surfaces. Due to the low and variable speeds, it is challenging to achieve a lambda ratio greater than the target minimum of 1.0 lambda ratio. Micropitting (Figure 2), a type of surface wear, is a common life-limiting factor in main bearings and is generally caused due to metal-to-metal contact on the asperity level in conjunction with variable loading and roller slip.

Figure 2. Micropitting on a Spherical Roller Bearing Inner Race

Therefore, as lambda ratio increases and surface interactions decrease, the risk of micropitting is also expected to decrease. Proper lubricant selection and regular relubrication intervals for mainshaft bearings is critical to mitigating surface-initiated damage like micropitting.

Composition of Grease

Grease is composed of a thickener uniformly distributed into a base oil. Additives are incorporated into the greases to improve performance in demanding applications. Unlike oil, which is fluid and flows easily through bearings, grease is a semi-solid and will not readily migrate back into the roller path once it is pushed away. This can lead to less entrained grease in the roller contact resulting in a lower film thickness.

Consistency and NLGI Grade

A grease's NLGI grade is a measure of the relative stiffness or consistency of a grease and affects grease migration in the bearing. For example, an NLGI Grade 2 grease may have the consistency similar to that of peanut butter, whereas a Grade 0 may be more like mustard. Grease formulations for wind mainshaft applications vary greatly. Most wind mainshaft bearing greases tend to use lithium-based thickeners while others could use a calcium sulfonate thickener.

Base Oil Type and Additive Packages

Along with viscosity, base oil type can also vary amongst greases. Many greases use synthetic base oils, but mineral oil or synthetic-mineral blends are also common. The base oil viscosity also varies greatly across common mainshaft greases, ranging from 130 to 680 cSt at 40°C. The oil separation, or the percentage of a grease's base oil that separates from its thickener, also varies greatly, from 1% to 5.5%. Further, there is a wide array of additive packages used in the greases, which makes it even more challenging to evaluate which grease is best for wind turbine mainshaft bearings. Under operation, oil bleeds from the thickener to lubricate the bearing, which will eventually lead to degradation of the grease's lubrication properties. Therefore, it is important to regularly relubricate the bearings to replenish the oil and maintain the proper consistency. In wind turbines, relubrication is typically performed manually every six months during regular maintenance or more regularly with an automatic lubrication system.

Engineered Coatings for Wear Mitigation

In addition to lubrication, some wind turbine main bearings may utilize engineered coatings to improve roller-race wear performance. Diamond-like carbon (DLC) coatings have been used in the wind industry to mitigate micropitting damage caused by thin oil films. Most commonly applied to spherical roller bearing (SRB) rolling elements, this coating helps to mitigate adhesive wear and progressive debris damage, and is intended to improve fatigue life.

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Previous Studies

In 2013, Dave Pierman published a paper in which he evaluated more than 10 typical wind turbine mainshaft greases [2]. He summarized the importance of specific tests and concluded that lubricant film thickness, oil separation, and bearing wear behavior are three of the most critical grease characteristics, regardless of the wind turbine environment. Obviously, low-temperature torque and grease pumpability are important tests to consider if turbines operate in cold weather conditions (-40°C). Bearing operating temperature is an important measurement that can be used to evaluate the oxidation rate and expected life of the grease. Corrosion protection is required in offshore applications and fretting protection is required to prevent fretting wear and false brinnelling on the raceways during oscillation events.

Kuldeep Mistry's 2019 paper [3] further evaluated the film thickness, traction coefficients, and bearing temperature, torque and grease migration for several of the greases from the Pierman work. His work confirmed that the greases with higher base oil viscosities resulted in larger film thicknesses. However, these greases also have higher traction coefficients and operating temperatures.

Testing also showed that the greases with lower base oil viscosity performed much better on the grease migration tests, allowing the grease to flow back into the bearing contact surfaces. This is expected to provide more effective lubrication at the contact surfaces. Additionally, although the lower-viscosity grease had a slightly higher traction coefficient, the bearing torque and temperature measured were significantly lower (35% less torque and 15% lower temperature (-10°C)). This confirms that a lower base oil-viscosity grease with the right chemistry can properly lubricate a bearing, regardless of the potentially lower calculated lambda ratio associated with the lower base oil viscosity.

All the properties discussed here are important to understand a suitable grease, but they cannot be evaluated during wind turbine operation to judge the lubrication effectiveness. Grease samples should be taken at regular intervals to evaluate the change in its properties over time. Bearing temperature is one of the simplest measurements on a wind turbine that can be related to bearing performance. Bearing torque and temperature increase with friction on a damaged or poorly lubricated bearing. It is a generally accepted guideline that approximately every 15°C temperature increase will double the oxidation rate of a lubricant, resulting in shorter (half) grease life.

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Test Grease Properties

As an extension of previous studies, six wind main bearing greases were selected for evaluation. These greases vary in formulation and popularity among the industry. The evaluated grease properties are highlighted in table 1.

Table 1. Grease Properties

Shaft Size	Grease A	Grease B	Grease C	Grease D	Grease E	Grease F
Viscosity at 40˚C	460	680	680	400	300	460
Base Oil Type	Synthetic	Synthetic	Synthetic	Synthetic	Synthetic	Synthetic
NLGI Grade	1.5	1.5	1	1.5	1	1
Thickener	Li Complex	Li Complex	Li Complex	CaS	Lithium Soap	Li Complex

Test Plan

The goal of this project was to evaluate existing and emerging mainshaft bearing greases' lubricant characteristics and tribological performance. The selected tests are described in table 2.

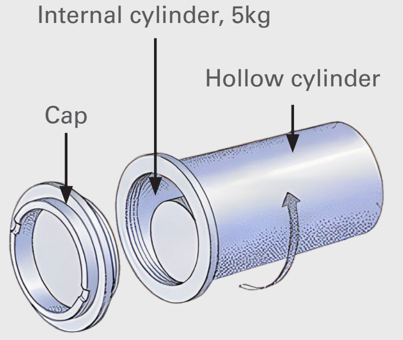
Table 2. Test Plan

Test	Grease F
Shear Roll Stability	ASTM 1831
Worked Penetration	ASTM D217A
Oil Separation	ASTM D6184
Four Ball Weld	ASTM D2596
Four Ball Wear	ASTM D2266
Fretting Wear Protection (DLC/Non-DLC)	ASTM D5707
Grease Migration	-
Traction Test	-
Optical Film Thickness	-

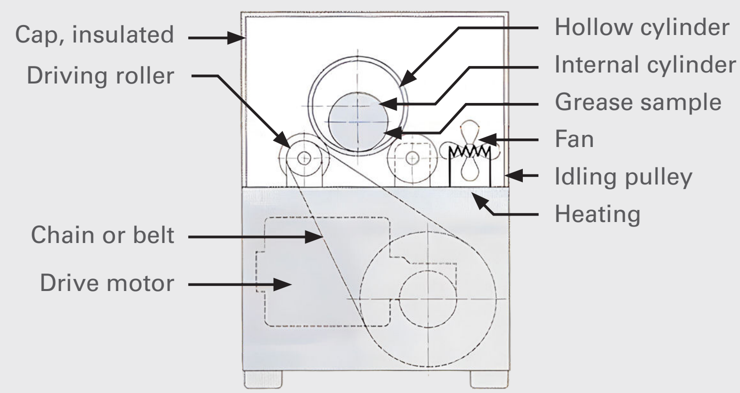
Shear Roll Stability

Shear roll stability is a common lubricant screening test to evaluate a grease's ability to withstand shear without mechanical degradation. Figure 3 shows an example of a roll stability test rig. The steel cylinders are filled with grease and rotated at a set speed to simulate a sliding contact which subjects the grease to shear. Performance in this test is measured based on the change in grease consistency before and after testing.

ROLL STABILITY TESTER



Specimen for roll stability tester



Test rig drawing

Figure 3. Shear Stability Test Schematic

For this evaluation, the test was run for 50 hours at a temperature of 65 C to mimic typical main bearing temperatures. Penetration, or consistency, values are measured using a penetrometer to determine the change in consistency before and after testing. Results of this test are recorded in table 3.

Table 3. Shear Roll Stability

	Unworked Penetration (NLGI)	Worked, 60x Penetration (NLGI)	Roll Stability [50 hrs] Penetration (NLGI)	Δ Penetration
Grease A	291.5 (2)	297.8 (2)	341.2 (1)	50
Grease B	269.6 (2)	282.2 (2)	311.8 (1)	42
Grease C	329.4 (1)	331.9 (1)	362.5 (0)	33
Grease D	286 (2)	295 (2)	302 (1)	16
Grease E	301 (1)	314 (1)	357 (0)	56
Grease F	290.5 (2)	307 (1)	356 (0)	49

In reviewing the change in penetration for each grease, there is minimal difference in the degree of softening indicated by the change in penetration values. However, Grease D experienced notably less softening compared to the other five greases in this study.

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Worked Penetration

In addition to shear roll stability, a second test is completed to measure resilience against mechanical working. In this worked penetration test, a grease undergoes 100,000 cycles using a reciprocating plunger (piston cylinder) with holes at room temperature. Like roll stability tests, this test is also used to measure the change in grease consistency but under different loading conditions. Table 4 summarizes the results of this test where the key metric once again is the change in penetration before and after testing.

Table 4. Worked Penetration

	Unworked Penetration (NLGI)	Worked, 60x Penetration (NLGI)	Δ Penetration
Grease A	291.5 (2)	313 (1)	22
Grease B	269.6 (2)	310 (1)	41
Grease C	329.4 (1)	344 (1)	15
Grease D	286 (2)	313 (1)	27
Grease E	301 (1)	363 (0)	62
Grease F	293 (1)	344 (1)	51

Results from this test show similar trends to that of roll stability. Grease E yields the highest change in penetration while Greases A and C show the least change in penetration.

Oil Separation

Figure 4. Oil Separation Test

In normal operation, the intent of grease is that the base oil bleeds from the thickener matrix to form an oil film to protect metal surfaces. One method of quantifying a grease's tendency to bleed oil is through an oil separation test. In this test, a set volume of grease is placed in a conical sieve which is then placed within a beaker. The grease dwells at 100˚ C which forces oil to weep from the thickener and through the sieve (figure 4). The measured oil weight collected in the beaker is then used to calculate the oil separation.

Timken's suggested oil bleed rate for main bearings is 0.5%-6.0%. Because oil bleed relates to how much oil separates from the thickener during operation, too low of a bleed rate could indicate that the grease does not release enough oil to form an oil film to separate metal surfaces. On the other hand, a grease with too high of a bleed rate could suggest that the grease releases too much oil such that it has difficulty returning to the thickener matrix leading to permanent separation and short grease life. Results of this test are shown in table 5.

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Table 5. Oil Separation

Sample	Bleed Rate
Grease A	1.62%
Grease B	1.85%
Grease C	4.41%
Grease D	1.60%
Grease E	9.40%
Grease F	4.07%

Of the six tested greases, all but Grease E yield a bleed rate within Timken's recommendation. Given a higher bleed rate, Grease E could benefit from more frequent relubrication cycles to replenish any oil that may permanently bleed from the thickener in operation.

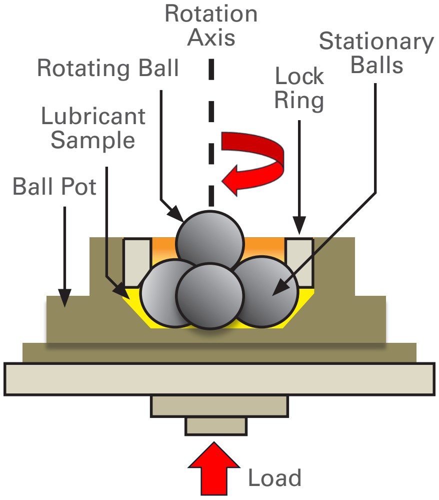

Figure 5. Four Ball Test

Four Ball Wear Tests

Wear performance and friction management are critical performance metrics for lubricating greases. Four Ball Weld and Four Ball Wear are two popular tests to measure a lubricant's ability to resist wear. The Four Ball Weld test is geared toward evaluating a lubricant's ability to mitigate adhesive wear. In this test, the driven ball operates at a set speed, but load is increased until ball seizure is achieved. The Four Ball Wear test is used as a general sliding wear test where the driven ball rotates at a constant load and speed. Both tests utilize similar instrumentation seen in figure 5. Three steel balls are stationary with the fourth ball being spun against the other balls under an axial load. The balls are submerged in the test lubricant of choice.

This test measures wear scar area and operating friction for a given grease. Results for Four Ball Weld and Four Ball Wear tests are reported in figures 6 and 7.

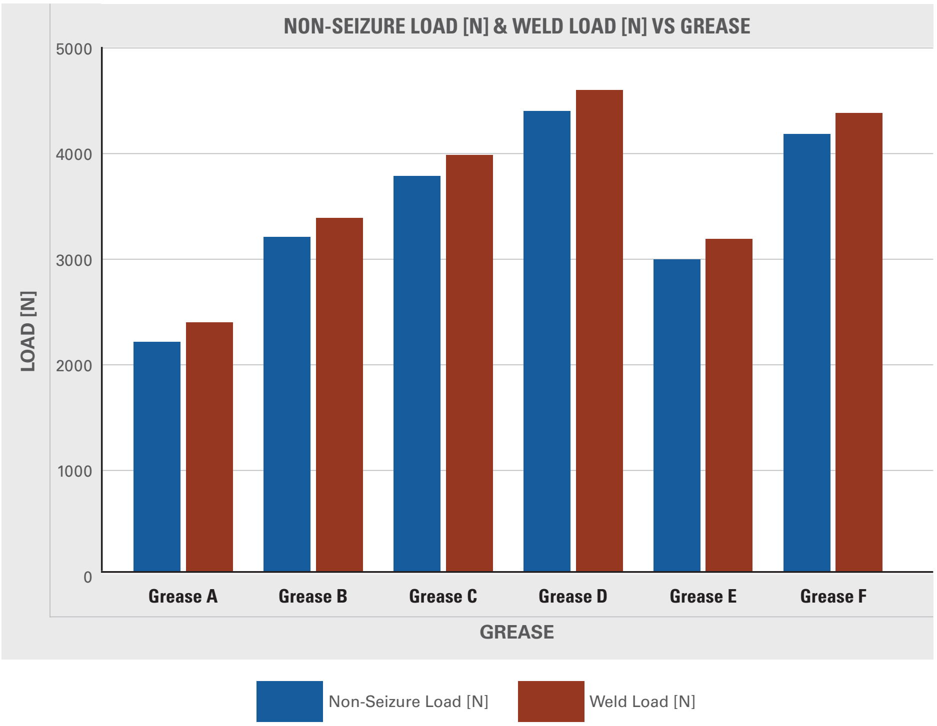

Figure 6. Four Ball Weld Test Results

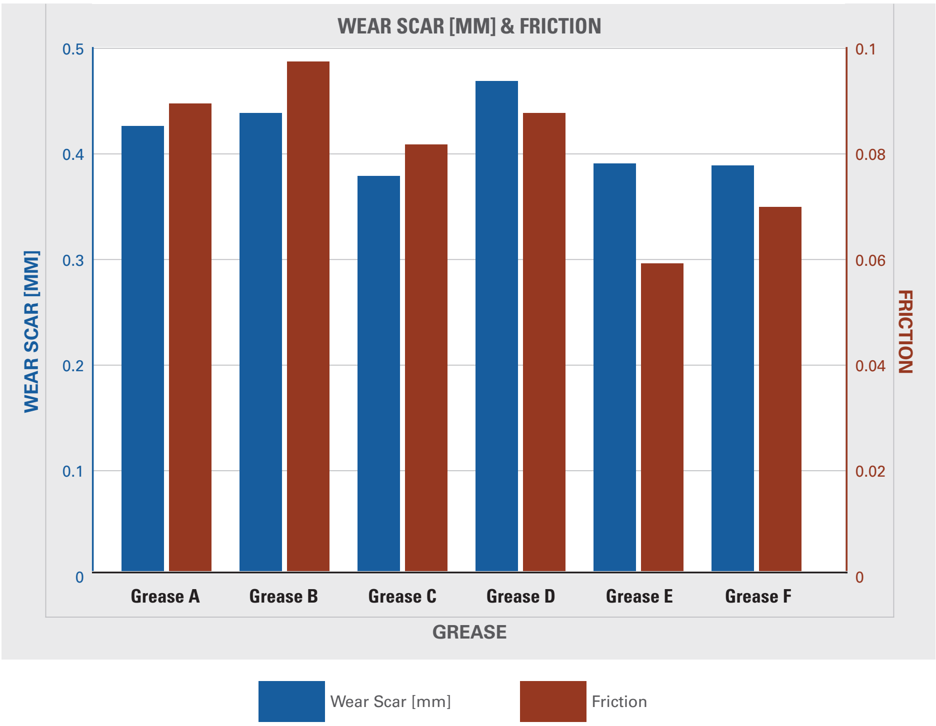

Figure 7. Four Ball Wear Test Results

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The recorded results for the Four Ball Weld tests are the weld load and non-seizure load. The weld load is the load at which ball seizure occurs while the non-seizure load is the highest load the lubricant can support without seizure. Four Ball Wear measures the post-test wear scar as well as the average operating friction. Greases D and F support the highest loads without seizure in the Four Ball Weld test. Greases F, C, and E perform well in the Four Ball Wear test in terms of wear scar with Grease E showing the lowest friction among tested greases.

Fretting Wear Protection

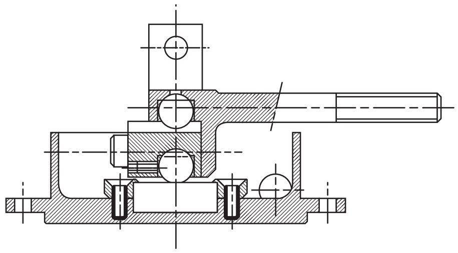

Figure 8. Ball on Disc Test Schematic

Given the variable, low speeds and frequent starts and stops experienced by wind turbines, main bearings are likely to operate with very thin oil films. Additionally, main bearings may experience axial vibrations due to variable axial loading from changing wind conditions and variable pitch control. A fretting wear protection test per ASTM D5707 is used to measure a grease's ability to protect metal surfaces under thin film sliding conditions. This test is a ball on disc oscillating test where an oscillating steel ball is loaded against a plate to measure friction and wear (figure 8).

Because main bearings often utilize DLC coatings, this test was completed with both 52100 ball and disc pairs as well as a 52100 ball against 52100 discs coated with DLC coating. DLC coating is a tungsten carbide/amorphous hydrocarbon wear resistant coating which has been used in several applications including wind turbine main bearings. Results for both Non-DLC and DLC tests are summarized in table 6 along with images of wear scars in figure 9.

Table 6. Fretting Wear Protection

Non-DLC				DLC			
	Normalized Wear Scar Area	Mean Friction	Mean Potential		Normalized Wear Scar Area	Mean Friction	Mean Potential
Grease A	3.21	0.094	4.63 mV	**Grease A**	1.20	0.085	52.87 mV
Grease B	4.22	0.093	0.17 mV	**Grease B**	1.24	0.093	52.05 mV
Grease C	1.20	0.087	48.51 mV	**Grease C**	1.33	0.085	47.57 mV
Grease D	1.44	0.103	48.95 mV	**Grease D**	1.47	0.101	53.00 mV
Grease E	1.38	0.091	2.32 mV	**Grease E**	2.20	0.093	0.61 mV
Grease F	3.86	0.098	3.67 mV	**Grease F**	1.46	0.091	37.25 mV

Figure 9. Fretting Wear Scars Non-DLC (left) DLC (right)

The results provided from this test are the normalized wear scar area, the average operating friction, and the contact potential for each grease. The wear scar is normalized by the hertzian contact area between the ball and the disc. Friction and wear are the two main results of interest while contact potential is a reference value only and simply provides more insight to the surface interaction. A low mean potential may indicate more metal-to-metal contact whereas a higher mean potential may indicate that the metal surfaces are separated to a greater extent by an oil film or tribofilm formation. Grease C performs the best in steel-on-steel tests with the lowest wear scar area and mean friction. Greases A, B, and F perform notably worse than other samples in terms of friction and wear. Aside from Grease E, tests conducted using DLC coated discs show less variation than steel-on-steel tests which may suggest that the DLC coating dominates the tribological performance of this test. Additionally, changes in friction between steel-on-steel and DLC tests are minimal in most cases which suggests that DLC coating does not have a negative effect on friction.

Optical Film Thickness

Optical film thickness tests utilize optical interferometry to measure oil film thickness. Film thickness is measured by passing light through disc of a visibly transparent material which is coated with a semi-reflective layer. This disc material is commonly glass or sapphire. A portion of the light passes through the semi-reflective layer and travels through the oil film and is reflected by a steel ball or roller and the film thickness is then calculated from the difference in phase between the light reflected by the semi-reflective layer and the light reflected from the rolling element (figure 10).

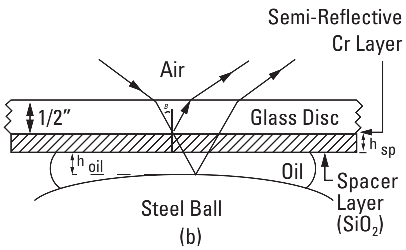

Figure 10. Optical Film Thickness Schematic

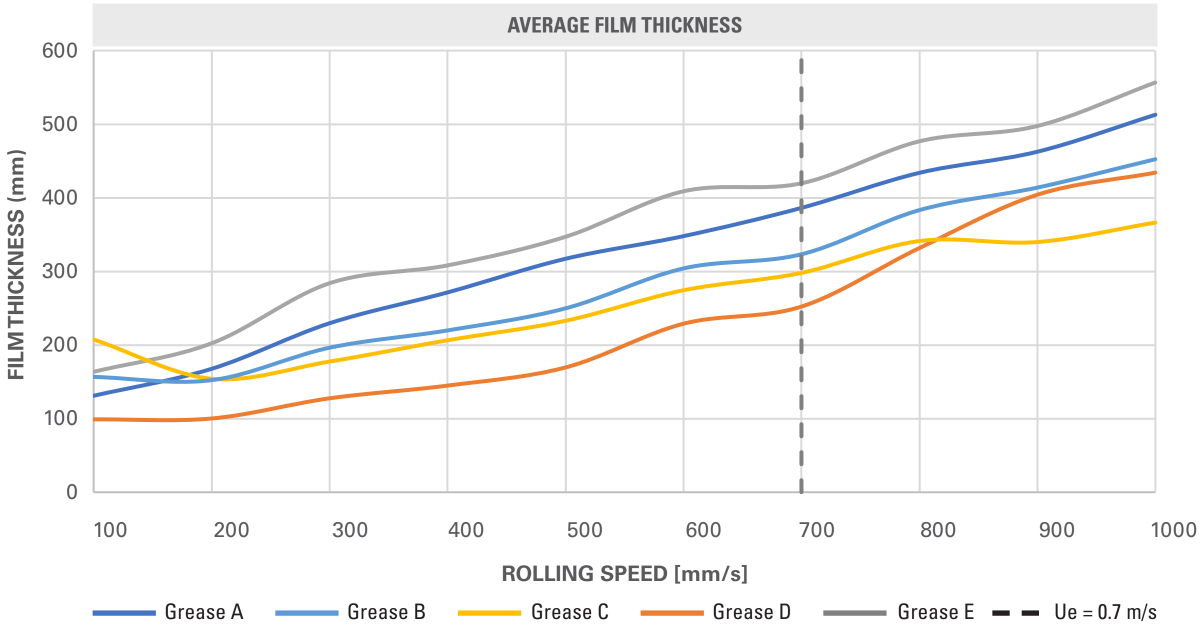

Figure 11. Optical Film Thickness Results

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From the test results, Grease B shows the lowest average film thickness at a nominal main bearing operating speed. Note that Grease B uses a 680 cSt base oil which is the heaviest base oil among tested samples. One potential reason for this unexpected result is thickener interaction. Because grease is a mixture of both base oil and thickener, it is possible to have thickener present within the oil film. Colorimetry of several greases provide insight into thickener interactions which could inflate film thickness measurements. For sake of comparison, colorimetry of Greases B and C as well as oil are shown in figure 12.

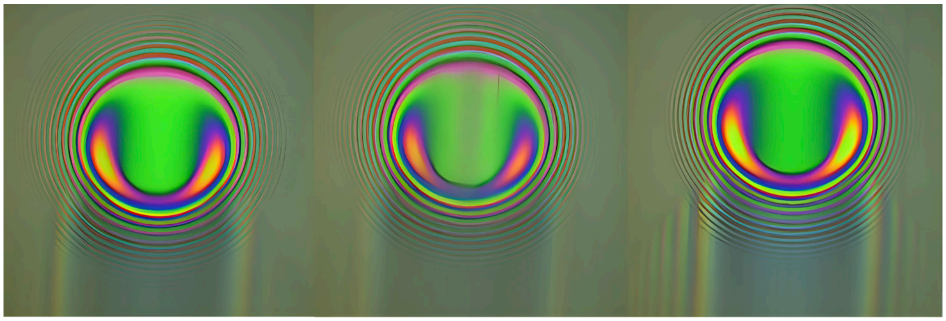

Figure 12. Colorimetry of Grease B (left), Grease C (center), and oil (right)

Lubricated roller contacts should yield colorimetry that resemble a horseshoe as seen with oil in figure 12. In the case of Grease C, smearing at the center of the contact indicates that there may be some thickener interaction or discontinuity in the oil film. By contrast, Grease B does not show the same degree of smearing as Grease C which suggests that there may be less thickener interaction in the rolling contact.

Grease Migration

Main bearings are often only relubricated on a six-month maintenance schedule. Therefore, it is critical that grease can flow within race-roller contacts and remain within the bearing assembly. To simulate grease migration and measure torque and temperature, the lubricant evaluation machine (LEM) was used to evaluate each lubricant. The test configuration is shown in figure 13.

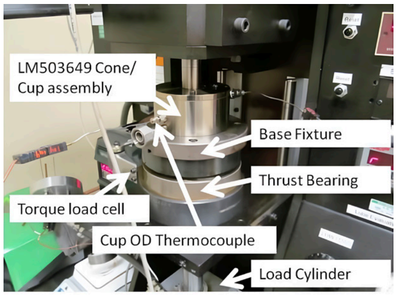

Figure 13. LEM Configuration

The test procedure involved weighing a tapered roller bearing (TRB) cone packed with grease and rotating the TRB for 12 hours at 300 RPM with an applied axial load of 1200 lbs. Operating torque and temperature are also measured for the duration of the test. Once the test is complete, the TRB cone is weighed again, and the post-test mass loss is assumed to be grease which has migrated out of the bearing assembly. Figure 14 shows percent grease loss while the chart in figure 15 shows the torque and temperature data. Note that Grease F was not available for this test and torque and temperature data is not available for grease D.

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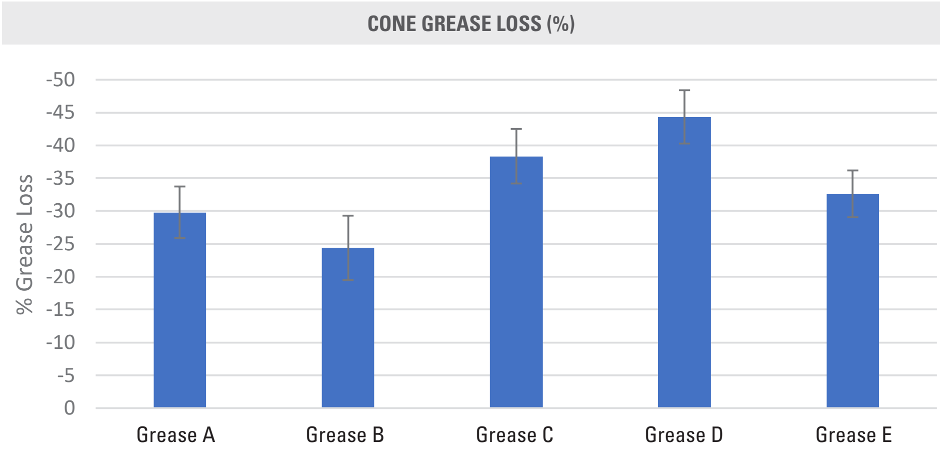

Figure 14. Percent Grease Loss

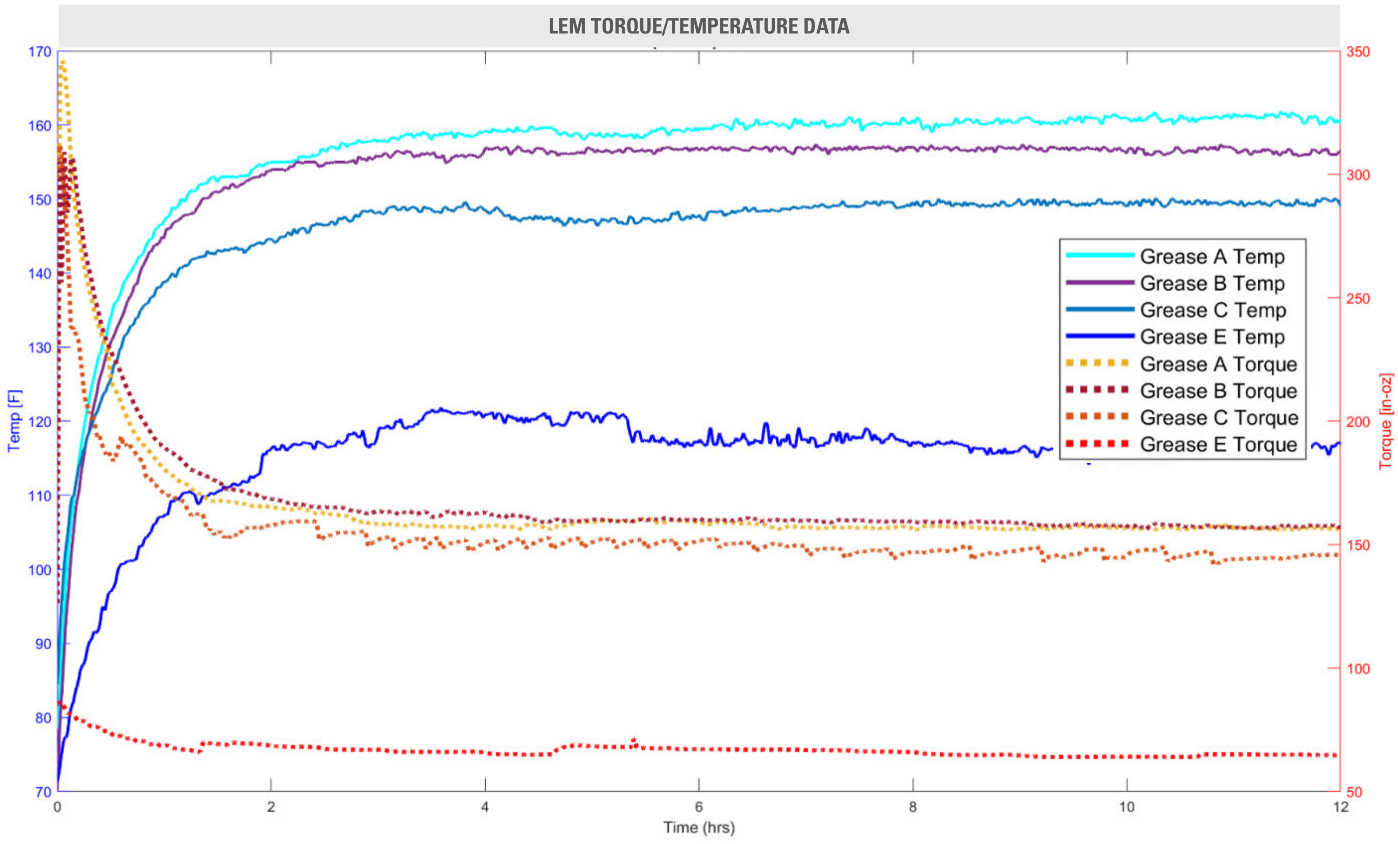

Figure 15. Operating Torque and Temperature

Figure 16. Post-Grease Migration Image of Grease B (top) and Grease C (bottom)

Grease B displays visible grease on the cage surface compared to Grease C which has flowed toward the large rib of the cone and away from the rolling elements. This could be caused by a difference in NLGI grade between grease B (NLGI 2) and grease C (NLGI 1) or tackiness which was not measured as part of this study.

Grease E yielded the lowest operating torque and temperature. Grease A, which aside from Grease E has the lowest base oil viscosity, yields the highest torque and temperature among measured samples. Based on the operating torque and temperature, there are no appreciable differences in required time to achieve equilibrium within the tested samples.

Stribeck Test

Friction at a lubricated contact is a function of speed, load, lubricant viscosity, and surface roughness. A Stribeck curve is commonly used to relate friction coefficient and lambda ratio for a lubricated contact (figure 17). In addition, this curve is also separated into three sections: boundary lubrication, mixed lubrication, and full film lubrication regimes. In boundary lubrication, film thickness is measured on the order of nanometers and friction is high due to asperity shear between contacting surfaces. As film thickness increases, mixed lubrication is achieved which is made apparent by a decrease in friction coefficient. While asperity contact may still occur, it is less likely than in the boundary regime. Full film lubrication suggests that contacting surfaces are entirely separated and the friction is caused primarily due to viscous losses within the oil film.

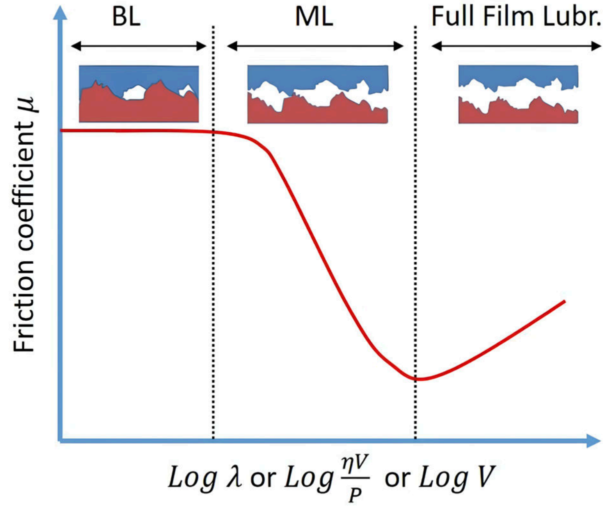

Figure 17. Example Stribeck Curve

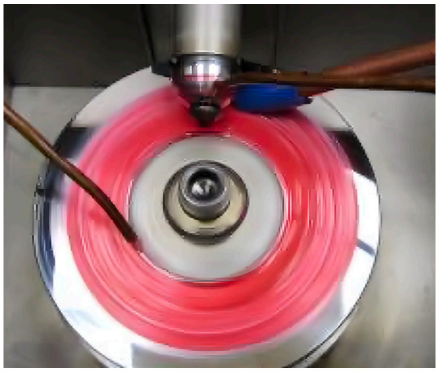

Figure 18. WAM6 Set-Up for Stribeck Test

The Stribeck test was included in this test to determine if there is an appreciable difference in friction when comparing the selected sample greases. This test was completed using a WAM6 ball on disc test machine (figure 18). Additionally, this test was completed using a steel-on-steel configuration as well as a steel ball on a DLC disc.

This test was run at a constant load with speeds varying from 0.1 to 1 m/s.

The surface roughness, load, and lubricant parameters for each test are held constant. Therefore, as speed increases, it would be expected that friction decreases as higher entrainment velocities drive thicker oil films and separate surface asperities. The results for both steel-on-steel and DLC tests can be seen in figures 19 and 20.

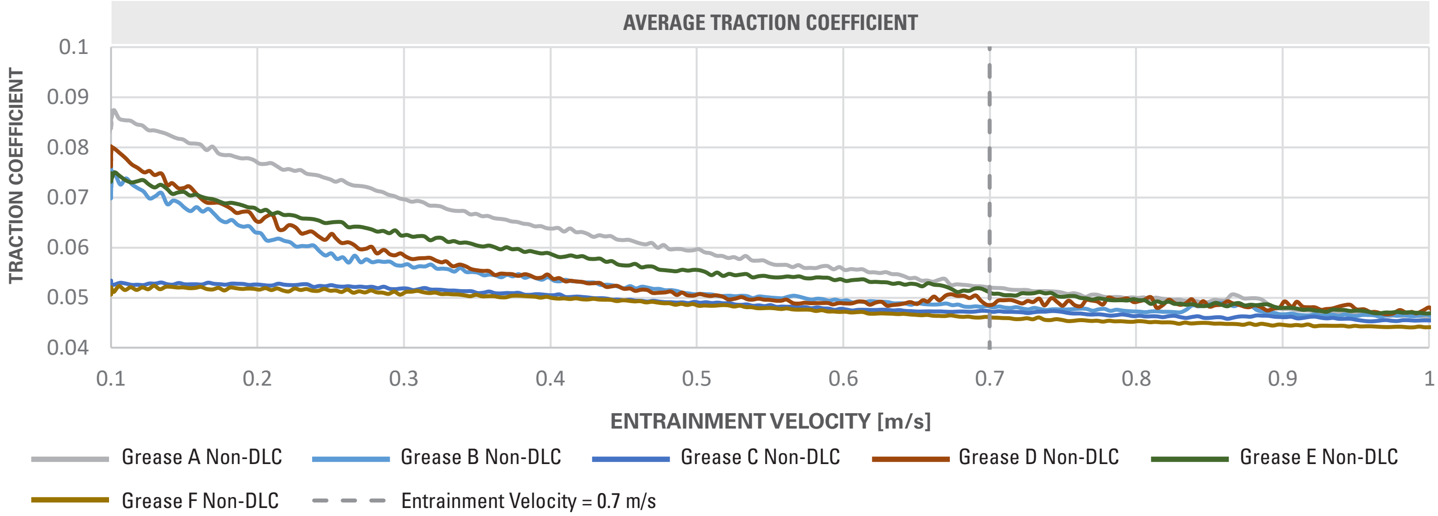

Figure 19. Steel-on-Steel Traction Results

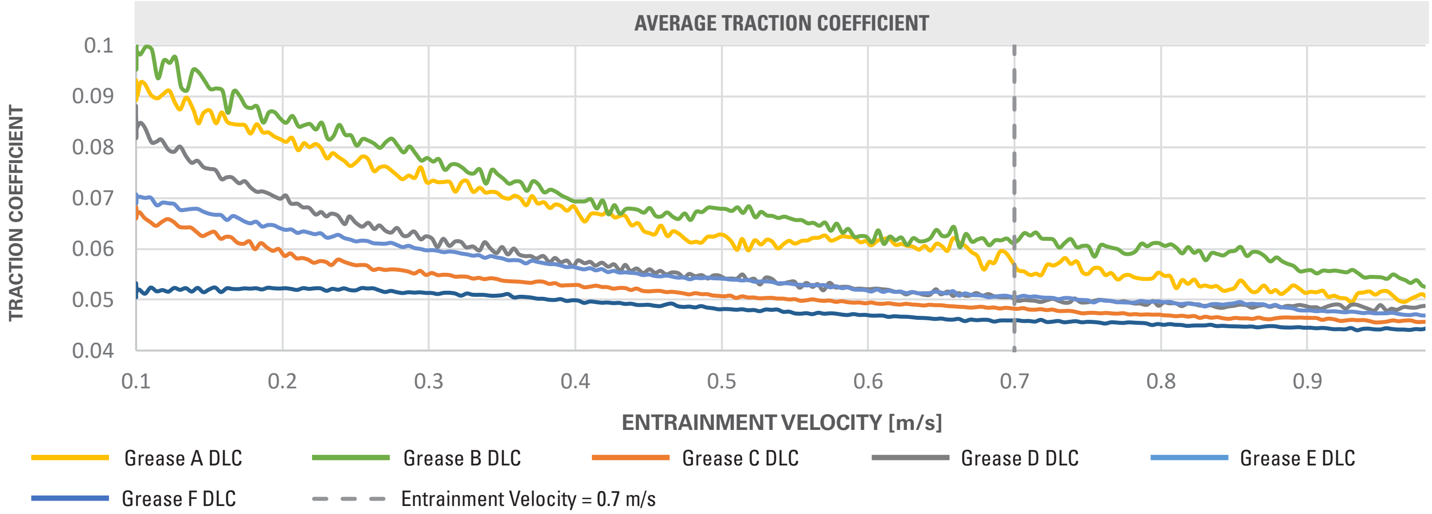

Figure 20. DLC Traction Results

The results of these traction tests suggest that DLC coatings do not decrease traction at a simulated race-roller contact. Additionally, Greases C and F have the lowest traction coefficients across all speeds compared to Greases A, B, and D which are among the highest friction greases for a given speed. The nominal wind mainshaft entrainment velocity of 0.7 m/s is included for reference.

Summary

Six common or emerging mainshaft greases were selected for this evaluation to better understand their performance characteristics. Each sample was evaluated based on its lubricant characteristics, wear protection, and friction management. DLC coated test samples were included for comparison for relevant tests. The results summary can be seen in table 7.

Table 7. Results Summary Table

Test	REF/Standard	Results Value	Grease A	Grease B	Grease C	Grease D	Grease E	Grease F
Shear Roll Stability	ASTM 1831	ΔPenetration	50	42	33	16	56	49
Oil Separation	DIN 51-817	Bleed Oil Sepr.	1.62%	1.85%	4.41%	1.60%	9.40%	4.07%
Shear Stability	ASTM D217A	ΔPenetration	22	41	15	27	62	51
Fretting Wear Protection	ASTM D5707	Normalized Wear Scar Area	3.21	4.22	1.20	1.44	1.38	3.86
		Mean Friction	0.094	0.093	0.087	0.103	0.0913	0.098
		Mean Potential*	4.63 mV	0.17 mV	48.51 mV	48.95 mV	2.32 mV	3.67 mV
Fretting Wear Protection (DLC)	ASTM D5707	Normalized Wear Scar Area	1.2	1.24	1.33	1.47	2.2	1.46
		Mean Friction	0.085	0.093	0.085	0.101	0.093	0.091
		Mean Potential*	52.87 mV	52.05 mV	47.57 mV	53 mV	0.61 mV	37.25 mV
Optical Film Thickness	None	Film Thickness @ 0.7 m/s	386 nm	252 nm	419 nm	323 nm	298 nm	-
Stribeck Traction Test	None	Traction Coef @ 0.7 m/s	0.052	0.048	0.047	0.049	0.046	0.051
Stribeck Traction Test (DLC)	None	Traction Coef @ 0.7 m/s	0.056	0.062	0.048	0.05	0.046	0.051
Grease Migration	None	Average %wt loss	-29.8%	-24.4%	-38.4%	-44.3%	-32.6%	-
Four Ball Weld	ASTM D2596	Non-Seizure Load	2200 N	3200 N	3800 N	4400 N	3000 N	4200 N
		Weld Load	2400 N	3400 N	4000 N	4600 N	3200 N	4400 N
Four Ball Wear	ASTM D2266	Wear Scar	0.43 mm	0.44 mm	0.38 mm	0.47 mm	0.39 mm	0.39 mm
		Friction	0.09	0.098	0.082	0.088	0.059	0.070

*mean potential is a reference value only

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Based on results from this set of tests, each grease has its own unique strengths and weaknesses. Grease C performed great in wear performance and friction management while showing notably high leakage in the grease migration test which may suggest that sealing is important when using this grease. Similarly, grease E showed the lowest friction in some tests but struggled in mechanical degradation tests. Grease E also had a notably higher bleed rate compared to other greases which may require more frequent relubrication if field issues are present. Due to the unique performance of each grease, it is important to review any potential field issues in order to make the best decision for which lubricant is best for a given fleet of turbines and their main bearings.

References

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