FULL LIFE WIND TURBINE GEARBOX LUBRICATING FLUIDS

DOE AWARD # DE-EE0001364

Recipient: **Dow Corning Corporation**

Teaming Members: **Clipper Windpower Technology, Inc**
University of Dayton Research Institute

FINAL REPORT

Submitted by Dow Corning Corporation to United States Dept. of Energy (DOE)

Glenn A. Lutz – Technical Project Director
Manfred Jungk – Principle Investigator
Becky Lauer – Government Contract Program Manager

Authorized Distribution Limitations: None

Patent Restrictions: None

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2. PROJECT BACKGROUND & SUBMISSION

Dow Corning has had a long history of lubricants innovation under the Molykote branded lubricants product lines. In 2005, Dow Corning actively started to work in the wind energy market by developing a series of solutions for both the pitch and yaw bearing systems. It was during these early phase industry lubrication developments that Dow Corning's technical professionals became aware of reliability issues in the gearbox. It became clear that as an industry, the wind energy supply chain was very much focused on data generation and predictive maintenance models, rather than an understanding of root cause analysis, an understanding of loads, and a total system approach of engineering design integrated with the science of tribology. With that industry analysis in mind, and a strong track record in innovation, Dow Corning started initial internal R&D studies into an alternative lubricating fluid to the well established synthetic oils. However it was clear that without industry partnership it was going to be very difficult to get traction on establishing a new lubricating fluid for a wind turbine gearbox. Whilst Dow Corning has the lubrication and tribology expertise, it was critical to establish a partnership with gearbox/engineering expertise, including the capability to generate gearbox test data.

In order to establish this partnership Dow Corning looked at the wind energy OEM supply chain with a view to understanding OEMs that had integrated gearbox supply chain capability. The project team established that Clipper Windpower Technologies was an excellent candidate partner and the team approached Clipper prior to the DOE project proposal submission to outline the concept around a "Full Life Lubricant for Wind Turbine Gearboxes". Clipper bought into the concept and agreed to our proposal for a joint submission according to FOA Number: DE-PS36-09GO99009 "20% Wind by 2030: Overcoming the Challenges" (see Appendix A). In discussions with the Clipper team we agreed on topic 1 - Supporting Wind Turbine Research and Testing.

As referenced in the Project Summary (See Appendix B), the proof and implementation of an alternative chemistry as a lubricating fluid will have a high impact on improving wind turbine performance and reducing its operating and maintenance cost.

The Technical Approach (see Appendix C for complete Project Narrative) included the establishing of a benchmark, the lubricant evaluation, the lab full scale gearbox trial and the economic evaluation. The optional field trial included 2 turbines in similar operating conditions, one filled with the benchmark and the other with the new lubricant.

The Project proposal was submitted in March 2009 and in July 2009 it was awarded with DOE funds of 745,189 USD, which did not cover a field trial. Additional cost share beyond the government award to be incurred by the recipient and teaming members. Contractual details were finalized through remainder of 2009 with a project kick off meeting taking place early January 2010.

Reference Appendix Documents for this Section:

*APPENDIX A:	DOE FUNDING OPPORTUNITY ANNOUNCEMENT
*** APPENDIX B:	PROJECT SUMMARY/ABSTRACT
*** APPENDIX C:	PROJECT NARRATIVE

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3. PROJECT OBJECTIVES

The most commonly used wind turbines are equipped with gearboxes to convert torque from the relatively slow turning blades to the fast rotating generator. Despite continued innovation in gearbox design and lubrication, fleet-wide gearbox maintenance issues persist resulting in higher lifetime operating cost which could ultimately reduce acceptance of wind as a renewable energy option. In recent years, the size of wind turbines has increased enormously and problems like gear scuffing or wear of the bearings due to vibrations have added to maintenance issues and gearbox failures. The goals of 20% Wind by 2030 are stated as increasing the use of offshore wind derived energy resulting in further increases in wind turbine size and an operating environment where technology challenges are more severe and maintenance harder to perform.

Despite the introduction of condition monitoring tools and tailoring of conventional gear oils, improvements in oil lifetime of between 2-4 years are all that have been achieved. Due to ever changing operating conditions, oil changes are often necessary after 6 months and gearbox changes after 5-7 years. This combination of short gearbox maintenance intervals and the difficult to service environment results in high operating and maintenance costs and illustrates the need for further innovation. The proposed solution is an innovative new lubricant that offers the opportunity for full life, increased gearbox reliability, and increased energy efficiency by reducing parasitic losses associated with the gearbox. This solution is believed to enable a paradigm shift in gearbox oil system design.

The goal of the project was to allow a "proof of concept" for an alternative chemistry as a lubricating fluid for a wind turbine gearbox. This concept we have termed as "Full Life Lubricating Fluid" and it is expected to have an operating lifetime comparable to the expected turbine life (~20 years). The alternative chemistry that Dow Corning proposes gives a fluid that is very robust, with excellent oxidative stability. This fluid would potentially allow a reduction in the dependence on condition monitoring/predictive maintenance models and a reduction in the mean time between failures for the gearbox during the turbine's lifetime. In partnership with Clipper Windpower a study has been designed to bench test and document the suitability of the new alternative chemistry lubricating fluid, versus a well know industry "standard" lubricating oil. The expectation of this study was to understand and quantify potential areas of benefit, such as the temperature-viscosity profile in use which is lower than existing oils, thus expected to reduce frictional wear on internal gears and components. Due to wind turbine operation parameters being at both upper and lower performance temperatures (extreme cold and warm weather conditions) the study is designed to measure the impact of improved power output efficiency across this operating temperature range. The potential for improved temperature-viscosity performance in use across a wide operating temperature range with reduced frictional wear is expected to positively impact the operating reliability of the gearbox and thus reduce overall turbine downtime linked to both scheduled and non-scheduled maintenance.

The overall objective of the project is to demonstrate a reduction in total wind turbine operation cost by using Perflouropolyether (PFPE) lubricant. Major tasks include: 1) Establish a benchmark of today's technology in order to define a target for the new fluid, 2) Lubricant evaluation of current and new technology in both fresh and used condition together with potential friction reduction in bench gear testing, 3) Lab gearbox trial to verify our concept on the actual component to set the stage for field trials, and 4) Economic evaluation of the cost savings anticipated. The evaluation of current oil technology gaps with respect to full life wind turbine gearbox lubricating fluid will set the target for the PFPE lubricant. The breakdown mechanism of currently used oils in high stressed gear contact zones will be determined and the behavior of PFPE lubricant will be observed to show lower friction in the contact zone resulting in less heat generation. The difference in performance will allow lubricant life time estimation. The objective of the lab gearbox trial is to evaluate PFPE lubricant in an accelerated gearbox life test to build confidence before entering the field trial. The optional field trial, while limited to only 12 months, is expected to provide adequate data to further evaluate performance and cost improvements. However that planned field trial with PFPE as the gearbox lubricant did not take place due to budget and timing constraints.

4. EXECUTIVE SUMMARY

Industrial gearbox lubricants typically are Mineral Oil based with considerable amounts of additives to overcome the lack of inherent base fluid properties like wear protection, oxidation stability, load carrying capacity, low temperature solidification and drop of viscosity at higher temperatures. For today's wind turbine gearboxes, the requirements are more severe and thus Synthetic Hydrocarbon oils are used which exhibit some improvement with respect to oxidative stability and viscosity profiles. Polyalphaolefins (PAO) present the majority of base fluids used today. For improved biodegradability, Ester based oils are used and for improved viscosity temperature profile, Polyalkyleneglycol (PAG) based lubricants are being considered. However, all such hydrocarbon based lubricants require significant amounts of Extreme Pressure (EP) additives to meet performance requirements. Perfluoropolyether (PFPE) fluids provide load carrying capacity as an inherent property.

During the course of the project with the main tasks of "Establish a Benchmark", "Lubricant Evaluation", "Full Scale Gearbox Trial" and "Economic Evaluation" the PAO Reference oil exhibited significant changes after laboratory gear testing, in service operation in the field and full scale gearbox trial. The chemical analysis of the PAO Reference oil revealed additive depletion and increase in water content not found in the PFPE fluids after laboratory gear and full scale gear testing. The lack of water in the PFPE fluid could be explained by its high specific gravity causing water to go to the fluids surface rather than to the sump as in the PAO Reference oil. The four hydrocarbon base oils selected for comparison in the benchmarking exercise showed variation with respect to meeting the requirements for the laboratory micro-pitting tests, while the PFPE fluid exceeded the requirements even with the material taken after the full scale gearbox trial. This is remarkable for a lubricant without EP additives. One note of interest is, that on the gears from the micro-pitting test there was oxygen (besides the elements found in steel) detected in the sliding surface area for the PFPE fluids, but not for the hydrocarbon based benchmark oils. Since oxygen was detected in the root surface for all tested oils/fluids, only the PFPE fluids create a tribofilm in the lubricating contact area. Laboratory bearing tests performed on the PFPE fluids before and after the full scale gearbox trial showed the results met requirements for the industry standard.

The PFPE fluid successfully completed the full scale gearbox test program which included 406 hours of baseline evaluation and an additional 282 hours of progressive staged load testing, simulating a period of 2 years according to the HALT conditions (8108 hours at 12% higher load simulating 20 years). The evaluation of gears showed no micro-pitting or objectionable wear. By the final stage, lubricant film thickness had been reduced to just 21% of its original value, this was by design and resulted in a lambda ratio of well below 1 that would predict possible asperity or surface to surface contact. This test design scenario of a low lambda ratio is a very undesirable lubrication condition for real world but creates the ability to test the lubricating fluids performance under the most extreme conditions. The PAO Reference oil also passed 298 hours of testing (baseline included) without any noticeable deterioration of the gear surface. However the PAO Reference oil was replaced midway through the progressive loading, as the lubricant was burned in an attempt to raise the sump temperature via an immersion heater. Both materials experienced a decrease of viscosity of approximately 20% after 688 hours runtime for the PFPE and 151 hours for the PAO, while the viscosity index decreased for the PAO there was a slight increase for the PFPE. The PFPE fluid at some load stages was run up to 100^{o}C compared to a maximum of 84^{o}C for the PAO Reference oil and the PFPE fluid had seen more dynamic loading caused by torque variation while reaching the operating limits of the test stand.

FZG laboratory gear tests and measurements of the drive motor's current during the full scale gearbox trial were made to characterize the relative efficiency between the PFPE fluid and the PAO Reference oil. In the FZG laboratory efficiency test, the PFPE fluids show much higher churning losses due to their higher viscosity and density. In the boundary lubrication region, PFPE fluids show reasonable results even without EP additives and in the Elastohydrodynamic (EHD) lubrication regime in loaded conditions (should be compared to actual operating conditions of the wind turbine gearbox) they show lower losses compared to the benchmark oils. Under high speed conditions, coefficients of friction are lower at similar film thicknesses for the PFPE fluids. The results of the full scale gearbox measurements varied as each material's properties and load stages were very different. The analysis seems to show that the efficiency correlates better to dynamic viscosity than any other of the measured metrics such as film thickness. In load stages where the load, speed and temperature are similar, the PFPE fluid has a greater film thickness and theoretical gear protection, but requires a larger current for the drive motor (lower efficiency) than the PAO. However in load stages where the film thickness is the same, the PFPE fluid's reduced dynamic viscosity gives it a slight efficiency advantage relative to the PAO reference oil. Ultimately, many factors such as temperature, rotational speed, and fluid viscosity combine in a complex fashion to influence the results. However, the PFPE's much lower change of viscosity with respect to temperature, allows variations in designing an optimum viscosity to balance efficiency versus gear protection. The original planned test of viscosity variants for the PFPE fluids were partially accomplished by the different temperatures during the full scale gearbox trial's load stages.

Economic analysis was done using Cost of Energy(COE) calculations. The results vary from 5.3% for a "Likely Case" to 16.8% for a "Best Case" scenario as potential cost improvement by using PFPE as the gearbox lubricating fluid. It is important to note the largest portion of savings comes in Levelized Replacement Cost, which is dictated by the assumption on gearbox reliability. Thus, verifying and quantifying the potential of PFPE fluid to effect gearbox reliability is the key assumption that would need to be further validated. The lifetime capability and efficiency of the PFPE fluid was investigated within the scope of the project.

In summary the proof of concept to use PFPE fluid as wind turbine gearbox lubricant was validated with this project. The increase in life time was qualitatively demonstrated and this supports the need for future activity of field trials and laboratory aging studies to quantify the predicted 20 year life. With micro-pitting being the major failure mechanism in the last years, recent publications show that white etch cracking of bearings seem to have the highest impact on wind turbine reliability. With its higher film thicknesses compared to PAO reference oils, PFPE fluids have the potential to reduce this failure occurrence as well.

5. TASK 1 RESULTS: ESTABLISH A BENCHMARK

5.1 Introduction

The benchmarking task was primarily focused on data which was already in existence and available at time of this project. New data was not produced specifically for this task although in later tasks there was data generated to satisfy specific requirements for the goals of those later tasks. The focus of the benchmarking was with respect to lubricants used in wind turbine gearboxes. Several different categories, or areas of interest, were identified and looked at in further detail as follows:

- Sample Population of Lubricating Oils: Selection of a small sample population of incumbent lubricating oils currently used by industry in wind turbine gearboxes. In addition, the new PFPE Fluid technology was also added to the population for comparison.
- Data Collection: Performance/test properties for the sample population identified above.
- Industry Requirements: As documented by specifications from various industry societies, organizations, technical committees or Original Equipment Manufacturers (OEM's).
- Viscosity Optimization: Selection of the optimal viscosity grade to use for the new PFPE fluid.

Given the above areas of interest, it was the project teams goal to most accurately identify what industry is currently using as a lubricant, collect performance data, and perform a gap analysis as to how to the performance data compares to what industry actually requires based on documented specifications. Interpretation and analysis tools are variables in this process and can impact the conclusions made.

The new PFPE fluid, which this entire project is focused on evaluating, has properties significantly different than many of the incumbent oils in use in wind turbine gearboxes today. This fact allowed for some "tailoring" of the fluid's viscosity from the industry norm, thus providing a more optimal gearbox efficiency and reliability, at least from a theoretical perspective.

5.2 Sample Population of Lubricating Oils

The selection of a particular lubricant for use in a gearbox can be made from many fluids generically identified as “gearbox oils”. The number of manufacturers and product offerings seems to be endless with many different products available around the globe. There are three key attributes which define the lubricating fluid and creates the main differences between each. They are: 1) Base Oil Technology 2) Viscosity Grade and 3) Additive Package.

5.2.1 Base Oil Technology

The base oil technology that is used in formulating a gearbox oil can be drawn from many different chemistry sets ranging from a basic hydrocarbon mineral oil to any one of many different synthetic fluids produced via special building of molecules through chemical synthesis. With different base oils, come different advantages and disadvantages such as thermal stability, oxidation stability, compatibility, water solubility, viscosity index and load carrying capability, just to name a few. It is well documented in the public domain the many different base oils and their features. This report will not go into those details. Choices were made in this project to select the most accurate sample population of base oils used currently in industry by wind turbine OEMs. Selections were made based on an unscientific surveying of OEMs by Dow Corning’s commercial group as well as Clipper’s knowledge and association with different lubricant manufacturers. It should be noted that a limited number of base oil technologies were selected to keep the scope of this project within budget while still providing the representative sample population desired. As a result of this, the following base oil technologies were chosen:

- Poly Alpha Olefin (PAO Synthetic Hydrocarbon)
- Poly Alkylene Glycol (PAG)
- Ester
- Perflouropolyether (PFPE)

5.2.2 Viscosity Grade

The viscosity grades available for gearbox oils can vary greatly from ISO Grade 10 (10 cSt @40°C) all the way up to ISO Grade 1500 (1500 cSt @ 40°C). The speed and load of the application are two important factors to consider when selecting the proper viscosity to use in a gearbox. It is primarily the viscosity that determines the ability for the base oil by itself to establish and maintain a proper film thickness between the surfaces of moving parts (bearings or gears) in a gearbox during its operation in order to prevent wear and delay ultimate failure for as long as possible, which in theory, is the design life of the equipment. Design life of Wind Turbines as a whole is stated to be 20 years by some OEMs.

Also worthwhile to note is the amount of change in a fluids viscosity with respect to temperature. This amount of change is quantified by a measurement called Viscosity Index (VI). In general, a low viscosity index indicates a higher amount of change in viscosity with respect to temperature and a high viscosity index indicates a lower amount of change in viscosity with respect to temperature.

$$\downarrow \text{VI} = \uparrow \text{Viscosity Change vs Temp}$$
$$\uparrow \text{VI} = \downarrow \text{Viscosity Change vs Temp}$$

VI can be a very important fluid property when different startup or operating temperatures are involved. High operating temperatures in very warm climates can cause the oil with a low VI to be thinner than desired and not provide the intended film thickness between moving components to protect wear from occurring. Thus over the years the ISO viscosity grade for hydrocarbon base oils for wind turbine gearboxes has been increased from 220 to 320 to potentially overcome micro-pitting problems. Cold conditions can also make pumping of the oil to be difficult because of the high viscosity which reduces overall efficiency but more importantly can again starve the required contact zones from being lubricated with the proper amount of oil, thus allowing surface to surface contact and accelerated wear leading to failure prior to the intended design life. Thus for wind turbine gearboxes, auxiliary heaters are being installed to allow for start up in cold climates. The ISO 320 grade is a compromise for providing minimum film thickness at operating temperature while remaining operable with auxiliary heating systems at cold temperatures; very little ability exists to either decreased or increased the viscosity grade with hydrocarbon base oil technology, primarily because of its Viscosity Index.

5.2.3 Additive Package

Nearly all lubricating oils produced, contain additives which are meant to enhance the performance of the base oil in some way. Certain additives are used to extend the life of the lubricating oil while others are used to protect metal surfaces or expand its function in some way into a multi-use oil. Specific additive packages used by manufactures are closely guarded and considered proprietary in nature so additional details regarding additive packages are not available.

Operating conditions under which the oil is used greatly effects the use able life and it’s the operating conditions that can greatly accelerate the depletion of the additive package in the oil. When additive depletion takes place, it leaves the oil in a state of reduced protection of the equipment for whatever property the particular additive was designed to enhance. While there may be other purposes for additives, the ones most primary for gearbox oils are those added to inhibit rust, inhibit oxidation of the base oil and improve either the Anti-Wear (AW) or the Extreme Pressure (EP) properties of the lubricant.

The term additive “package” is used because historical technology is such that a manufacturer will start with a base oil which contains no additives and simply add the additives to it, mixing or processing in some way to blend it together.

The innovative PFPE technology (Reference Patent Application WO2009/141284 A1) as the focus of this project is unique in that there are no additives blended into the base oil. In this case, the enhancements to the base oil typically provided by the additives are functionalized as part of the molecular structure. This eliminates the historical problem of additive depletion and allows the fluid to perform theoretically for the entire life of the design or beyond.

5.2.4 Candidates for Benchmark

Table 5.2.4-1 shows the candidate lubricating fluids selected for study focus. These were again selected unscientifically through surveying of OEMs by Dow Corning's commercial group as well as Clipper's knowledge and association with lubricant manufacturers. From this information, determination was made as to popularity of use and representative cross section of current leading technology. As a result of this, the following base oil technologies were selected:

Base Oil Technology	Viscosity Grade	Generic Reference
PAO Synthetic Hydrocarbon	ISO 320	PAO Ref
PAO Synthetic Hydrocarbon	ISO 320	PAO Alt
PAG Polyalkylene Glycol	ISO 320	PAG
Synthetic Ester	ISO 320	Ester
PFPE	ISO ≈220	PFPE

Table 5.2.4-1

5.3 Data Collection

The three tables which follow are a collection of data which previously existed for the materials selected for the sample population of lubricating oils from 5.2.4. No data in this section, 5.3, was generated as part of this project. Manufacturers of the different oils were solicited to provide data for the properties and characteristics shown. The intended use for inclusion as benchmarking data for this DOE project was communicated in the solicitation. Various levels of responses were received. Data provided here was obtained through the solicitation and/or material data sheets. Many cells are shown in the tables as *NA (Not Available).* For cells with *NA*, the manufactures either chose to not provide data in order to protect their commercial interest, or the data simply was not available because it had not previously been tested. In each case the exact reason for *NA* is unknown.

All data is shown "as received" from the manufacturer and one can quickly determine that many gaps exist in the data. Even those tests identified in what will be explained later as the best leading industry standard, do not have all data supplied by manufacturers. In addition, there are often times multiple variations of similar tests. Slightly different parameters were used for those similar tests which allows for little ability to make legitimate comparisons between the different lubricating fluids.

5.3.1 Physical Property Data

Table 5.3.1-1 lists basic physical properties. These are not specifically identified in any industry standard but rather are listed here only to frame the fundamental properties of lubricating oils currently used in the wind turbine industry and how the PFPE used for the Full Scale Laboratory Gearbox Trial in Section 7 of this report compares.

Property	Method	Units	PAO Ref	PAO Alt	PAG	Ester	PFPE
Viscosity @40C	ASTM D 445	cSt	330	335	320	320	212
Visc @100C	ASTM D 445	cSt	33	38.3	54	37	61
Viscosity Index	ASTM D2270	-	140	164	237	160	339
Pour Point	ASTM D 97	°C	-36	-38	-39	-40	-58
Specific Gravity	@15.6 °C	kg/m^3	870	860	1051(25°C)	958	1840
Flash Point		°C	220	242	*NA*	270	Not Flammable
Pressure Visc Coefficient	@38°C	GPa^{-1}	*NA*	*NA*	11	*NA*	16
Pressure Visc Coefficient	@ 99°C	GPa^{-1}	*NA*	*NA*	8	*NA*	13
NA = Data not available or not provided							

Table 5.3.1-1 Benchmark Physical Property Data

5.3.2 Application Testing

Table 5.3.2-1 lists results for those tests which are most closely related to an application or most representative of how the lubricant will perform in a gearbox.

Property	Method/ Conditions	Units	PAO Ref	PAO Alt	PAG	Ester	PFPE
FZG Load Stage			*NA*	*NA*	*NA*	*NA*	*NA*
FZG Pitting			*NA*	*NA*	*NA*	*NA*	*NA*
FZG Scuffing	A/16.6/90 DIN 51354 (modified)	Failed Load Stage	*NA*	14+	*NA*	*NA*	*NA*
FZG Scuffing/Scoring (intensified)	A/16.6/140 DIN 51354	Failed Load Stage	>12	*NA*	*NA*	*NA*	*NA*
FZG Scuffing/Scoring	A/16.6/120	Failed Load Stage	*NA*	*NA*	*NA*	*NA*	*NA*
*FZG Scuffing/Scoring	A/90/8.3 ISO 14635-1	Failed Load Stage	*NA*	*NA*	*NA*	>14	>12
FZG Scuffing/Scoring	A/90/16.6	Failed Load Stage	*NA*	*NA*	*NA*	12	*NA*
**FVA No. 54 micro-pitting	@90°C		>10 (high)	10 (high)	*NA*	>=LS10	>10
**FVA No. 54 micro-pitting	@60°C GT-C/8.3/60	Failed Load Stage	*NA*	*NA*	>10	>=LS10	>10
Filterability	SKF Method		*NA*	*NA*	Pass 12µ	*NA*	Pass
Filterability	Hydac Method		*NA*	*NA*	*NA*	*NA*	*NA*
Filterability	ISO 13357-2 (modified) (Stage1/Stage2)		*NA*	*NA*	*NA*	Pass/Pass	*NA*
Filterability	Wet Pall Filterability		*NA*	Pass (3µ filter 1% water contamination, 25PSI pressure drop)	*NA*	*NA*	*NA*
Foaming Test	Flender Foam Test		*NA*	*NA*	*NA*	8	*NA*
FE 8 Bearing Test	D 7.5/80-80 DIN 51354-2		4.5 Roller 43.2 Cage	*NA*	*NA*	*NA*	*NA*
*FE 8 Bearing Test Stage1	D 7.5/80-80 DIN51819		*NA*	1	Pass	7.0/No/Small (Rollers/Rippling/Micro-pitting)	5
*FE 8 Bearing Test Stage2	D 7.5/800-80 DIN51819		*NA*	1.5	Pass	Pass/14 mg (Running Time/Roller Wear)	*NA*
**FE 8 Bearing Test Stage3	L11, 700h		*NA*	1	Pass	>700 h (FAG Internal)	*NA*
**FE 8 Bearing Test Stage4	FE8-WKA		*NA*	1.1	Pass	Pass 1 (FAG Internal)	Pass
*Elastomeric Compatability	NBR & FKM		*NA*	*NA*	Pass	Pass/Pass/Pass/Pass (DIN53538) (Vol/Hardness/Elong/Tensile)	*NA*

* Included in IEC 61400-4 Table 37 Standardized Test Methods for Evaluating WT Fluids (fresh oil)
** Included in IEC 61400-4 Table 38 Non-Standardized Test Methods for Lubricant Performance (fresh oil)
NA = Data not available or not provided

Table 5.3.2-1 Application Testing Benchmark Data

5.3.3 Lubricant Testing

Table 5.3.3-1 lists results for those tests which are predictive tests and attempt to provide a best case simulation of how the lubricant will perform under conditions likely to be seen in the application.

Property	Method/Conditions	Units	PAO Ref	PAO Alt	PAG	Ester	PFPE
SRV Test	ASTM D7421-08 (PO Mean)	MPa	*NA*	*NA*	3577	*NA*	*NA*
SRV Test	300/50C/122/ball/surface/2h		*NA*	*NA*	*NA*	*NA*	*NA*
SRV Friction Coefficient			0.055	*NA*	0.08	*NA*	*NA*
SRV Ball Scar Wear		mm	0.50	*NA*	*NA*	*NA*	*NA*
SRV Profile Depth PT (wear)		µm	1.0	*NA*	*NA*	*NA*	*NA*
***Copper Corrosion**	3hr @100°C ASTM D130		1a	*NA*	1a	1	<1a
***Corrosion (Ferrous)**	SKF EMCOR ISO 11007		*NA*	*NA*	0/0	0/1, 3/3 (Dist Water. 0.5% NaCl)	0/0
***Corrosion (Ferrous)**	Distilled water ASTM D665		*NA*	*NA*	Pass	0-A	Pass B
***Corrosion (Ferrous)**	Sea water ASTM D665		*NA*	Pass	Pass	0-B	*NA*
4 Ball Wear Test	1800RPM 20kg 54°C 60 min ASTM D4172	mm	*NA*	0.25	*NA*	0.33	*NA*
4 Ball Wear Test	1200RPM 40kg 75°C 60 min ASTM D4172	mm	*NA*	*NA*	.51	*NA*	*NA*
Water Seperability	Time to 40/37/3 @ 82°C ASTM D1401	min	*NA*	10	*NA*	40@82°C	*NA*
Foaming Characteristics	ASTM D 892, Seq I		*NA*	*NA*	0/0 10Min	0/0 Flender Foam test (20C/60C) 6%/4%	20/0
Foaming Characteristics	ASTM D 892, Seq II	ml/ml	*NA*	0/0	0/0 10Min	20/0	240/0
Foaming Characteristics	ASTM D 892, Seq III		*NA*	*NA*	0/0 10Min	0/0	0/0
Oxidative Stability Ageing Behavior	ASTM D2893	%	*NA*	*NA*	*NA*	2	*NA*
***Shear Stability**	CEC L-45-A99 KLR Shear Test		*NA*	*NA*	*NA*	Shear Stable no VI improver	*NA*
Sludging Tendancy			*NA*	*NA*	*NA*	*NA*	*NA*

***** Included in IEC 61400-4 Table 37 Standardized Test Methods for Evaluating WT Fluids (fresh oil)
****** Included in IEC 61400-4 Table 38 Non-Standardized Test Methods for Lubricant Performance (fresh oil)
NA = Data not available or not provided

Table 5.3.3-1 Lubricant Testing Data

5.4 Industry Requirements

Specification requirements for wind turbine gearbox lubricants were sought after in this task. If specific requirements could be identified, a gap analysis could then be performed against actual test and performance data to identify where candidates for benchmark either do not meet the requirement or are not even tested for that particular requirement, which could indicate a complete design characteristic not being accounted for.

OEM's either don't have established internal lubricant specifications, or they are not willing to share as this can be considered proprietary information and create a competitive advantage over competitors.

Conclusions were drawn that the design and rating standards for wind turbine gearbox design are very general in nature, especially for lubrication requirements. This is because there is not any one accepted dominant standard which exists that is followed exclusively by wind turbine OEMS, lubricant manufactures, or suppliers such as bearing manufacturers or gearbox designers. However, trends are slowly evolving to where a global standard, and a more broad industry agreement, is getting closer to reality. The IEC 61400 Wind Turbines Series is the standard which seems to be leading this effort. In particular, IEC 61400-4 "Design Requirements for Wind Turbine Gearboxes" still remains in draft form but was considered by this project team to be the most technically advanced with specific requirements documented for lubricants in the area of discrete properties and associated specification ranges. As such, IEC 61400-4 would be the standard most relied upon to perform this benchmarking exercise.

Additional reference may be made to industry standards in an article published in Gear Technology by B. Bradley, July 2009 [1]. See also Appendix 1A.

With the goal of being consistent in making comparisons, analysis tools were used which commonly exist and are used in industry. Specific to this project, Table 5.4-1 provides a combined listing of all Industry Standards and Analysis Tools used in varying degrees for the work throughout this entire project. As will be commented on later in this report, some tools accurately model the new PFPE Fluid and others do not as compared to the modeling tools available which are standardized for traditional hydrocarbon fluids.

Industry Standards and Analysis Tools for Lubricants in Wind Turbine Gearboxes

1. GL wind 2005	"Guideline for the Certification of Offshore Wind Turbines"
2. ANSI/AGMA 6006/A03	"Standard for Design and Specification of Gearboxes for Wind Turbines"
3. KissSoft Software	Release 04/2010
4. ISO 6336-1 &-2	"Calculation of Load Capacity of Spur and Helical Gears"
5. ISO/CD TR 15144-1	"Calculation of micro-pitting load capacity of cylindrical spur and helical gears."
6. DIN/ISO 281-4	"Rolling bearings – Dynamic load ratings and rating life"
7. DIN IEC 61400-4 (Draft)	"Design requirements for Wind turbine Gearboxes"
8. ISO 81400	Part of IEC 61400 Series
9. AGMA 9005-E02	Industrial Gear Lubrication
10. DIN 51517-1	Lubricants – Lubricating Oils-Part 1: Lubricating oils C; Minimum Requirements
11. ISO 12925-1	"Specification for lubricants for enclosed gear systems

Table 5.4-1

See also Appendix Documents 1B, 1C, 1D & 1E for additional info in Industry Standards and Analysis Tools.

References:

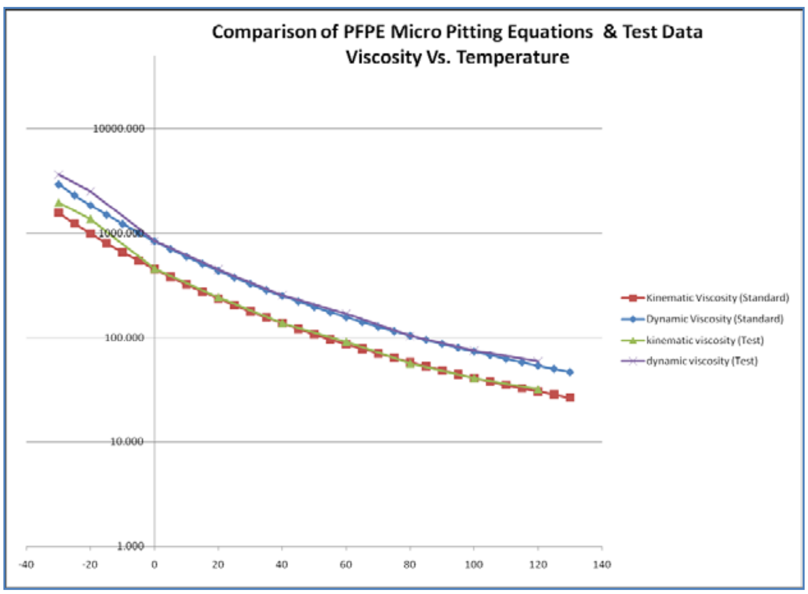
1. Bradley, B. “An International Wind Turbine Gearbox Standard”, Gear Technology, July 2009. http://www.geartechnology.com/issues/0709x/wind.pdf

5.5 PFPE Viscosity Optimization

Within the scope of this project, the team had the opportunity to select and manufacture a PFPE fluid specifically tailored for the viscosity deemed most appropriate. This fluid would then be used in the Full Scale Gearbox Trial as defined in Task 3 of the Project (Section 7 of this Report). Prior to manufacture, a determination had to be made as to what the most appropriate viscosity should be. Prior to going through the summary analysis which follows, a viscosity of approximately 137 cSt at 40°C was selected for the PFPE based on engineering judgment simply to provide a starting point for analysis. This is significantly different vs. the traditional PAO used by industry of 320 cSt at 40°C. The viscosity models were confirmed to be accurate and subsequently several additional factors were analyzed via mathematical models including Micro-pitting, Flank Safety-Pitting and Efficiency. As a result of this further analysis, a decision was made by the team to fine tune the target viscosity of the PFPE fluid to 224 cSt at 40°C.

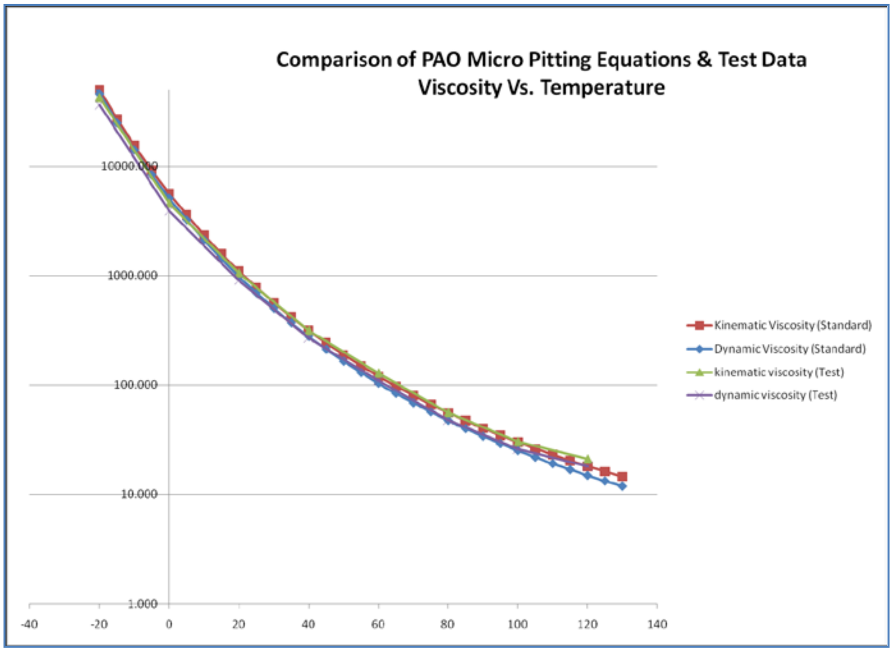
See also Appendix 1F

Graph 5.5-1 shows the equations in ISO TR 15144-1:2010 closely model theoretical viscosity vs. actual test viscosity.



Graph 5.5-1 PFPE Viscosity Theoretical vs. Actual

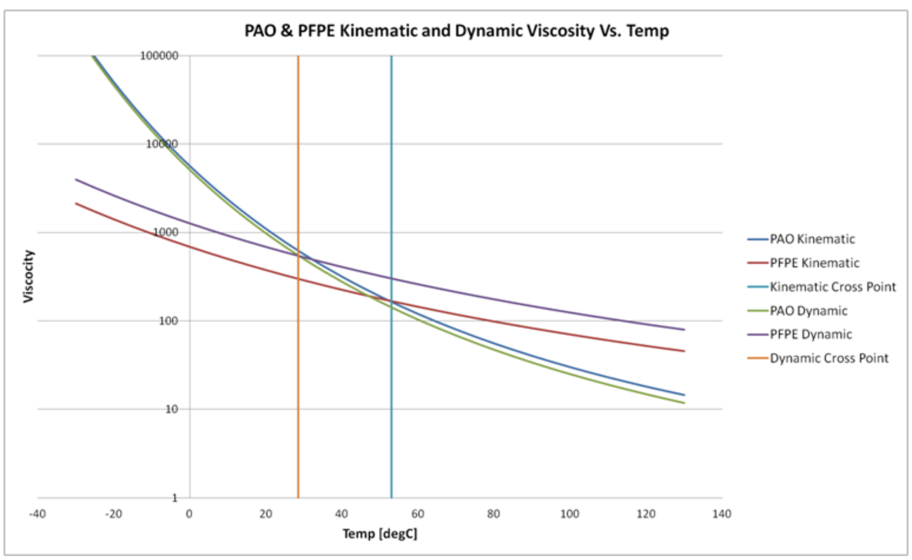
Graph 5.5-2 shows the equations in ISO TR 15144-1:2010 closely model theoretical viscosity vs. actual test viscosity.



Graph 5.5-2 PAO Viscosity Theoretical vs. Actual

Graph 5.5-3 shows the comparison of PFPE and PAO Viscosities. Two particular points to note are:

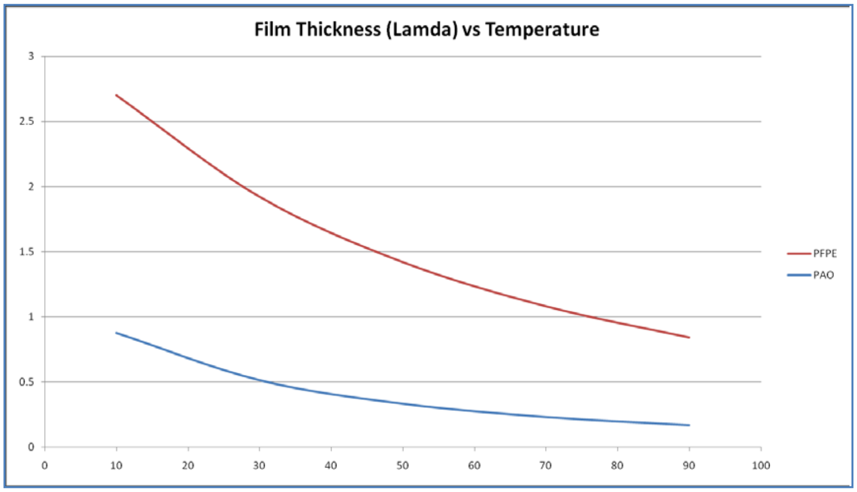
- PFPE Dynamic Viscosity is greater than PAO above 28°C
- PFPE Kinematic Viscosity is greater than PAO above 53°C



Graph 5.5-3 PAO & PFPE Viscosity vs. Temperature

5.5.1 Micro-pitting

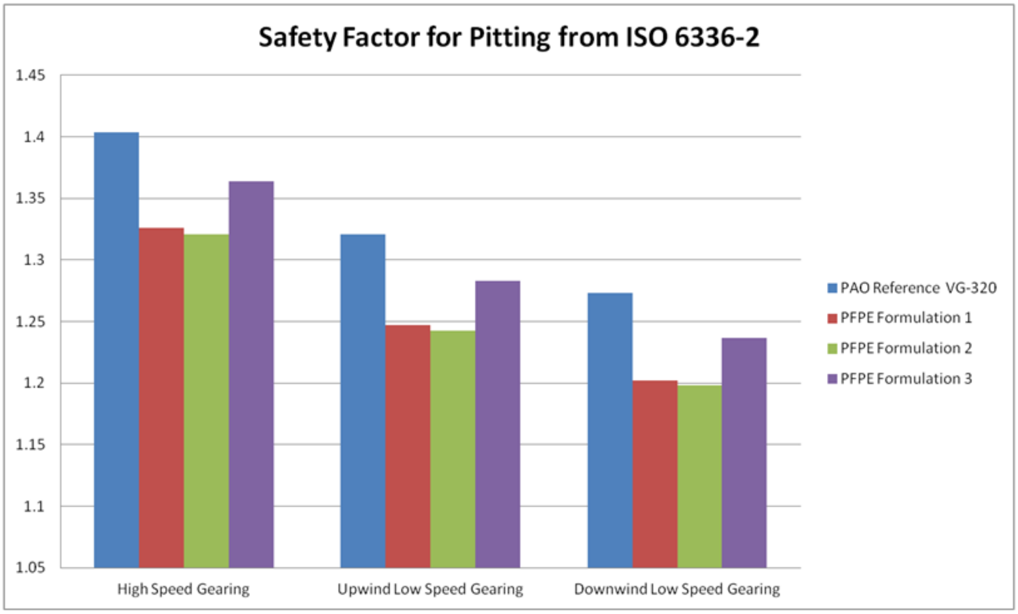
Micro-pitting calculations were performed per ISO 15144-1:2010. The specific temperature used in the calculation for film thickness is the local flash temperature in the contact points of the gear mesh. This temperature is always much greater than the bulk temperature of the oil. Since the PFPE has a superior viscosity profile compared to the PAO there is a significant safety margin improvement for film thickness (lambda ratio) over the entire temperature range as compared to the PAO. Graph 5.5.1-1 shows this in graphical form.



Graph 5.5.1-1 PAO vs PFPE Film Thickness

5.5.2 Flank Safety – Pitting

Flank safety or pitting of wind turbine gearing is performed in accordance with ISO 6336-2. The rating includes a factor, Z_L, to account for the viscosity grade of the lubricating fluid. The Z_L factor is calculated using the viscosity of the fluid at 50°C. A lower viscosity results in a lower Z_L and thus a lower safety factor for pitting. This simplistic calculation does not take into account the viscosity index of the fluid. As such, lower viscosity formulations of PFPE are penalized in the pitting calculation even though the calculated film thicknesses are above or beyond that of the baseline fluid. The final test formulation of PFPE had a kinematic viscosity of 178 at 50°C which is in line with standard ISO VG 320 oils. Future testing can be done to increase the Z_L to account for the increased viscosity index and film thickness of the PFPE, this allows for a greater safety factor for pitting than the baseline. Graph 5.5.2-1 shows the effect of reduced Z_L factor of the pitting safety factors for the PFPE formulations.



Graph 5.5.2-1 Safety Factor for Pitting

5.5.3 Efficiency

Efficiency is looked at as a secondary consideration with both micro-pitting and pitting protection taking higher priority in selecting the appropriate viscosity. Because the viscosity profiles are so greatly different between the PFPE and PAO, there is in essence a tradeoff to where there exists higher theoretical micro-pitting and pitting protection and possibly lower efficiency because of the higher viscosity of the PFPE vs. the PAO at the higher temps due to the higher VI, even though the overall viscosity grade is lower. To error on the side of increased lubrication protection for the gearbox, a higher viscosity was chosen compared to choosing a lower viscosity with safety factors comparable to the PAO reference oil.

Reference Appendix Documents for this Section:

** APPENDIX 1A: AN INTERNATIONAL WIND TURBINE GEARBOX STANDARD

*** APPENDIX 1B: OIL STANDARD INVESTIGATION

** APPENDIX 1C: ANSI 6006-A03 LUBRICATION ANNEX F

** APPENDIX 1D: ANSI 6006-A03 LUBRICATION SECTION 7

** APPENDIX 1E: DIN IEC 61400-4 PARTIAL

*** APPENDIX 1F: PFPE VISCOSITY OPTIMIZATION & LAMBDA

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6. TASK 2 RESULTS: LUBRICANT EVALUATION

6.1 Introduction

The evaluation included the PFPE sample candidates along with currently used synthetic based oils utilizing analytical and performance bench testing. During the project kick-off meeting in January 2010 there was some modifications to the proposed subtasks as indicated in the following sub bullets:

- Task 2.1: Base line analytical testing of 6 fresh fluids, currently used oil and 5 PFPE fluids (2 fluids used in preliminary trials and 3 viscosity variants):
    - Further preliminary PFPE trials continued to show a promising PFPE candidate and the viscosity variants were dropped and replaced with a viscosity of PFPE at the higher end of the spectrum so there would be a maximum micro-pitting safety factor for the full scale gearbox trial. Thus the number of total fluids was still used to include one more PAO, one Ester and one PAG based oil to increase benchmark data.

- Task 2.2: Monitoring of currently used oil from 3 reference wind turbines in field, includes online lubricant sensors and analytical testing.
    - A total of 4 field turbines were selected for in service monitoring, but online lubricant monitoring was not possible.

- Task 2.3: Stressing of 3 PFPE fluids (viscosity variants) with gear and bearing tests.
    - Since 1 viscosity was chosen it was decided to have the PFPE fluid after full scale gearbox trial tested on bench gear and bearing conditions.
    - In addition it was also decided to test all 5 synthetic benchmark fluids for micro-pitting.

- Task 2.4: Analytical testing of all 5 PFPE fluids.
    - Was performed on PFPE candidate at various testing stages.

- Task 2.5: Surface Analysis of gear and bearing surfaces from PFPE stress experiments.
    - Preliminary trials on bearing test did not show any surface changes so analysis was limited to gears.

- Task 2.6: FZG gear efficiency test of currently used oil compared to PFPE selected candidate with subsequent analytical testing.
    - FZG gear efficiency test was chosen for PAO, PAG, PFPE fresh and after full scale gearbox trial, though analytical testing is warranted after longer test duration, hence the FZG micro-pitting test from Task 2.3 above.

- Task 2.7: Monitoring of PFPE fluid in use at Lab Gearbox Trial, includes online lubricant sensors and analytical testing.
    - Online instrumentation was limited to temperatures, but besides sampling PFPE at different load stages it also includes sampling of subsequent run with PAO.

Over the course of the project the following reports were produced:

- University of Dayton Research Institute (UDRI)
    - Final Report on Analysis of In-Service Turbine Oil Samples
    - Gearbox Oil Samples
    - Fresh and Laboratory Gear Test Stressed Oils/Fluids
    - Teeth Surface Analysis from Laboratory Gear Tests
- Technical University Munich
    - Report No 4203: Expertise on the micro-pitting load capacity on PFPE fresh laboratory sample.
    - Report No 4364: Expertise Investigation of the Efficiency of PAO, PAG, PFPE fresh and PFPE used Lubricants
- Ruhr- University Bochum
    - Report G 1259: Micro-pitting Test on PAO reference
    - Report G 1262: Micro-pitting Test on Ester
    - Report G 1265: Micro-pitting Test on PAG
    - Report G 1269: Micro-pitting Test on PAO alternative
    - Report G 1276: Micro-pitting Test on PFPE fresh
    - Report G 1286: Micro-pitting Test on PPFE used
- Dow Corning Wiesbaden
    - FE 8 Bearing test according stage 1 – DIN 51819
        - PFPE lab sample
        - PFPE before gearbox trial
        - PFPE after gearbox trial
    - FE 8 Bearing test according stage 2 – DIN 51819
        - PFPE lab sample
        - PFPE after gearbox trial
    - FE 8 Bearing test according stage 3 – FAG test
        - PFPE lab sample
        - PFPE after gearbox trial

In order to evaluate all test data and determine suitability of candidate lubricant against wind turbine requirements the data from above mentioned reports are pulled together in the following subsections:

- Analytical data on Reference PAO
- Analytical data on Candidate PFPE
- Analytical data on other benchmark fluids
- FZG Micro-pitting tests
- FZG Efficiency test
- FE 8 Bearing tests

6.2 Analytical Data on Reference PAO

Table 6.2-1 shows analytical data of fresh PAO reference materials extracted from various reports written by project sub contract partner University of Dayton Research Institute (UDRI), these reports are provided as Appendix documents 2A, 2B & 2C. Notable differences exist in the Pour Point, Water and Silicon content. The later could be an indication of different levels of Anti-Foam additive or contamination. For further analysis the data were averaged in order to compare them with samples from different test procedures and in service. The last column shows viscosity measurement prior to the FZG Efficiency testing indicating some laboratory variation.

	ASTM No.	PAO procured in the US	PAO procured in Germany	PAO provided by Clipper	Average	Viscosity data of Germany procured material measured by University of Munich
Appearance			Clear			
Color			Reddish-Brown			
Viscosity (cSt)	D445					
40°C		324	330.1	328.3	327	349
100°C		34.2	32.9	33.9	34	37
Viscosity Index	D2270	149	140	146	145	153
Pour Point	D97		-18°C	-40°C		
Karl Fischer Water (ppm)	D6304	395	146	297	279	
Acid Number (mg KOH/g of oil)	D664	3.4	3.7	3.4	3.5	
ICP (ppm)	D5185					
Fe		2	2.6	2.4	2.3	
Cu		<1	3.1	2.5	2	
Si		12	15.7	0.9	10	
B (additive)			5	31	18	
Ca (additive)		16	20	18	18	
Mo (additive)		968	1447	1393	1269	
P (additive)		1138	1227	1297	1221	
Zn (additive)		1163	1659	1747	1523	
RULER	D6971					
Active ZDDP		100%	100%	100%	100%	
Phenolic Antioxidant			100%	100%	100%	

Table 6.2-1: Analytical Data from Fresh PAO Reference Oil

Table 6.2-2 lists the analytical data produced at UDRI for the in-service oil samples obtained from four wind turbines designated: T-A, T-B, T-C and T-D in November 2010, April 2011 and September 2011.

Except for the November 2010 T-C sample which contained high levels of water and a visible residue, all of the analyzed in-service oil samples are considered acceptable based on the guidelines (see that column) suggested by ISO 81400-4:2005 (E). Cautionary water levels were detected in the November 2010 T-D sample and in all of the September 2011 samples.

All of the in-service oils had viscosities at 40 and 100°C similar to the new PAO Reference oil except for the April and September 2011 T-A samples. Whether the lower viscosities of the April and September T-A oils were due to shear, wrong top-off oil, additive depletion, etc. is unknown, however, the viscosity index was stable for all of the in-service oils compared to the new oil.

Compared to the new oil, all of the in-service oils had lower acid numbers (due to additive depletion). The only acid number increases that occurred during the project were due to oil top-offs (acidic additives) and not oxidation (carboxylic acids).

All of the in-service oil samples had much lower ZDDP levels than the new oil. The ZDDP levels increased with use indicating significant oil top-offs.

ICP analyses detected low levels of iron (Fe) and copper (Cu) wear metals in all of the analyzed samples. Filtration with 3 microns did not isolate wear particles from any of the in-service oils, only additive particles from the November 2010 T-C oil. Additional 0.45 micron filtration of the September samples, isolated additive particles from the different oils except for T-B (clean filter). The 0.45 micron filters also isolated detectable submicron Fe (contained inside pores) from the T-A and T-B samples. Potential larger wear particles are probably removed from the oil by the 7 micron inboard filter.

ICP and XRF analyses indicate that the additive levels in the new and in-service oils are very similar except for the low levels of molybdenum (Mo) in the T-B oil series (T-B Mo levels increased with use indicating significant oil top-offs).

Service records of the turbines indicate that in most cases there was a significant top up of oil a week or so after the initial sample. The records are a little bit unclear as the maintenance contract for these turbines changed within the sampling period, so in summary the takeaway is significant amounts of oil are being added to the gearboxes and could account for the increasing levels of ZDDP in the samples. Therefore a quantitative interpretation of the analytical data down to an exact correlation would be too speculative. However to draw some qualitative conclusion from the field sampling an average of 12 in-service oils presents statistically a good reference for in use status, which will be discussed in comparison to gearbox and gear bench testing.

		Turbine A			Turbine B			Turbine C			Turbine D			Guidelines[a]		average
Tests	ASTM No.															
		Sample	Sample	Sample	Sample	Sample	Sample	Sample	Sample	Sample	Sample	Sample	Sample	Caution	Bad	
		Nov-10	Apr-11	Sep-11	Nov-10	Apr-11	Sep-11	Nov-10	Apr-11	Sep-11	Nov-10	Apr-11	Sep-11			
Viscosity (cSt)	D445													% Change		
40°C		324	307	308	328	326	325	324	324	318	322	323	322	>10	>20	321
100°C		33.4	32.4	32.1	34	33.9	33.4	32.6	33	32.7	33.3	33.2	32.9			33
Viscosity Index		145	147	145	147	147	145	141	143	144	146	145	143			145
Karl Fischer Water (ppm)	D6304[b]	494	173	787	312	442	731	1102	495	821	504	450	890	>500	>1000	600
Acid Number (mg KOH/g of oil)	D664	3.1	2.1	3.1	1.5	1.5	1.9	3	2.2	2.8	3	1.9	2.5	% Increase 40	% Increase 75	2
3 µm Filter[c]	None													None	Visible	
Photograph		Clean	Clean	Clean	Clean	Clean	Clean	Residue	Clean	Clean	Clean	Clean	Clean			
Metal Particles		No	No	No	No	No	No	No	No	No	No	No	No			
XRF		-	-	-	-	-	-	Mo, Zn	-		-	-				
SEM/EDS		-	-	-	-	-	-	Mo, Mg, Zn, P, S	-		-	-				
ICP (ppm)[d]	D5185															
Fe (wear)		38	31	39	63	37	39	26	20	19	17	13	15	75-100	>200	30
Cu (wear)		2	2	2	4	3	3	2	2	2	1	1	1	50-75	>75	2
Si (dirt)		13	10	6	10	7	3	12	9	3	13	8	3	15-20	>20	8
Mo (additive)		847	845	1040	152	330	354	717	813	852	811	683	765	-	-	684
Ca (additive)		13	14	19	2	5	6	11	11	14	19	15	23	-	-	13
Zn (additive)		1150	1246	1673	1145	1137	1290	1036	1069	1184	1202	1038	1271	-	-	1203
P (additive)		1142	1243	1200	1022	1022	813	1060	1041	812	1240	1040	908			1045
RULER	D6971															
Active ZDDP		<1%	5%	11%	<1%	14%	0%	<1%	9%	10%	<1%	5%	10%			6%
0.45 µm Filter[e]	None															
Photograph		-	-	Stain	-	-	Clean	-	-	Stain	-	-	Stain			
Metal Particles		-	-	No	-	-	No	-	-	No	-	-	No			
XRF		-	-	Fe, Mo, Zn	-	-	Fe, Mo	-	-	-	-	-	Mo, Zn			

Table 6.2-2: Analytical Data from Reference Oil Field Samples

In order to study degradation, Table 6.2-3 shows the analytical data for the PAO reference oil in fresh condition and after laboratory FZG micro-pitting gear, full scale gearbox and turbine field stressing.

Appearance was only recorded before and after the micro-pitting test and showed no impact. The viscosity increased slightly after the micro-pitting test and decreased slightly for the averaged data from the field samples. However, after the Gearbox Trial the viscosities of the stressed samples decrease with increasing test time (18 – 22%) along with the viscosity index.

The micro-pitting test and the averaged field data had no effect on the pour points while it increased after the gearbox trial with test time. Water content increased for all 3 stress methods except after the load stage 4 which can be attributed to the overheating that took place (see gearbox trial section). The Micro-pitting test had no effect on the acid numbers while it decreased after the gearbox trial and with the averaged field samples. The decrease of the acid numbers is in line with the decrease of active ZDDP anti wear additive levels, which are responsible for higher acid number in the fresh material. The depletion and removal of the acidic anti-wear additive zinc dialkyldithiophosphate (ZDDP) is determined by the RULER analyses (reference).

Except for the November 2010 T-C sample which contained a visible residue all other analyzed PAO Reference oils showed no residues after passing through a 3 micron pore size filter. Therefore, filters with a 0.45 micron pore size were used in an attempt to isolate any submicron particles from the stressed oils. The x-ray fluorescence (XRF) analyses of the insoluble particles removed from the stressed oils indicate that the particles were formed by degradation of the ZDDP and a Molybdenum containing additive.

To further characterize the small (< 3 micron) particles present in the oil after the Stage 4 test, a section of the filter was removed and analyzed by scanning electron microscope/energy dispersive spectrometry (SEM/EDS). The EDS elemental analyses of the isolated particles indicate that they are products of additive degradation (C, O, Mg, Mo, P, S and Zn) and do not contain wear particles (Cu or Fe). Surprisingly, the particles do contain a significant (~20%) concentration of F (contaminant from previous PFPE gearbox test?). The Fourier Transform Infrared (FTIR) spectra overlay of the oil series shows that only the 1700 – 1500 wave number region of the spectra (thought to be related to the oils' additive package) had a detectable (decreased) change with increasing test time, i.e., additive depleted, while base stock of oils unaffected, by increasing test time. The FTIR spectra of the Stage 4 residue confirmed this.

The Inductively Coupled Plasma (ICP) atomic emission analyses (detect particles below 6 microns) of all the PAO reference samples showed minimal wear particles, except for higher Cu levels after the micro-pitting test. Determination of additive levels with ICP shows depletion for the load stage 4 and for the averaged field samples.

Tests	ASTM No.	Before FZG	After FZG	Before Gearbox Trial	After Load Stage 3	After Load Stage 4	Average fresh oil	Average field samples
Appearance		Clear	Clear					
Color		Reddish-Brown*	Reddish-Brown*					
Viscosity (cSt)	D445							
40°C		330.1	336.1	328.3	307.3	267.1	327.5	320.9
100°C		32.9	34.9	33.9	32.3	25.5	33.7	33.1
Viscosity Index	D2270	140	148	146	146	123	145	145
Pour Point	D97	-18°C*	-18°C	-40°C	-40°C	-30°C		
Karl Fischer Water (ppm)	D6304[a]	146	202	297	449	333	279	600
Acid Number (mg KOH/g of oil)	D664	3.7	3.7	3.4	2.7	2.4	3.5	2.4
Filter Photograph [b]	None	Clean	Stain	Clean	Stain	Dark		stain
Filter Elemental Content 0.45 um	None							
XRF		Ca, Fe	Ca, Cu, Fe, Mo, Zn	Fe, Mo	Mo, Zn	Mo, Zn		Mo, Zn
SEM/EDS				-	-	C(55%), O(20%), F(21%), S(1.5%), Mg, Mo, P, Zn (<1%)		
ICP (ppm) [c]	D5185							
Fe (wear)		2.6	3.7	2.4	3	2.4	2.3	29.8
Cu (wear)		3.1	28.2	2.5	1.9	1.2	2.2	2.1
Si (dirt)		15.7	5.8	0.9	0	0	9.5	8.1
B (additive)		5	1	31	7	6	18	
Ca (additive)		20	19	18	20	14	18	13
Mo (additive)		1447	1246	1393	1473	940	1269	684
P (additive)		1227	1081	1297	1419	956	1221	1045
Zn (additive)		1659	1514	1747	1955	1370	1523	1203
RULER	D6971							
Active ZDDP		100%	80%	100%	20%	0%	100%	8%
Phenolic Antioxidant		100%	90%	100%	100%	100%	100%	

Table 6.2-3: PAO Reference Oil Analysis After Various Stressing

6.3 Analytical Data on Candidate PFPE

Table 6.3-1 compares the data obtained for the various PFPE samples. The first data column shows the fresh material before micro-pitting test. The second column shows viscosity data from the same sample measured at the Technical University Munich for comparison with notable variance in the viscosity index. The third column represents the data after the micro-pitting test. The fourth column shows data after28 hours of base line run gearbox testing, basically at the beginning. The fifth column represents the data after the end of the base line gearbox run and with the sixth column after the gearbox test was completed. That sample's viscosity was measured for comparison at the Technical University of Munich with a notable difference of the lower temperature viscosity as shown in the eights column. Column nine shows the data of the sample that has undergone the full scale gearbox trial with a subsequent micro-pitting test.

The PFPE fluid underwent changes in appearance/color (milky/white) during the Micro-pitting test, as well as during the gearbox trial (hazy opaque/gray) compared to the fresh fluid (clear/colorless). The used PFPE fluid from the previous full-scale gearbox trial underwent minimal changes in appearance and color after another Micro-pitting test.

Significant thinning occurred with a viscosity drop of 17% from the Micro-pitting test and 19% after the gearbox trial with another 15% after an additional Micro-pitting test. However the viscosity index increases slightly for the PFPE fluid series. Thus, the high viscosity index advantage of the PFPE fluids over the PAO reference oil increased during the gearbox test.

Due to the very low pour point of the PFPE fluids, the -70°C low temperature bath was insufficient to determine the pour point of the fresh fluids, after 28h base line gearbox testing and of the sample after the full scale gearbox trial before Micro-pitting test. However, the fluids after Micro-pitting tests had measurable pour points of -57 and -65°C and also the samples after 420h base line and after completed gearbox trial had measurable pour points of -66 and -63°C. Some samples required 3-4 seconds before a flow was observed (effect of insolubles?) whereas the new “before” fluid flowed immediately. Having a decrease in pour point of the same sample shown in columns six and seven might indeed be an effect of different storage conditions.

The water levels were below the detection limits of the Fourier Transform Infrared (FTIR) analyses which were used to confirm the fluids were dry since they fouled the Karl Fisher technique. All non-PFPE materials in this study did contain water in ppm levels from the low hundreds to over 4000.

The PFPE fluids experienced minimal acid accumulation during the gearbox test. A modified RULER acid test confirmed that the PFPE fluids contained a very low level of acids. However, the micro-pitting test appeared to have an effect on the PFPE fluid acidity as can be seen by an increase in the RULER numbers. Further evidence was found by some corrosion observed on the inner lids of the cans containing the fluids before and after the micro-pitting test, which might be caused by enclosed gases over the fluid level to react with the cans tin plating. However, washing of the PFPE fluids with water did not detect any HF in the water.

X-ray Fluorescence (XRF) analyses (detect particles below 20 microns) of the PFPE fluids detected no metal content (20 – 30 ppm detection limits), i.e., minimal wear particles detected in fluids.

None of the PFPE fluids contained particles larger than 3 microns. Therefore, filters with a 0.45 micron pore size were used in an attempt to isolate any submicron particles from the stressed fluids.

XRF analyses of the PFPE fluid filters detected a slowly increasing, although very low, level of Fe content with increasing test time. In addition to the Fe wear product, Ca was detected which is not explainable other than a contamination as well as Zn. Cu in the filters of the samples after the micro-pitting test could be explained by either wear from bearing cages or contamination. The detection of K can be explained by the partially functionalizing the fluid as well as can be the detection of P in SEM/EDS.

Further characterization of a small (< 3 micron) particles present in the PFPE fluid after the Stage 6 test, a section of the filter was removed and analyzed by SEM/EDS. The SEM microphotograph of the particles in Figure 3 shows that the particles are submicron (a gelatinous material was also isolated) and cover the entire surface of the sectioned piece of filter. The EDS elemental analysis of an isolated particle of the fluids deposit also indicates the extraction of the partly functionalized fluid.

Tests	ASTM No.	Before FZG	Result from Munich before efficiency	After FZG	After 28 hours of Baseline Run	After 420 hours of Baseline Run	After Load Stage 6	From Gearbox Trial before FZG	Result from Munich before efficiency	From Gearbox Trial after FZG
Appearance		Clear		Milky	Clear	Hazy	Opaque	Hazy		Opaque
Color		Colorless		White	Colorless	Tan	Gray	Gray		Gray
Viscosity (cSt)	D445									
40°C		220.1	225	184.9	237.8	190.7	183.5	182.8	196	156.7
100°C		62.5	69	51.6	61.4	54.1	52.7	51.8	56	43.5
Viscosity Index	D2270	337	356	327	317	332	334	330	335	321
Pour Point	D97	<-67°C		-57°C	< -67°C	-66°C	-63°C	<-67°C		-65°C
Karl Fischer Water (ppm)	D6304[d]	-		-	Fouled Electrode	Fouled Electrode	Fouled Electrode	-		-
Acid Number (mg KOH/mL of oil)	RULER Test/D664	0.7		1.4	0.3	0.3	0.3	0.8		1.3
Filter Photograph[e]	None	Clean		Stain	Clean	Clean	Slight Residue	Stain		Stain
Filter Elemental Content[e]	None									
XRF		-		Ca, Cu, Fe, Zn	Ca, Fe, K	Ca, Fe	Ca, Fe	Ca, Fe		Cu
SEM/EDS		-		Ca,Cu,F, Fe,K,P, Zn	-	-	C(81%), O(15%), F(4%)	-		-
XRF of Fluid[f]	None									
Fe (wear)		BDL[g]		BDL.	BDL[g]	BDL.	BDL	BDL		BDL
Cu (wear)		BDL.		BDL.	BDL.	BDL.	BDL.	BDL.		BDL
Si (dirt)					BDL.	BDL.	BDL.			

Table 6.3-1: Analytical Data of Fresh & Stressed PFPE Fluid

6.4 Analytical Data on other Benchmark Fluids

Table 6.4-1 shows the analytical data obtained for PAO Alternative, Ester and PAG oils before and after micro-pitting test. None of the analyzed oils or fluids contained particles larger than 3 microns. Therefore, filters with a 0.45 micron pore size were used in an attempt to isolate any submicron particles from stressed oils.

For all the other benchmark oils, the gear test caused minimal changes in the viscosity (0 – 3%), appearance (clear to slightly hazy) or color (slightly darker) of each tested oil.

For the PAO Alternative oil, the micro-pitting test had no effect on the pour points with a slight increase in the water content of the stressed oil. For the Ester oil, the pour point increased slightly while the water content decreased. In contrast, for the PAG the pour point decreased (improved) as the high water content of the "before" oil was decreased by the micro-pitting test.

For all of the other benchmark oils, the micro-pitting test had minimal effect on the acid numbers of the oils. In agreement with the acid numbers, the micro-pitting test had minimal effect on the additive contents of the oils.

Inductively Coupled Plasma (ICP) atomic emission analyses (detect particles below 6 microns) only detected significant Fe in the PAG "after" oil. X-ray Fluorescence (XRF) analyses (detect particles below 20 microns) of the oils showed minimal wear particles.

Therefore, an additional set of thinned oils were passed through 0.45 micron filters to isolate submicron particles (only 8-14 grams of thinned oil/fluid could be passed through prior to clogging the filters). The qualitative XRF analyses detected a low level of Fe in the PAO alternative "after" filter. In contrast to the PAO alternative oil, the XRF analyses of the Ester and PAG oils did not detect any metallic species on the "after" filters. Therefore, the Fe detected in the PAG "after" oil by the ICP must be present as dissolved species since Fe was not retained by the 0.45 micron filter.

The FTIR spectra of the "before" and "after" oils are very similar.

		PAO Alternative		Ester		PAG		
Tests	ASTM No.	Before	After	Before	After	Before	Result from Munich before efficiency	After
Appearance		Clear	Hazy	Clear	Clear	Clear		Hazy
Color		Pale Yellow	Yellow	Yellow	Orangish-Brown	Yellow		Organish-Brown
Viscosity (cSt)	D445							
40°C		330.3	329.5	332.9	332.8	301.4*	314	309.9
100°C		37.5	37.2	36.9	36.6	52.9	55	51.5
Viscosity Index	D2270	162	161	159	157	241	242	231
Pour Point	D97	-30°C	-30°C	-24°C*	-18°C	0°C*		-9°C
Karl Fischer Water (ppm)	D6304[a]	143	193	362	272	4,194		3,754
Acid Number (mg KOH/g of oil)	D664	0.73	0.58	0.85	0.78	2.4		2.5
Filter Photograph[b]	None	Clean	Clean	Clean	Clean	Clean		Stain
Filter Elemental Content[b]	None							
XRF		-	Fe	-	-	Ca		-
ICP (ppm)[c]	D5185							
Fe (wear)		<0.1	2.1	<0.1	0.7	0.1		57
Cu (wear)		1.2	1.1	0.1	0.6	5.2		7.4
Si (dirt)		11.7	<0.1	<0.1	<0.1	<0.1		<0.1
B (additive)		<0.1	<0.1	47	30	6.2		16
Ca (additive)		0.8	0.9	1.8	2.1	3.9		42
Mo (additive)		4.8	1.9	<0.1	5.3	1.2		42
P (additive)		368	348	125	171	1254		4107
Zn (additive)		25	21	2.3	15.2	14		99
RULER	D6971							
Aminic Antioxidant				-	-	100%		100%
Active ZDDP		-	-					
Phenolic Antioxidant		100%	75%	100%	100%	-		-

Table 6.4-1: Others Benchmark Oils Before & After Micro-pitting Test

6.5 FZG Micro-pitting Tests

About the subject micro-pitting, as stated by FZG (Technical University of Munich):

"Micro-pitting occurs on tooth flanks with a high surface hardness under unfavorable lubrication conditions. Oil viscosity, additives, peripheral speed, tooth flank surface (processing, roughness), heat treatment as well as temperature influence the damage occurrence, development and intensity. First indications of the occurrence of micro-pitting are determined often after few load cycles at very low loads; that indicates the beginning of damages of wear type. Cracks and material pits in the further damage progression point out the fatigue character of the micro-pitting failure. Micro-pitting occurs on the flank surfaces normally first on the dedendum flank of the driving gear. It propagates gradually over the flank and covers in extreme cases the whole active flank surface."

"Forms of micro-pitting failure and consequential damages:"

- "Micro-pitting is a surface damage. It can cause profile deviations of wear type on the active flanks (Figure 6.5-1). In this case the dynamical additional forces and the gear noise can increase."
- "The immediately resulted small pits, that give the flank its grey appearance, can build large flat pits."
- "Some of the many small cracks can propagate and ramify. As a result, large deep triangular particles can break out. That is the way of occurrence of pitting and spalling, that in many cases, can reach the tip edge of the gear."

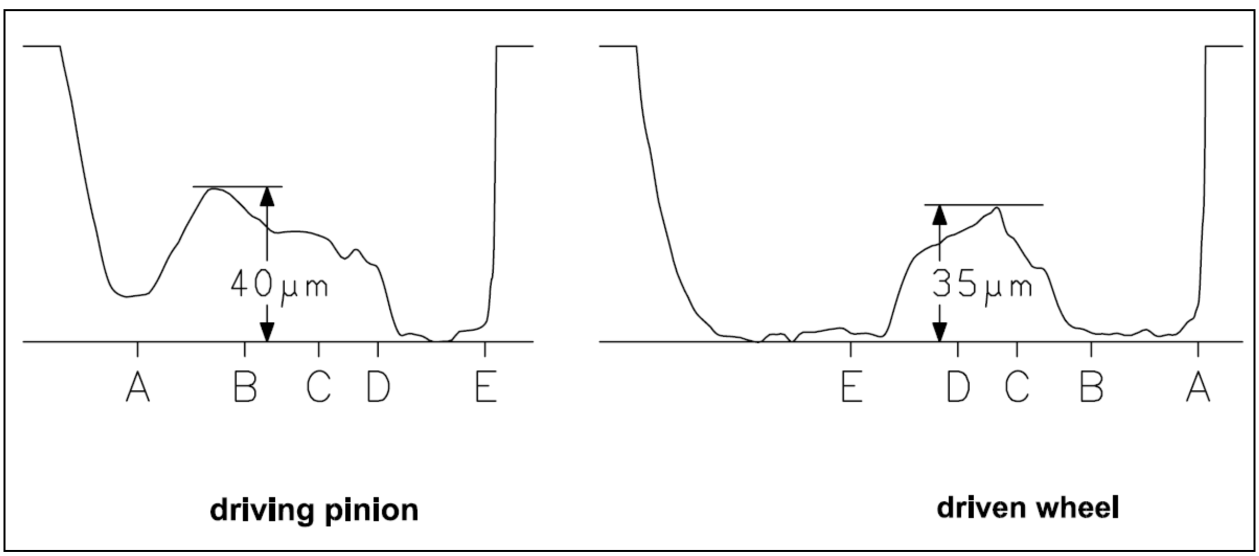


Figure 6.5-1: ***The profile deviation caused by the micro-pitting after approx. 10^8 load cycles at Hertzian contact stress pc of approx. 1300 N/mm^2 (measured on the involute measurement machine), A and E are the contact points, C is the center with no sliding – example from appendix***

"The FZG-micro-pitting test provides a quantitative evaluation of the influence of lubricant (especially additives), lubricant temperature and other parameters on the occurrence of micro-pitting. The micro-pitting test differentiates oils according to their micro-pitting load capacity and enables the choice of a lubricant with a sufficient micro-pitting resistance. The FZG-micro-pitting test consists of a load stage test with incremental increasing of the contact stress and an endurance test. A detailed description of the test method according to FVA 54 can be found in Appendix 2E."

It should be noted that the condition described above for micro-pitting is tested for in the laboratory under controlled circumstances using specified test gears and procedure. It is the results and discussion of these laboratory tests which follows:

Test reports from the University of Bochum for the 6 materials described in above chapters are in Appendix documents 2G-2L, for ease of presenting the results the 3 values taken were:

- Average Profile Form Deviation (as indicated in Figure 6.5-1)
- Micro-pitting area (the lower dedendum section in Picture 6.5-3)
- Weight loss

Summarized in Table 6.5-2 are the results from the 6 reports mentioned above, while a reference report (Appendix 2E) from the University of Munich has been added in the 5th data column.

Note: University of Munich performed preliminary testing. Lab availability and scheduling required University of Bochum to be used as a second external laboratory to perform the primary comparative testing for micro-pitting in this project task.

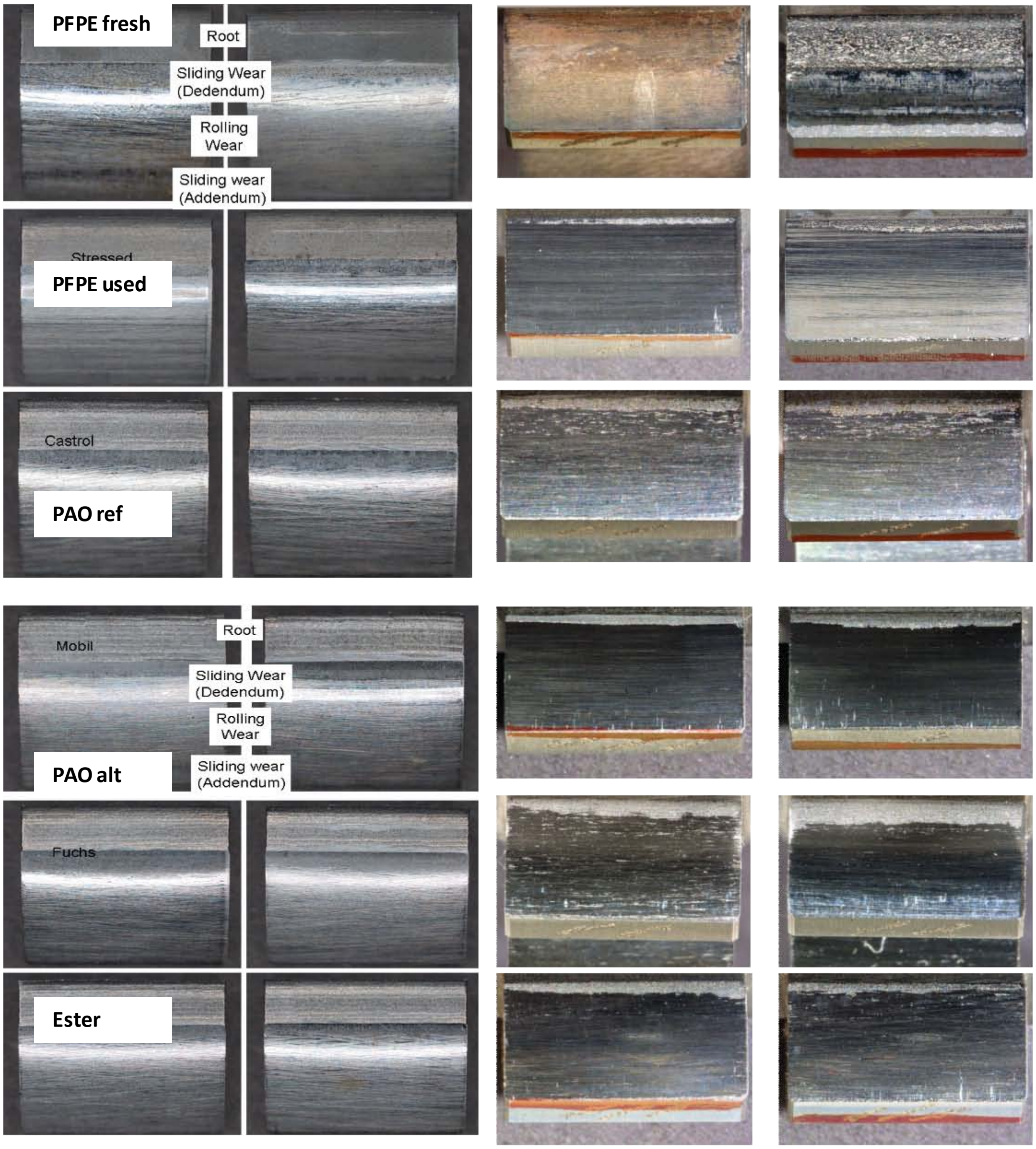
	PAO ref	Ester	PAG	PAO alt	PFPE*	PFPE**	PFPE***
Average Profile Form Deviation [ug]							
Load stage test 1	8.5	8.7	7.3	11.3	5.1	4.8	5.7
Load stage test 2	5.2	8.3	6.0	11.0	4.8	6.0	7.0
Endurance test	8.0	10.7	7.3	12.3	12.7	16.7	10.8
Micropitting % Tooth Flank Area							
Load stage test 1	12.0	15.0	11.0	9.0	5.0	5.0	5.0
Load stage test 2	11.0	13.0	7.0	10.0	7.0	5.0	6.0
Endurance test	14.0	25.0	11.0	12.0	22.0	31.0	11.0
Weight Loss Amounts [mg]							
Load stage test 1	11	14	12	14	15	3	4
Load stage test 2	6	13	12	11	15	13	6
Endurance test	17	26	97	21	36	51	24
Resulting Failure Load Stage	10	10	>10	9	>10	>10	>10
Resulting Endurance GFT Class	High	High	High	High	High	High	High
* Fresh PFPE laboratory sample tested in Munich							
** Fresh PFPE, Results reported but Endurnace Test Aborted due to Macropitting							
*** Used PFPE from Clipper Gearbox Trial							

Table 6.5-2: Summary of Micro-pitting Test Results

UDRI carried out surface analysis for the teeth cut from gears used in the micro-pitting gear test performed at the University of Bochum. In an effort to both qualitatively and quantitatively measure the surface wear/damage of the selected gear teeth, the following surface analysis techniques were used:

- Photography
- Scanning Electron Microscope/Energy Dispersive Spectrometer (SEM/EDS)
- Optical Profilometer

The gears were unwrapped and initially examined for surface damage. For each gear, the tooth with the highest level of apparent wear was cut at the root diameter of the gear and then length-wise producing two surfaces suitable for analysis. The pictures of each tooth side are put next to the pictures taken at the University of Bochum for all the six different oils/fluids tested are summarized in Picture 6.5.-3.



PAG

Picture 6.5-3: Gear Tooth Images taken at UDRI & University of Bochum

At the start of the micro-pitting test, 3 gears are marked that are used for the inspection. After the first load stage test (3rd column of images in Picture 6.5-3) the gears are turned on the shaft, so the opposite side is used for the 2nd load stage and endurance test (4th column of images in Picture 6.5-3). While the images in columns 1 and 2 are random, they show a section of the tooth section at the top of each picture, the images from columns 3 and 4 show the micro pitted area at the top of each picture.

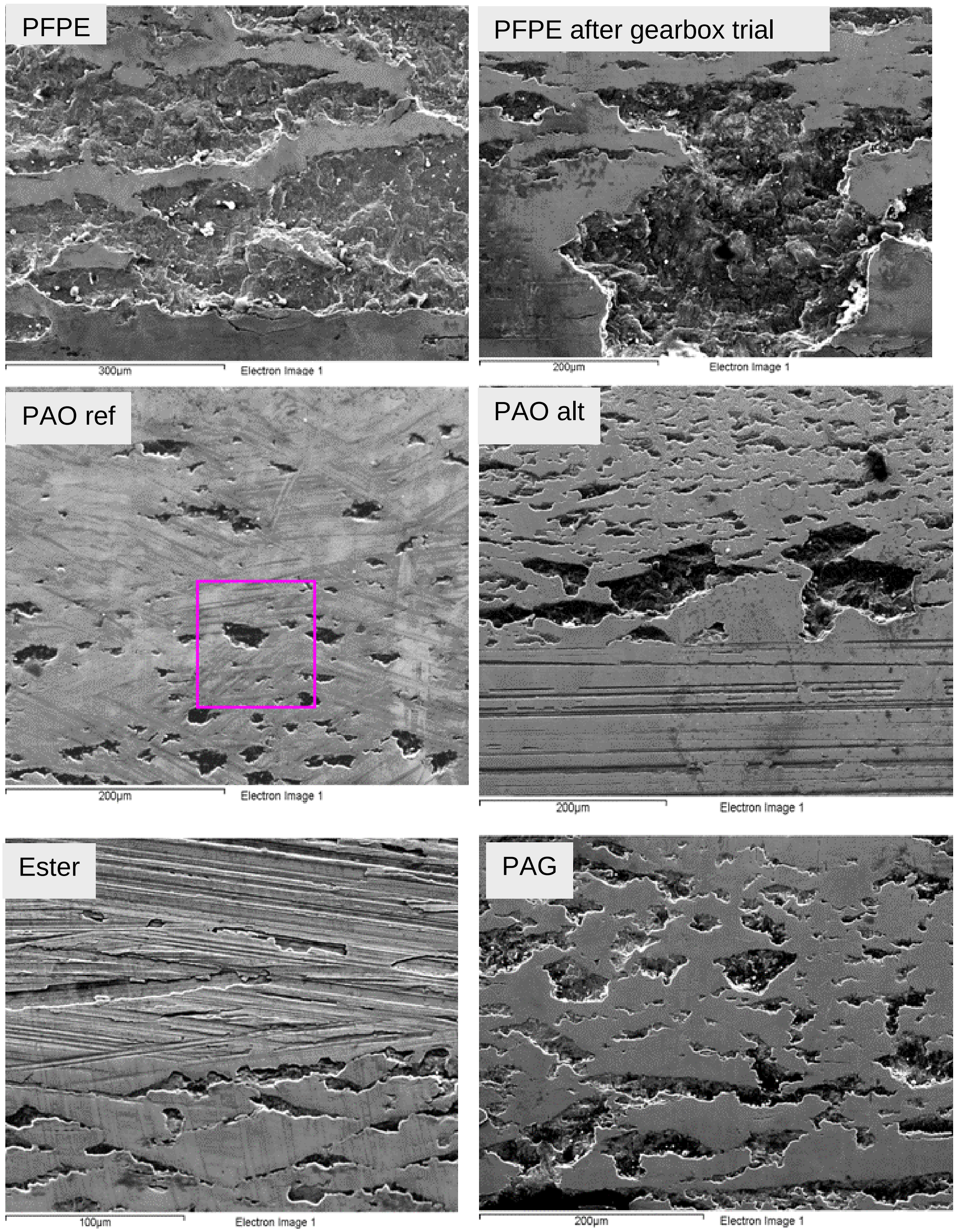


Figure 6.5-4: ***SEM Microphotographs of the Sliding Wear (Dedendum) Region of the Gear Tooth Face***

Under higher magnification in Figure 6.5-4, all of the sliding wear regions contain a significant number of pits though it is not fully visible if the micro pitted area is chosen for the close up. For all of the tested oils and fluids, the high magnification SEM microphotographs in Figure 6.5-5 show that the teeth have much lower levels of wear/no pits in the rolling contact regions. The wear in the rolling contact regions appears as bidirectional lines in the higher magnification SEM microphotographs in Figure 6.5-5.

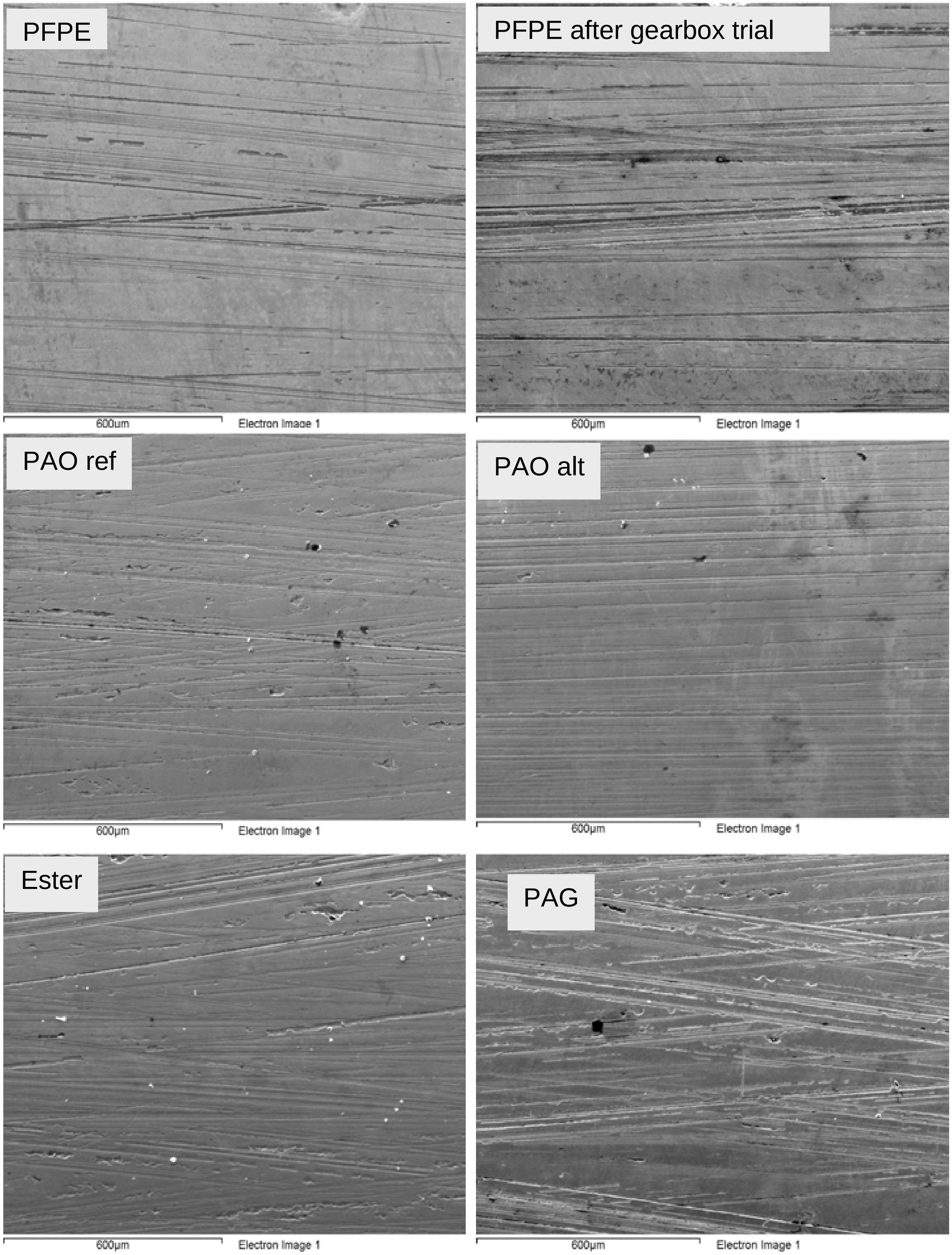
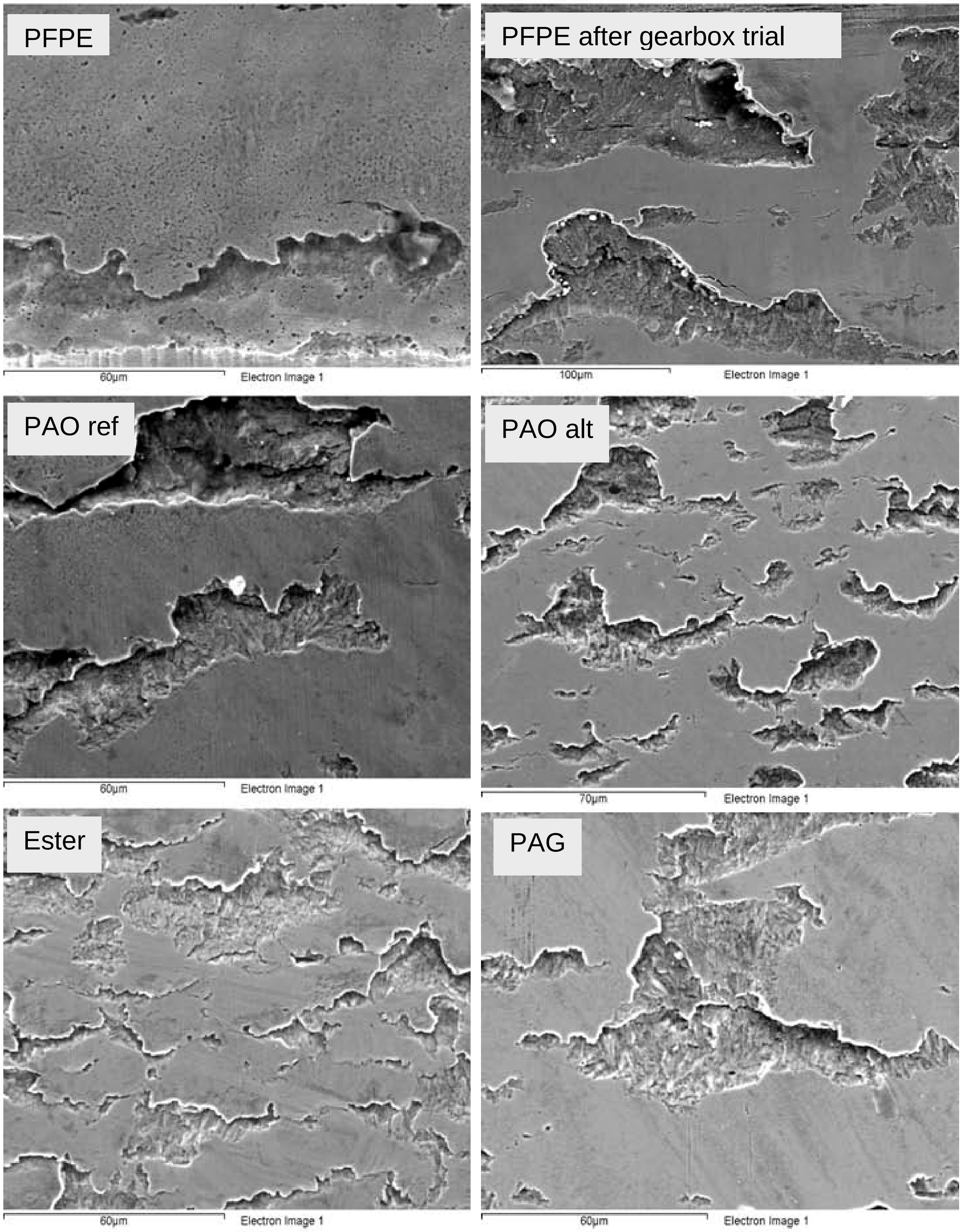


Figure 6.5-5: SEM Microphotographs of the Rolling Contact Region of the Gear Tooth Face

Similar to the dedendum sliding wear regions, the high magnification SEM microphotographs in Figure 6.5-6 show that the teeth have extensive wear/pits in the addendum sliding wear regions for all of the tested oils and fluids.



Picture 6.5-6: ***SEM Microphotographs of the Sliding Wear (Addendum) Region of the Gear Tooth Face***

In summary all tested oils and fluids have high micro-pitting load carrying capacity. For materials with medium or low micro-pitting load carrying capacity the gears would show micro-pitting in the addendum, dedendum and rolling area, as schematically shown in Figure 6.5-1. The key difference between the PFPE fluid and the other 4 benchmark oils is that the PFPE fluids do not contain Extreme Pressure additive and bring the micro-pitting load carrying capacity from the inherent base fluid performance. In addition the test results for the PFPE fluid that has undergone the full scale gearbox trial performs even slightly better than the fresh material.

Since the gear tooth surfaces for the PFPE fluids had a difference appearance than those from the other 4 oils, EDS elemental maps were performed. Table 6.5-7 summarizes the Atomic percentages found for all oils and fluids of the tooth and sliding (dedendum) surfaces.

	Atomic %											
	PAO Ref		**PAO Alt**		**Ester**		**PAG**		**PFPE**		**PFPE***	
	Root	***Slide***	***Root***	***Slide***	***Root***	***Slide***	***Root***	***Slide***	***Root***	***Slide***	***Root***	***Slide***
Element												
C	35.1	53.4	27.7	33.4	38.7	46.0	19.3	30.9	19.0	45.1	21.5	39.3
O	19.3		20.9		18.9		26.8		38.6	18.6	46.2	20.9
F									22.4		11.7	
P									5.2	0.7	6.1	1.3
K									0.1			
Ca											0.1	
Cr	1.1	0.8	1.2	0.9	1.0	0.7		0.8	0.3	0.5	0.5	0.5
Mn	0.8	0.7	1.2	0.8	1.0	0.7		0.8	0.3	0.5	0.3	0.5
Fe	43.7	45.2	48.0	65.0	40.4	52.6	54.0	67.5	13.7	34.1	13.1	37.2
Cu										0.3	0.3	0.3
Zn									0.3		0.2	
* material after full scale gearbox trial												

Table 6.5-7: EDS Elemental Map analysis of the Root and Sliding Wear (Dedendum) surfaces

The root areas for both PFPE fluids have a phosphate/oxide coating (P: 5-6% and O: 38 – 46%) while the root areas of the benchmark oils only contain oxide layers (O: 18 – 26%). In addition to the phosphate/oxide coating, the root areas of the PFPE fluid gear teeth also contain fluorides, higher for the new fluid (F: 22%) than the stressed fluid (F: 11%).

The sliding wear regions produced by the PFPE fluids have a thin phosphate (P: 0.5 – 1%) with a thicker oxide layer (O: 18 – 21%). In contrast, the dedendum sliding wear regions of the benchmark oils are metallic in nature (O not detected), so no tribological film could be detected.

The gear surface analysis report from UDRI (Appendix 2D) contains optical profilometer measurements to obtain width and depth on pits present in a 2 x 5 millimeter (mm) area of each tooth surface (root, sliding wear (dedendum) and rolling contact areas). The FZG-micro-pitting test procedure uses stylus profilometers which generate linear (2-dimensional) depth profiles over the entire tooth width of approximately 12 mm to track micro-pitting progression of 3 distinguished gear teeth. Without reference to the distinguished tooth with its micro pitted section it was difficult to correlate the 3-D images generated by optical profilometer (taken randomly) with those from the reports of the University of Bochum.

6.6 FZG Efficiency Test

The frictional losses of cylindrical gears are measured in a modified back-to-back gear test rig. The test pinion and the test gear are mounted on two parallel shafts which are connected to the slave gear stage. In the slave gear stage, compared to the standard FZG test rig, two identical gears to the test gears are mounted, so that two equal stages are closing the power circle. The pinion shaft consists of two separate parts, which are connected by the load clutch. By twisting the load clutch using defined weights (load stages) on the load lever a defined static torque is applied. The electric motor has only to compensate the frictional losses in the power circle. For the measurement of the loss torque a torque meter shaft is mounted between the electric engine and the slave gearbox. The applied load is measured with a load torque meter shaft next to the load clutch. During the test, different operating conditions are applied, the circumferential speed is set to vt = 0.5 m/s, 2.0 m/s, 8.2 m/s or 20 m/s, the load is set to no load, Hertzian stress of pC = 960 N/mm², 1340 N/mm² or 1720 N/mm² and the temperature is set to oil = 40 °C, 60 °C, 90 °C or 120 °C. The test method uses dip lubrication. A detailed description of the test method can be taken from the Information sheet in the annex of the Appendix 2F).

For this project the PAO ref oil, the PAG oil, the PFPE fresh fluid and the PFPE used (after the full scale gearbox trial) fluid were evaluated. Table 6.6-1 compares the different viscosities and densities of the materials with the FZG reference FVA3A (Mineral oil plus 4 % additive).

Lubricant	***η_{40} [mPas]***	***η_{100} [mPas]***	***ν_{40} [mm^2/s]***	***ν_{100} [mm^2/s]***	***ρ_{15} [kg/m^3]***	***VI***
PAO ref	298	30	349	37	870	153
PAG	328	55	314	55	1060	242
PFPE	408	123	225	69	1830	356
PFPE*	354	99	196	56	1820	335
FZG ref	84	9	95	11	885	95
* material after full scale gear box trial						

Table 6.6-1: Viscosity Data of Materials Tested for FZG efficiency

The measured loss torques of the different lubricants for an oil temperature of oil = 90 °C are shown in Figure 6.6-2 versus the pitch line velocity. The torque values of 3 load stages are plotted, whereas the torque values at no load are already subtracted and only the losses related to bearing and gear friction are displayed.

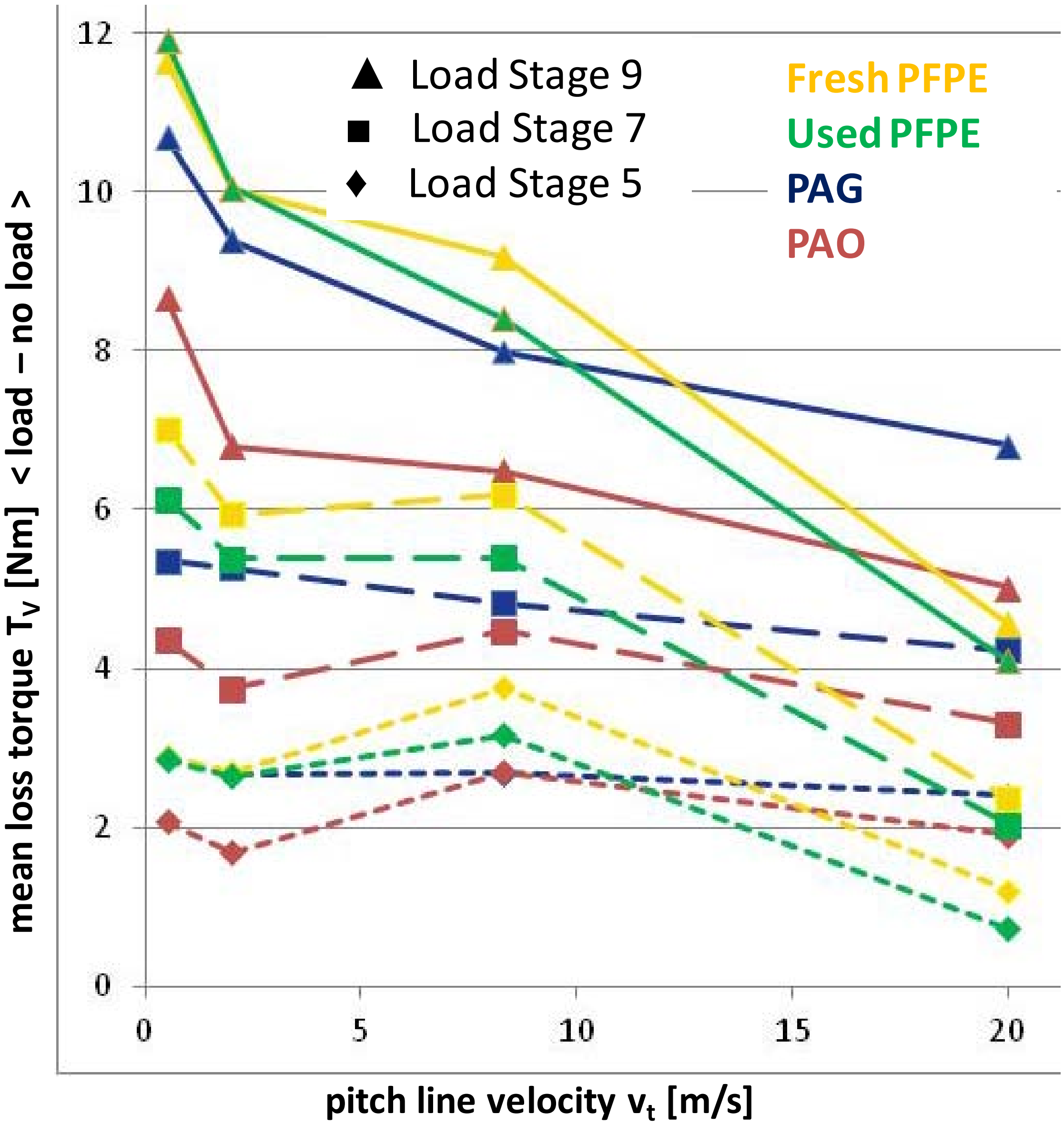


Figure 6.6-2: Measured Loss Torque at 90°C

In Figures 6.6-3, 6.6-4 and 6.6-5 the loss coefficients $X_{L0}(\ )$, $X_{LL}(\ )$ and $X_{LG}(\ )$, which indicate the relative losses of the candidate oils compared to the FZG reference oil FVA3A, are shown.

The no load loss coefficient $X_{L0}(\ )$ indicates the relative no-load losses compared to FZG ref. For lubricants with a higher viscosity than FZG ref the no-load loss coefficient usually shows values $X_{L0}(\ ) > 1$, for lubricants with lower viscosity values $X_{L0}(\ ) < 1$ are derived. In Figure 6.6-3, the measured no-load loss coefficients $X_{L0}(\ )$ of the 4 candidate lubricants depending on the temperature are shown. This factor reflects the lubricant viscosity in relation to the viscosity of the FZG reference oil as the main influence. Because of their different viscosities the four candidate oils show different loss behavior and slightly decreasing losses with rising temperatures. Candidates PFPE and PFPE* show highest no load losses with up to 330 % higher losses than the FZG reference oil, which has a much lower viscosity and density than the candidates.

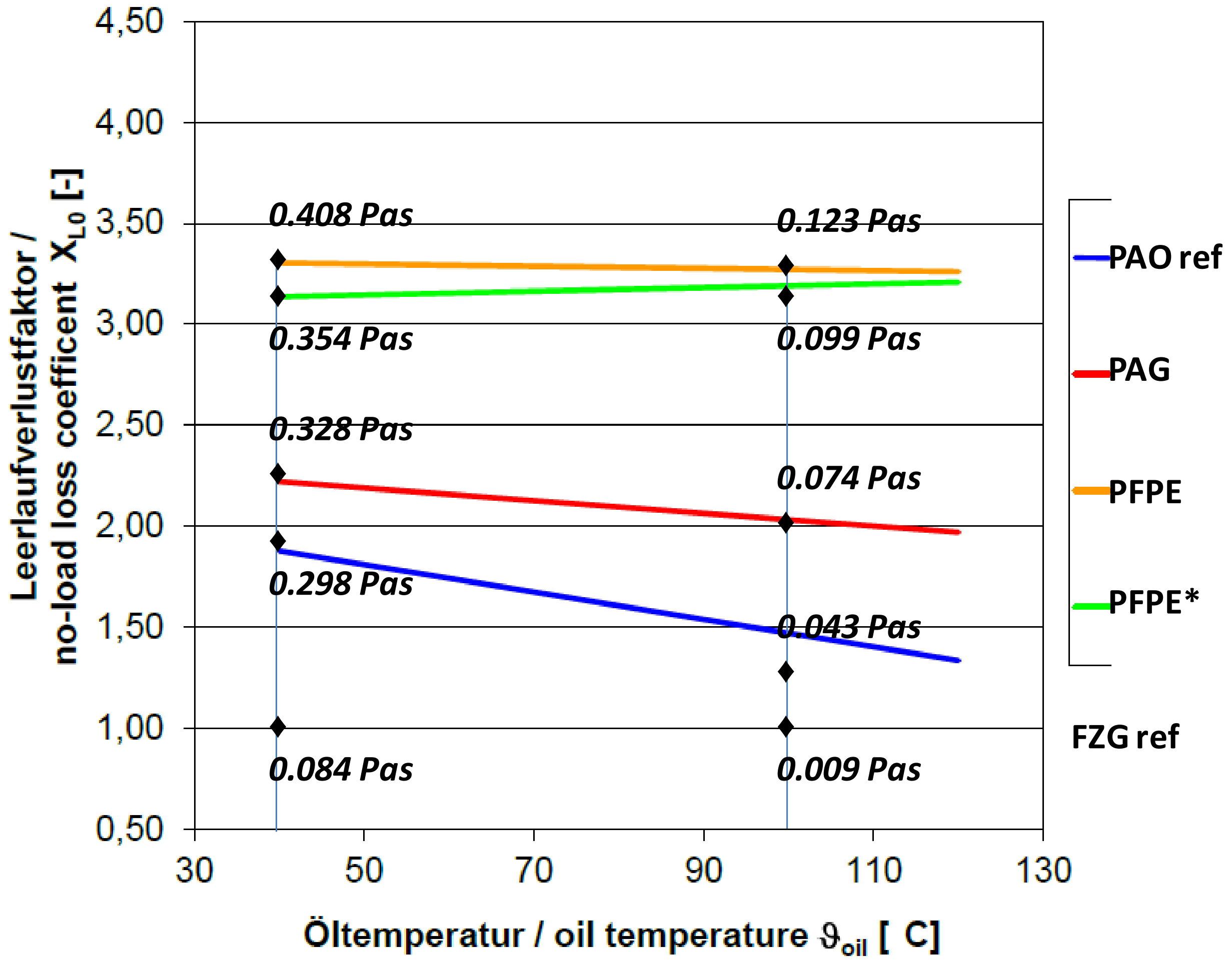


Figure 6.6-3: No Load Losses

The load loss coefficient X_{LL}( ) describes the frictional behavior at predominantly EHD-lubrication conditions and is calculated from the load dependent losses at operating conditions in the mixed lubrication and EHD-regime compared to the comparable operating conditions with the FZG reference oil. The load loss coefficient X_{LL}( ) expresses mainly the influence of the base oil on the frictional behavior of the lubricant. In Figure 6.6-4 the measured load loss coefficients XLL of the candidate lubricants are shown. Due to the type of base oil all candidates show loss coefficients lower than 1 decreasing with rising temperatures. The candidate PFPE fresh shows highest losses of all candidates about 2 % lower than the FZG reference oil at 40 °C. At 120°C oil temperature PFPE used shows lowest losses and PAO shows highest losses of all candidates, 31 to 12 % lower than the FZG reference oil. A significant difference between the candidates can be found at all temperatures.

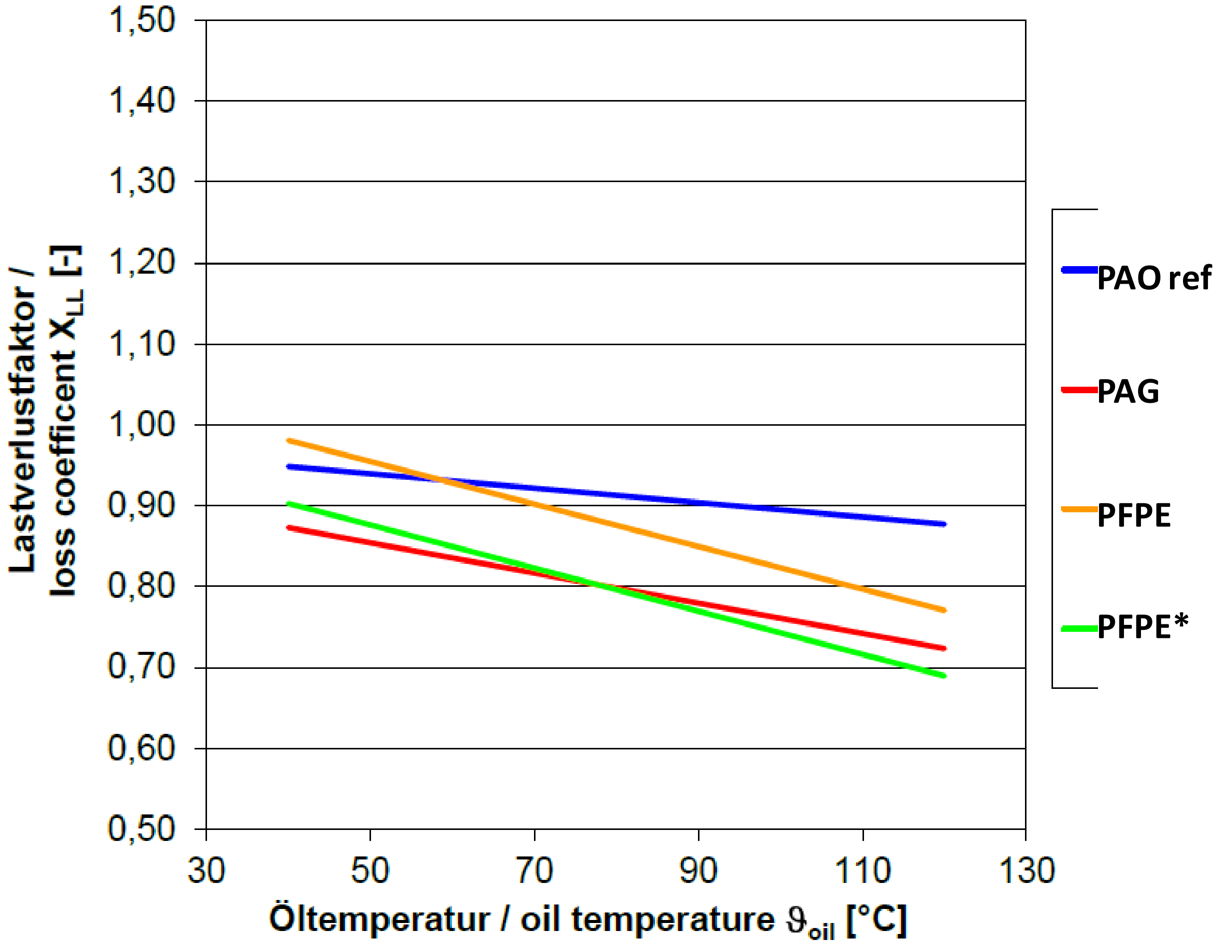


Figure 6.6-4: **Load Losses at Mixed and EHD Lubrication Conditions, each curve is fitted from 7 measurements**

The loss coefficient X_{LG}( ) describes the relative load dependent losses compared with the reference data of FZG reference oil at operating conditions where usually boundary lubrication occurs. The boundary loss coefficient X_{LG}( ) expresses mainly the influence of the additive system on the frictional behavior of the lubricant, but for higher viscous oils where also mixed lubrication can occur at the referring operating conditions, also the type of base oil influences this loss coefficient. In Figure 6.6-5 the measured boundary lubrication loss coefficients X_{LG}( ) of the candidate lubricants are shown. All candidates show the same frictional behavior at boundary lubrication with rising temperature. Overall PFPE fresh shows the highest losses and PAG shows the lowest losses of all candidates compared to the FZG reference oil. In most cases a significantly lower coefficient of friction in the boundary lubrication regime is hence stated in comparison to the Sulfur-Phosphorus-additive system of the FZG reference oil. This is even more remarkable in the case of the PFPE fluids which contrary to the PAO and PAG oils do not contain any Extreme Pressure additives and the boundary friction behavior is a result of the inherent base fluid chemistry.

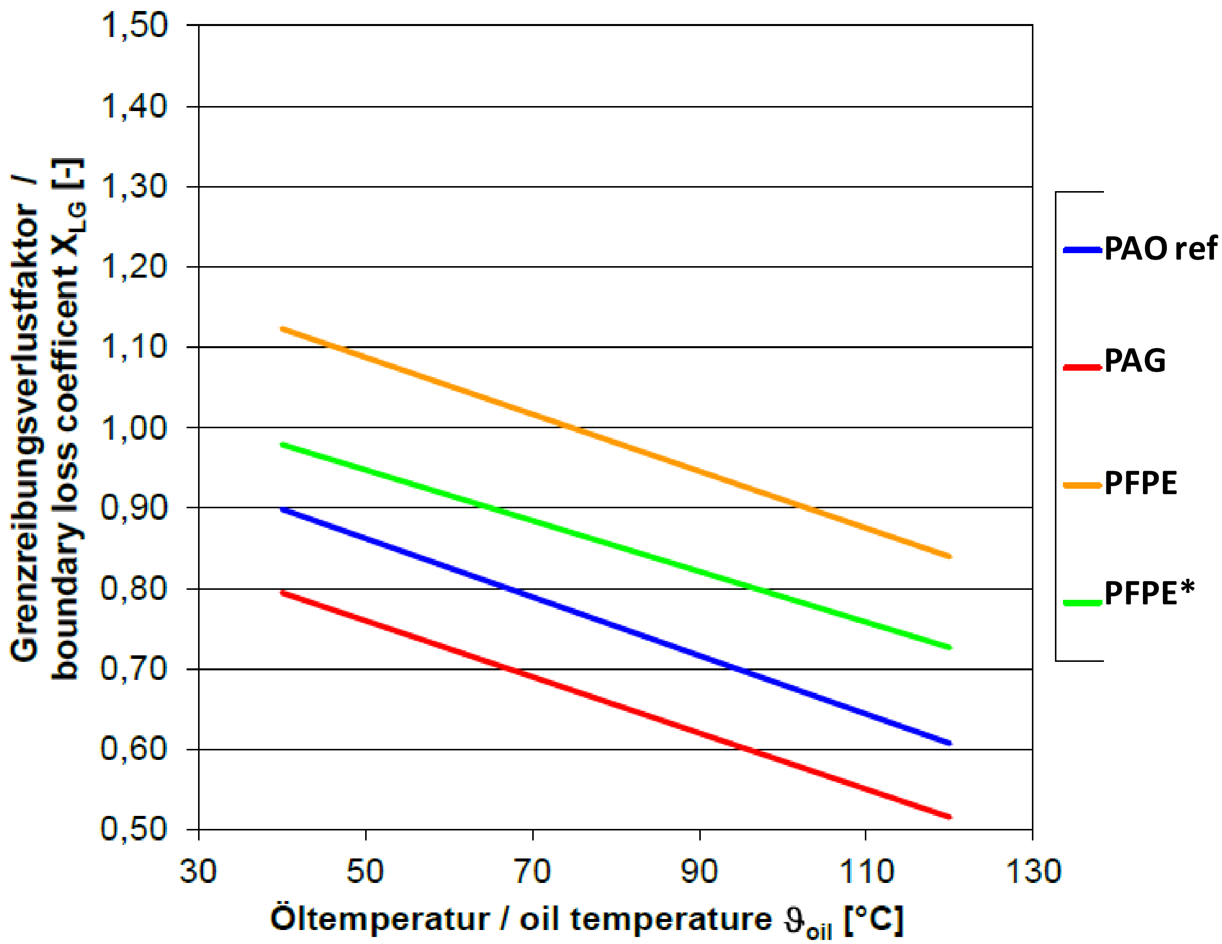


Figure 6.6-5: Load Losses at Boundary Lubrication Conditions, each curve is fitted from 4 measurements

In Figure 6.6-6 the measured coefficient of friction is shown for different lubrication regimes. For boundary lubrication conditions at low speeds the coefficient of friction is usually independent of the film thickness and decreases in the transition from boundary to mixed lubrication; for EHD-lubrication condition at higher speeds it is increasing with the film thickness (compare to Stribeck curve). At the referring conditions (load stage 7) all candidates, according to lubricating regime definition from film thickness criteria are expected to act in the boundary ($\lambda < 0.7$) and mixed lubrication regime ($0.7 < \lambda < 2.0$) for $v_t = 0.5$ m/s. At higher speeds ($v_t = 8.3$ m/s) they reach the mixed lubrication regime ($0.7 < \lambda < 2.0$) and EHD lubrication regime ($\lambda > 2.0$). As it can be seen in Figure 6.6-6 the measured frictional behavior in the individual lubrication areas slightly differs from the defined characteristics. Because of the increase of μ_{mz} for $v_t = 0.5$ m/s partly EHD conditions seem to prevail.

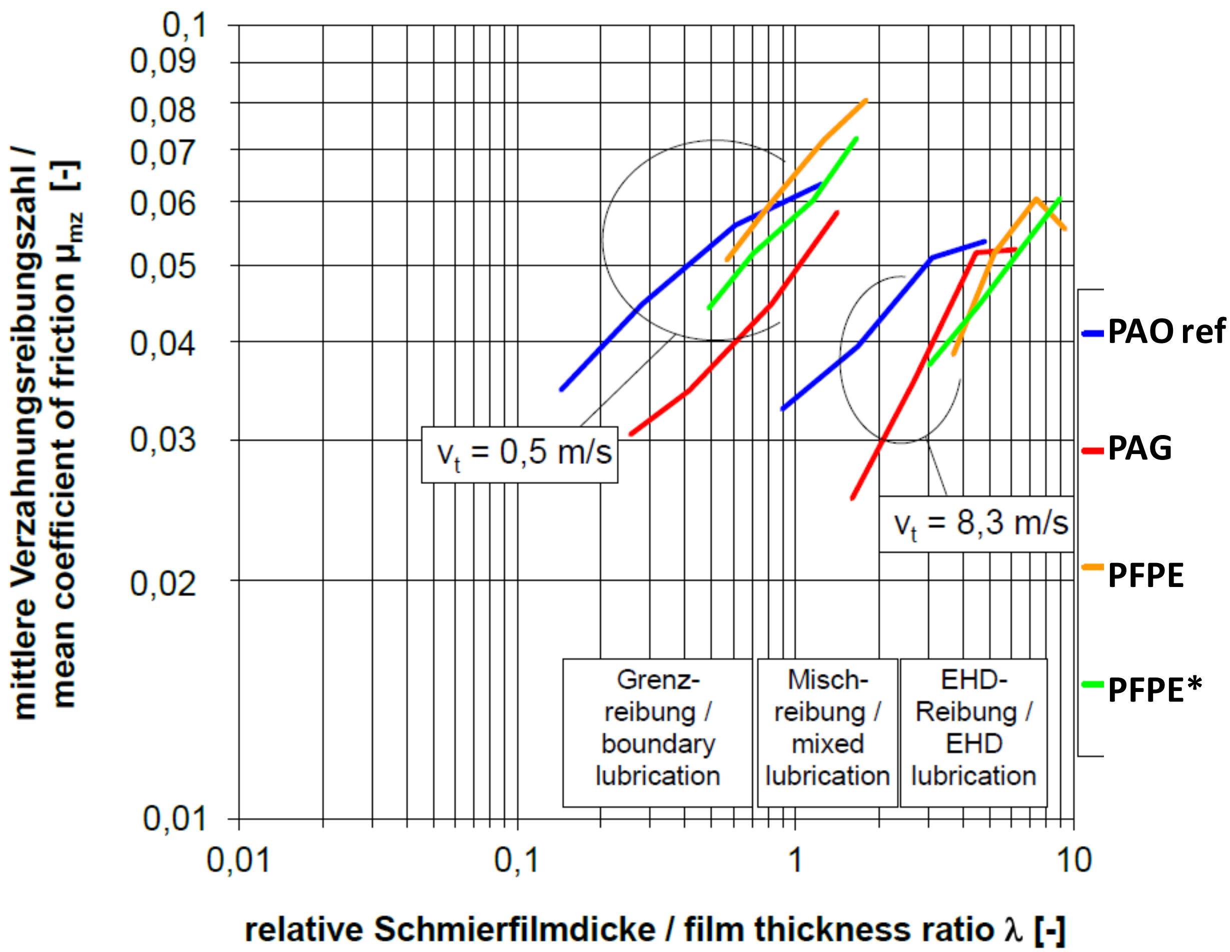


Figure 6.6-6: **Influence of film thickness ratio on the frictional behavior at load stage 7 and varying temperatures.**

In summary the PFPE fluids show much higher churning losses due to their higher viscosity and density. In the boundary lubrication region they show reasonable results even without Extreme Pressure additives and in the EHD lubrication regime in loaded conditions (should be compared to actual operating conditions of the wind turbine gearbox) they show lower losses compared to the benchmark oils. As shown in the Figure 6.6-6 for the EHD-conditions, coefficients of friction are lower at similar film thicknesses.

6.7 FE 8 Bearing tests

In order to check the suitability of the PFPE fluid for wind turbine gearbox lubrication bearing tests according to the draft DIN IEC 64100-4 were carried out and are summarized in Table 6.7-1.

	rpm	***load [kN]***	***run time [h]***	***temp [^{o}C]***	***roller wear [mg], negative value = increase of mass***	
draft DIN IEC 61400-4) <= 30mg roller wear (FE 8 stage 1 - DIN 51819)						
PFPE lab sample	*7.5*	*100*	*80*	*80*	*0*	*0*
PFPE before gearbox trial	*7.5*	*100*	*80*	*80*	*5*	*5*
PFPE after gearbox trial	*7.5*	*100*	*80*	*80*	*2*	*3*
draft DIN IEC 61400-4) <= 30mg roller wear (FE 8 stage 2 - DIN 51819)						
PFPE lab sample	*75*	*100*	*800*	*70*	*29*	*23*
PFPE after gearbox trial	*75*	*100*	*800*	*70*	*16*	*29*
draft DIN IEC 61400-4) <= 600 h run time (FE 8 stage 4 - FAG test)						
PFPE lab sample	*750*	*60*	*600*	*100*	*-3*	*-6*
PFPE after gearbox trial	*750*	*60*	*600*	*100*	*-21*	*-15*

Figure 6.7-1: Bearing Test Results

FE 8 stage 1 is aimed to test in boundary lubrication conditions and the PFPE fluids before and after the gearbox trial pass. FE 8 stage 4 is aimed at mixed lubrication conditions and the PFPE fluid after the gearbox trial as well as a laboratory sample with a similar composition passes as well. The same is valid for the FE 8 stage 4 which is aimed at EHD lubrication conditions where the required run time was reached.

Reference Appendix Documents for this Section:

*** APPENDIX 2A:	FINAL REPORT ON ANALYSIS OF IN-SERVICE TURBINE OIL SAMPLES
*** APPENDIX 2B:	GEARBOX OIL SAMPLES
*** APPENDIX 2C:	FRESH & LABORATORY STRESSED OILS FLUIDS
*** APPENDIX 2D:	TEETH SURFACE ANAYSIS FROM LABORATORY GEAR TESTS
*** APPENDIX 2E:	REPORT 4203 MICRO-PITTING LAB PFPE
*** APPENDIX 2F	REPORT 4364 EFFICIENCY TEST FZG
*** APPENDIX 2G	REPORT G 1259 MICRO-PITTING ON PAO REFERENCE
*** APPENDIX 2H	REPORT G 1262 MICRO-PITTING ON ESTER
*** APPENDIX 2I	REPORT G 1265 MICRO-PITTING ON PAG
*** APPENDIX 2J	REPORT G 1269 MICRO-PITTING ON PAO ALTERNATIVE
*** APPENDIX 2K	REPORT G 1276 MICRO-PITTING ON PFPE FRESH
*** APPENDIX 2L	REPORT G 1286 MICRO-PITTING ON PFPE USED
*** APPENDIX 2M	WIESBADEN FE8 BEARING TEST COMBINED
*** APPENDIX 2N	TASK 2 IMAGES AND TABLES

*** *Document contains PROPRIETARY information and is withheld from the public version of this report.*

7. TASK 3 RESULTS: FULL SCALE LAB GEARBOX TRIAL

7.1 Introduction

This document summarizes the outcome of testing of the performance of a new lubricant fluid in a Quantum Drive Gearbox at Clipper Windpower's Gearbox Test Stand. The document also outlines the test process and deviations to the test plan (Appendix 3A Test Plan Reference Document "Test Plan Lifetime Lubricant Evaluation" dated December 17, 2010), describes the data and results from the testing, and provides data analysis and conclusions from the test program. The PFPE fluid under test is a new formulation designed with an optimized viscosity profile to provide greater gear protection through a wider range of temperatures than existing fluids used in wind turbines and other heavy industrial equipment. As a point of comparison the new fluid was evaluated against a PAO Reference. The new fluid was subjected to an extended period of baseline performance testing equal to approximately one year of equivalent torque cycles on a typical gearbox. In addition, each fluid underwent rigorous stage testing, with operating conditions of increasing severity over those that a field turbine would normally experience; rotational speed and lubricant temperatures were adjusted to achieve increasingly tribologically stressful conditions until the limits of the test stand were reached.

7.2 Background and Objectives

7.2.1 Background

The PFPE lubricant's relatively consistent viscosity, its wear protection, and its projected long lifespan, all attained without the use of sacrificial additives prone to depletion, are considered groundbreaking. Preliminary testing in a laboratory confirmed the advantages, leading to a test to validate its capability in a full scale gearbox.

Wind turbine gearboxes, located in often inhospitable environments, were deemed suitable candidates- while seeing high operational torque loads, they also operate at a variety of speeds and through nearly every weather pattern. The gearboxes are conventionally filled with lubricants needing additive packages to combat specific environmental conditions and wear mechanisms. Additives will eventually deteriorate and need to be replenished in order to protect the gearing; this requires oil changes throughout the life of the gearbox. Such maintenance requires downtime and can be difficult in harsh environments.

Clipper Windpower provided a gearbox (GB) and gearbox test stand (GTS) for use in full scale testing as well as test support and operational personnel.

7.2.2 Objectives

The test program documented in this report was to determine the capability of the PFPE as a lubricating fluid in industrial gearboxes. The goals were to make an assessment of the proposed lubricant properties and to determine the relative performance when compared to the existing lubricant in a controlled setting. Evaluation criteria were degradation of the lubricant, degradation of the gear and bearing surfaces in the gearbox, and a comparison of the relative efficiency.

7.2.2.1 Verify PFPE durability

The PFPE Fluid had never been produced in large quantities, nor tested in any application other than small-scale laboratory tests, a goal of this test program was to establish a real world baseline of the PFPE lubricant performance. The baseline evaluation test called for approximately 500 hours of high torque running in a production gearbox on the Clipper GTS at load levels near a previously run Highly Accelerated Life Testing (HALT). Assuming no visible lubricant degradation or abnormal gear wear, additional testing with increasing loads would take place.

7.2.2.2 Verify PFPE gear wear protection through exposure to intense load conditions

After the extended period of baseline evaluation, the limits of the PFPE fluid were to be explored through staged testing of increasing severity. The stages were designed such that adjustments of the test stand load, speed, and temperature conditions would reduce the film thickness of each stage increasing the likelihood of surface asperity contact induced failures such as micro-pitting. The goal of this staged testing was to determine if the limits of the lubricants protective abilities could be reached.

7.2.2.3 Establish gear wear performance differential between two lubricants

In an effort to make comparisons to a known standard, the PAO Reference fluid would also undergo the same staged testing as the PFPE fluid. Expectations were that noticeable micro-pitting would occur on one or both high speed pinion (HSP) sets. By subjecting both fluids to similarly harsh conditions and similar film thickness reductions, a failure such as noticeable micro-pitting on gears utilizing one lubricant and not the other could indicate superior performance.

7.2.2.4 Determine potential efficiency differences between two lubricants

Another area of possible differentiation between the lubricants is efficiency. At all points during the test, the current draw from the Variable Frequency Drive (VFD) was recorded. As the test stand is a power re-circulating torque loop, the VFD current at steady state is proportional to the power lost in the test stand gearboxes. By comparing the required current for similar load conditions the relative efficiency can be determined.

Alternatively, temperature can be used as an indicator of efficiency. When used in the same hardware, a more efficient lubricant would be expected to have less friction or shear losses and would not build up as much internal temperature. The sump, manifold, and radiator inlet steady state temperatures and rate of temperature build could give an indication of lubricant efficiency. Similarly the steady state surface temperature of the gearbox as recorded by thermal imagery could give and indication of efficiency.

7.3 Executive Summary for Full Scale Laboratory Gearbox Trial

The PFPE lubricant successfully completed the full test program including 406 hours of baseline evaluation and an additional 282 hours of staged testing. The evaluation gears showed no micro-pitting or objectionable wear. Fluid degradation was limited to discoloration associated with a slight level of cross contamination (due to repurposing a previously used gearbox). By the final stage, the film thickness had been reduced to just 21% of its original value; this was by design, resulting in a lambda ratio of 0.44 and a mixed lubrication regime with asperity or surface to surface contact. This test design scenario of a low lambda ratio is a very undesirable lubrication condition for real world but creates the ability to test the lubricating fluids performance under the most extreme conditions.

The PAO Reference lubricant also passed 23 hours of baseline testing and an additional 275 hours of staged testing without any noticeable deterioration of the gear surface. Unfortunately the PAO Reference lubricant was replaced midway through the progressive loading, as the lubricant was burned in an attempt to raise the sump temperature via an immersion heater in a similar a fashion which was previously performed successfully for the PFPE. The gearbox was drained and filled with fresh PAO Reference oil and the load stage was repeated at a lower speed and temperature. The new fill subsequently successfully completed several additional load stages. This fresh fill also replenished any wear additives that had been depleted for the final load stages of the PAO testing. By the final stage, the film thickness had been reduced to just 44% of its original value; again this was by design, resulting in a lambda ratio of 0.42 and a mixed lubrication regime with asperity contact.

In addition to wear and micro-pitting performance, an attempt was made to characterize the relative efficiency between the new fluid and the baseline lubricant. The results are varied as each fluid's properties and load stages were very different. The analysis seems to show that the efficiency correlates better to viscosity than any other of the measured metrics such as film thickness. In load stages where the load, speed and temperature are similar the PFPE has a greater film thickness, and theoretical gear protection, but requires a larger VFD current (lower efficiency) than the PAO. However in load stages where the film thickness is the same, the PFPE's reduced viscosity gives it a slight efficiency advantage (lower VFD current per unit Torque) relative to the PAO (refer to Section 7.9.1). Ultimately, many factors such as temperature, rotational speed, and fluid viscosity combine in complex fashion to influence the results. The additional temperature and thermal image methods of evaluating efficiency were deemed inconclusive due to uncontrolled variables.

Fluid samples were collected throughout the test and are being evaluated independently (refer to Section 6 for results). Additionally, one HSP from each lubricant set was sent to a third party metallurgical laboratory for a detailed independent analysis of its condition. Physical inspection of the pinion gears and bearings were also inspected by an external gear design consultant. Full reports by these external parties are provided in Appendix 3G but can be summarized as follows:

"The exam of the tested pinions indicates that both lubricants provided sufficient protection to the surfaces to prevent damage due to micro-pitting, macro pitting and scuffing. During the test the temperature was raised to induce low Lambda ratios but this did not result in micro-pitting of the pinion. This was true for both the PAO and the PFPE lube products."

"In general, the PFPE performed as well as the current PAO lubricant wit h the exception that the level of surface scratches is greater on the PFPE lubricated test articles." The surface scratches are later describes as being caused by installation error.

7.4 Methods

The methods described below were used to achieve the test objectives:

7.4.1 Lubricant durability and micro-pitting protection

Lubricant degradation and gear wear were initially evaluated via prolonged use of PFPE in a gearbox installed on Clipper Windpower's GTS. To facilitate this effort the torque level was set to consume the largest percent of design life of the gearbox as possible. The design life of a Clipper gearbox is 20 years under typical conditions experienced in the field, however running at a constant high speed and using an increased torque the equivalent design life of the gearbox can be reduced to a duration suitable for a bench test. During this test the load level of 6300 Nm, approximately 12% over nominal, reduces the design life of the gearing to 8108 hours. The speed and sump temperature targets were set to be as near to the operating speed and temperature of an operating turbine.

The bounds of the lubricant durability and gear wear protection were then explored through the use of increasingly harsh load stages. The goal of each stage was to reduce the film thickness in the gear mesh by approximately 20% each stage until the limits of the lubricant or the test stand were reached.

The film thickness in the gear mesh is a cushion of fluid that separates the contacting surfaces and protects the gearing from surface contact related failure mechanisms of pitting, micro-pitting, scuffing, and wear. The film thickness is a function of the load transferred in the mesh, the speed of the gears, and the fluid properties of the lubricant. By increasing the load, reducing the speed and / or increasing the sump temperature of the test article the film thickness can be reduced.

The film thickness values were analytically calculated using the equations from the standard ISO 15144-4 entitled "Calculation of micro-pitting load capacity of cylindrical spur and helical gears -- Part 1: Introduction and basic principles" and used to determine the severity of each load stage. The load, speed, and temperature parameters of each stage were designed such that the analytical film thickness reduced by approximately 20% each stage. The initial calculations of the test proposal were revised as the limits of test stand were reached. The final stage of each fluid test was as the highest load, lowest speed and the highest temperature that was deemed practical to demand of the test stand and as such the smallest film thickness for each fluid was achieved. Each load stage was run for a minimum of 1.44e6 load cycles which is equal to the cycles per stage of the FZG standardized test for micro-pitting.

Figures 7.4.1-1, 7.4.1-2 and 7.4.1-3 below show the parameters and analysis results from the KISSsoft gear analysis program that was used to perform the calculations.

Figure contains PROPRIETARY information and is withheld from the public version of this report.

Figure 7.4.1-1 KISSsoft Example Calculation Input Window

Figure contains PROPRIETARY information and is withheld from the public version of this report.

Figure 7.4.1-2 KISSsoft Example Result Output Graph

Figure contains PROPRIETARY information and is withheld from the public version of this report.

Figure 7.4.1-3 KISSsoft Example Output Results

7.4.2 Inspections

The pinions and gears were inspected and lubricant samples were taken prior to testing, regularly throughout, after each stage was completed, and at the end of the test to ensure the onset or progression of gear wear mechanisms, such as micro-pitting, were captured. High speed bearing rollers were also inspected visually in situ during the test and a posthumous destructive inspection was planned for each HS cartridge to identify the magnitude of potential skidding or other bearing failure mechanisms.

Pinion inspections originally included all of the following: documentation using a high power camera, capable of extreme zoom; documentation using a macro lens in order to capture any issues not located in our designated locations; and manual tape and dental pick evaluation. The tape and dental pick methods were abandoned due to ineffectiveness on the smooth surface of the teeth; anytime a visual inspection resulted in something of note, they were attempted with little efficacy.

The result of each load stage was to either be pass or fail, based primarily on the presence of visible contact related failures.

7.4.3 Determine relative lubricant efficiency

In an attempt measure the relative efficiency of the PFPE, the input current necessary to drive the test specimen for the baseline and stage testing was monitored. In addition, sump, radiator-line, and tooth-root lubricant temperatures were also recorded to determine any possible decrease in waste heat generation. Thermal images of the steady state baseline load stage operation were taken for both tests.

7.5 Test Article Information

7.5.1 Component Parts

The test article was a Clipper C96 gearbox, manufactured at the Clipper's Cedar Rapids, Iowa factory. The Gearbox was previously installed in a wind turbine but was returned after light use when an external defect was found. The gearbox housings, intermediate gears and bull gear were inspected for condition and wear prior to the test and found acceptable. These parts were not the focus of this test effort but were subject to further in-situ inspections throughout the test as permitted with the components installed in the gearbox

The test plan uses the surface of the high speed pinion gears for the majority of the assessment of the lubricant performance. The pinions and HS bearings used in this evaluation were brand new and thoroughly inspected. The test used 2 sets of pinions, one for each lubricant. The test article HSP were slightly different from the production pinions that utilize a special "diamond-like" hard coating to safeguard against surface contact initiated failures such as micro-pitting. The pinions were ordered without the diamond coating to increase the opportunity for any wear mechanism to appear.

A detailed bill of materials fully documenting the configuration of the test article is included in Appendix E. The test article part and serial numbers of the major components can be found in Table 7.5.1-1 below:

<table>
<tr><th rowspan="2"></th><th rowspan="2">Gearbox</th><th rowspan="2">Housing</th><th rowspan="2">Bull gear</th><th rowspan="2">Intermediate Gears</th><th>High Speed Pinions</th><th>High Speed Pinions</th></tr>
<tr><th>PFPE</th><th>PAO Reference</th></tr>
<tr><td rowspan="2">Part Numbers</td><td rowspan="2">10-040836-02</td><td>10-005104-01</td><td rowspan="2">20-004976-01</td><td>10-004959-02</td><td rowspan="2">20-122673-03</td><td rowspan="2">20-122673-03</td></tr>
<tr><td>10-005103-01</td><td>10-004960-02</td></tr>
<tr><td rowspan="4">Serial Numbers</td><td rowspan="4">N/A</td><td rowspan="4">N/A</td><td rowspan="4">N/A</td><td>60-2363</td><td>22133849-236</td><td>22133849-279</td></tr>
<tr><td>60-2362</td><td>22133849-244</td><td>22133849-234</td></tr>
<tr><td>59-2316</td><td>22133849-229</td><td>22133849-342</td></tr>
<tr><td>59-2338</td><td>22133849-249</td><td>22133849-242</td></tr>
</table>

Table 7.5.1-1 Test Article Part and Serial Numbers

7.6 Lubricant Data

7.6.1 PFPE

The basic lubricant parameters and viscosity profiles that were used in the calculations are shown below in Table 7.6.1-1 and Figure 7.6.1-1.

PFPE		
Parameter	Value	Unit
Base	PFPE – Perfluoropolyether	-
Viscosity at 40 °C	224	mm^2/s
Viscosity at 100 °C	70	mm^2/s
Viscosity Index	340	-
Density	1845	kg/m3
Pour Point	-65	°C
Upper Service Limit	None	°C
Flash Point	No Flash Point	°C
Scuffing Test	Load Stage 14	-
Micro-pitting Test	Load Stage 10	-

Table 7.6.1-1 PFPE Basic Lubricant Data

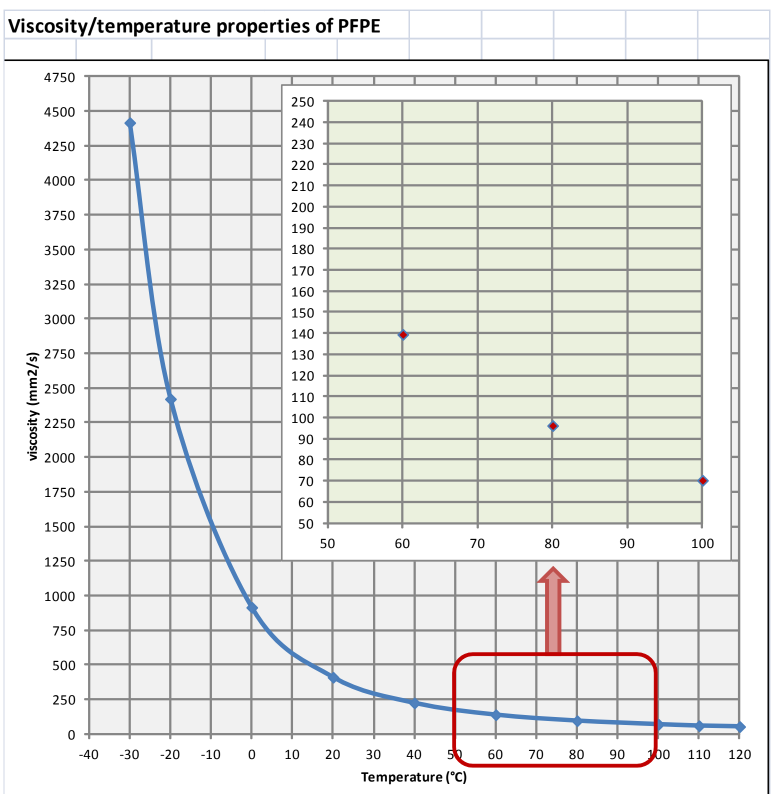


Figure 7.6.1-1 PFPE Viscosity Profile

7.6.2 PAO Reference

The basic PAO lubricant parameters and viscosity profiles that were used in the calculations are shown below in Table 7.6.1-1 and Figure7.6.1-1.

PAO Reference		
Parameter	**Value**	**Unit**
Base	Synthetic Oil Based on Polyalphaolefin	
Viscosity at 40 °C	330	mm^2/s
Viscosity at 100 °C	33	mm^2/s
Viscosity Index	140	-
Density	870	kg/m3
Pour Point	-30	°C
Upper Service Limit	95	°C
Flash Point	205	°C
Scuffing Test	Load Stage 14	-
Micro-pitting Test	Load Stage 10	-

Table 7.6.2-1 PAO Reference Basic Lubricant Data

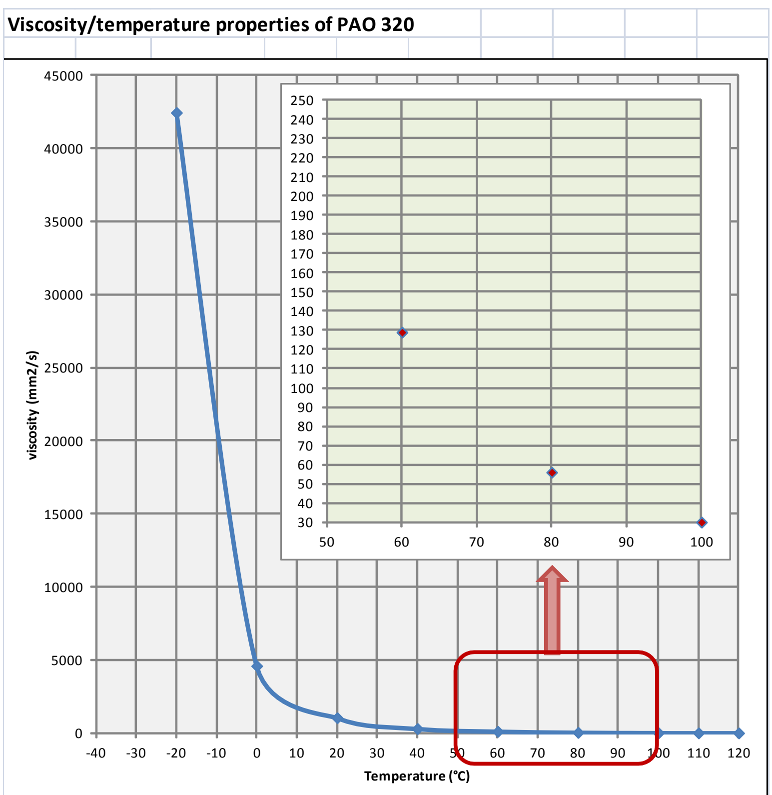


Figure 7.6.2-1 PAO Reference Viscosity Profile

7.7 Test Setup

This section will detail the manner and ideology behind the test setup. See Appendix 3A "Test Plan Lifetime Lubricant Evaluation" dated December 17, 2010 for further details.

7.7.1 Equipment

Figure contains PROPRIETARY information and is withheld from the public version of this report.

Figure 7.7.1-1 Test Stand Configuration Schematic

7.7.1.1 Gearbox Test Stand (GTS), Data Acquisition Equipment

The GTS is essentially two gearboxes mounted back-to-back in a torque loop; the driver box rotates in reverse of a standard gearbox, while the test specimen rotates in the conventional direction. The two gearboxes are connected at the mainshaft / hub attachment point, and hydraulic units provide torque on the high speed pinions. The electronic control unit constantly monitors the torque at each test pinion, targeting a single value set by the technicians for the specific test at hand.

The driver box was unmodified for the test. Eventually, due to high vibration when operating at such high torque and speed, the feedback system was rebalanced and bearings replaced. The test specimen, as previously mentioned, was constructed of re-purposed components, with the exception being the HSP and associated bearings.

Test conditions and data are monitored and recorded by an IMC Cronos PL-3 unit. Additional channels were required for this evaluation, so a second Cronos unit and dedicated computer were added. Channels monitored by the standard GTS unit are numerous; those pertinent to this evaluation are described in table 7.7.1-1, while section 7.7.1.2 "Additional Equipment" details the second Cronos unit and extra equipment added for this evaluation.

Channel	Sub-channels	Units	Sample Rate	Notes
Date / Time	n/a		1/3 Hz	Standard
Gearbox speed	n/a	rpm	1/3 Hz	Standard
Pinion Torque	A, B, C, D (1 for each pinion)	Nm	1/3 Hz	Standard
GB Bearing Temperature	1, 2 (1 each for test box, driver bo	ºC	1/3 Hz	Monitored for system health
VFD Current	n/a	amps	1/3 Hz	Unique to this test

Table 7.7.1.1-1 Gearbox Test Stand Data Collection Channels

This evaluation not only attempted to find the edge of the lubricant capabilities; it also pushed the GTS to the bounds of its abilities. The test stand is typically used to test gearboxes to 100% torque, not above and beyond. As such, some "dialing in" of the GTS was necessary; ultimately, the sweet spot for continual runtime was found to be 6300 Nm, seemingly regardless of rotational speed targets. Also the maximum operational speed was found to be 850 RPM.

7.7.1.2 Additional Equipment

Understanding the uniqueness of the proposed lubricant was not left to chance. Equipment added specifically for this evaluation include a cooling system (flushed of contamination), a flow meter and (eventually) an immersion heater, plumbed in line with the external oil lines. As mentioned in section 7.7.1.1, a second Cronos unit was added to monitor additional temperatures and lubricant flow not necessary for a typical gearbox evaluation. These channels are described in table 7.7.1.2-1. The locations of the measurement channels and the direction of lubricant flow can be seen in the schematic in Figure 7.7.1.2-1.

Channel	Units	Sample Rate
Flow Rate	gpm	1/3 Hz
Pinion Tooth Root Temperature	ºC	1/3 Hz
Lubricant Manifold Temperature	ºC	1/3 Hz
Lubricant Radiator Inlet Temperature	ºC	1/3 Hz
Pressure	psi	1/3 Hz

Table 7.7.1.2-1 Additional Gearbox Test Stand Data Collection Channels

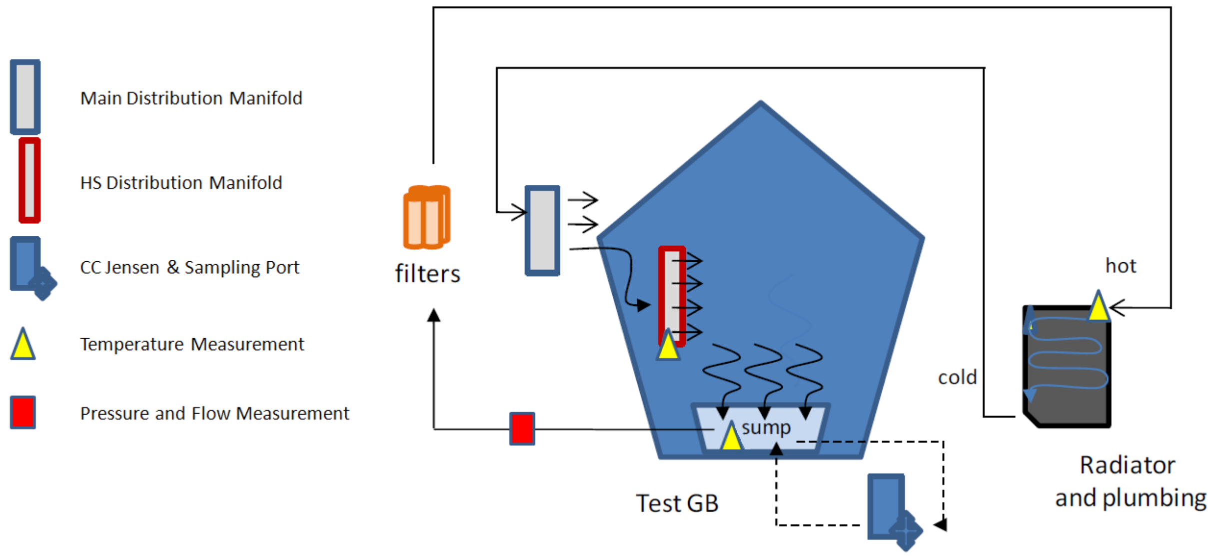


Figure 7.7.1.2-1 Lubrication, Cooling and Measurement Schematic

The coolant system is essentially the same as used on Liberty turbines. It was flushed 3 times prior to testing the PFPE lubricant. It activates at a user-defined point, but de-activates at an internal set point; typically, this was approximately 5^{o} C cooler than when it activated.

The flow meter, provided by Dow Corning, was a Micro Motion CFM300, capable of handling 54 gallons / minute.

A commercially available immersion heater was used in the latter portion of the test to maintain lubricant temperature. This heater was purchased at McMaster-Carr, model number 35705K132. Further, the radiator lines were wrapped in flexible strip heaters and insulation to help maintain heat loss to the ambient air through the relatively thin, exposed lubricant lines.

7.7.2 Gear Inspections

Understanding the initial condition of each pinion set was critical to understanding whether any micro-pitting occurred due to test conditions. Each was examined visually prior to assembly, with any potentially aberrant issues documented using a digital camera with macro lens capability, as well as a Hirox digital microscope with recording capability. Four teeth on each pinion, spaced 90^{o} apart, were identified as teeth of interest. Three locations on each tooth, approximately 20mm from each end and at the midpoint, were examined at three distinct zoom levels. A digital level and a test-specific fixture were used to establish consistent placement from pinion to pinion and across inspection dates.

Mid-test inspections included removing the pinions for microscope documentation after the first four hours of run time, and thereafter alternating visual inspections while the HSP remained in their installed positions, and every other test day conducting microscope inspections. Before and after photos, along with select intra-stage images, for each stage are documented in the directories as documented in Appendix 3D. A minimum of 144 images per inspection were collected; any notable damage or wear outside of the identified locations was documented in addition to the regular locations.

An example of the image quality and detail of the Hirox digital microscope at each level of zoom is shown below.

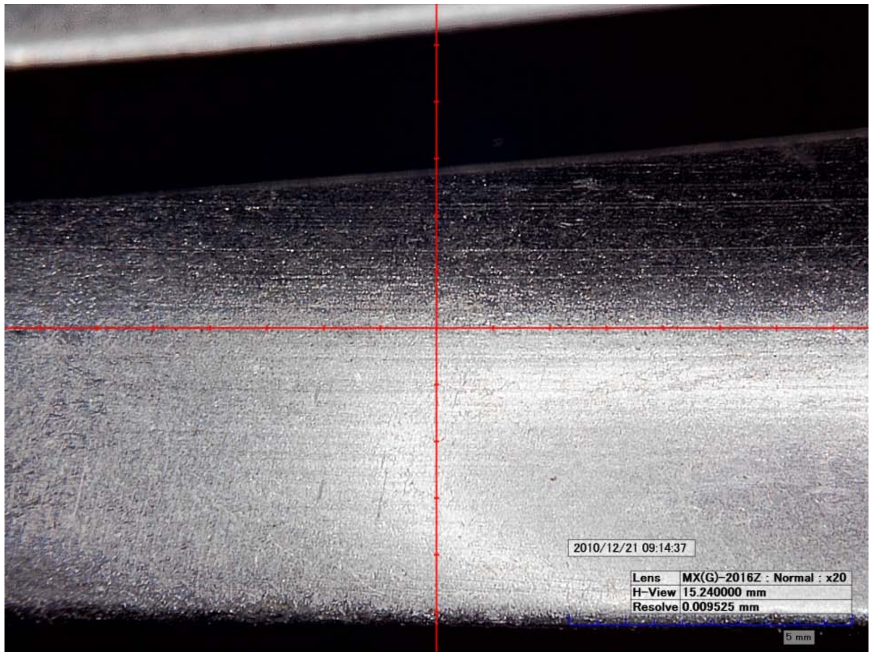


Image 7.7.2-1. PFPE P1DWNA001 pre-test. The nomenclature refers to the image location on Pinion 1 20mm from the downwind end, at the widest zoom. Other images in this position (002 and 003) zoom in an additional 60x each.

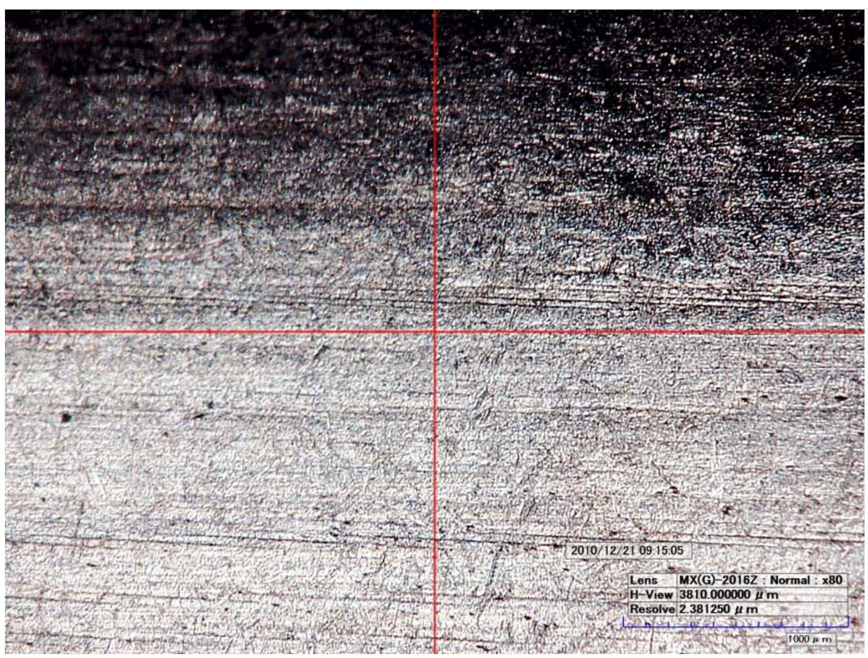


Image 7.7.2-2. PFPE P1DWNA002 pre-test. An example of the middle field of view.

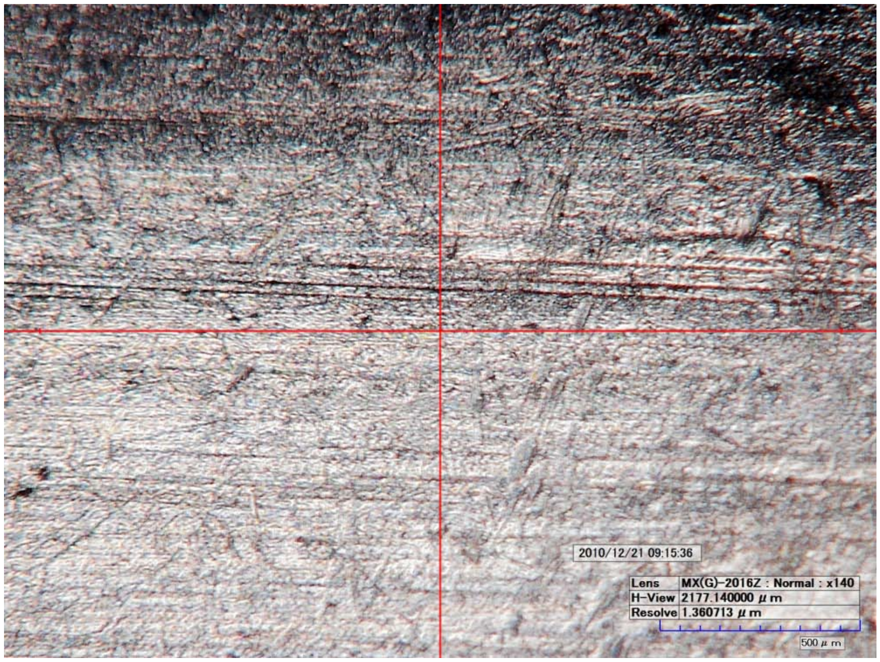


Image 7.7.2-3. PFPE P1DWNA003 pre-test. The narrowest field of view used in the inspection photographs.

Additionally, surface quality was to be monitored by both objective and subjective means. A Hommel profilometer was utilized before and after each test (not in between stages, however, due to time constraints). While fixturing this device was not practical, the same points (approximately) as the Hirox inspections were used. The same individual collected data on all of the pinions and teeth to ensure as consistent a measurement as possible. Subjective measurements were attempted- utilizing graphite powder and tape to identify surface imperfections did not work well with the curved geometry of the teeth and the finely polished tooth surfaces; eventually, this method was abandoned due to its inefficacy and the time associated with the activity.

7.7.3 Calibration

Torque calibration was not conducted during this test. Time was of the essence, and would have added a significant delay. The GTS was operating within Clipper Windpower's approved calibration timeframe, and test personnel monitored all channels for any sign of obvious inaccuracy. The RTDs used to monitor lubricant temperature were "baked" in a chamber along with a calibrated control device. Their response was compared to the control and each other, and all were found to be accurate as seen in table 7.7.3-1.

Time	Thermocouple T1	Thermocouple T2	R1 - Wire A Ω	R1 - Wire B Ω	R2 - Wire A Ω	R2 - Wire B Ω	Ambient Temp °C
1:50 PM	22.5	22.5	110.2	110.2	110.2	110.2	22.0
3:15 PM	31.2	31.2	112.5	112.5	112.6	112.6	46.0
4:15 PM	38.3	38.3	115.2	115.2	114.9	115.0	56.0
5:35 PM	45.0	45.0	118.6	118.5	117.8	117.8	60.0
7:30 AM	56.5	57.6	123.5	123.6	123.9	123.9	59.0
R1 = Temp Probe #279 - Pinion #1 with no coupler							

Table 7.7.3-1 RTD Calibration and Response Measurements

The flow meter was checked per manufacturer instructions.

7.7.4 Lubricant Sampling

Samples of each fluid were collected prior to testing and at each inspection point, including at test completion. These were collected to aid in diagnosing lubricant quality and any potential degradation. Results of multiple lubricant sample analysis are provided at various respective locations in Section 6 of this report.

7.7.5 Between Test Flush

The two lubricants under evaluation are immiscible; as such, contamination became a concern. Recycling a previously used gearbox and radiator system posed some concern, particularly as the PFPE fluid was tested first. To counter this concern, the gearbox was cleaned thoroughly using approved citrus cleaner during its assembly. The radiator system was flushed 3 times using a low molecular weight formulation of the PFPE fluid. Finally, a high pressure air hose was fitted to the inlet in order to encourage any remaining fluid to evacuate the system.

Following the PFPE testing, the gearbox was drained; an approximate volume of drained fluid was noted and compared to the 130 gallons originally in the system. The gearbox was subsequently filled with several gallons of low molecular weight PFPE fluid, and run through the oil pump and lubricant lines. Samples of the contaminated mixture of PFPE and low molecular weight version of the fluid were collected, poured on a Petri dish, weighed and baked in an oven at temperature of 150°C. Upon baking for approximately 30 minutes, the dish was re-weighed; the process repeated until before and after weights were equal. At that point, only the test-weight PFPE remained in the dish.

When it was calculated that only 1.3 gallons of the PFPE remained, a sacrificial quantity of PAO Reference was introduced to the gearbox. This quantity was pumped through the lubrication lines, and then drained. A 25mL sample was spun on a centrifuge, and the samples evaluated for the cloudiness indicative of contamination. Once the fluid sample was confirmed to have separated completely (no residual cloudiness, but rather two separate fluids) the size of the droplet was measured, and the concentration calculated. The gearbox was filled and drained one more time for good measure, and then filled with the test quantity of the PAO Reference fluid.

7.8 Results

7.8.1 Test Summary

Tables 7.8.1-1 and 7.8.1-2 summarize the loading parameters and measurements of the full test campaign for each lubricant. The values presented are the averages of each measured parameter while the system was under test. The duration was calculated based on the number of data points collected while the system was under test, with any data points at which the specified test parameters were not met excluded (data points collected during warm up, for example). Film thickness and viscosity values were calculated according to the equations contained in ISO IEC 15144-1 "Calculation of Micro-pitting…".

PFPE			Baseline	Progressive Stages							
				Stage 1	Stage 2	Stage 3	Stage 4	Stage 5A	Stage 5B	Stage 6	Stage 7
Test Parameters	HS RPM	RPM	850.1	850.1	750.1	450.0	549.8	399.9	450.0	350.0	200.0
	Torques Averaged	Nm	6228.4	6232.3	6232.4	6191.1	6233.9	6214.0	6230.5	6221.8	6227.0
	VFD current	A	612.3	583.3	545.9	524.9	511.7	525.7	508.6	511.2	536.9
Duration	Hours	hr	406.32	17.29	48.54	33.85	22.08	35.33	23.17	41.35	60.43
	# of Cycles	-	4.15E+07	1.76E+06	4.37E+06	1.83E+06	1.46E+06	1.70E+06	1.25E+06	1.74E+06	1.45E+06
	% of Fzg Cycles (1.44e^6)	%	2878%	122%	303%	127%	101%	118%	87%	121%	101%
Temperatures	Sump Temp	°C	58.81	72.13	85.37	74.95	87.32	80.37	96.04	96.04	98.98
	Manifold Temp	°C	45.63	62.38	71.23	67.40	79.51	73.44	83.99	83.38	81.77
	Radiator Inlet Temp	°C	54.61	68.23	82.07	71.91	83.49	77.32	91.41	89.69	91.53
	Pinion Tooth Root Temp	°C	66.53	75.66	85.60	78.16	93.32	84.32	95.77	93.74	90.65
Lubricant	Flow Rate	GPM	30.33	37.26	33.67	20.83	25.17	16.61	20.63	16.14	9.38
	Pressure	psi	189.47	193.42	141.68	80.07	88.90	63.56	63.14	47.16	24.65
Calculated Values	Film Thickness	μm	0.628	0.506	0.382	0.319	0.303	0.273	0.233	0.197	0.132
	Lambda Ratio	-	2.092	1.687	1.273	1.064	1.009	0.991	0.776	0.657	0.44
	Viscosity	cst	150.04	113.60	91.32	109.41	88.20	98.03	74.62	74.62	72.26
	Dynamic Viscosity	$N \cdot s/m^2$	272.40	205.14	164.14	197.42	158.41	176.47	133.50	133.50	129.17
	% Film Thickness Reduction	%	-	19%	25%	16%	5%	2%	23%	15%	33%
	Efficiency Metric	Nm/A	10.17	10.68	11.42	11.79	12.18	11.82	12.25	12.17	11.60
Success Criteria	Pass/Fail	-	Pass	Pass	Pass	Pass	Pass	Pass	Pass	Pass	Pass

Table 7.8.1-1: PFPE Test Summary

PAO Reference			Baseline	Progressive Stages							
				Stage 1	Stage 2	Stage 3	Stage 4A	Stage 4B	Stage 5A	Stage 5B	Stage 6
Test Parameters	HS RPM	RPM	850.1	450.0	350.0	249.8	800.1	549.8	400.0	350.0	249.8
	Torques Averaged	Nm	6297.0	6294.0	6298.2	6299.8	6295.3	6298.0	6300.3	6299.2	6299.8
	VFD current	A	579.8	539.9	549.7	570.3	514.3	510.8	511.1	518.4	551.2
Duration	Hours	hr	22.96	30.55	35.14	47.63	15.03	40.96	15.57	49.41	39.76
	# of Cycles	-	2.34E+06	1.65E+06	1.48E+06	1.43E+06	1.44E+06	2.70E+06	7.47E+05	2.08E+06	1.19E+06
	% of Fzg Cycles (1.44e^6)	%	163%	115%	103%	99%	100%	188%	52%	144%	83%
Temperatures	Sump Temp	°C	57.50	55.10	55.65	56.36	83.95	70.36	68.68	66.82	59.61
	Manifold Temp	°C	50.29	48.64	44.75	38.66	72.26	65.01	62.15	59.78	52.54
	Radiator Inlet Temp	°C	54.17	53.05	52.26	51.93	78.41	67.20	64.43	62.76	56.25
	Pinion Tooth Root Temp	°C	65.71	64.31	63.87	60.46	89.32	78.90	77.47	75.73	69.81
Lubricant	Flow Rate	GPM	37.84	20.73	16.19	11.60	33.59	25.32	18.65	16.54	12.06
	Pressure	psi	130.16	64.03	55.45	50.68	57.65	51.63	35.89	32.90	28.51
Calculated Values	Film Thickness	μm	0.286	0.215	0.184	0.144	0.134	0.154	0.13	0.127	0.127
	Lambda Ratio	-	0.953	0.716	0.614	0.481	0.447	0.514	0.434	0.423	0.423
	Viscosity	cst	140.75	153.82	153.82	140.75	52.35	79.78	86.05	92.98	129.05
	Dynamic Viscosity	$N \cdot s/m^2$	118.51	129.73	129.73	118.51	43.13	66.39	71.73	77.64	108.48
	% Film Thickness Reduction	%	-	25%	14%	22%	7%	-7%	16%	18%	0%
	Efficiency Metric	Nm/A	10.86	11.66	11.46	11.05	12.24	12.33	12.33	12.15	11.43
Success Criteria	Pass/Fail	-	Pass	Pass	Pass	Pass	Fail	Pass	Pass	Pass	Pass

Table 7.8.1-2: PAO Reference Test Summary

Of note, sub-stages, denoted #A and #B, were utilized for both the PFPE and PAO Reference tests during Stage 5 in order to facilitate attaining the desired film thickness reduction while using alternate speed and temperature combination. These additional runs were required as the internal heat generation of the gearbox did not generate sufficient heat to reach the target sump temperature and a significant decrease in film thickness was not achieved.

PAO Reference Stage 4 was run twice as the use of an inline heater burned the fluid in the first attempt. The gearbox was refilled, at which point the replacement fluid successfully completed a secondary stage at equivalent film thickness, though at dramatically reduced speed and temperature targets. The two stages were also noted as 4A and 4B versions.

The load conditions of each stage were meant to reduce the film thickness and increase the likelihood of surface contact initiated failures. The original target was to methodically reduce the film thickness by approximately 20% per stage. Figure 7.8.1-3 outlines the achieved reduction of film thickness through the test campaign. The flattening out of the curves was a result of the limits of the test stand being reached.

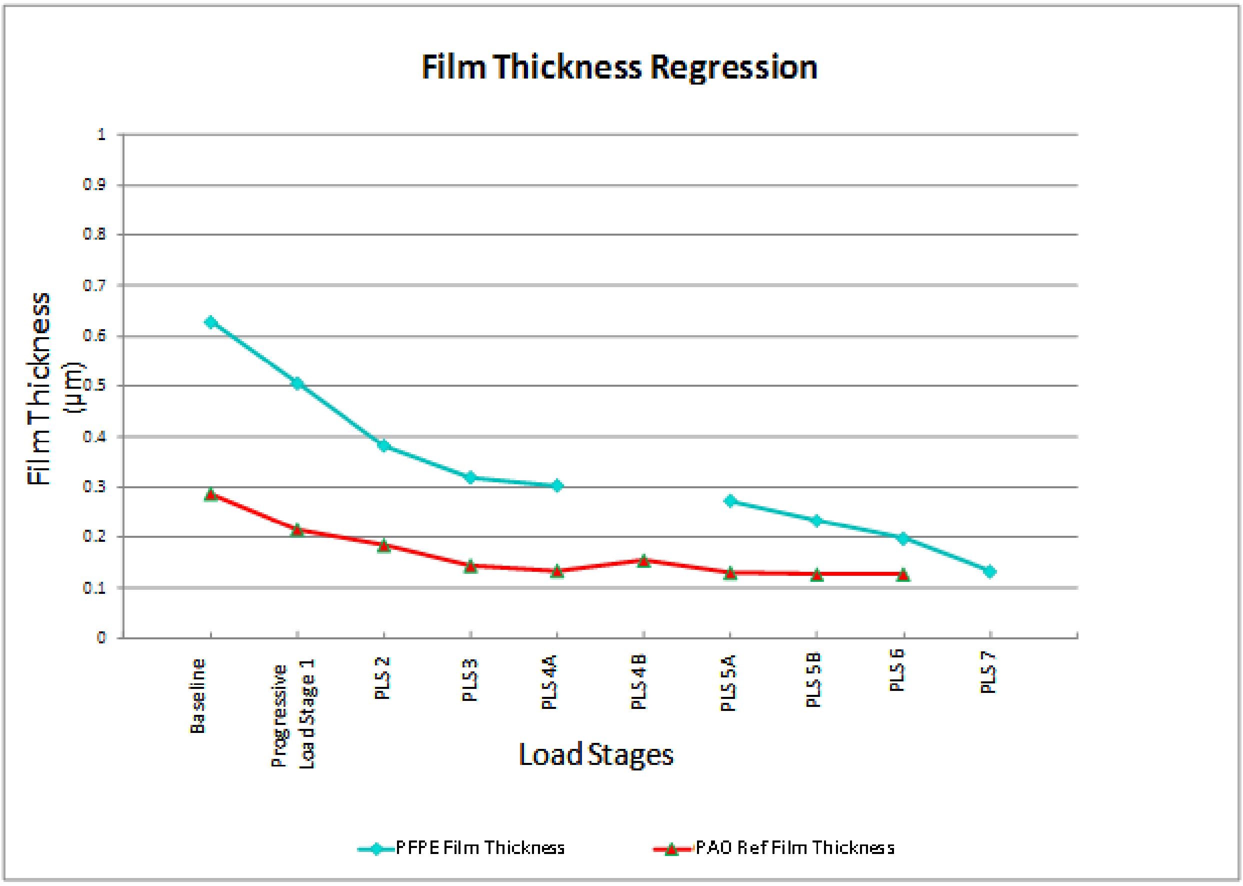


Figure 7.8.1-3. Stage Comparison of Film Thickness for PFPE and PAO Reference Fluids

7.8.2 PFPE Results

7.8.2.1 PFPE Durability and Baseline Stage

The baseline stage for the PFPE was designed to evaluate its long term performance at load levels typical of what would be seen by an operational turbine.

Table 7.8.2.1-1 shows the average values attained for the lubricant as it underwent this “baseline” testing. For this stage, all cooling equipment was set to activate at production values (60º C), the target torque was 6300 Nm, and rotational speed was set to 850 rpm. The calculated film thickness at these test conditions was 0.628 micrometers (µm). The target duration of 400 hours was achieved.

Further breakdown of the stage is available in “Test Assessment and Analysis” in Section 7.7.9.

PFPE			Baseline
Test Parameters	HS RPM	RPM	850.1
	Torques Averaged	Nm	6228.4
	VFD current	A	612.3
Duration	Hours	hr	406.32
	# of Cycles	-	4.15E+07
	% of Fzg Cycles (1.44e^6)	%	2878%
Temperatures	Sump Temp	°C	58.81
	Manifold Temp	°C	45.63
	Radiator Inlet Temp	°C	54.61
	Pinion Tooth Root Temp	°C	66.53
Lubricant	Flow Rate	GPM	30.33
	Pressure	psi	189.47
Calculated Values	Film Thickness	µm	0.628
	Lambda Ratio	-	2.092
	Viscosity	cst	150.04
	Dynamic Viscosity	$N\cdot s/m^2$	272.40
	% Film Thickness Reduction	%	-
	Efficiency Metric	Nm/A	10.17
Success Criteria	Pass/Fail	-	Pass

Table 7.8.2.1-1: PFPE Baseline Test Stage Summary

7.8.2.1.1 Baseline Stage Notables

After only four hours of runtime, during the first inspection of the PFPE testing, a thin layer of “coagulated” fluid had formed in the sump. The top layer of fluid was skimmed; approximately 2 oz. of cloudy lubricant was removed from the gearbox, including the vast majority of the odd-appearing surface layer. During subsequent inspections, the coagulation never returned. Experimentation with agitating different quantities of each lubricant mixed within sample jars replicated the coagulated effect reasonably well. The cause was determined to be low level contamination of the PFPE with the PAO Reference lubricant, despite thorough flushes of all previously used equipment and the test was resumed. Samples of the affected lubricant were collected to allow analysis at a later time outside of the scope of this project.

7.8.2.1.2 Hirox Inspection Results

As mentioned in Section 7.7.2, 144 images of each pinion set were taken every other test day. The entire collection of photos is archived as described in Appendix 3D. As a representative example the before and after images of the widest view for a consistent pinion position is presented here.

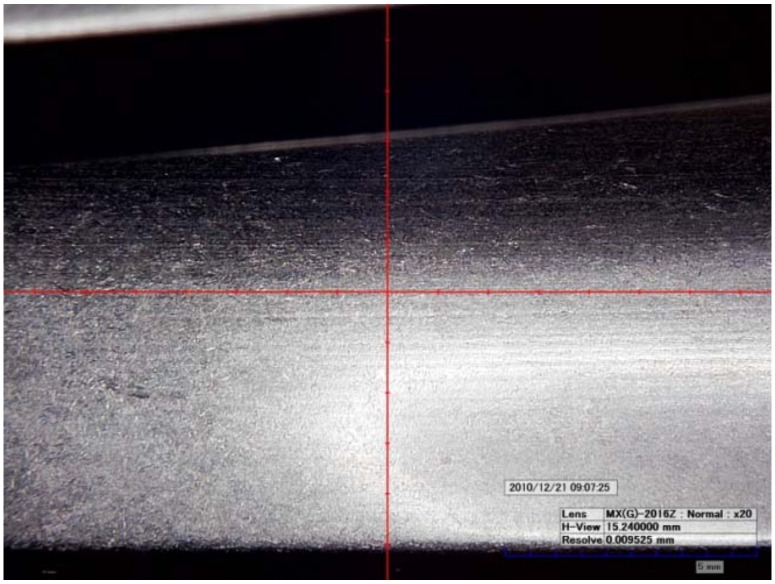


Image 7.8.2.1.2-1 P1MIDC001 PFPE Initial

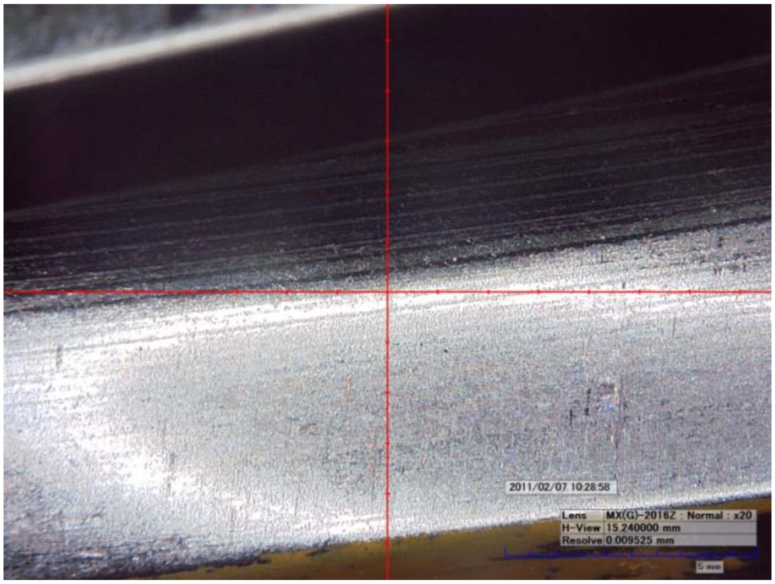


Image 7.8.2.1.2-2 P1MIDC001 PFPE Baseline Stage End

7.8.2.2 Progressive Load Stage (PLS) 1

Table 7.8.2.2-1 shows the average values attained for the lubricant as it underwent Progressive Stage 1.

PFPE			Progressive Stage 1
Test Parameters	HS RPM	RPM	850.1
	Torques Averaged	Nm	6232.3
	VFD current	A	583.3
Duration	Hours	hr	17.29
	# of Cycles	-	1.76E+06
	% of Fzg Cycles (1.44e^6)	%	122%
Temperatures	Sump Temp	°C	72.13
	Manifold Temp	°C	62.38
	Radiator Inlet Temp	°C	68.23
	Pinion Tooth Root Temp	°C	75.66
Lubricant	Flow Rate	GPM	37.26
	Pressure	psi	193.42
Calculated Values	Film Thickness	µm	0.506
	Lambda Ratio	-	1.687
	Viscosity	cst	113.60
	Dynamic Viscosity	$N \cdot s/m^2$	205.14
	% Film Thickness Reduction	%	19%
	Efficiency Metric	Nm/A	10.68
Success Criteria	Pass/Fail	-	Pass

Table 7.8.2.2-1: PFPE Progressive Load Stage 1 Summary

7.8.2.2.1 Progressive Load Stage (PLS) 1 Notables

This stage proceeded without any notable occurrences. The image identified as P1MIDC001 (pinion 1, middle of tooth C, widest zoom) is shown below as an example of the condition of the gear teeth.

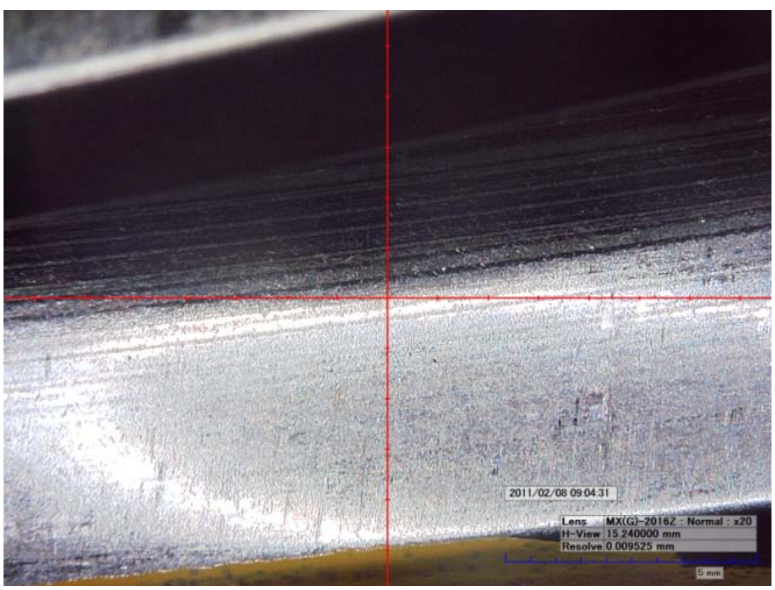


Image 7.8.2.2.1-1 P1MIDC001 Post PFPE Stage 1

7.8.2.3 Progressive Stage 2

Table 7.8.2.3-1 shows the average values attained for the lubricant as it underwent Progressive Stage 2.

PFPE			Progressive Stage 2
Test Parameters	HS RPM	RPM	750.1
	Torques Averaged	Nm	6232.4
	VFD current	A	545.9
Duration	Hours	hr	48.54
	# of Cycles	-	4.37E+06
	% of Fzg Cycles (1.44e^6)	%	303%
Temperatures	Sump Temp	°C	85.37
	Manifold Temp	°C	71.23
	Radiator Inlet Temp	°C	82.07
	Pinion Tooth Root Temp	°C	85.60
Lubricant	Flow Rate	GPM	33.67
	Pressure	psi	141.68
Calculated Values	Film Thickness	µm	0.382
	Lambda Ratio	-	1.273
	Viscosity	cst	91.32
	Dynamic Viscosity	$N \cdot s/m^2$	164.14
	% Film Thickness Reduction	%	25%
	Efficiency Metric	Nm/A	11.42
Success Criteria	Pass/Fail	-	Pass

Table 7.8.2.3-1: PFPE Progressive Load Stage 2 Summary

7.8.2.3.1 Progressive Load Stage (PLS) 2 Notables

Progressive Stage 2 passed without any notables. Again the P1MIDC001 image is shown below.

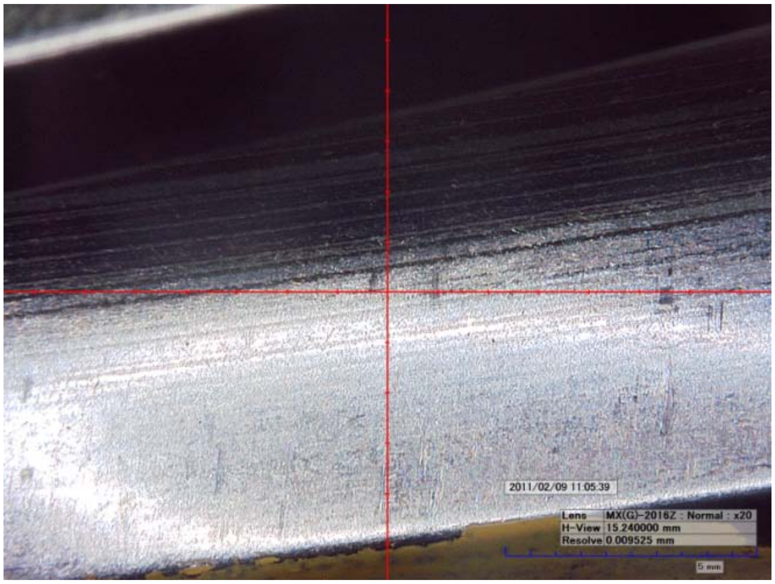


Image 7.8.2.3.1-1 P1MIDC001 Post PFPE Stage 2

7.8.2.4 Progressive Stage 3

Table 7.8.2.4-1 shows the average values attained for the lubricant as it underwent Progressive Stage 3.

PFPE			Progressive Stage 3
Test Parameters	HS RPM	RPM	450.0
	Torques Averaged	Nm	6191.1
	VFD current	A	524.9
Duration	Hours	hr	33.85
	# of Cycles	-	1.83E+06
	% of Fzg Cycles (1.44e^6)	%	127%
Temperatures	Sump Temp	°C	74.95
	Manifold Temp	°C	67.40
	Radiator Inlet Temp	°C	71.91
	Pinion Tooth Root Temp	°C	78.16
Lubricant	Flow Rate	GPM	20.83
	Pressure	psi	80.07
Calculated Values	Film Thickness	µm	0.319
	Lambda Ratio	-	1.064
	Viscosity	cst	109.41
	Dynamic Viscosity	$N{\cdot}s/m^2$	197.42
	% Film Thickness Reduction	%	16%
	Efficiency Metric	Nm/A	11.79
Success Criteria	Pass/Fail	-	Pass

Table 7.8.2.4-1: PFPE Progressive Load Stage 3 Summary

7.8.2.4.1 Progressive Load Stage (PLS) 3 Notables

After the inspection of progressive stage 3 small pits were found near the root of the tooth on 2 of the 4 high speed pinions. As this was expected to be the onset of a failure mode, the pits were documented with photographs to help determine their progression, if any, in the next load stages. The initial hypothesis was the low rotational speeds of this stage contributed to the pit formation. To verify this theory, the originally proposed load conditions of stage 4 were altered to utilize a higher speed and temperature combination while maintaining a film thickness approximately the same as stage 3. The test was resumed.

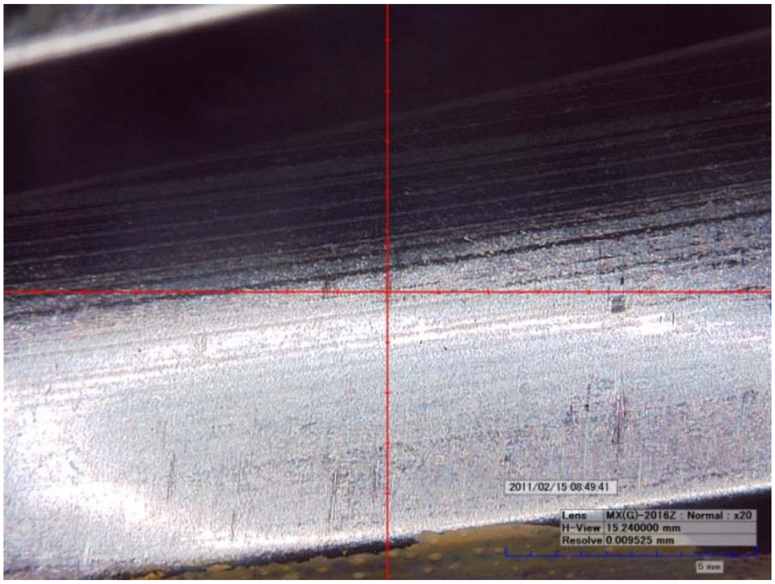


Image 7.8.2.4.1-1 P1MIDC001 Post PFPE Stage 3

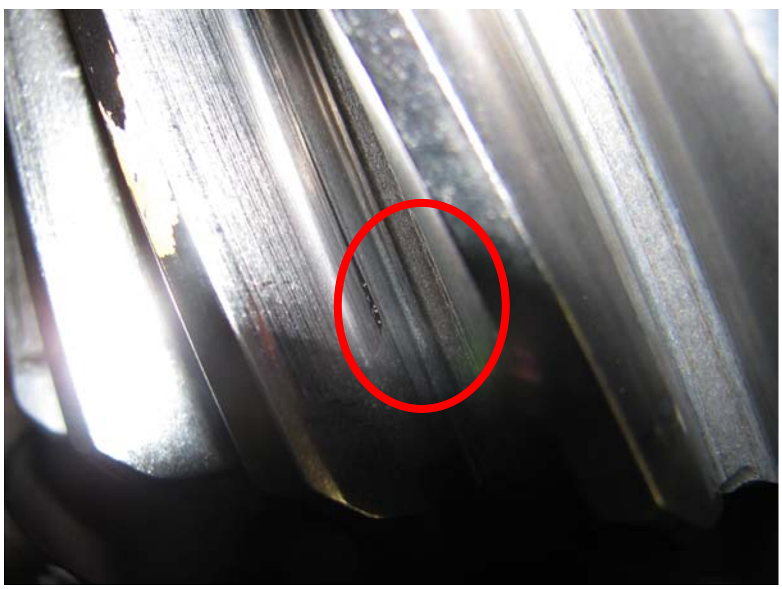


Image 7.8.2.4.1-2 Images of pits found near root of tooth

Image 7.8.2.4.1-3 Close up of pits found near root of tooth

7.8.2.5 Progressive Stage 4

7.8.2.5.1 Stage Performance

Table 7.8.2.5-1 shows the average values attained for the lubricant as it underwent Progressive Stage 4.

PFPE			Progressive Stage 4
Test Parameters	HS RPM	RPM	549.8
	Torques Averaged	Nm	6233.9
	VFD current	A	511.7
Duration	Hours	hr	22.08
	# of Cycles	-	1.46E+06
	% of Fzg Cycles (1.44e^6)	%	101%
Temperatures	Sump Temp	°C	87.32
	Manifold Temp	°C	79.51
	Radiator Inlet Temp	°C	83.49
	Pinion Tooth Root Temp	°C	93.32
Lubricant	Flow Rate	GPM	25.17
	Pressure	psi	88.90
Calculated Values	Film Thickness	µm	0.303
	Lambda Ratio	-	1.009
	Viscosity	cst	88.20
	Dynamic Viscosity	$N{\cdot}s/m^2$	158.41
	% Film Thickness Reduction	%	5%
	Efficiency Metric	Nm/A	12.18
Success Criteria	Pass/Fail	-	Pass

Table 7.8.2.5-1: PFPE Progressive Load Stage 4 Summary

7.8.2.5.2 Progressive Load Stage (PLS) 4 Notables

The “notable” following this stage is actually the lack of notable items- no further progression of pits incurred in the previous stage was observed. Image of location P1MIDC001 Stage 4 End:

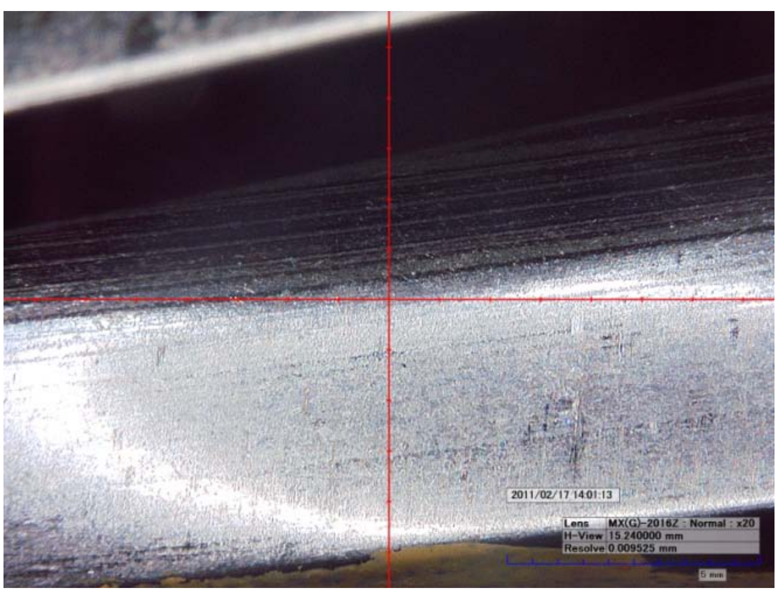


Image7.8.2.5.1-1 P1MIDC001 Post PFPE Stage 4

7.8.2.6 Progressive Stage 5

Table 7.8.2.6-1 shows the average values attained for the lubricant as it underwent Progressive Stage 5.

PFPE			Progressive Stage 5A	Progressive Stage 5B
Test Parameters	HS RPM	RPM	399.9	450.0
	Torques Averaged	Nm	6214.0	6230.5
	VFD current	A	525.7	508.6
Duration	Hours	hr	35.33	23.17
	# of Cycles	-	1.70E+06	1.25E+06
	% of Fzg Cycles (1.44e^6)	%	118%	87%
Temperatures	Sump Temp	°C	80.37	96.04
	Manifold Temp	°C	73.44	83.99
	Radiator Inlet Temp	°C	77.32	91.41
	Pinion Tooth Root Temp	°C	84.32	95.77
Lubricant	Flow Rate	GPM	16.61	20.63
	Pressure	psi	63.56	63.14
Calculated Values	Film Thickness	µm	0.273	0.233
	Lambda Ratio	-	0.991	0.776
	Viscosity	cst	98.03	74.62
	Dynamic Viscosity	$N{\cdot}s/m^2$	176.47	133.50
	% Film Thickness Reduction	%	2%	23%
	Efficiency Metric	Nm/A	11.82	12.25
Success Criteria	Pass/Fail	-	Pass	Pass

Table 7.8.2.6-1: PFEP Progressive Load Stages 5A and 5B Summary

7.8.2.6.1 Progressive Load Stage (PLS) 5 Notables

The initial attempt at Stage 5 was to run at 400 RPM and 90°C, however the internal heat generation of the gearbox was not enough to overcome the convective heat loss from the gearbox surfaces. The first run settled at 80°C, well below the target and resulted in a negligible reduction in film thickness. The stage was re run at 450 RPM and 95°C after an immersion heater, inline heaters and insulation described in section 7.7 were installed. With this additional hardware sufficient sump temperatures could be maintained and the test was resumed.

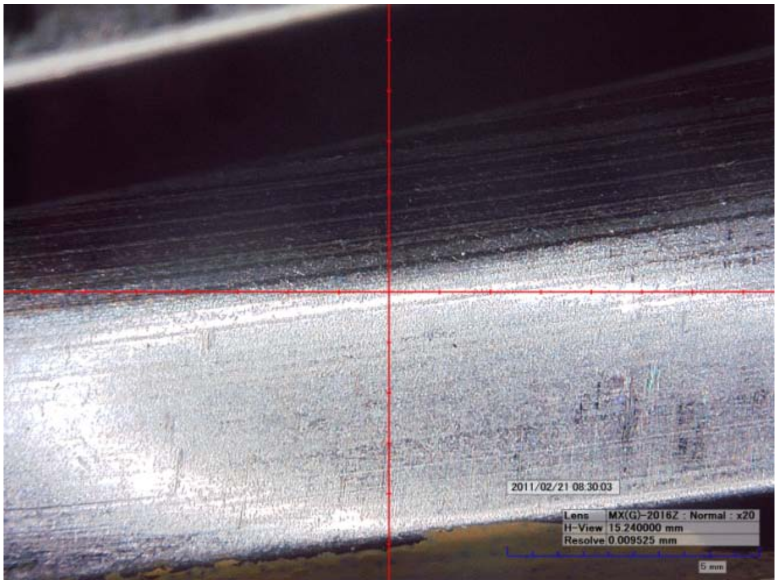


Image 7.8.2.6.1-1 P1MIDC001 Post PFPE Stage 5A

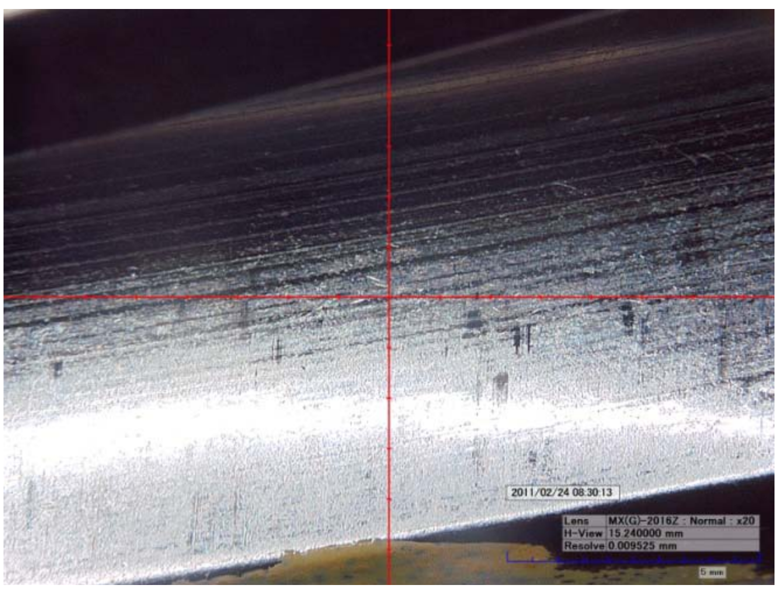


Image 7.8.2.6.1-2 P1MIDC001 Post PFPE Stage 5B

7.8.2.7 Progressive Stage 6

Table 7.8.2.7-1 shows the average values attained for the lubricant as it underwent Progressive Stage 6.

PFPE			Progressive Stage 6
Test Parameters	HS RPM	RPM	350.0
	Torques Averaged	Nm	6221.8
	VFD current	A	511.2
Duration	Hours	hr	41.35
	# of Cycles	-	1.74E+06
	% of Fzg Cycles (1.44e^6)	%	121%
Temperatures	Sump Temp	°C	96.04
	Manifold Temp	°C	83.38
	Radiator Inlet Temp	°C	89.69
	Pinion Tooth Root Temp	°C	93.74
Lubricant	Flow Rate	GPM	16.14
	Pressure	psi	47.16
Calculated Values	Film Thickness	µm	0.197
	Lambda Ratio	-	0.657
	Viscosity	cst	74.62
	Dynamic Viscosity	$N{\cdot}s/m^2$	133.50
	% Film Thickness Reduction	%	15%
	Efficiency Metric	Nm/A	12.17
Success Criteria	Pass/Fail	-	Pass

Table 7.8.2.7-1: PFPE Progressive Load Stage 6 Summary

7.8.2.7.1 Progressive Load Stage (PLS) 6 Notables

Stage 6 resulted in nothing of note.

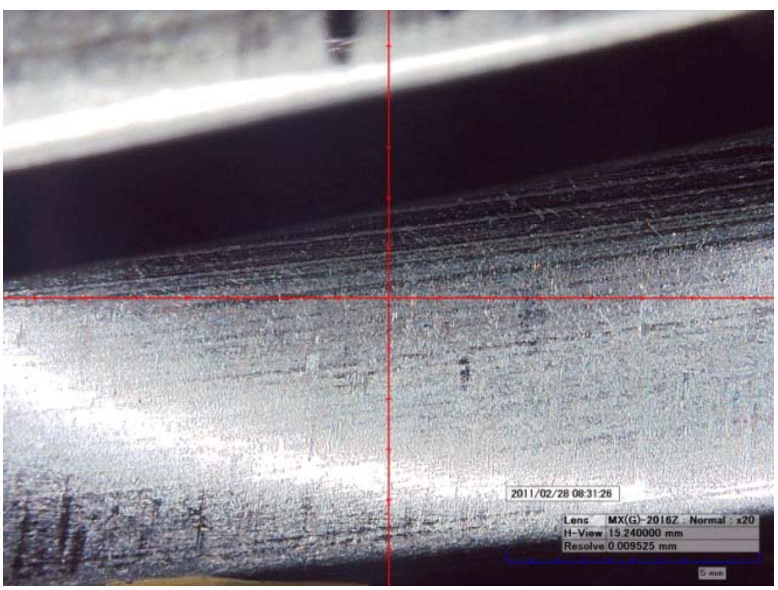


Image 7.8.2.7.1-1 P1MIDC001 Post PFPE Stage 6

7.8.2.8 Progressive Stage 7

Table 7.8.2.8-1 shows the average values attained for the lubricant as it underwent Progressive Stage 7.

PFPE			Progressive Stage 7
Test Parameters	HS RPM	RPM	200.0
	Torques Averaged	Nm	6227.0
	VFD current	A	536.9
Duration	Hours	hr	60.43
	# of Cycles	-	1.45E+06
	% of Fzg Cycles (1.44e^6)	%	101%
Temperatures	Sump Temp	°C	98.98
	Manifold Temp	°C	81.77
	Radiator Inlet Temp	°C	91.53
	Pinion Tooth Root Temp	°C	90.65
Lubricant	Flow Rate	GPM	9.38
	Pressure	psi	24.65
Calculated Values	Film Thickness	µm	0.132
	Lambda Ratio	-	0.44
	Viscosity	cst	72.26
	Dynamic Viscosity	$N{\cdot}s/m^2$	129.17
	% Film Thickness Reduction	%	33%
	Efficiency Metric	Nm/A	11.60
Success Criteria	Pass/Fail	-	Pass

Table 7.8.2.8-1: PFPE Progressive Load Stage 7 Summary

7.8.2.8.1 Progressive Load Stage (PLS) 7 Notables

The lower bounds of the GTS speed capabilities were reached- thus, though smaller film thicknesses may be calculated, the fluid was tested to the minimum possible film thickness available hardware achieve. Throughout the stages, the gear teeth achieved a much more polished appearance, broken up only by certain blemishes such as debris damage, slight directional wear indicators which did not seem to wear further after their initial appearances and the pits observed after stage 3. Tooth surface appeared in good shape overall, despite the abusive conditions of the test.

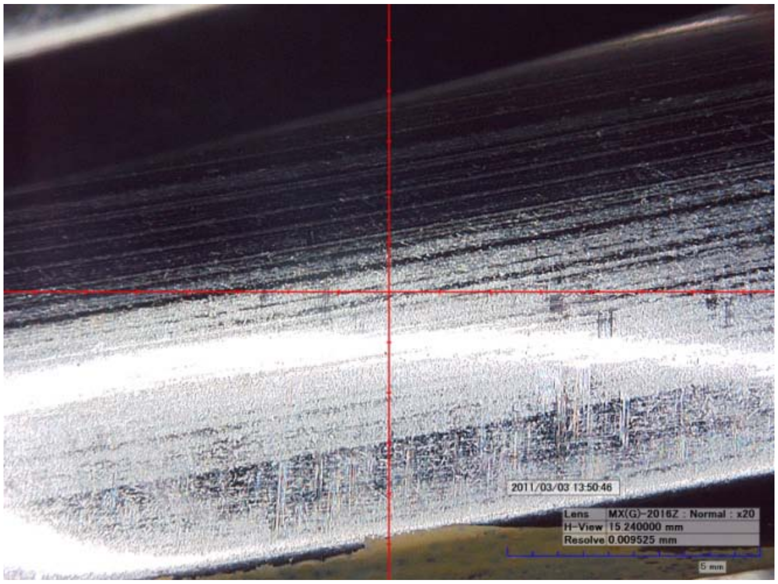


Image 7.8.2.8.1-1 P1MIDC001 Post PFPE Stage 7, End of Test

7.8.3 PAO Reference Results

7.8.3.1 PAO Reference Baseline Stage

Table 7.8.3.1-1 shows the average values attained for the lubricant as it underwent this "baseline" testing.

PAO Reference			Baseline
Test Parameters	HS RPM	RPM	850.1
	Torques Averaged	Nm	6297.0
	VFD current	A	579.8
Duration	Hours	hr	22.96
	# of Cycles	-	2.34E+06
	% of Fzg Cycles (1.44e^6)	%	163%
Temperatures	Sump Temp	°C	57.50
	Manifold Temp	°C	50.29
	Radiator Inlet Temp	°C	54.17
	Pinion Tooth Root Temp	°C	65.71
Lubricant	Flow Rate	GPM	37.84
	Pressure	psi	130.16
Calculated Values	Film Thickness	µm	0.286
	Lambda Ratio	-	0.953
	Viscosity	cst	140.75
	Dynamic Viscosity	$N \cdot s/m^2$	118.51
	% Film Thickness Reduction	%	-
	Efficiency Metric	Nm/A	10.86
Success Criteria	Pass/Fail	-	Pass

Table 7.8.3.1-1: PAO Reference Baseline Load Stage Summary

7.8.3.1.1 Baseline Stage Notables

The baseline inspection was uneventful.

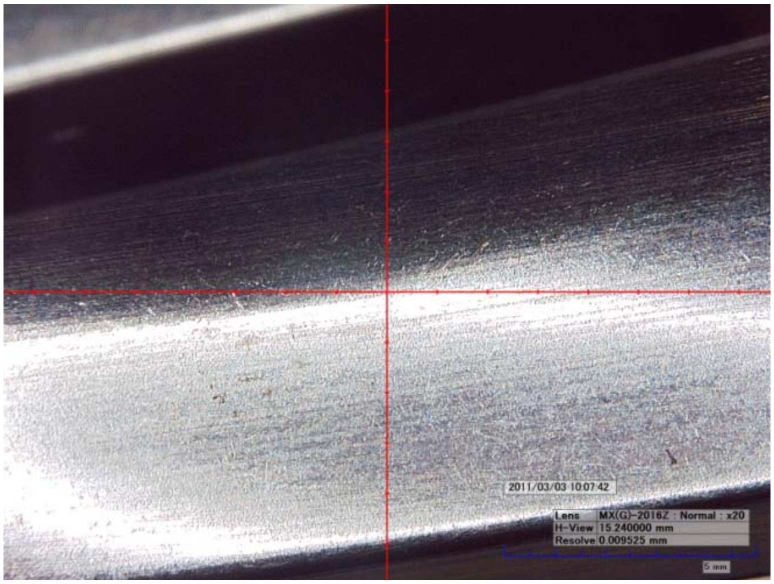


Image 7.8.3.1.1-1 P1MIDC001 PAO Reference Pinion Pre Test

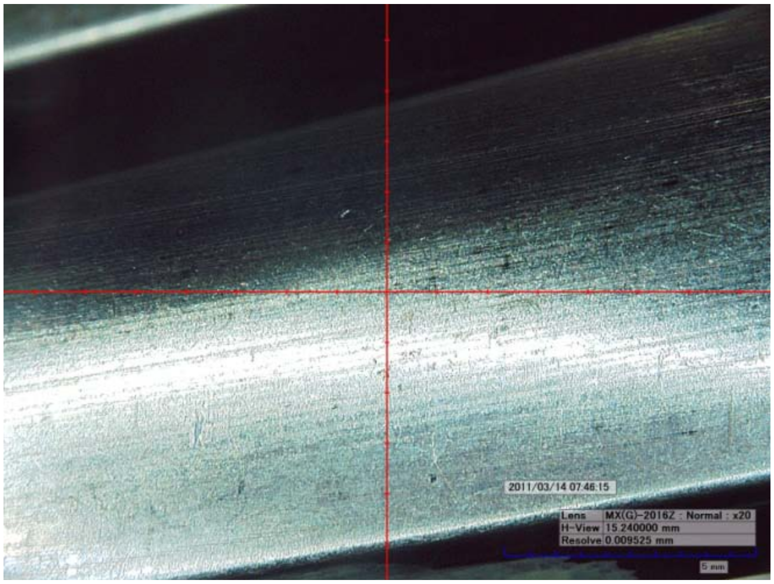


Image7. 8.3.1.1-2 P1MIDC001 Post PAO Reference Baseline Stage

7.8.3.2 PAO Reference Stage 1

Table 7.8.3.2-1 shows the average values attained for the lubricant as it underwent Progressive Stage 1.

PAO Reference			Progressive Stage 1
Test Parameters	HS RPM	RPM	450.0
	Torques Averaged	Nm	6294.0
	VFD current	A	539.9
Duration	Hours	hr	30.55
	# of Cycles	-	1.65E+06
	% of Fzg Cycles (1.44e^6)	%	115%
Temperatures	Sump Temp	°C	55.10
	Manifold Temp	°C	48.64
	Radiator Inlet Temp	°C	53.05
	Pinion Tooth Root Temp	°C	64.31
Lubricant	Flow Rate	GPM	20.73
	Pressure	psi	64.03
Calculated Values	Film Thickness	µm	0.215
	Lambda Ratio	-	0.716
	Viscosity	cst	153.82
	Dynamic Viscosity	$N{\cdot}s/m^2$	129.73
	% Film Thickness Reduction	%	25%
	Efficiency Metric	Nm/A	11.66
Success Criteria	Pass/Fail	-	Pass

Table 7.8.3.2-1: PAO Reference Progressive Load Stage 1 Summary

7.8.3.2.1 Progressive Load Stage (PLS) 1 Notables

The surface of the PAO Reference pinions appeared very similar to the PFPE pinions, slight marks are noted visually, but not via dental picks or graphite powder. P1MIDC001 PAO end stage 1

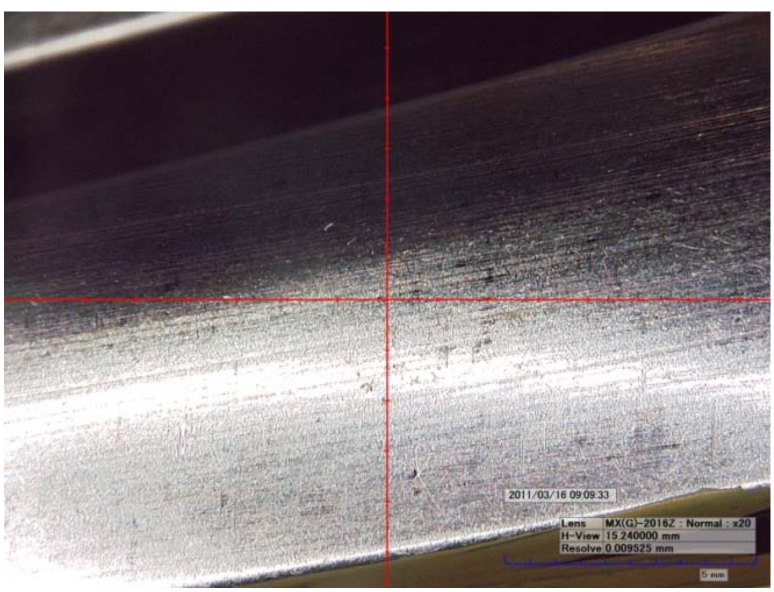


Image 7.8.3.2.1-1 P1MIDC001 Post PAO Reference Stage 1

7.8.3.3 PAO Reference Stage 2

Table 7.8.3.3-1 shows the average values attained for the lubricant as it underwent Progressive Stage 2.

PAO Reference			Progressive Stage 2
Test Parameters	HS RPM	RPM	350.0
	Torques Averaged	Nm	6298.2
	VFD current	A	549.7
Duration	Hours	hr	35.14
	# of Cycles	-	1.48E+06
	% of Fzg Cycles (1.44e^6)	%	103%
Temperatures	Sump Temp	°C	55.65
	Manifold Temp	°C	44.75
	Radiator Inlet Temp	°C	52.26
	Pinion Tooth Root Temp	°C	63.87
Lubricant	Flow Rate	GPM	16.19
	Pressure	psi	55.45
Calculated Values	Film Thickness	µm	0.184
	Lambda Ratio	-	0.614
	Viscosity	cst	153.82
	Dynamic Viscosity	$N\cdot s/m^2$	129.73
	% Film Thickness Reduction	%	14%
	Efficiency Metric	Nm/A	11.46
Success Criteria	Pass/Fail	-	Pass

Table 7.8.3.3-1: PAO Reference Progressive Load Stage 2 Summary

7.8.3.3.1 Progressive Load Stage (PLS) 2 Notables

Nothing of note.

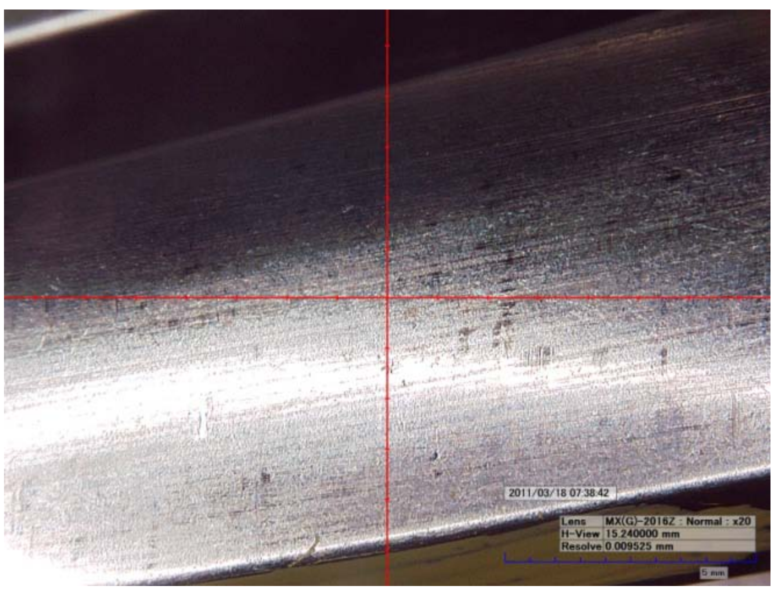


Image7.8.3.3.1-1 P1MIDC001 Post PAO Reference Stage 2

7.8.3.4 PAO Reference Stage 3

Table 7.8.3.4-1 shows the average values attained for the lubricant as it underwent Progressive Stage 3.

PAO Reference			Progressive Stage 3
Test Parameters	HS RPM	RPM	249.8
	Torques Averaged	Nm	6299.8
	VFD current	A	570.3
Duration	Hours	hr	47.63
	# of Cycles	-	1.43E+06
	% of Fzg Cycles (1.44e^6)	%	99%
Temperatures	Sump Temp	°C	56.36
	Manifold Temp	°C	38.66
	Radiator Inlet Temp	°C	51.93
	Pinion Tooth Root Temp	°C	60.46
Lubricant	Flow Rate	GPM	11.60
	Pressure	psi	50.68
Calculated Values	Film Thickness	µm	0.144
	Lambda Ratio	-	0.481
	Viscosity	cst	140.75
	Dynamic Viscosity	$N \cdot s/m^2$	118.51
	% Film Thickness Reduction	%	22%
	Efficiency Metric	Nm/A	11.05
Success Criteria	Pass/Fail	-	Pass

Table 7.8.3.4-1: PAO Reference Progressive Load Stage 3 Summary

7.8.3.4.1 Progressive Load Stage (PLS) 3 Notables

Nothing of note.

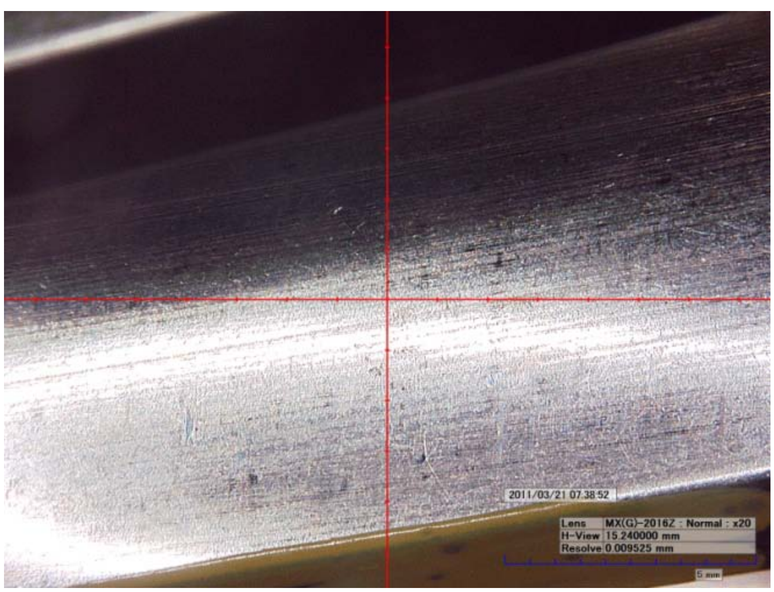


Image 7.8.3.4.1-1 P1MIDC001 Post PAO Reference Stage 3

7.8.3.5 PAO Reference Stage 4

Table 7.8.3.5-1 shows the average values attained for the lubricant as it underwent Progressive Stage 4.

PAO Reference			Progressive Stage 4A	Progressive Stage 4B
Test Parameters	HS RPM	RPM	800.1	549.8
	Torques Averaged	Nm	6295.3	6298.0
	VFD current	A	514.3	510.8
Duration	Hours	hr	15.03	40.96
	# of Cycles	-	1.44E+06	2.70E+06
	% of Fzg Cycles (1.44e^6)	%	100%	188%
Temperatures	Sump Temp	°C	83.95	70.36
	Manifold Temp	°C	72.26	65.01
	Radiator Inlet Temp	°C	78.41	67.20
	Pinion Tooth Root Temp	°C	89.32	78.90
Lubricant	Flow Rate	GPM	33.59	25.32
	Pressure	psi	57.65	51.63
Calculated Values	Film Thickness	µm	0.134	0.154
	Lambda Ratio	-	0.447	0.514
	Viscosity	cst	52.35	79.78
	Dynamic Viscosity	$N\cdot s/m^2$	43.13	66.39
	% Film Thickness Reduction	%	7%	-7%
	Efficiency Metric	Nm/A	12.24	12.33
Success Criteria	Pass/Fail	-	Fail	Pass

Table 7.8.3.5-1: PAO Reference Progressive Load Stages 4A and 4B Summary

7.8.3.5.1 Progressive Load Stage (PLS) 4 Notables

PAO Reference Stage 4 was run twice due to the inline heater seemingly “burning” the fluid in the first attempt. Further lubricant analysis was undertaken, and presented in an independent report which may increase the team’s knowledge of the burned lubricant condition. The gearbox was refilled, at which point the replacement fluid successfully completed a secondary stage at equivalent film thickness, though at dramatically reduced speed and temperature targets.

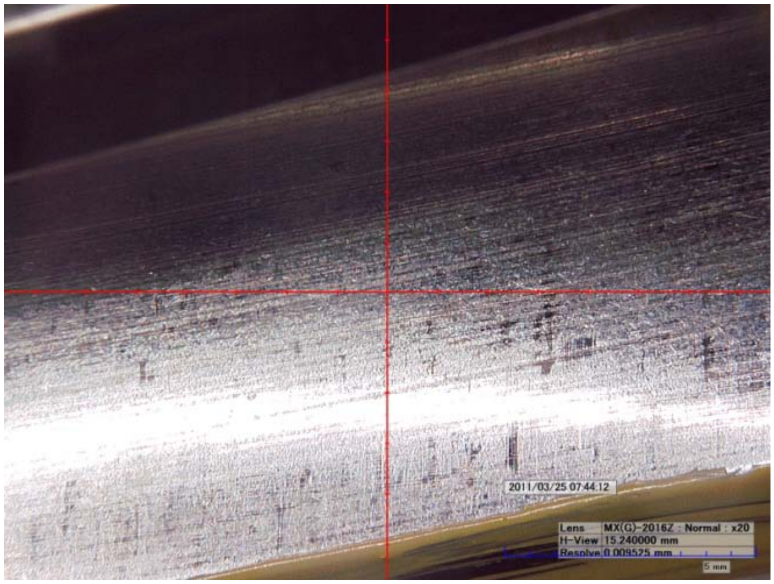


Image 7.8.3.5.1-1 P1MIDC001 Post PAO Reference Stage 4A

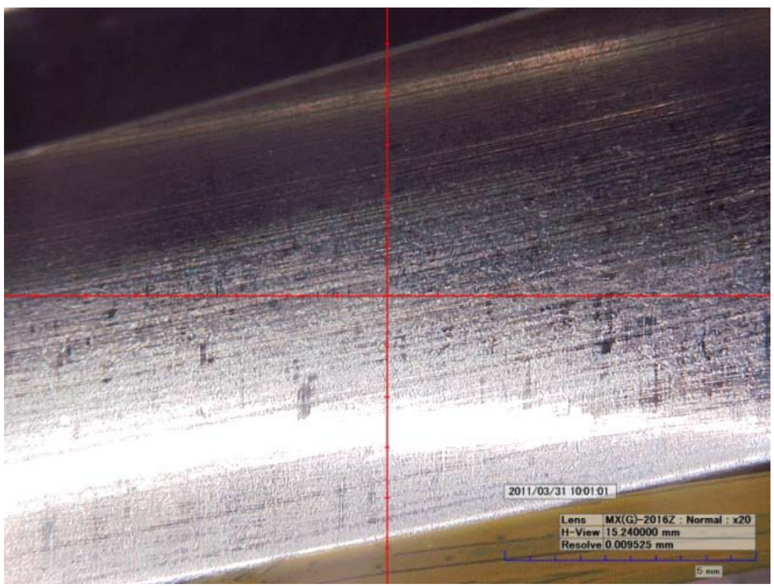


Image 7.8.3.5.1-2 P1MIDC001 Post PAO Reference Stage 4B

7.8.3.6 PAO Reference Stage 5

Table 7.8.3.6-1 shows the average values attained for the lubricant as it underwent Progressive Stage 5.

PAO Reference			Progressive Stage 5A	Progressive Stage 5B
Test Parameters	HS RPM	RPM	400.0	350.0
	Torques Averaged	Nm	6300.3	6299.2
	VFD current	A	511.1	518.4
Duration	Hours	hr	15.57	49.41
	# of Cycles	-	7.47E+05	2.08E+06
	% of Fzg Cycles (1.44e^6)	%	52%	144%
Temperatures	Sump Temp	°C	68.68	66.82
	Manifold Temp	°C	62.15	59.78
	Radiator Inlet Temp	°C	64.43	62.76
	Pinion Tooth Root Temp	°C	77.47	75.73
Lubricant	Flow Rate	GPM	18.65	16.54
	Pressure	psi	35.89	32.90
Calculated Values	Film Thickness	µm	0.13	0.127
	Lambda Ratio	-	0.434	0.423
	Viscosity	cst	86.05	92.98
	Dynamic Viscosity	$N \cdot s/m^2$	71.73	77.64
	% Film Thickness Reduction	%	16%	18%
	Efficiency Metric	Nm/A	12.33	12.15
Success Criteria	Pass/Fail	-	Pass	Pass

Table 7.8.3.6-1: PAO Reference Progressive Load Stages 5A and 5B Summary

7.8.3.6.1 Progressive Load Stage (PLS) 5 Notables

Stage 5 was adjusted mid way through as the system was having trouble maintaining temperature and a significant decrease in film thickness was not achieved. As such stage 5A was halted and replaced with stage 5B, in which the same film thickness was achieved by utilizing slower rotation. This was necessary to create the desired 20% reduction in film thickness from the previous stage.

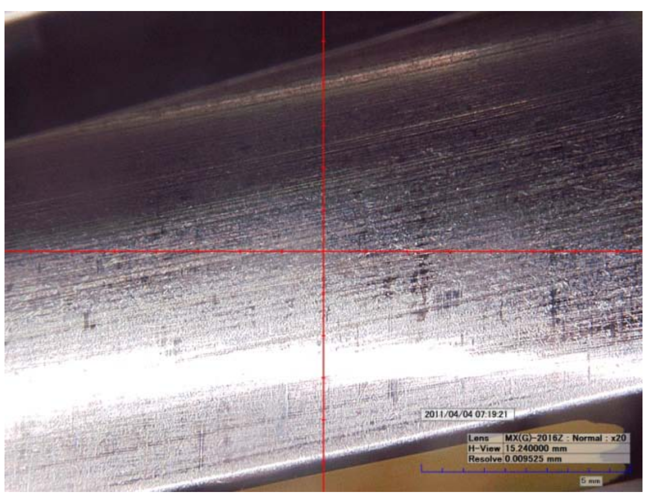


Image 7.8.3.6.1-1 P1MIDC001 Post PAO Reference Stage 5B

7.8.3.7 PAO Reference Stage 6

Table 7.8.3.7-1 shows the average values attained for the lubricant as it underwent Progressive Stage 6.

PAO Reference			Progressive Stage 6
Test Parameters	HS RPM	RPM	249.8
	Torques Averaged	Nm	6299.8
	VFD current	A	551.2
Duration	Hours	hr	39.76
	# of Cycles	-	1.19E+06
	% of Fzg Cycles (1.44e^6)	%	83%
Temperatures	Sump Temp	°C	59.61
	Manifold Temp	°C	52.54
	Radiator Inlet Temp	°C	56.25
	Pinion Tooth Root Temp	°C	69.81
Lubricant	Flow Rate	GPM	12.06
	Pressure	psi	28.51
Calculated Values	Film Thickness	µm	0.127
	Lambda Ratio	-	0.423
	Viscosity	cst	129.05
	Dynamic Viscosity	$N \cdot s/m^2$	108.48
	% Film Thickness Reduction	%	0%
	Efficiency Metric	Nm/A	11.43
Success Criteria	Pass/Fail	-	Pass

Table 7.8.3.7-1: PAO Reference Progressive Load Stage 6 Summary

7.8.3.7.1 Progressive Load Stage (PLS) 6 Notables

The limits of speed and temperature were reached for the PAO Reference lubricant. Without the ability to supplement the sump heaters with an immersion heater, for risk of burning the fluid, such slow rotation speed did not generate sufficient heat to reduce the film thickness any further.

This point marked the end of the test. Image P1MIDC001 PAO End of Test:

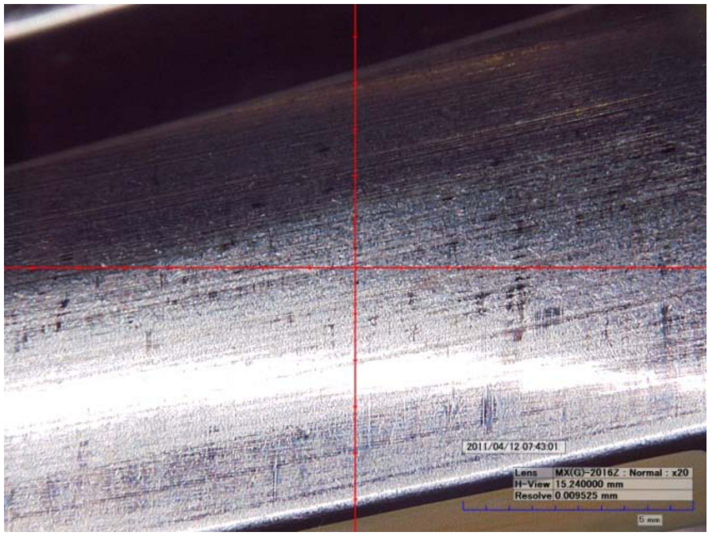


Image 7.8.3.7.1-1 P1MIDC001 Post PAO Reference Stage 6, End of Test

7.8.4 Surface Finish Results

Surface finish was checked before and after the full test to determine the level of polishing or running in of the gears. Unfortunately the precision of the profilometer was changed such that the initial values of the PAO pinions were only recorded to 1 significant digit. This significantly limits the ability to make comparisons between the results, as in most cases the margin of error of the initial values are greater than the change in roughness.

The initial values of average roughness are used in the calculations to determine the lambda ratio of the fluid and determine the severity of the load stage. Since there was not a large change in the average roughness the results and calculated values are accurate.

The results indicate that some "running in" of the gears occurred as in most of the cases the pinions average surface roughness decreased from the initial values.

This result is very subjective by this method and the actual post test profile deviation of the gears for both PFPE and PAO was performed by a third party metallurgical laboratory and in addition, the results were reviewed by an external gear design consultant for which the conclusion was; "There are no indications of measurable wear". The full reports are included in Appendix G.

PFPE Comparison							
Pre-test values less Post-test values							
Pinion 1		Upwind (UPW)		MID		Downwind (DWN)	
		R_A	R_k	R_A	R_k	R_A	R_k
	A	-0.02	0.29	0.07	0.52	0.05	0.44
	B	0.15	0.47	0.08	1.99	0.04	0.99
	C	0.15	0.77	0.10	0.23	0.05	0.13
	D	0.03	0.90	0.00	0.16	0.02	0.45
Pinion 2							
	A	0.05	0.77	0.08	0.24	0.07	-0.07
	B	0.10	0.81	0.05	-0.36	0.08	0.53
	C	0.05	0.15	0.07	0.49	0.03	-0.59
	D	0.09	0.07	0.02	0.37	0.03	0.04
Pinion 3							
	A	0.11	0.20	0.07	0.08	-0.01	0.08
	B	0.04	0.85	0.07	0.52	-0.03	0.01
	C	0.00	-0.16	0.10	0.54	0.00	-0.12
	D	0.08	0.24	0.00	0.27	0.02	-0.66
Pinion 4							
	A	0.10	0.16	0.07	0.39	0.00	-0.02
	B	0.03	-0.04	0.07	0.44	0.00	-0.23
	C	0.01	-0.38	0.10	0.38	0.04	0.05
	D	0.06	0.25	0.03	-0.09	0.03	0.19

Table 7.8.4-1: PFPE Surface Finish Values

PAO Reference Comparison							
Pre-test values less Post-test values							
Pinion 1		Upwind (UPW)		MID		Downwind (DWN)	
		R_A	R_k	R_A	R_k	R_A	R_k
	A	-0.1	-0.4	0.0	0.4	0.0	0.2
	B	0.0	-0.1	-0.2	-0.1	-0.1	0.5
	C	0.2	0.3	0.3	0.5	0.2	0.5
	D	0.2	0.5	0.3	0.4	0.2	0.2
Pinion 2							
	A	-0.1	-0.1	-0.1	-0.1	0.1	0.4
	B	-0.1	-0.2	-0.2	-0.2	-0.1	-0.4
	C	0.2	0.5	0.2	0.5	0.3	0.5
	D	0.2	0.6	0.2	0.2	0.3	0.4
Pinion 3							
	A	-0.1	0.1	-0.2	-0.3	0.0	0.1
	B	-0.1	0.0	0.0	-0.1	-0.1	0.2
	C	0.2	0.5	0.3	0.8	0.2	0.3
	D	0.2	0.3	0.3	0.8	0.4	0.8
Pinion 4							
	A	-0.1	0.0	0.0	0.1	-0.1	-0.1
	B	0.0	0.6	-0.1	-0.3	-0.1	-0.4
	C	0.2	0.3	0.3	0.7	0.2	0.7
	D	0.2	0.3	0.2	0.8	0.2	0.4

Table 7.8.4-2: PAO Reference Surface Finish Values

7.8.5 Load distribution results, Kf

At the end of the test the load distribution of the high speed pinions were checked to ensure the test gearbox was within the production specification of a Clipper gearbox. The results are presented in Figure 7.8.5-1 below and indicated the test box was within the production specification limits.

Figure contains PROPRIETARY information and is withheld from the public version of this report.

Figure 7.8.5-1: Test Article Load Distribution Factors

7.9 Test Assessment and Analysis

7.9.1 Durability and Micro-pitting Protection

7.9.1.1 PFPE

The durability of the new PFPE lubricant was tested through prolonged evaluation at "baseline" load levels. The elevated torque allowed 406 hours of runtime to consume the equivalent of 1 year of design life of the gearing. The fluid passed this testing without any signs of gear problems or wear.

The staged testing added an additional 282 hours runtime on the fluid and subjected the gearing to much harsher conditions. Again the evaluation gears showed no micro-pitting or objectionable wear. By the final stage, film thickness had been reduced to just 21% of its original value, resulting in a film thickness of 0.132 and lambda ratio of 0.44. Such values are indicative of mixed lubrication regime where asperity contact should be occurring.

Over the whole test campaign there were only a few mentionable signs of changes to the gear surface. The pinions showed some isolated debris damage from contaminates in the gearbox. There were small signs of polishing and scuffing that were found to be in line with normal gear running. Also there were a few unique markings on one pinion from contact with the corner of another gear during improper installation. Fluid degradation was limited to discoloration associated with a slight level of cross contamination (due to repurposing a previously used gearbox). Further description of the damage marks and general surface condition of the gears can be found in Appendix G but the situation raised no concerns as evaluate by an external gear design consultant.

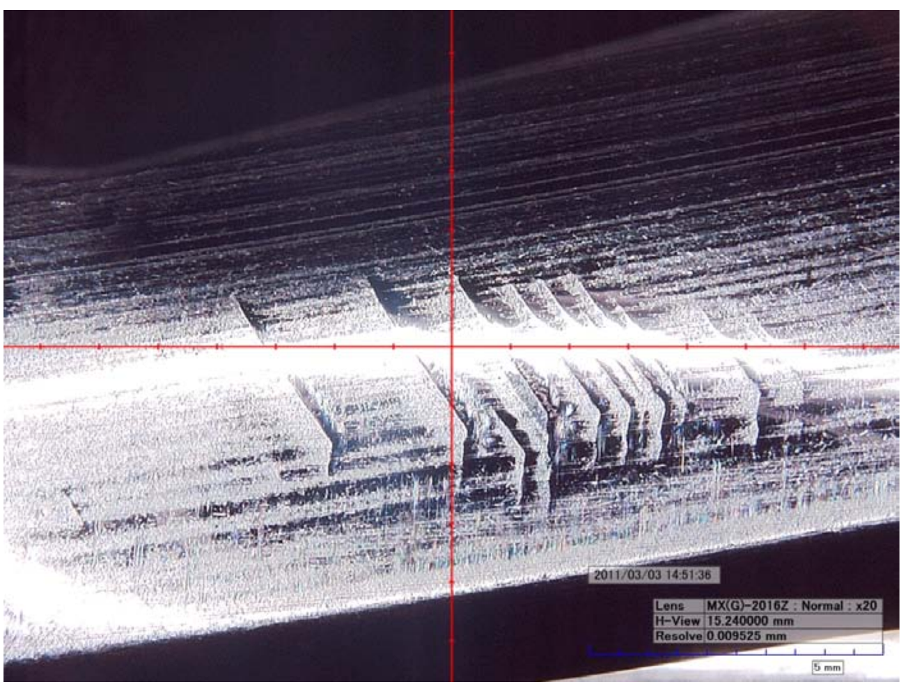


Figure 7.9.1.1-1: Image of Installation Damage Marks

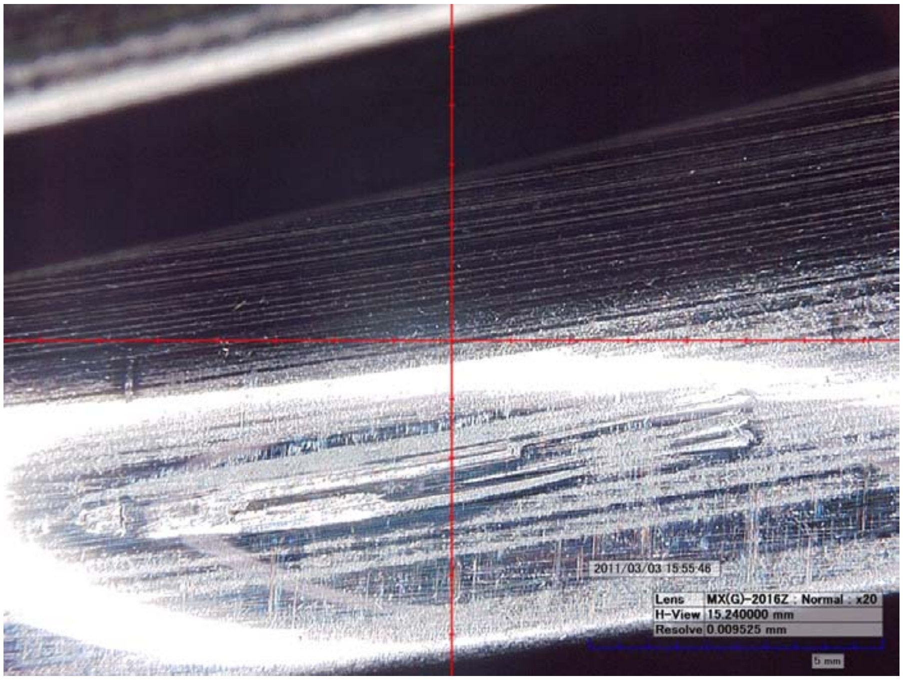


Figure 7.9.1.1-1: Image of Damage Marks

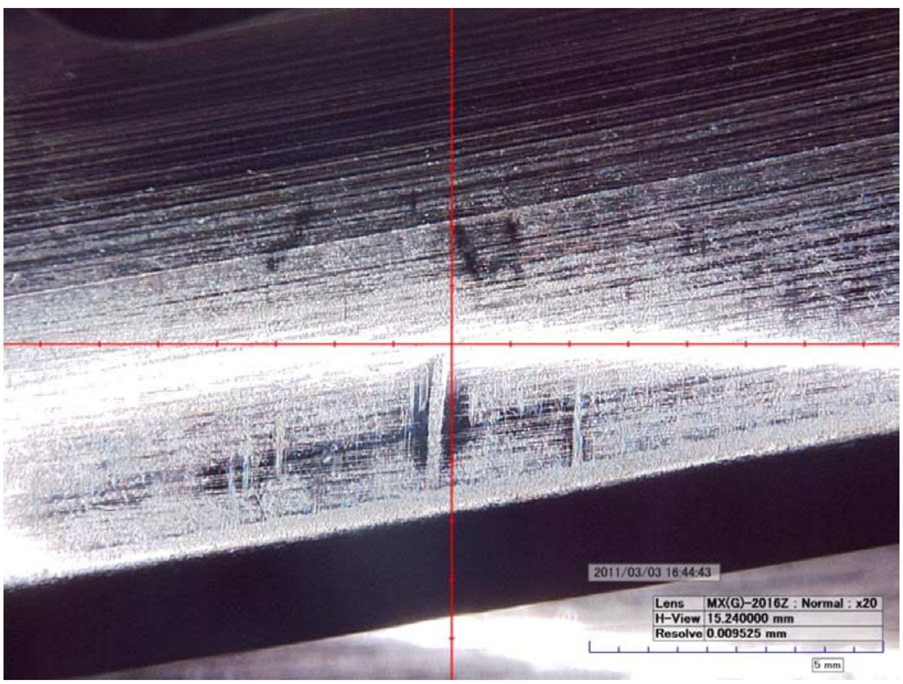


Figure 7.9.1.1-1: Image of Damage Marks

The only major point of concern was the pitting near the root of the tooth that occurred during stage 3 the PFPE testing. However, the occurrence was only evident on some teeth and did not progress, even while being subjected to progressively harsher load conditions and was thus deemed an anomaly.

The duration and severity of the testing seems to indicate that the lubricant could make a suitable industrial gearbox lubricant. Unfortunately the limits of the test equipment were reached before the limits of the fluid.

7.9.1.2 PAO Reference

In similar vein the PAO Reference fluid successfully passed all the load stages that it was subjected to. The total test accounted for 297 hours runtime on the fluid. Again the evaluation gears showed no micro-pitting or objectionable wear. By the final stage, film thickness had been reduced to just 44% of its original value, resulting in a film thickness of 0.127 and lambda ratio of 0.423. Such values are again indicative of mixed lubrication regime where asperity contact should be occurring.

7.9.1.3 Full Life

The full life durability of the PFPE cannot be quantified in the short duration of this test campaign. The test results show the PFPE survived approximately 800 hours of runtime which would result in approximately 2 years of operation on a wind turbine gearbox. However the torque load design life of a gearbox cannot be compared directly with the life of a lubricating fluid.

Further information regarding the full life durability of the fluids will be evaluated via testing and sampling of the PFPE and the PAO Reference in scaled laboratory tests as well as field sampling of wind turbine gearboxes with PAO Reference. The fluid samples were analyzed for degradation as well as additive loss and the results are included in Section 6 of this report.

7.9.1.3 Additional Comments

The high temperature running of the PFPE shows it has good resistance to degradation at high heat levels.

7.9.2 Efficiency- VFD Current

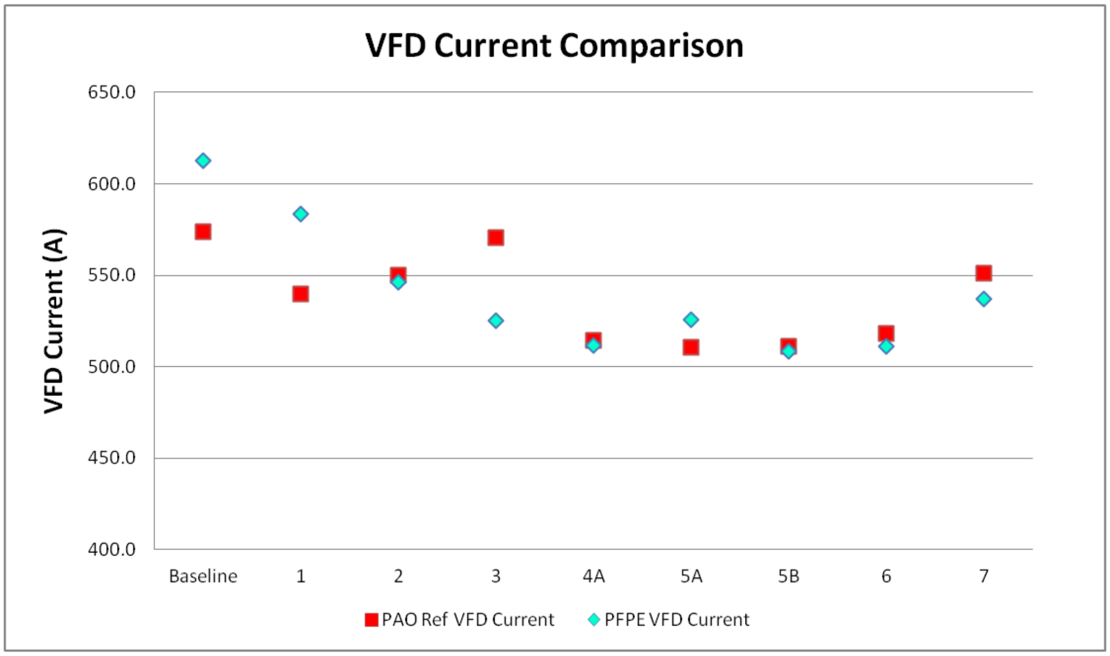


Figure 7.9.2-1: Stage Comparison of VFD Current for PFPE and PAO Reference

In addition to wear and micro-pitting performance an attempt was made to characterize the relative efficiency between the new fluid and the baseline lubricant. The results are varied as each fluid's properties and load stages were very different. A cursory examination of Figure 9.2-1 reveals that the PFPE lubricant, in three stages, required less drive current. In other stages, including baseline, the PAO Reference lubricant required significantly lower current for the target torque. Of note, Figure 9.2-1 does not take into consideration difference in load conditions and discrepancies in speed, temperature targets, and achieved torque significantly influence the VFD current.

For each stage the torque was divided by the current draw required to create an "Efficiency Metric" that could then be used to determine if any trends existed to any of the other measured parameters. The equation of the Efficiency metric (EM) is show here:

Efficiency Metric (Nm/A) = Average Torque (Nm) / Average Current Draw (A)

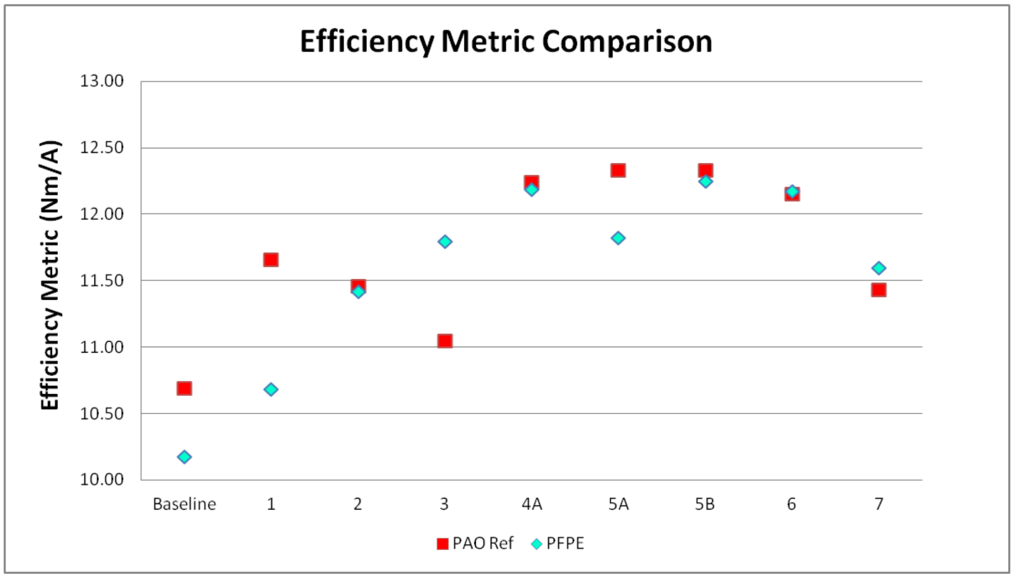


Figure 7.9.2-2: Stage Comparison of EM for PFPE and PAO Reference

The plot of efficiency metric vs. film thickness seems to show a trend that thicker film thicknesses result in reduced efficiency.

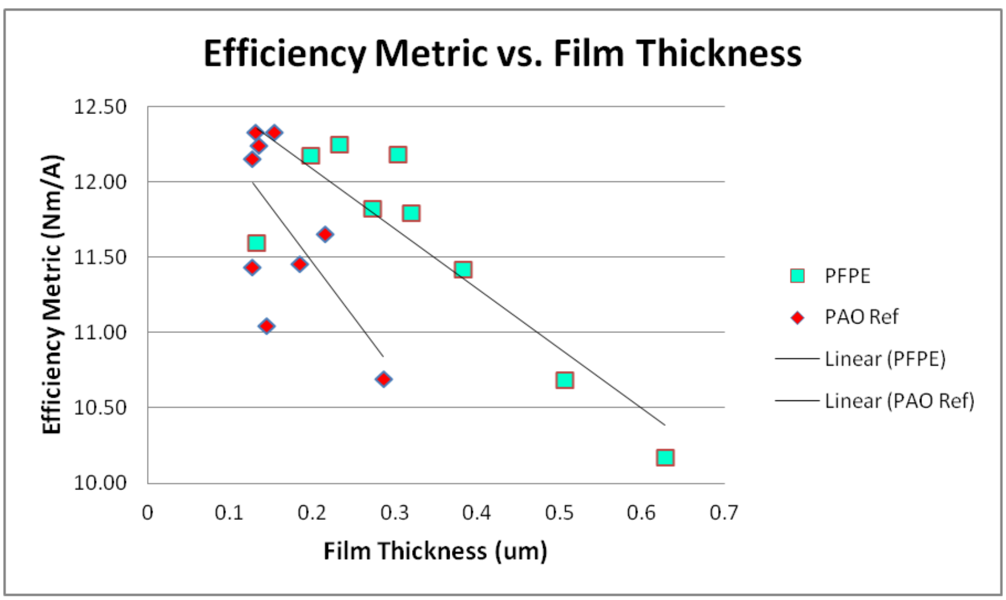


Figure 7.9.2-3: Trend Analysis EM vs. Film Thickness

Plots of efficiency vs. viscosity show a clear trend between viscosity and efficiency. This correlates well with the plot above as a high viscosity tends to create a thicker film.

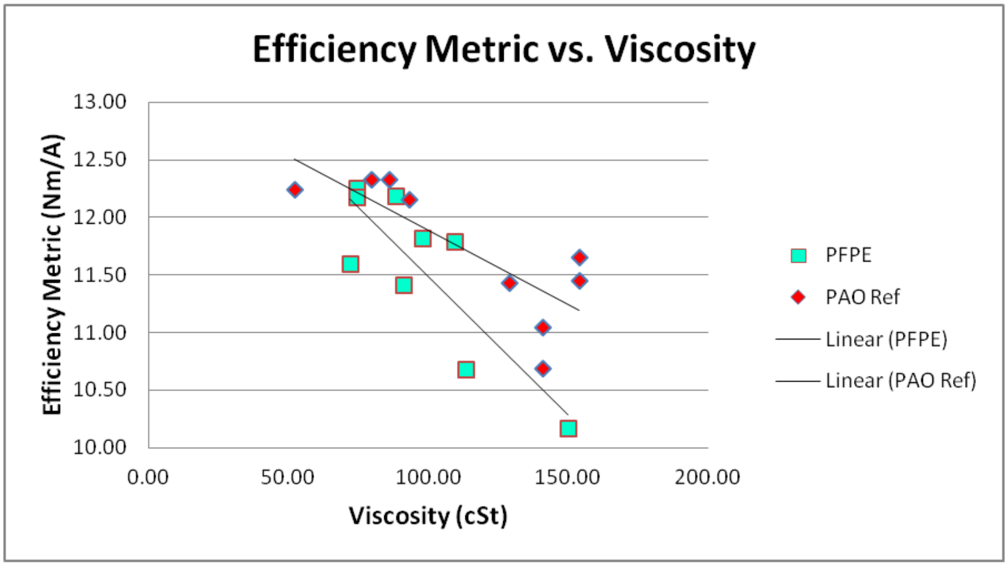


Figure 7.9.2-4: Trend Analysis EM vs. Viscosity

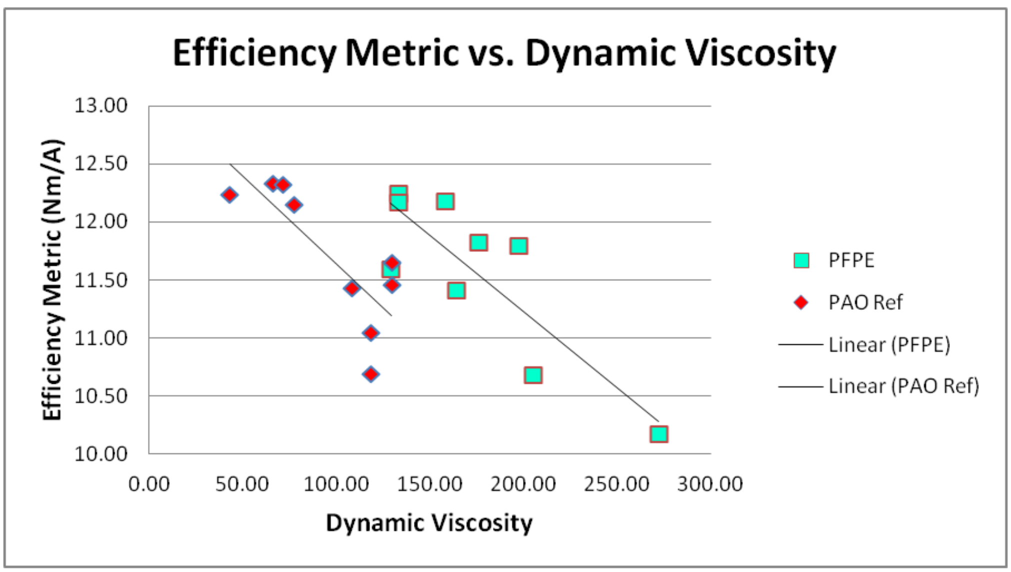


Figure 7.9.2-5: Trend Analysis EM vs. Dynamic Viscosity

An initial look at the baseline stage of each fluid shows that under similar load speed and temperature conditions the PAO Reference has a lower current draw and a higher efficiency. However it is important to note that under these conditions the lambda ratio of the PFPE is more than double that of the PAO Reference, which would seemingly indicate greater gear surface protection.

Efficiency Comparison		PFPE Baseline	PAO Ref Baseline	% Difference
HS RPM	RPM	850.1	850.1	0.0%
Pinion Torques Averaged	Nm	6228.4	6297.0	-1.1%
Hours	hr	406.3	23.0	1669.7%
Sump Temp	°C	58.8	57.5	2.3%
Manifold Temp	°C	45.6	50.3	-9.3%
Radiator Inlet Temp	°C	54.6	54.2	0.8%
Pinion Tooth Root Temp	°C	66.5	65.7	1.2%
Flow Rate	GPM	30.3	37.8	-19.8%
Pressure	psi	189.5	130.2	45.6%
Film Thickness	µm	0.628	0.286	119.6%
Lambda Ratio	-	2.09	0.95	119.5%
Viscosity	cst	150.0	140.7	6.6%
Dynamic Viscosity	$N{\cdot}s/m^2$	272.4	118.5	129.9%
VFD current	A	612.3	579.8	5.6%
Efficiency Metric	**Nm/A**	**10.2**	**10.9**	**-6.4%**

Table 7.9.2-1: Efficiency Comparison of Baseline Stage

In order to further evaluate the efficiency differences between the fluids it was important to look at stages with the same film thickness. Table 7.9.2-2 shows the relative Film Thicknesses of each stage, ordered from largest to smallest. It provides a visual representation of which stages, based on film thickness alone, are comparable.

PAO Reference		PFPE	
Stage #	Film Thickness (µm)	Film Thickness (µm)	Stage #
		0.628	baseline
		0.506	1
		0.382	2
		0.319	3
		0.303	4
baseline	0.286		
		0.273	5A
		0.233	5B
1	0.215		
		0.197	6
2	0.184		
4B	0.154		
3	0.144		
4A	0.134		
		0.132	7
5A	0.130		
6	0.127		
5B	0.127		

Table 7.9.2-2: Ordered Stage Comparison of Film Thickness

Good points for comparison were identified as:

- PAO Reference Stage 1 and PFPE Stage 5B
- PAO Reference Stage 6 and PFPE Stage 7

These points shared similar load, and speed conditions as well as similar values of film thickness. The load conditions, current requirements and film thicknesses are shown in the tables below.

The results seem to indicate that cases where the film thickness and load conditions are similar, the PFPE has a higher efficiency. However in conditions such as the baseline, where load and temperature are the same, the PAO Reference has a lower film thickness but higher efficiency.

Efficiency Comparison		PFPE Stage 5B	PAO Ref Stage 1	% Difference
HS RPM	RPM	450.0	450.0	0.0%
Pinion Torques Averaged	Nm	6230.5	6294.0	-1.0%
Hours	hr	23.2	30.6	-24.1%
Sump Temp	°C	96.0	55.1	74.3%
Manifold Temp	°C	84.0	48.6	72.7%
Radiator Inlet Temp	°C	91.4	53.0	72.3%
Pinion Tooth Root Temp	°C	95.8	64.3	48.9%
Flow Rate	GPM	20.6	20.7	-0.5%
Pressure	psi	63.1	64.0	-1.4%
Film Thickness	µm	0.233	0.215	8.4%
Lambda Ratio	-	0.78	0.72	8.4%
Viscosity	cst	74.6	153.8	-51.5%
Dynamic Viscosity	$N{\cdot}s/m^2$	133.5	129.7	2.9%
VFD current	A	508.6	539.9	-5.8%
Efficiency Metric	**Nm/A**	**12.2**	**11.7**	**5.1%**

Table 7.9.2-3: Efficiency Comparison of PFPE 5B and PAO Ref 1 Stages

Efficiency Comparison		PFPE Stage 7	PAO Ref Stage 6	% Difference
HS RPM	RPM	200.0	249.8	-19.9%
Pinion Torques Averaged	Nm	6227.0	6299.8	-1.2%
Hours	hr	60.4	39.8	52.0%
Sump Temp	°C	99.0	59.6	66.0%
Manifold Temp	°C	81.8	52.5	55.6%
Radiator Inlet Temp	°C	91.5	56.3	62.7%
Pinion Tooth Root Temp	°C	90.6	69.8	29.8%
Flow Rate	GPM	9.4	12.1	-22.2%
Pressure	psi	24.7	28.5	-13.5%
Film Thickness	µm	0.132	0.127	3.9%
Lambda Ratio	-	0.44	0.42	4.0%
Viscosity	cst	72.3	129.1	-44.0%
Dynamic Viscosity	$N{\cdot}s/m^2$	129.2	108.5	19.1%
VFD current	A	536.9	551.2	-2.6%
Efficiency Metric	**Nm/A**	**11.6**	**11.4**	**1.5%**

Table 7.9.2-4: Efficiency Comparison of PFPE Stage 7 and PAO Ref Stage 6

7.9.3 Efficiency- Temperature

In another attempt to investigate the efficiency of the two fluids, the operating temperatures of the sump, manifold, and radiator inlet were analyzed for steady state temperature, cooling system duty cycle and rate of temperature increase.

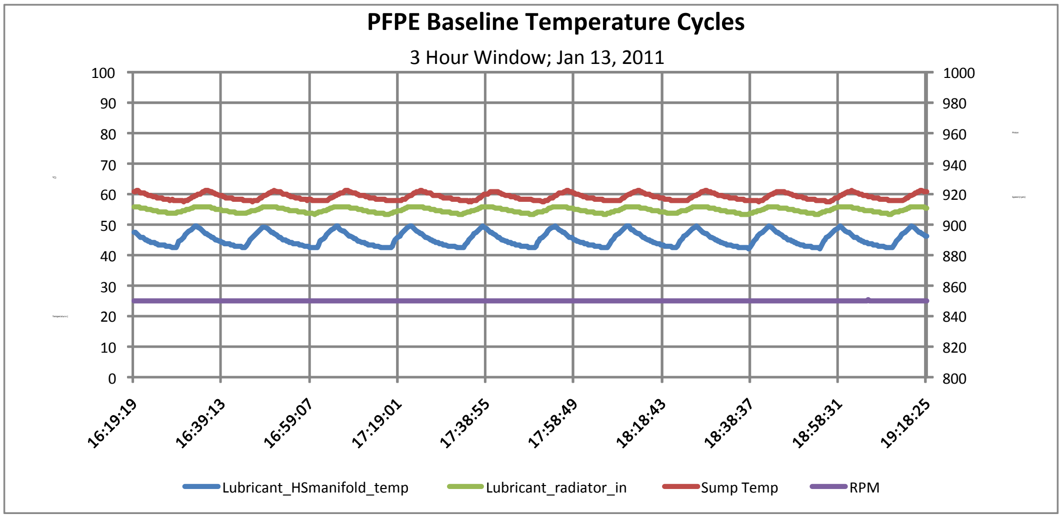


Figure 7.9.3-1: Steady State Temperature Plot of PFPE Baseline Stage

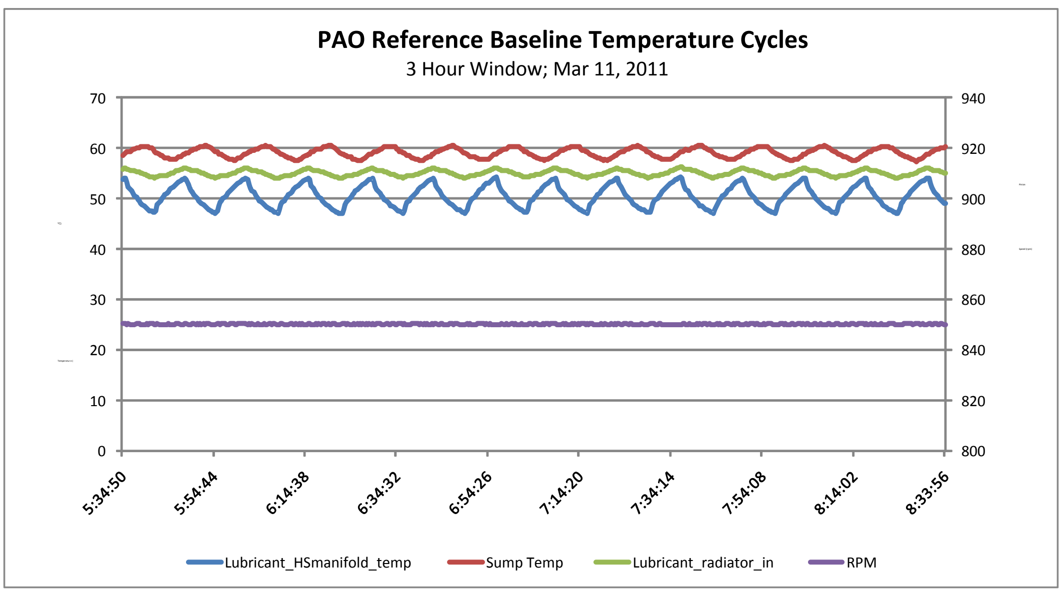


Figure 7.9.3-2: Steady State Temperature Plot of PAO Reference Baseline Stage

Unfortunately there were too many uncontrolled variables (ambient temperature, insulation, load stage differences, specific heat capacity of the fluids, etc.) to come to any conclusive results.

7.9.4 Efficiency- Thermal Images

Thermal images were taken of the gearbox housing surfaces during the baseline load stage for both tests. However similar to the temperature- efficiency analysis in the previous section there were too many variable to draw any conclusions from the thermal images.

Figure 7.9.4-1: PFPE Baseline Stage High Speed Pinion Thermal Image

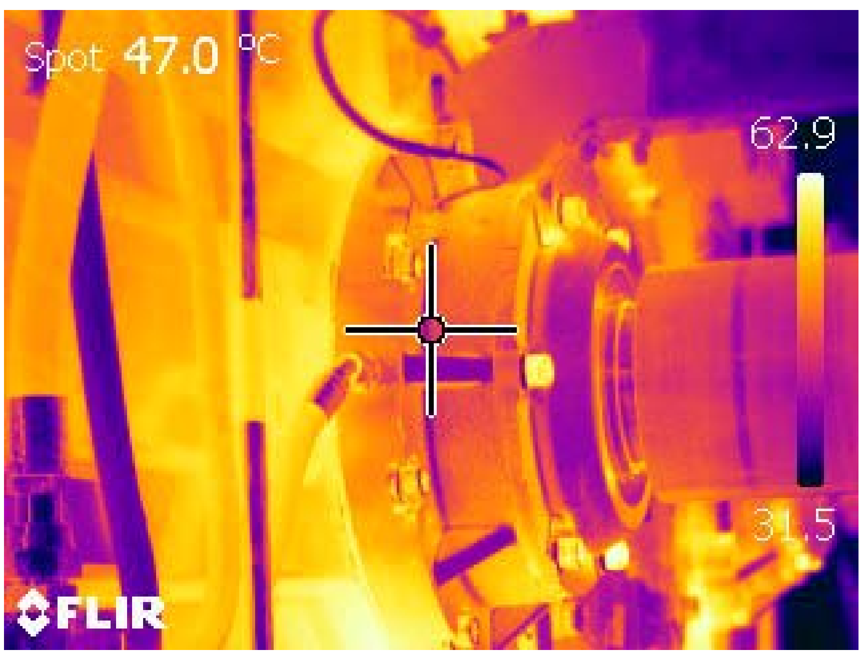


Figure 7.9.4-2: PAO Reference Baseline Stage High Speed Pinion Thermal Image

7.9.5 Torque Oscillation

During the data analysis phase an unexpected torque oscillation was found, the variation in torque for the duration of the PFPE testing was much greater than that of the PAO Reference. The figures below show the torque levels for a typical time segment during the baseline evaluation stages. The graphs are plotted on the same scale and over the same time period. Further investigation of the PFPE data shows peaks of up to 7000 Nm (+12%) and minimums of 5600 (-11%) in some stages. This variation is much greater than expected. Alternatively the PAO Reference fluid torque levels remain relatively constant throughout the test with only slight 1-2% variation.

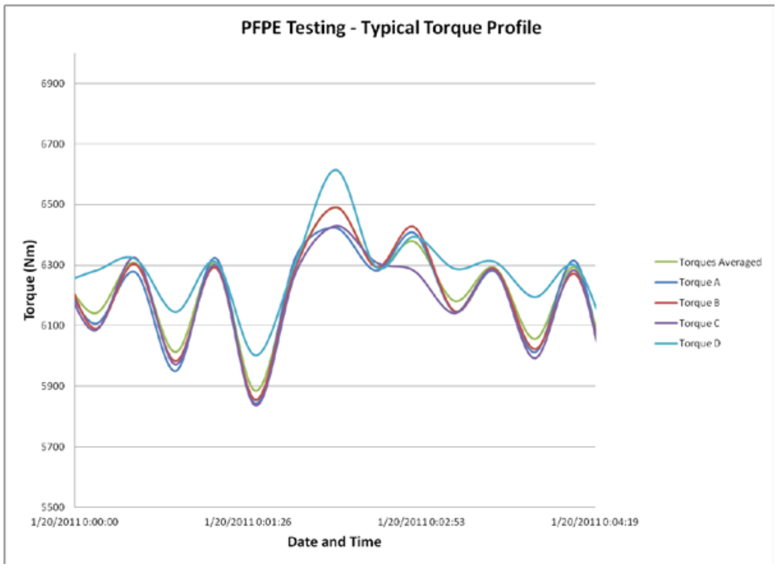


Figure 7.9.5-1: Typical Torque Profile during PFPE Testing

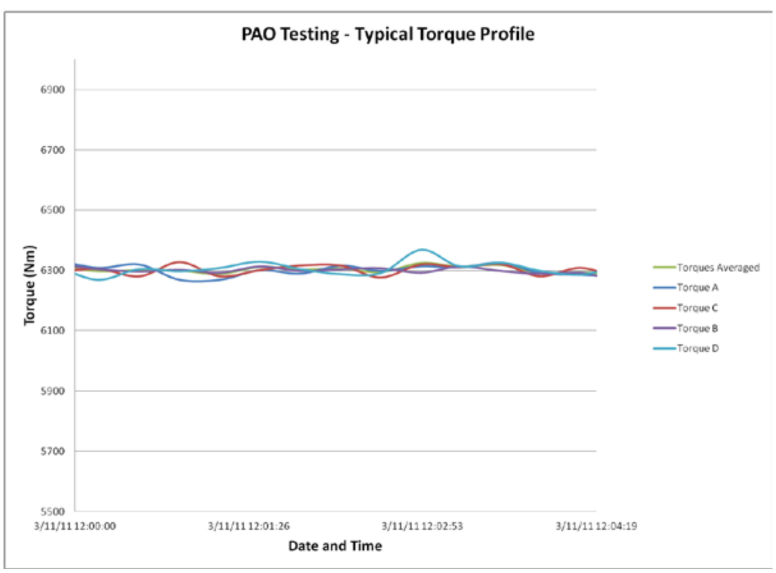


Figure 7.9.5-2: Typical Torque Profile during PAO Reference Testing

The test stand control system regulates the four high speed shaft torques using feedback from independent torque measurements of the four test stand collector gearbox output shafts. For the duration of the PFPE testing there was a continuing problem with the test stand torque regulation system that caused a periodic torque fluctuation with a cycle of approximately 40 seconds. This fluctuation occurred because the test stand hydraulic accumulator pressure was regulated to too low of a level during this time period, which interfered with test stand torque regulation. These torque fluctuations also resulted in faults and shutdowns of the system. Between the two tests the pressure set point value was increased in order to eliminate the faults and shut downs of the test stand. This change also eliminated the torque oscillation for the PAO Reference testing.

The result of this oscillation was a much more severe dynamic loading condition than was present for the PAO Reference test.

7.10 Conclusions

Durability / Wear Protection

In terms of wear protection, both lubricants passed their respective stages based on visual and microscopic evaluation. The duration and severity of the testing seems to indicate that the PFPE lubricant would be a suitable industrial gearbox lubricant on par with the industry standard. Unfortunately the limits of the test equipment were reached before the limits of the fluids. The high temperature running of the PFPE shows it has good resistance to degradation at high heat levels.

Efficiency

An attempt to evaluate the relative efficiency of the PFPE Fluid using the current draw of the test stand ended with mixed conclusions. The results seem to indicate that cases where the film thickness and load conditions are similar, the PFPE has a higher efficiency. However in conditions such as the baseline, where load and temperature are the same, the PAO Reference has a lower film thickness but higher efficiency. Ultimately, the numerous variables such as temperature, rotational speed, and fluid properties combine in a complex fashion to influence the results and make the small differences in efficiency difficult to compare.

Temperature and thermal imagery efficiency results were considered inconclusive due to uncontrolled variables.

Final Assessment

The DOE sponsored, Clipper and Dow Corning full scale gearbox test program successfully proved that PFPE is suitable for use in a wind turbine gearbox. While the test equipment was not able to show a large benefit over the PAO Reference, the PFPE performance is on par with current industry offerings. Further testing, including a full scale uptower test, will be required to further validate the performance and the benefits of this advanced fluid.

Reference Appendix Documents for this Section:

*** APPENDIX 3A:	TEST PLAN DOW CORNING CLIPPER LIFETIME LUBRICANT STUDY FULL SCALE TEST
*** APPENDIX 3B:	TEST LOG
*** APPENDIX 3C:	TEST DATA DIRECTORIES
*** APPENDIX 3D:	INSPECTION PHOTO DIRECTORIES
*** APPENDIX 3E:	GEARBOX BOM
*** APPENDIX 3F	PFPE FULL SCALE GEARBOX BENCH TEST REPORT
*** APPENDIX 3G	EXTERNAL LABS POST TRIAL PINION ANALYSIS
*APPENDIX 3H	CLIPPER LIBERTY TURBINE BROCHURE

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8. TASK 4 RESULTS: ECONOMIC EVALUATION

8.1 Introduction

This document summarizes the potential economic benefits of utilizing Dow Corning's advanced PFPE Lubricating fluid, in a multi megawatt scale wind turbine gearbox. An economic analysis was performed assuming that Clipper Windpower's C-96 Liberty Turbine would be the standard machine used for comparison. In order to asses any potential economic benefits gained from using PFPE, a complete lifecycle cost of energy (COE) analysis was performed for multiple alternative lubricating scenarios.

8.2 Cost of Energy Methodology

Two steps were used to perform the economic analysis of the new lubricant. The first step was to identify potential cost/benefit areas the new lubricant using a brainstorming technique. Then the second step was to perform COE calculations for a baseline and a few scenarios utilizing the new lubricant to quantify the impact of the potential cost/benefit areas that were identified in the brainstorming.

8.2.1 Value Word Equations Exercise

A technique called "value word equations" was used to structure the brainstorming exercise. The various components of COE were discussed and split into main categories where the team felt there may be an impact from the new lubricant. Then the categories were broke down further to get to the underlying constituents of potential value of the new lubricant. Finally, several equations in words were developed to help quantify the value of each constituent. See Appendix A for the result of the value word equation exercise.

8.2.2 COE Calculation Method

In this analysis, COE is used as a metric to compare the economic merits of the PFPE lubricated gearbox versus the standard PAO lubricated machine. This metric, expressed in $/kWh, is not meant to be a precise prediction of the cost to produce electricity but rather an indicator of the expected economic advantages judged through a relative comparison.

The COE is calculated using the equation outlined in Figure 8.2.2-1 below. This equation is consistent with the methods outlined in the NREL Report, Wind Turbine Design Cost and Scaling Model [2].

The NREL System Advisor Model (SAM) was used to estimate the COE for the Liberty C96 wind turbine. Details of this model can be found at https://sam.nrel.gov/.

(1)	COE	=	$\frac{(DR+IWF) \times ICC + LRC + O\&M}{AEP_{net}}$
where:	COE	≡	Levelized Cost of Energy ($/kWh) (constant dollars)
	DR	≡	Discount Rate (1/yr)
	IWF	≡	Insurance, Warranty and Fees (1/yr)
	ICC	≡	Initial Installed Capital Cost ($)
		≡	$TUR_{cc} + BOS_{cc}$
	TUR_{cc}	≡	Turbine Capital Cost ($)
	BOS_{cc}	≡	Balance of System Capital Cost ($)
	LRC	≡	Levelized Replacement/Overhaul Cost ($/yr)
	O&M	≡	Levelized O&M Cost ($/yr)
	AEP_{net}	≡	Net Annual Energy Production (kWh/yr)

Figure 8.2.2-1: Cost of energy Equation & Variables

8.2.3 Analysis Scenarios

The detailed COE analysis was performed for three separate cases; a baseline case and two cases using the new lubricant.

The first case, “The Baseline”, evaluated the COE when the Liberty’s Quantum Drive Gearbox is lubricated using the standard PAO Reference lubricating fluid. This case provides a basis for comparison between the new PFPE fluid and standard lubricating conditions.

The second and third cases involve a COE evaluation for the Liberty Quantum Drive while lubricated with the advanced PFPE fluid. The distinction between these two scenarios is that the second case is a prediction of the “Most Likely” economic advantages of the PFPE fluid while the third case is considered the “Best Case” where all benefits of the PFPE fluid are fully realized. The two alternative PFPE cases were considered to provide a possible range of COE benefit compared to the baseline.

8.2.4 COE Assumptions

The following sections provide the key driving assumptions for each scenario; also see Appendix 4B for additional assumptions listed on the calculation pages.

Description	Units	Baseline 2500 kW Turbine	Proposed 2500 kW Turbine	
Scenario Name		Baseline Case	Likely Case	Best Case
Gearbox Oil Type		PAO	PFPE	PFPE
Oil Cost	$/gal	37	650	650
Oil Quantity	gal	110	80	55
Sump Heater Cost	$k	2	0	0
High Speed Pinion Coating Expense	$k	2	0	0
Gear Super Finish Expense	$k	15	15	0
Cooling Fan Cost	$k	2	1	0
Radiator Cost	$k	3	1.5	0
Misc. Other Cost (Casting, filter, etc)	$k	10	5	0

Table 8.2.4.1: Initial Capital Cost

Description	Units	Baseline 2500 kW Turbine	Proposed 2500 kW Turbine	
Scenario Name		Baseline Case	Likely Case	Best Case
Gearbox Oil Type		PAO	PFPE	PFPE
Scheduled Oil Change Cost	$k/yr	2.3	0	0

Table 8.2.4.2: Levelized O&M Cost

Description	Units	Baseline 2500 kW Turbine	Proposed 2500 kW Turbine	
Scenario Name		Baseline Case	Likely Case	Best Case
Gearbox Oil Type		PAO	PFPE	PFPE
Frequency of Gearbox Replacement	1/yr	0.12	0.08	0.02
Expected Gearbox Life	yr	8	13	50
Total Cost/Gearbox Replacement	$k	731.2	767.6	729.2
Annual Gearbox Replacement Cost	$k/yr	87.7	61.4	14.6

Table 8.2.4.3: Levelized Replacement/Overhaul Cost

Description	Units	Baseline 2500 kW Turbine	Proposed 2500 kW Turbine	
Scenario Name		Baseline Case	Likely Case	Best Case
Gearbox Oil Type		PAO	PFPE	PFPE
Drivetrain Efficiency Increase	%	0	0	2
Gross Annual Energy Production	MWh	10,233	10,233	10,346
Net Annual Energy Production	MWh	8,749	8,749	8,846
Net Capacity Factor	%	39.95	39.95	40.39

Table 8.2.4.4: Net Annual Energy Production

8.3 Cost of Energy Results

The following table summarizes the results from the detailed COE analysis:

Description	Units	Baseline 2500 kW Turbine	Proposed 2500 kW Turbine			
Scenario Name		Baseline Case	Likely Case		Best Case	
Gearbox Oil Type		PAO	PFPE		PFPE	
Turbine Capital Cost	$/kWh	0.0238	0.0241	+0.6%	0.0235	-0.5%
Balance of System Cost	$/kWh	0.0089	0.0089	0%	0.0088	-0.2%
O&M Cost	$/kWh	0.0054	0.0053	-0.3%	0.0052	-0.4%
Levelized Replacement Cost	$/kWh	0.0157	0.0127	-5.6%	0.0072	-15.7%
Total System COE	$/kWh	0.0538	0.0510	-5.3%	0.0448	-16.8%

Table 8.3.1: Results from Detailed COE Analysis

Note: For the proposed technology the % represents impact on Total System COE.

The results indicate that the new lubricant has the potential to improve COE by 16.8% under the "Best Case" scenario using optimistic assumptions providing an upper bound on the potential improvement. In the "Likely Case" scenario which represents a more conservative set of assumptions, the improvement is 5.3%.

It is important to note the largest portion of savings comes in Levelized Replacement Cost, which is dictated by the assumption on gearbox reliability. Thus, verifying and quantifying the potential of new lubricant to effect gearbox reliability is the key assumption that would need to be validated to support the economic analysis and was outside the scope of the project. The lifetime capability and efficiency of the new lubricant was investigated within the scope of the project, however further validation should also be performed on these assumptions.

References:

2. Fingersh, L., Hand, M., and Laxson, A.”Wind Turbine Design Cost and Scaling Model”, National Renewable Energy Technical Report: NREL/TP-500-40566, December 2006. http://www.nrel.gov/docs/fy07osti/40566.pdf

Reference Appendix Documents for this Section:

*** APPENDIX 4A: VALUE WORD EQUATIONS
*** APPENDIX 4B: COST OF ENERGY CALCULATION SHEETS
*** APPENDIX 4C: PFPE ECONOMIC ANALYSIS FULL CLIPPER REPORT
*** APPENDIX 4D: DC COST OF ENERGY DYNAMIC SPREADSHEET ANALYSIS

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9. APPENDIX DOCUMENTS

The Appendix Documents associated with this report could not be included in the public version of this report due to the various reasons cited below. This same information is also reiterated at the end of each section of the report for which Appendix Documents are referenced. Sorry for any inconvenience this may cause.

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