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ARTICLE TYPE

An experimental characterization of the friction coefficient of a wind turbine gearbox lubricant

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Abstract

In this study, a method to predict the contact friction coefficients at the rolling–sliding contacts of isotropic super-finished and axially ground gear and bearing surfaces in wind turbine gearboxes is proposed. A two-disc test rig was used to measure friction coefficient values within a slide-to-roll ratio range of -0.8–0.8, rolling velocities of 2–12 m/s, at oil temperatures of 50, 70 and 100 °C and Hertzian contact pressures of 1.2, 1.6 and 2.0 GPa. A Polyalphaolefin (PAO) International Standards Organization (ISO) viscosity grade (VG) 320 was the lubricant. Data from ninety individual friction coefficient tests were combined to develop two closed-form friction coefficient formulae. A multivariable linear regression analysis was used to derive the regressed formulae. Using the developed regressed formulae, the predicted friction values follow the trend of measured values at all of the operating conditions. The results show that the regressed formulae are close to experimental friction coefficient values. The friction coefficient predictions confirm that the regressed formulae are capable of simulating contact friction values for the lubricated contact conditions considered.

KEYWORDS:

Wind turbine gearboxes, rolling–sliding contacts, regressed formulae, friction coefficient, PAO ISO VG-320 lubricant

1 | INTRODUCTION

The three-point suspension, two-main-bearing suspension, integrated drivetrain, and hub support drivetrain with a flexible coupling are the four categories of wind turbine drivetrain configurations that are typical for geared wind turbine drivetrains¹. Figure 1 compares the basic physical features of a geared drivetrain and a direct-drive drivetrain. The geared drivetrain as shown in Figure 1a, consists of the main shaft, main bearing(s), generator coupling (not shown) and the generator. A significant challenge with the design of a geared drivetrain (see Figure 1a) is the efficiency and the reliability of the gearbox as compared to direct-drive systems (see Figure 1b). The geared drivetrain relies on the lubrication system of the gearbox, to provide sufficient oil to the rolling–sliding contacts to form an elastohydrodynamic (EHD) lubricant film. The purpose is to reduce asperity contact as well as sliding friction. In addition, maintaining a well-conditioned lubrication system in the gearbox provides adequate cooling for all rotating components. Because wind turbines usually operate under severe conditions, the effective lubrication of the rolling element bearings and gears in the wind turbine gearbox is key to realizing the expected design life of the geared drivetrain².

Recent data on the subsystem reliability of wind turbines have proven that a vast majority of the gearboxes in wind turbines fail to achieve their predicted design life³. The performance and accompanying reliability issues related to the wind turbine gearbox have led to increasing interest in direct-drive wind turbines. The direct-drive systems reduce the number of lubricated contacts by doing away with the gearbox (see Figure 1b), they are characterized by larger generator size, a larger number of coils, and are yet to be proven reliable and efficient⁴. Although direct-drive systems

FIGURE 1 Comparison of a wind turbine with and without gearbox

could possibly lead to high savings on drivetrain repairs due to premature gearbox failures, direct benefits from the direct-drive wind turbines over their counterpart geared wind turbines are still not conclusive^{4,5}.

Wind turbine drivetrains are subject to high loading and severe unsteady operating conditions such as braking event, grid engagement, torque reversals, misalignment, wind gusts, and grid faults. Such conditions can lead to the possible breakdown of the lubricant film, which consequently increases total friction loss and contributes to rolling element bearing and gear failure. In addition, the idling conditions of wind turbines usually result in low operational loads that may cause slippage and skidding damage⁶. Furthermore, due to the speed requirements for the low-speed shaft (the rotor shaft between the main bearings and the gearbox) through the high-speed shaft (the gearbox output shaft located between the gearbox and the generator), the resultant lubrication condition experienced in the gearbox can vary from boundary lubrication through mixed lubrication. At the low-speed shaft where the rolling speed is low and the load is high, boundary lubrication occurs because the effective film thickness at the contact is too small to support the applied load. Thus, the total load is mainly supported by the contacting asperities. Meanwhile, at the high-speed shaft where the rolling speed is moderately high, the film thickness at the contact may be about the same order of magnitude as the asperity heights. Under this condition, the fluid film and local asperity interactions coexist, thus load applied to the interacting surface is carried partly by the fluid film and the direct asperity interactions at the contact.

As such, any study involving the EHD lubrication of mating rolling element bearings and gear teeth operating under the severe contact condition, such as that present in wind turbine gearboxes, requires rheological parameters that can correctly describe the pressure dependence of viscosity. This is particularly true because at the Hertzian zone of an EHD contact, the central thickness of the lubricant film determines the shear rate, a parameter which must be known to accurately calculate friction. Therefore, without a correct description of the pressure dependence of viscosity, the predicted friction values would not fit the measured data unless viscosity is adjusted^{7,8,9,10}. The lubricant viscosity and pressure-viscosity coefficients needed for such a quantitative EHD prediction of friction for wind turbine lubricants are now unavailable in the literature^{11,12}. The database for lubricants contained in the American Gear Manufacturers Association (AGMA) Information Sheet AGMA 925-A03 is a remarkable source for determining viscosity and pressure-viscosity coefficients¹³. The AGMA 925-A03 provides pressure-viscosity coefficients derived from optical interferometry for a list of approved ISO viscosity grade industrial lubricants. The data tabulated in the AGMA 925-A03¹³ have been fitted to a viscosity-pressure relationship presented by Gold et al.¹⁴. The viscosity-pressure relationship provided by Gold et al.¹⁴, although convenient for estimating pressure-viscosity coefficients, (accurate enough for the minimum film thickness which is mainly determined by what happens in the inlet region of an EHD contact where pressure is relatively low) is limited to a maximum pressure level of 0.8 GPa. However, in EHD contacts the maximum Hertzian pressures are extremely high (1–4.8 GPa)¹⁵, and the pressure dependence of viscosity cannot be accurately described by the viscosity-pressure relationship provided by Gold et al.¹⁴. Therefore the most realistic approach to predicting friction coefficients in wind turbine gearboxes is to rely on directly measured friction coefficient data under Hertzian contact conditions that are representative of an actual wind turbine gearbox.

At an EHD contact, the generated pressure is typically high and the elastic deformation present at such contacts may be greater than the film (thin) thickness. In addition, because non-conformal steel surfaces are characterized by large elastic modulus, any typical EHD contact pressure can substantially increase the viscosity at the Hertzian contact region. Friction properties at any EHD contact is characterized by what happens at the Hertzian region. Therefore, the lubricant properties must be known at the high pressures and temperatures that prevail at the Hertzian region. Sliding friction (characterized by the thin film and high viscosity at the Hertzian region of the EHD contact) increases the bulk temperature, in

addition, it generates friction forces which contribute to energy losses. In regards to gearbox efficiency, the product of the traction force and the rate of slip at a rolling–sliding contact is a direct measure of the sliding mechanical power loss at the rolling element bearing and race contact¹⁶ and also at the gear mesh contact^{17,18,19,20}.

Although wind turbine gearboxes tend to fail predominately at critical bearing locations because of white structure flaking (WSF)², several recent studies on wind turbine gearbox rolling element bearings have shown that lubrication conditions can also have an impact on rolling contact fatigue (RCF)^{21,22,23}. Similarly, it has been shown elsewhere that, the boundary and mixed lubrication regimes together with frictional energy should be used as key input parameters in tribological experiments that aim to reproduce actual service operating conditions that initiate WSF^{24,25,26}. The high friction coefficients present at such EHD lubrication conditions are known to raise subsurface shear stresses closer to the surfaces of bearings, which possibly lead to WSF in wind turbine bearings²⁷. However, there still is no study available in the literature on how friction coefficients in wind turbine EHD contacts can be predicted. The present study will bridge this gap by proposing a family of closed-form friction coefficient formulae for an ISO VG 320 wind turbine gearbox lubricant and contact conditions that represents wind turbine gearbox contact parameters and operating conditions.

Some previous studies on the prediction of friction coefficient values of lubricants at rolling–sliding contacts have been reported by Benedict and Kelly²⁸ and Drozdov and Gavrikov²⁹. Benedict and Kelly proposed a formula using measured data from the Society for Automotive Engineers (SAE) 10W,30 and 50 oils. The formula considers the variations of sliding speed, rolling velocity and the normal load at a spur gear pair EHD contact to predict friction coefficients. The formula by Benedict and Kelly predicts its largest friction coefficient (shows a singularity) at the pitch point (where slide-to-roll ratio $SR=0$) on the line of action²⁸. Meanwhile, according to EHD theory, only nearly zero friction coefficients (only a small amount of rolling friction) exists at $SR=0$. In addition, the Benedict and Kelly's formula predicts low friction coefficients when SR is increased, a behavior which was not evident in the measurements presented in this paper. Other studies have used EHD lubrication theory to theoretically predict friction^{19,30,31,32,33}. However, this approach requires significant computational effort to predict gear tooth contact friction. In terms of design optimization, the prediction of friction coefficient values in real-time becomes unrealistic. For example, the prediction of mechanical power loss of a gear pair using a numerical EHD model will require several hundreds of EHD analyses of the defined gear pair. To circumvent this issue, Xu et al.¹⁷ and Li and Kahraman³⁴ have used smooth contact and mixed EHD lubrication theory to derive friction coefficient formulae for automotive (SAE 75W90) and aerospace (Mil-L23699) lubricants respectively. Recently, Masjedi and Khonsari³⁵ used a line-contact EHD methodology to derive a curve-fit formula for predicting friction coefficients in a rough EHD line-contact lubricated with a non-detergent SAE 30.

All the above-reviewed studies that have provided friction coefficient formulas for the prediction of friction coefficients at rolling–sliding contacts are limited to operating conditions that exist in automotive and aerospace applications. The contact conditions for the automotive and aerospace applications are non-representative of wind turbines. In addition, the lubricants used are not representative of a wind turbine gearbox lubricant. The viscosity grade for the oils used in previous studies^{28,29,17,34,35} are too low ($< 120 \text{ mm}^2/\text{s}$ and $< 27.6 \text{ mm}^2/\text{s}$ at 40°C respectively for the automotive and aerospace lubricants) to allow film-forming capabilities at the Hertzian region of the rolling–sliding contacts in wind turbine gearboxes. In fact He et al.³⁶ recently presented friction test data for a gear oil Mobil 600 XP 68 whose dynamic viscosity at oil temperature of 40°C $0.0597 \text{ Pa}\cdot\text{s}$ is similar to that of SAE 75W90. The measurement results have demonstrated that the frictional characteristics of these two oils under similar test conditions are in fact different. Therefore using friction formulas derived with automotive oils for EHD contact simulations in wind turbine gearboxes is certainly unrealistic.

Greaves³⁷ and Fernandes et al.³⁸ provides measured friction coefficient data for some wind turbine fluids. However, the friction coefficient values reported are only for a very limited range of operating conditions. Meanwhile, a typical wind turbine gearbox sees a quite diverse range of operating conditions. To date there is no formula existing in the literature that can predict the actual friction coefficient values at the EHD contacts of a wind turbine gearbox. This study will bridge this gap by making available closed-form friction coefficient μ formulae for the instantaneous description of friction coefficients in rolling–sliding contacts of a wind turbine gearbox.

The objective of this study is to present a methodology to develop a family of friction coefficient formulae for a wind turbine gearbox lubricant and contact conditions. The derived friction coefficient formulae employed data from an experimental database of friction coefficient values for rough and smooth contacts and a PAO ISO VG 320 wind turbine gearbox lubricant. The derived formulae will be correlated to the measured test data and then simulated to assess the accuracy of the regressed formulae in predicting friction coefficients.

2 | EXPERIMENTAL MEASUREMENT OF FRICTION COEFFICIENT

2.1 | Test set-up

A roller and disc (twin-disc) test rig as shown in Figure 2a is employed in this study to experimentally evaluate the friction coefficient at the rolling–sliding contacts of two example test specimens. A similar twin-disc test rig of different configuration has been successfully used in rolling contact

FIGURE 2 Two-disc test rig

studies that are typical to wind turbine EHD contacts^{25,39}. Figure 2b is a close-up view of the test compartment showing the intimate view of the non-conforming surfaces of the roller and disc in contact.

A pair of rolling elements bearing supports a rigid shaft coupled to the disc test specimen. The roller shaft was coupled to the pivoted loading arm (see Figure 2b) through a pair of high precision ball bearings (on one side) and a needle bearing (on the other). The roller was axially fixed through an 8 mm nut and washer fixture at about 200 μm distance from the disc. Under operational conditions, the loading arm pressed the roller to create contact at the disc circumferential interface, at the same time initiating an axial misalignment of the roller shaft axis.

To accommodate this axial misalignment, a helical flex coupling as shown in Figure 2b is used to connect the roller drive side and the roller shaft to the loading arm. The loading arm applied a normal force through a pneumatic cylinder with up to a 550 kPa pressure capacity, corresponding to a 4450 N normal load. The applied load was properly monitored by a button head-type load cell located between the pneumatic cylinder piston and the loading arm. Illustrated in Figure 2a are two 10 HP, 3-phase AC drive motors, coupled to a 2:1 ratio timing belt to allow the roller and the disc to be independently driven at rotational speeds ω_1 and ω_2 .

The disc was mounted and axially constrained with a locking nut, on a rigid shaft connected to the disc side AC motor. Given the diameters d_1 and d_2 respectively for the roller and disc, the test machine allows different tangential surface velocities $u_1 = \frac{1}{2}d_1\omega_1$ and $u_2 = \frac{1}{2}d_2\omega_2$ to be continuously varied through ω_1 and ω_2 to allow relative rolling and sliding at the roller and disc contact. From this test set-up, for which the kinematics of the twin-disc is described in Figure 3, the rolling u_r and sliding u_s velocities of the contact are given by

$$u_r = \frac{1}{2}(u_1 + u_2) \quad (1)$$

$$u_s = u_1 - u_2 \quad (2)$$

The corresponding relative sliding at the contact, which is defined by the dimensionless slide-to-roll ratio SR as

$$SR = \frac{u_s}{u_r} = \frac{2(u_1 - u_2)}{u_1 + u_2} \quad (3)$$

was varied in the SR range during the entire test. A lubricant reservoir positioned vertically on the horizontal platform of the test machine conveys preheated lubricant through a flexible hosing to a jet nozzle, to allow the roller and disc contact interface to be vertically lubricated within a flow rate range of 0.50-3.75 lpm. During the entire test run, a constant flow rate of 0.75 lpm was maintained to allow the test lubricant to be supplied at the contact inlet, for different user defined oil inlet temperatures. The test set-up consists of three K-type thermocouples. Two located in the measurement chamber (for bulk temperature measurement of the test specimen), and one in the oil reservoir (for oil supply temperature measurement). See Figure 2a. The entire test was monitored and recorded using a LabVIEW program.

FIGURE 3 Twin-disc parameters.

2.2 | Test specimens and lubricant properties

The specimens selected for the traction test comprised of two discs with different profile curvatures (see Figure 3). The smaller component of the contact pair referred to as the roller, has a pure cylindrical shape with an outer radius $r_1 = 15.875$ mm, a face width of 7.6 mm and no lead crown. The larger of the two discs referred to as the disc, had an outer radius of $r_2 = 28.575$ mm, a face width of 6.3 mm, and a circular lead crown of radius 75 mm. Given the geometrical parameters for the test specimen used in this study, the contact between the crowned disc and the roller is represented by an elliptical point contact. When these two discs are loaded an elliptical contact patch is created with a Hertzian contact ellipticity $b/a = 3.74$ where a and b are the Hertzian half-widths of the contact in the rolling (tangential) and axial directions respectively.

Two sets of rollers with different surface roughnesses were procured in order to investigate the effects of surface roughness. The procured test specimen used in this study had a surface roughness that has been processed to resemble the direction lay of actual gear contacts in relation to the direction of sliding. The rough contact test specimen has axial grinding marks across the face width whereas the isotropic finishing (chemically polished) has a non directional low roughness texture. The material properties and the surface roughness of the test specimens used in the study are presented in Table 1. The selected gear steel is typical to automotive, aerospace and wind turbine applications.

Before the commencement of the actual traction test, it was necessary to inspect all test specimens. This inspection included the circumferential surface roughness measurements using a Taylor Hobson surface profiler (Form Talysurf i60) and an optical surface profiler. Figure 4 (a) and (b) shows an example measured initial surface roughness traces for the roller of the chemically polished and axially ground specimens respectively. A composite surface roughness defined approximately as $R_q = \sqrt{R_{q1}^2 + R_{q2}^2}$. Here R_{q1} and R_{q2} are the RMS roughness amplitudes for roller and disc in the rolling direction respectively. The initial baseline R_q values obtained for the roller and disc surface roughness have been presented in Table 1. This roughness values are typical to axially ground and chemically polished gear tooth surface finishes in the above-mentioned application fields. It is worth noting that each specimen was initially case carburized to achieve a surface hardness of 60-62 HRC representing typical wind turbine and aerospace gear tooth hardness.

A PAO ISO VG 320 gear oil, a commercially available fully-synthetic wind turbine lubricant (Mobil SHC-XMP-320) was used in the tests reported in this study. Table 2 lists its properties at three different temperature levels. The pressure-viscosity coefficient in this study were calculated according Gold et al.¹⁴ where $\alpha_0 = s \cdot v_0^t$. Here $s = 0.010326$ and $t = 0.0507$, these constant lubricant coefficients were obtained from a large range of regression analysis of PAO ISO VG 320 lubricants¹³.

2.3 | Test procedure and test matrix

Table 3 presents the traction test matrix. Each surface roughness type with the same wind turbine lubricant was tested at three contact pressure levels $p_h = 1.2, 1.6,$ and 2.0 GPa. These loads are justified, considering, that a 600 μm displacement of an intermediate shaft from a 1.5 MW gearbox can produce a transient raceway stress exceeding 3.1 GPa^2 . Three oil temperatures of 50, 70, and 100 $^\circ\text{C}$, five rolling velocities of $u_r =$

TABLE 1 Specimen material properties and surface roughness

Surface treatment	Axially ground (Gd)	Chemically polished (CP)
Material	ANSI 9310	
Young's modulus (Pa)	$207 \cdot 10^9$	
Poisson's ratio	0.3	
Density (kg/m)	7850	
Specific heat capacity (J/kg °C)	475	
Thermal conductivity (W/m °C)	46.6	
RMS surface roughness R_{q1} (μm)	0.470	0.032
RMS surface roughness R_{q2} (μm)	0.450	0.031
Composite RMS roughness R_q (μm)	0.650	0.045

FIGURE 4 Measured surface roughness profiles in circumferential direction for (a) chemically polished (b) axially ground specimens.**TABLE 2** Basic parameters of the Mobil SHC-XMP 320 gear oil

Temperature [°C]	α_0 [GPa^{-1}]	η_0 [Pa s]	ν_0 [cSt]	ρ_0 [kg/m^3]
50	13.5	0.18	210.7	842.3
70	13.0	0.08	95.8	832.5
100	12.4	0.03	38.3	818.3

2, 4, 8, 10, and 12 m/s, and an SR range range [-0.8, 0.8] were specified for the entire test. These ranges of applied normal loads, rolling speeds, and temperature range were chosen to emulate typical operating conditions of a complete wind turbine transmission. The loads and speeds were selected based on a previous study conducted by one of the authors in conjunction with the Department of Energy (DOE) USA, under the Wind & Water Power Program⁴⁰. In this previous study, a complete 750 kW wind turbine gearbox was molded to investigate the micro-pitting of wind turbine gearbox components. In terms of the oil temperature specification, the selected oil temperature range used in the test were specified based on experience (where 50 °C represents the temperature at normal ambient temperatures and > 80 °C representing extreme oil temperature). As such the low and high oil temperature range were set at 50–100 °C. With these parameters, the test matrix consisted of 45 individual test for each

TABLE 3 Traction test matrix

Surface treatment	Axially ground [Gd]	Chemically polished [CP]
Lubricant	PAO ISO VG 320	
Temperature [°C]	50,70,100	
u_r [m/s]	2,4,8,10,12	
p_h [GPa]	1.2,1.6,2.0	

roller-disc pair, adding up to 90 traction tests to constitute a database of contact conditions for axially ground and chemically polished surfaces for the given lubricant.

The disc was preheated on an induction heater to allow the bore to be fitted to the disc shaft. This was necessary since the disc was designed to have a minor interference to prevent any relative motion between the disc and shaft. The roller shaft assembly was secured by applying an axial force to the roller shaft to allow a lock washer and nut to be threaded onto the assembly to secure the inner ring of the ball bearings. A retaining component was secured at either end of the arm to secure the bearing outer ring axially. The roller shaft assembly was then moved to the testing platform and a miniature button-like load cell was inserted into the end of the loading fixture and the pneumatic cylinder. A cylindrical pin was inserted to form the pivot point at the base of the loading fixture, located below the mesh point of the disc and roller. The roller shaft was then connected to the drive shaft through a flexible helical coupling. Using a flexible coupling in this set-up, allowed no further axial motion of the roller drive shaft.

Before commencing the test for each roughness type, both specimen were cleaned with ethyl alcohol. Prior to an actual traction test, the test device was warmed up until thermal stability is reached for a desired user-defined oil temperature. In addition, it was critical to perform trial traction tests to optimize the test duration with sufficient data points without drastic surface roughness changes. Consequently, a three hundred and sixty second (six minutes) test duration was adapted as a sufficient test time after steady-state friction was reached at one thousand two hundred seconds (twenty minutes). This initial test run, ensured a mild run-in before the commencement of an actual friction coefficient test for each load case.

Given a user-defined normal load F_n (Hertzian contact pressure, p_h), the resulting torque provided to the disc drive, which in this test is equal to the total traction torque, experienced by the disc drive is measured by a torque transducer. The measured total traction torque T_t consists of the traction torque of the disc (contact friction torque) T_d and the disc drive bearing viscous torque T_b , defined as $T_t = T_d + T_b$. The value T_b , was measured by running a non-loaded ($F_n = 0$) traction test resulting in $T_t \approx T_b$ for each non-loaded test case. After several successful traction tests, the traction torque of the disc was obtained as $T_d = T_t - T_b$ for a given range of oil temperatures. Next, the friction coefficient of the contact is determined as

$$\mu = \frac{T_d}{F_n r_d} \quad (4)$$

where r_d is the radius of the disc and F_n is the applied normal load at the mesh contact. To estimate the lubrication regimes corresponding to the measured friction coefficient data and the test conditions, the lambda Λ ratios (the ratio of the minimum film thickness to the composite roughness amplitude of the contacting surfaces) were approximated based on the minimum film thickness h_{min} predictions in accordance to Dawson and Hamrock⁴¹. The calculated minimum film thickness and Λ ratio for all the test conditions discussed in this study are presented in Table 4. Figure 5 presents a set of measured friction coefficient curves as a function of slide-to-roll ratio SR. In Figure 5 the rolling velocities $u_r = 2, 4$, and 8 m/s were varied for both test specimen for an oil temperature $T_0 = 70$ °C and Hertzian contact pressure $p_h = 2.0$ GPa. It is observed that the measurement data covers the entire lubrication regimes from boundary and mixed lubrication through full-film lubrication condition (see Table 4).

Unless stated otherwise, the acronyms Gd and CP are used here to represent axially ground and chemically polished surfaces respectively. In Figure 5, after pure rolling conditions are achieved, the slide-to-roll ratio was varied from SR [-0.8, 0.8] over a 6 minutes time interval. The same test settings was used to generate all the friction data presented here. The measured data was collected starting with the lowest load case first and highest load last, and the highest speed first and the lowest speed case last. This was necessary in order to minimize any damage that may result from the high load and low speed operating conditions.

TABLE 4 Predicted minimum film thickness $h_{\min}$ and lambda Λ ratio values for tests with the PAO ISO VG 320 gear oil

Surface roughness	Temperature (°C)	u_r (m/s)	p_h (GP)	$h_{\min}$ (μm)	Λ (-)	Lubrication regimes
Gd	70	2	2.0	0.42	0.64	Boundary
Gd	100	2	1.2	0.23	0.36	Boundary
Gd	100	2	1.6	0.22	0.34	Boundary
Gd	100	2	2.0	0.21	0.32	Boundary
Gd	100	4	1.2	0.38	0.58	Boundary
Gd	100	8	1.2	0.60	0.92	Boundary
Gd	50	2	1.2	0.83	1.27	Mixed
Gd	50	4	1.2	1.33	2.04	Mixed
Gd	70	4	1.6	0.70	1.08	Mixed
Gd	70	4	2.0	0.67	1.03	Mixed
Gd	50	2	1.6	0.78	1.20	Mixed
Gd	50	2	2.0	0.74	1.14	Mixed
Gd	70	8	2.0	1.07	1.65	Mixed
Gd	100	10	1.2	0.70	1.08	Mixed
Gd	100	12	1.2	0.79	1.22	Mixed
Gd	100	12	2.0	0.71	1.09	Mixed
CP	50	2	1.2	0.83	18.59	Full-film
CP	100	2	1.2	0.23	5.26	Full-film
CP	100	2	1.6	0.23	4.94	Full-film
CP	100	2	2.0	0.21	4.71	Full-film
CP	100	4	1.2	0.38	8.43	Full-film
CP	50	4	1.2	1.33	29.79	Full-film
Gd	50	8	1.2	2.13	3.27	Full-film
CP	50	8	1.2	2.13	47.73	Full-film
CP	100	8	1.2	0.60	13.50	Full-film
Gd	50	10	1.2	2.47	3.80	Full-film
CP	50	10	1.2	2.47	55.55	Full-film
CP	100	10	1.2	0.70	15.71	Full-film
Gd	50	12	1.2	2.80	4.31	Full-film
CP	50	12	1.2	2.80	62.88	Full-film
CP	100	12	1.2	0.79	17.79	Full-film
CP	100	12	2.0	0.71	15.92	Full-film
CP	50	2	1.6	0.78	17.47	Full-film
CP	50	2	2.0	0.74	16.64	Full-film
CP	70	2	2.0	0.42	9.39	Full-film
CP	70	4	1.6	0.70	15.80	Full-film
CP	70	4	2.0	0.67	15.05	Full-film
CP	70	8	2.0	1.07	24.10	Full-film

3 | DERIVATION OF NEW FRICTION COEFFICIENT FORMULAE

The derivation of a new family of friction coefficient formulae based on the experimental traction data is demonstrated next in this section. Here, the entire test database that includes 90 friction coefficient tests for the MOBIL SHC-XMP 320 lubricant were combined to develop two closed

FIGURE 5 A comparison of measured μ values for $u_r = 2, 4$, and 8 m/s at $p_h = 2.0$ GPa and $T_0 = 70$ °C for (a) axially ground surface (Gd) (b) chemically polished (CP).

form μ formulae. According to the specifications of some wind turbines, the rotor speeds can be as low as 10 rpm, as a result the rolling speed at the EHD contacts of the rolling element and gear can be < 2 m/s. Lower rolling speeds are possible but combining these low speeds and the high Hertzian contact pressure would only lead to complete destruction of the roller and disc contact. The reader is hereby advised to ensure, before use that all operating conditions fall within the boundaries of their target application. The full numerical prediction of the friction based on the lubrication conditions predicted for the sample test results presented in Figure 5 and in Table 4 does not seem possible in the current study, mainly because such a quantitative EHD model would require not only a very accurate knowledge of the local conditions of pressure and shear rate within the EHD contact but also accurate, measurable rheological properties of the liquid. In addition, the pressure dependence on viscosity must be known at the local pressures and temperatures that prevail at the Hertzian zone where friction is predicted. However, the actual viscosity for the local conditions of pressure and temperature in wind turbine EHD contacts is unavailable in the literature^{11,13}.

In accordance with the methodology proposed by Xu et al.¹⁷, a multiple linear regression analysis was performed using a matrix of contact conditions defined in Table 3. Accordingly, the first order of non-dimensional parameters selected to describe the response $\ln \mu$ of the multiple linear regression are given as:

$$\bar{H} = \frac{p_h}{E'}; \bar{\eta} = \frac{\eta_0^2}{E' R_n^2 \rho_0}; \bar{U} = \frac{u_r \eta_0}{E' R_n^2}; \bar{G} = \alpha_0 E'; \quad (5)$$

where p_h is the the Hertzian contact pressure, E' is the equivalent modulus of elasticity, α_0 is the pressure-viscosity coefficient of the lubricant, u_r is the rolling speed, η_0 is the ambient viscosity of the lubricant, ρ_0 is the density of the lubricant. Here the parameter $R_n = 0.01$ m was chosen as the normalizing parameter. The Minitab Statistical Software is used to conduct, a weighted multiple regression analysis. For this analysis the weights are defined as $\sigma_i^2 \approx (\text{fits})_i^2$. Here the fits are the resulting fitted values of an initial ordinary linear regression. Given the weights, w_i , the first order non-dimensional parameters were log transformed to achieve a constant variance for the regression formulae. The obtained regression formulae yielded a 91.25 % and 91.14 % R-square (adjusted) respectively for the chemically polished (CP) and axially ground (Gd) test specimens. The resultant friction coefficient formulae obtained from the regression is given as

For axially ground surfaces:

$$\mu = \exp \left[a_0 + SR \left(a_1 \frac{\bar{\eta}}{\bar{U}} + a_2 \bar{H} \right) \right] |SR|^{a_3} \bar{H}^{a_4} \bar{U}^{a_5} \bar{G}^{a_6} \quad (6)$$

For chemically polished smooth surfaces:

$$\mu = \exp \left[b_0 + b_1 SR \cdot \bar{U} \cdot \log_{10} \bar{\eta} \right] |SR|^{b_2} \bar{H}^{b_3} \bar{\eta}^{b_4} \bar{U}^{b_5} \bar{G}^{b_6} SR \quad (7)$$

where, the slide-to-roll ratio $SR = u_s/u_r$ the correlation coefficients $a_0 - a_6$ and $b_0 - b_6$ listed in Table 5, are representative of the considered PAO ISO VG 320 wind turbine lubricant.

TABLE 5 Numerical values of the coefficients in Eqs. (6) and (7)

i	a_i	b_i
0	-34.75	4.763
1	20.20	93795713
2	-71.44	0.06937
3	0.07338	1.7714
4	0.9803	0.18060
5	-0.28842	-0.24077
6	3.765	-0.03689

3.1 | Correlation between regression formulae and measured traction data

To illustrate how well the newly developed μ formulae (Eqs. 6 and Eqs. 7) can predict the measured friction data, a sample correlation analysis is presented below. A direct correlation is essential to provide confidence of the formulae applicability to predicting μ at wind turbine gearbox EHD contacts. Note that the PAO ISO VG 320 gear oil includes additive packages, whose chemical compositions are proprietary to the lubricant supplier. The study presented here does not intend to discuss detailed chemical compositions with respect to friction coefficients due to unavailability of such informations. Figure 6 presents a comparison of the fitted μ values from the regression analysis and the actual traction data at three rolling speeds and a contact pressure $p_h = 1.2$ GPa. In Figure 6 the variation of friction coefficient is given as a function of slide-to-roll ratio. It is observed that the regressed μ formulae predicts the measurement data very well over the entire transition from the boundary and mixed to full-film lubrication. It is evident from Figure 6 that as the rolling speed increases the film thickness h_m also increases (see Table 4). Consequently, the predicted μ values and the measured friction coefficient data continue to decrease. This is true for both Gd and CP contact surfaces. In fact, for Gd contacts the friction coefficient transitions from mixed to full-film lubrication as indicated by the Λ ratios inscribed on Figure 6 or Tabulated in Table 4, meanwhile friction coefficients at the CP contacts continues to stay in the full-film lubrication condition. This continuous reduction of friction is related to the limiting shear stress, which is known to be a function of temperature at EHD contacts. Here it is noted that as the speed increases the contact temperature increases and consequently the limiting shear stress decreases. This behavior is evident in the rheological model of Bair and Winer⁴² and also collaborated by the numerical simulations of Zhu et al.³².

In Figure 7 a set of friction coefficient curves similar to those presented above is given. Here the applied pressure is held at $p_h = 2$ GPa and the oil temperature maintained at $T_0 = 100$ °C. From Figure 7 it seen that although μ is observed to decrease with increase in speed, the h_m values for the μ curves are much lower than presented above. Here a more intense contact condition is observed for the Gd contacts as they transition from boundary to mixed lubrication regime. It is observed that, when the speed is low (2 m/s) and the contact interface is in the boundary lubrication regime, the μ is noticeably higher than that presented in Figure 6. Here the increase in speed from 2 m/s through 10 m/s is observed to have negligible effect on friction under the boundary lubrication regime as shown in Table 4. However, as the rolling speed reaches 10 m/s, the lubricant film thickness starts to increase gradually (see Table 4) until 12 m/s where the contact friction transitions to mixed lubrication regime. Also, as observed from Figure 7, the CP contacts, while still in full-film lubrication regime have much lower Λ ratios as compared to the results presented in Figure 6.

A reasonably good agreement between the predicted and measured friction curves is observed in the boundary and mixed lubrication to full-film lubrication regimes. Although some discrepancies are observed, the global difference is below 10 %. However, the extent of the difference is dissimilar for each case. The most obvious case is the inflexion of μ curves at higher SR in some of the measured data. While this behavior could be due to thermal effects or the dynamics of the test rig, it is still uncertain which of the two is dominating. Nonetheless, it is apparent from Figure 6 and Figure 7 that friction predictions using Eqs. 6 and 7, indeed capture the absolute trends and values of μ suggesting that they are in good agreement with the actual measurement data.

3.2 | Prediction of friction coefficient in sliding-rolling contacts

In order to investigate the predicting capability of the newly developed closed-form μ (Eqs. 6 and 7), different operating conditions are employed in the simulations to show their effect on μ . One major advantage of the regressed formulae is that, it allows one to conduct parameter sensitivity studies in for example efficiency and frictionally excited contact mechanics problems. Also, these equations would be useful in cumulative frictional energy predictions in wind turbine rolling contact fatigue analyses. Meanwhile, it is widely recognized that using raw measurement data for such

FIGURE 6 Comparison between measured and predicted μ curves at $p_h = 1.2$ GPa for axially ground surface (Gd) and chemically polished surfaces (CP) at $T_0 = 50$ °C.

analyses restricts one to conduct studies requiring the prediction of contact friction instantaneously. This is because the measurement data is usually a series of discrete points.

3.3 | Effect of load

Figure 8 shows the effect of load on the friction coefficient. It illustrates the variations of the friction coefficient μ with slide-to-roll ratio SR for three different Hertzian contact pressure p_h values. The rolling speed and the oil temperature are kept at $u_r = 2$ m/s and $T_0 = 50$ °C respectively. As observed from Figure 8, under the given contact conditions the Gd contacts are seen to operate in the mixed lubrication regime for all the three

FIGURE 7 Comparison between measured and predicted μ curves at $p_h = 2$ GPa for axially ground surface (Gd) and chemically polished surfaces (CP) at $T_0 = 100$ °C.

p_h values. Here it is apparent that as the contact pressure is increased from 1.2 through 2.0 GPa, the hydrodynamic fluid film and local asperity interactions are both present and that as the p_h increases it creates more metal-to-metal interaction, which creates greater asperity friction and consequently the friction coefficient increases at the Gd contact. For the CP contacts, it is observed from Figure 8 that an appreciable amount of oil film still exist at the contacts in manner that is allows the CP contacts to operate in the full-film lubrication regime. Here it is observed for both Gd and CP contacts that a higher p_h results in higher friction coefficients, which is due to the significant influence of p_h on oil viscosity and shear rate. An increase in p_h increase the shear stress. Meanwhile, as is observed from Figure 8, regardless of the surface roughness, at higher SR values the contact temperature tends to reduce the viscosity of the thin film, and therefore friction coefficient starts to decrease. This behavior

FIGURE 8 Effect of load on the friction coefficient μ at $u_r = 2$ m/s and $T_0 = 50$ °C for axially ground surface (Gd) and chemically polished surfaces (CP).

is more pronounced for higher loads because at high contact stresses the heat generation is higher compared to the lower load cases where heat generation is minimal or almost negligible.

3.4 | Effect of speed

Figure 9 shows the effect of rolling speed on the friction coefficient. These results illustrate how the friction coefficients vary with a slide-to-roll ratio in regards to u_r within the range of 2 through 12 m/s. Here the Hertzian contact pressure $p_h = 1.2$ GPa and the oil temperature $T_0 = 50$ °C are kept constant. Evident from Figure 9 is the increase in μ for the lowest rolling speed $u_r = 2$ m/s. Here for both Gd contacts and CP contacts μ continuously increases as SR increases because the contact temperature is negligibly small. However, for higher rolling speeds ($u_r = 4$ through 12 m/s), an increase in SR results in an increase in μ until a plateau is reached and μ begins to decrease gradually due to the rise in contact temperature. As is shown in Figure 9 at the lowest rolling speeds, $u_r = 2$ m/s and $u_r = 4$ m/s the Gd contacts operate in mixed lubrication regime. Here at these u_r values the thin film present at the contact is so small that it's unable to support the load, hence more metal-to-metal interaction, which creates greater asperity friction and consequently the friction coefficient increases. However, as the speed increases an increase in film thickness at the contact results in less local asperity interaction which as indicated (see Λ ratios) for the Gd contacts, μ begins to transition from mixed lubrication to full-film lubrication. A similar behavior is observed for CP contacts. Here CP contacts although are in full-film lubrication, Λ ratios for this contact are shown (see Figure 9) to increase with speed. Another effect of speed on μ is that at low speed and hence less heat generation at the contact interface μ tends to drop less with increasing SR, however, as the speed increases and the contact temperature rises a gradual drop in μ is evident with increasing SR. This continuous decrease in μ with speed can be attributed to the decrease in the lubricant limiting shear stress, which is a function of temperature. On the other hand, it should be noted that as speed increases, the contact temperature increases as well, and hence a decrease in the lubricant viscosity. Because at higher SR, the increase in contact temperature tends to dominate, the viscosity decrease with temperature leads to a decrease in μ at high speeds.

3.5 | Effect of temperature

Figure 10 shows the effect of oil temperature on the friction coefficient. These results illustrate how the friction coefficients vary with a slide-to-roll ratio in regards to T_0 at 50 °C and 100 °C. Here $p_h = 1.6$ GPa and $u_r = 4$ m/s are kept constant. It is observed from Figure 10 that as the oil temperature increases from $T_0 = 50$ °C to 100 °C, the Gd contacts transition from mixed lubrication regime to boundary lubrication. In the mixed lubrication regime oil temperature increases appreciably, resulting in a decrease in oil viscosity. Because oil viscosity decreases, the lubricant film at the contact becomes negligibly small such that the lubricant film can no longer separate the surfaces, thus the load is now supported by the asperities at the contact. This metal-to-metal interaction leads to asperity contact dominating the friction coefficient (see Figure 10). Under the same operating conditions as discussed previously, the CP contacts continue to operate in full-film lubrication regime, as indicated by the Λ ratios displayed on Figure 10. In full-film lubrication, it is observed that increasing the oil temperature will result in a lower limiting shear stress, and the corresponding decrease of μ at the CP contacts.

FIGURE 9 Effect of speed on the friction coefficient μ at $p_h = 1.2$ GPa and $T_0 = 50$ °C for axially ground surface (Gd) and chemically polished surfaces (CP).

FIGURE 10 Effect of speed on the friction coefficient μ at $p_h = 1.2$ GPa and $T_0 = 50$ °C for axially ground surface (Gd) and chemically polished surfaces (CP).

4 | CONCLUSIONS

Experimental investigations of the friction coefficient curves for lubricated elliptical point contacts are conducted in the present study by using an already existing twin-disc test rig for rolling–sliding contacts. In the experiments, the twin-disc test rig was operated in a rolling speed range of 2 to 12 m/s. The applied contact pressures were 1.2 GPa, 1.6 GPa and 2.0 GPa. Two different specimens and surface roughness were studied using a PAO ISO VG 320 wind turbine lubricant within a slide-to-roll ratio range of -0.8 to 0.8. Given the above stated contact conditions, the friction coefficient curves were measured for the entire transition of lubrication regime from the boundary and mixed lubrication to full-film lubrication. Two new closed-form friction coefficient formulae were obtained by performing a multiple linear regression analysis to the entire friction coefficient measurement database.

Based on the obtained experimental and the regression results the following conclusions can be drawn:

1. The regressed formulae capture the same trends with respect to all operating conditions as the measured values. Thus, they can be used to predict contact friction in rolling–sliding contacts and generate complete friction coefficient curves for lubricated contact conditions studied for this particular lubricant.
2. Both the test data and the regressed formulae show friction coefficient decreasing with increasing film thickness as the contact transitions from one lubrication regime to the other.

3. The roughest specimen (axially ground contacts) generates a higher friction coefficient. Friction at the axially ground contacts transitioned from the boundary and mixed lubrication to full-film lubrication regime.
4. Friction in full-film lubrication is dominated by the hydrodynamic action of the lubricant at both the chemically polished and axially ground contacts. Thus allowing the influence of temperature at the contact to be more apparent.
5. Under the test conditions studied, the roughest specimen (axially ground contacts) generates a higher friction coefficient than the smooth specimen (chemically polished contacts). This is due to the load supported by the asperity contact. The rough surfaces appear to cause asperity contacts than the smooth surfaces under the same operating condition.
6. Friction is dominated mostly by the severity of the asperity interactions. However, as the lambda ratio surpasses 3, the specimen with axially ground contacts begun transitioning from mixed lubrication regime to full-film lubrication regime.
7. At increasing speeds and high temperature, the surface asperity continued to dominate friction making the influence of oil viscosity almost negligible for the axially ground contacts.
8. Under low lambda ratio (0.9 and below), in the boundary lubrication regime, it has been found that friction is not constant. Thus, conditions such as load, rolling speed, and sliding speed can affect friction in the boundary lubrication regime.

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CONFLICTS OF INTEREST

The authors declare that there are no conflicts of interest regarding the publication of this paper.

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