A friction-wear correlation for four-ball extreme pressure lubrication

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A Friction-Wear Correlation for Four-Ball Extreme Pressure Lubrication

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ABSTRACT

A first-ever friction-wear model for Four-Ball Extreme Pressure (EP) Lubrication test (ASTM D2783) is presented in this work. The model considers the rate of entropy generation and dissipation within the lubricated tribosystem to establish the friction-wear correlations for 12 lubricating oils comprising minerals, esters and other formulated oils. The correlations can be used to calculate the probability to pass/fail in the EP lubrication. The probability has similar trend as load-wear index from ASTM D2783 method. Besides, the friction-wear correlations allows quick estimation of EP performance of an unknown lubrication, upon comparing with that of an established one. The methods demonstrated here will help researchers or lubricant technologist to characterize the EP behavior quickly without over-relying on tribotester.

Keywords: weld point; extreme pressure; friction-wear correlation; Four-Ball; ASTM D2783

1. Introduction

Tribological modelling has significantly contributed to the development of various applications: lubrication and friction, wear minimization, coating, additivation, functional lubricants, and surface material. It also helps in exploring new tribology area such as nano-tribology, bio-tribology, and nature-inspired adhesive contact. In order to keep up with the above developments, tribological modelling has advanced from a simple contact mechanic approach to a complex multiphysical approach, involving different time scale, length scale and conditional state (e.g. discrete, continuous) [1]. Despite the advancements, the fundamentals of friction and wear are still not fully understood. This is due to a lack of fundamental knowledge to describe and to validate the phenomena, and also due to the inability of experimental setup to observe the sliding interface in situ [1, 2]. The foregoing gaps call for the establishment of new tribological models to: a) shorten the computational time for predictions of high precision, and, b) prevent numerical simulations from becoming black boxes where the nuances of the phenomena are lost.

In lubricated tribology, the spectrum of lubrication models is widespread; but they all essentially depend on the mechanical characteristics of the system, lubrication regime, and the lubricant properties [3]. Traditionally, Reynolds Equation has been employed to analytically solve the classical contact problems such as Hydrodynamic Lubrication and Elasto-Hydrodynamic Lubrication [4-6]. The model aims to simulate friction coefficient, lubricant film thickness, and other rheological properties that account for the interactions between contact interfaces in specific lubricated conditions [4-7]. Hitherto, the model has been improvised to address non-Newtonian effect, surface roughness effect [8], surface topography effect [9], thermal effect [10, 11], and boundary lubrication behaviors [12]. The abovementioned model capabilities require the incorporation of other model frameworks and advanced approaches, e.g. asperity contact model to address the roughness effect in lubricated contact [8], finite element method to solve the lubricated contact problem involving complex domain of surface contacts [9, 13], and molecular dynamics simulation to investigate tribological phenomena in multiscale modelling [14, 15]. Other than Reynolds Equation and its variants, other lubrication models in the literature revolve around regression analysis based on statistical approaches [16-19], empirical model driven by experimental observations [20, 21], and models derived from first principles [22-24].

The modelling of wear is usually performed after friction and lubrication modelling [25-27], although it can be conducted independently when the information of asperity contacts (or models), the mechanical properties of the surfaces, the wear mechanism and the lubricant actions are known [28-30]. As the understanding of wear mechanism even at macroscopic level is limited and incomplete [1], wear modelling is mostly conducted based on empirical approaches. Apart from the regression models and the complex model driven by hypothesized mechanism [26, 27, 30-32], one famous wear model is the Archard's wear model [33], or alternatively known as the Holm's wear model [30]. This model describes sliding wear based on the theory of asperity contacts. It is often characterized by the Archard wear coefficient, which can be obtained from experiments [33]. This model is usually integrated into multiphysics modelling to study the wear behaviors which result from lubricant-triboparts interaction.

In the literature, most friction and wear models were developed in such a way that they do not have the desired level of generality [34]. These models are undoubtedly useful to describe certain manifestations in tribology, but among themselves, the mutual dependence and the correlation between friction and wear remain unclear [34]. Since tribology can be seen as a thermodynamic system that deteriorates irreversibly with dissipation of energy [35-39], efforts have been made to unify the mechanisms of friction and wear based on the theory of Non-Equilibrium Thermodynamics (NET). The NET approaches have been employed in the study of advanced surface materials [36, 40, 41] and sliding systems [35, 37, 42]. One of the simplest approach of NET is based on the assumption of local equilibrium [36], which states that in any small region of the occurring tribological system, its thermodynamic properties are related to the state variables in the same manner as in equilibrium. Another NET approach is known as the second law analysis [37, 43], which states that the lost available energy is directly proportional to the entropy production due to irreversibility in the process. The next type of NET is based on the observation of the degree of irreversibility of any condition in the system [36, 41, 42], in which nonlinear flux-force relations are invoked to characterize a process that operates far from global equilibrium. Although NET offers a systematic approach to characterize a tribological system, more work is needed to incorporate real quantitative variables for practicality. Additionally, the approach should be lightweight, sufficiently robust and useful for the prediction of critical responses with *a priori* known precision.

In this work, for the first time the friction and the wear behaviors of Four-Ball extreme pressure (EP) test (ASTM D2783) are described using a single-equation model based on NET principles. The aims is to establish practical correlation and methods to estimate EP performance of lubricants in practice. This work presents the theoretical framework of the model in Section 2. Following that, a case study involving 12 lubricant samples is elucidated in Section 3, followed by the results validated over a broad range of lubrication conditions in Section 4. Finally, conclusions to this study are given in Section 5.

2. Modelling framework

This model is constructed based on the following premises as shown in Fig. 1. Consider a tribosystem which consists of a pair of solid surfaces or triboparts, which are sliding against each other with lubrication at any operating condition. In this manner, the tribosystem is forced to operate at a non-equilibrium state. According to the second law of thermodynamics, the entropy production in the tribosystem is always positive. In this case, due to the generation of frictional energy during the sliding activity, the value of entropy in the tribosystem increases with time, in order to drive the tribosystem to the equilibrium state [40, 44]. If the tribosystem is not able to dissipate the frictional energy, it will then reach equilibrium state and thereafter its triboparts will be stopped or jammed. Naturally, there exists entropy dissipation mechanisms that take place within the tribosystem to compensate for the effect of entropy generation due to friction, so that the sliding activity prevails. The desired entropy dissipation mechanisms stem from the lubricant action, such as the conduction of frictional heat, the formation of highly ordered lubricant layer and the tribofilm formation due to additivation [45]. On the other hand, the undesired entropy dissipation is the wearing process, which could damage the triboparts and cause seizure [40]. The above dissipation mechanisms drive the tribosystem into a stationary state with minimal entropy production possible [45]. By balancing the rate of entropy production and dissipation within the tribosystem, a simple relationship between friction and wear can be established.

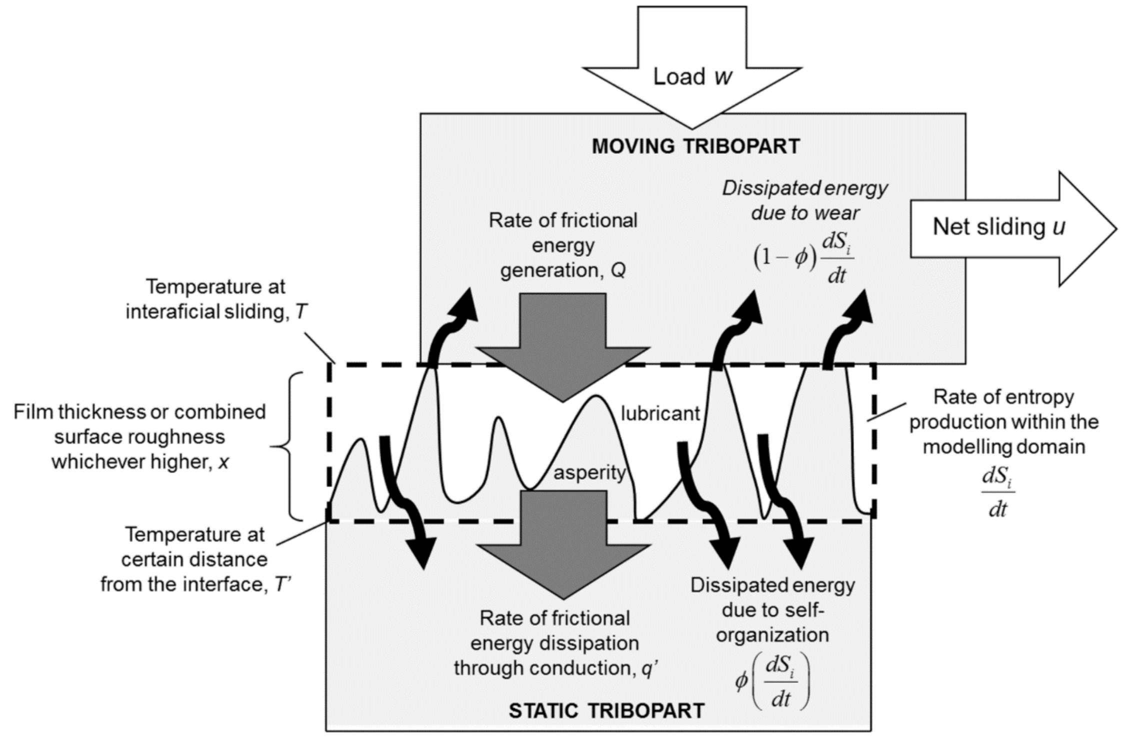

Fig. 1: Schematic diagram of Thermodynamic friction-wear model

2.1. Entropy balance

The total entropy production, dS of a lubricated tribosystem under any operating conditions has the following components [38, 40]:

$$dS = dS_i + dS_e + dS_m - |dS_f| - |dS_w| \tag{1}$$

where dS_i is the entropy production due to frictional sliding; dS_e is the change of entropy due to heat transfer with the surrounding; dS_m is the entropy production due to the generation of wear particles during friction; dS_f is the decrease in entropy due to lubricant action; and dS_w is the decrease in entropy as a result of the wearing process. Few assumptions are made to simplify this expression. First, it is assumed that the tribosystem is at a stationary state under constant operating condition (dS/dt=0) [38]; Second, that the lubricant is kept within an enclosed cavity and there is no bulk movement of lubricant. As such, the heat transfer across the system (triboparts-lubricant system) is assumed to be negligible (dS_e=0) as the magnitude of energy generated and dissipated within the system through friction and wear during short duration (i.e. 10 s) is expected to be significantly larger than that of the heat loss to the surroundings (e.g. static air and neighbouring connection parts); and finally dS_m is insignificant since the mass of wear particles are extremely low as compared to the overall mass of the triboparts [40]. With that, the rate of entropy production of the tribosystem at a stationary state can be approximated as follow.

$$\frac{dS}{dt} \approx \frac{dS_i}{dt} - \left|\frac{dS_f}{dt}\right| - \left|\frac{dS_w}{dt}\right| \approx 0 \tag{2}$$

In practice, determination of dS_f is difficult due to the complex lubricant-triboparts interaction. As such, dS_f is taken as a portion of the total frictional energy dS_i, shown below:

$$\frac{dS_f}{dt} = \frac{dS_i}{dt}(\phi) \tag{3}$$

The friction and the wear components can be correlated by substituting Eq. (3) into Eq. (2), as follows:

$$\frac{dS_w}{dt} = \frac{dS_i}{dt}(1-\phi) \tag{4}$$

where ϕ is the dissipated portion of the total frictional energy due to the lubricant action.

2.2. Entropy generation due to friction

The expression for the rate of entropy generation due to friction, dS_i/dt in Eq. (4) depends on the modelling domain of the tribosystem. The modelling domain considered here is the volume bounded by the interfacial contact area of triboparts and a certain depth of x of the subsurface layer, which can be the lubricant film thickness or the combined surface roughness, as illustrated in Fig. 1. Following that, few assumptions are made similar to those of [40, 46]. First, it is assumed that the ratio of frictional energy flux generated, q to the dissipated flux through conduction, q' is directly proportional to the ratio of the temperature at the sliding interface, T to the temperature at the certain sublayer depth x, T'; and second, the conduction of frictional heat leaving the domain is driven by the average thermal conductivity of the domain, k in the direction of x. With that, the rate of energy entering the domain, Q and that leaving the domain, Q' can be expressed as Eq. (5) and Eq. (6), respectively.

$$Q = \mu wv = UA_c(T - T') \tag{5}$$

$$Q' = \left(1 - \frac{Q}{UA_c T}\right)Q \tag{6}$$

where μ is the friction coefficient; w is the normal load; v is the sliding speed; U is the heat transfer coefficient of the domain, which equals k/x; A_c is the interfacial contact area; and T is the lubrication temperature. Subtracting Eq. (6) with Eq. (5) and dividing by T, the rate of entropy built up within the domain can be expressed as:

$$\frac{dS_i}{dt} = \frac{(\mu wv)^2}{A_c U T^2} \tag{7}$$

Eq. (6) characterizes the deterioration of tribosystem based on the heat transfer coefficient U, similar to those reported in [36, 40, 46], which represents the efficiency of the tribosystem (domain) to conduct the frictional heat away from the system.

2.3. Entropy dissipation due to wear

Since wear is usually accompanied by a change in the interfacial surface area, the use of surface energy to describe wear behaviour of the tribosystem is appropriate and can be found in [35, 37, 43, 45]. Surface energy is the change of free energy when the surface area of a medium is increased by unit area. Using this approach, the rate of entropy dissipation due to wear can be described as follows:

$$\frac{dS_w}{dt} = \frac{2\gamma}{T}(b \times k_w wv) \tag{8}$$

where γ is the surface energy of the worn material; b is the ratio of worn surface area to the worn volume; and k_w is the specific wear rate. The ratio b depends entirely on the geometry of contact (e.g. flat-on-flat, ball-on-flat). The ratio b is needed for practicality to convert the model input from a worn area basis to a worn volume basis. The specific wear rate k_w in Eq. (8) is usually determined experimentally by dividing the total wear volume with the normal force (load) and sliding distance (mm^3/Nm).

2.4. Friction-wear correlation in lubricated tribosystem

A simple and direct equation relating friction and wear can be obtained by correlating Eq. (4, 7, and 8) as follows.

$$k_w = \frac{\mu^2}{cU_w} \tag{9}$$

where c is the physical characterization parameter of the tribosystem ($c = 2A_c\gamma bTw^{-1}v^{-1}$), and U_w is the dissipative coefficient of tribosystem ($U_w = U(1-\phi)^{-1}$). In this expression, the parameter c can be calculated based on the physical property of the tribosystem and the operating conditions, whereas U_w is the proportionality constant between the entropy dissipation capability of the lubrication and that of wear. In reality, both U and ϕ are difficult to measure since they are interfacial properties. For simplicity, these parameters are lumped together as U_w to denote the tendency of the tribosystem to cause wear.

3. Case study

This model was applied to study Four-Ball extreme pressure (EP) test for lubricating fluids (ASTM D2783) [47]. The model was validated based on the EP test results of 12 lubricants comprising of mineral oils, ester oils, and formulated lubricants. The detail of the experiments and the related model parameters are shown in Table 1. For the modelling in Four-Ball configuration, it is sensible to consider only one of the point contacts, i.e. the contact between the top spinning ball and one of the bottom stationary ball. The procedure to calculate the model parameters such as friction, wear and other constants for the Four-Ball EP test are shown in Appendix A. Note that this model is constructed to evaluate the internal entropy generation and dissipation within the triboparts-lubricant system, without considering the heat transfer between the system and the surrounding as described previously in Section 2.1. This is because the rate of entropy dissipation through wear is expected to be significantly larger than that through heat loss especially in such short duration (10 s). For the wear region in the Four-Ball EP test, the wear condition that is close to the last non-seizure load is mild (k_w = 1×10^{-8} to 1×10^{-6} mm^3/Nm); and the wear condition within the incipient seizure and immediate seizure regions can be classified as moderate/severe wear region (k_w = 1×10^{-6} to 1×10^{-2} mm^3/Nm) and they usually have wear scar diameter above 1 mm. Although the wear regions can be categorized based on Archard Dimensionless Wear Coefficient as reported in [48, 49], in this study the wear regions were categorized based on the specific wear rate k_w because the two parameters have the same magnitude. They can be inter-converted based on the hardness of the triboparts material (order of 1×10^9 Pa for steel ball).

Table 1: The EP test results and the model inputs

lubricant	viscosity at 40°C (cP) [a]	machine load, L (kg)	average frictional torque, F_t (kg-mm)	wear scar diameter, d_w (mm) [b]	friction coefficient, μ	specific wear rate, k_w (mm^3/(Nm))
G1 SN150 (mineral)	26.4	50	24.57	0.35	0.1093	3.4E-08
		63	38.94	1.72	0.1375	7.9E-05
		80	112.03	2.65	0.3116	3.5E-04
		100	100.51	2.63	0.2236	2.7E-04
		126	101.43	5.75	0.7381	5.0E-03
G1 SN500 (mineral)	99.7	50	26.71	0.38	0.1188	8.9E-08
		63	29.18	0.63	0.1030	1.2E-06
		80	70.08	1.92	0.1949	9.7E-05
		100	88.33	2.22	0.1965	1.4E-04
		126	453.92	5.70	0.8015	4.8E-03
G1 BS150 (mineral)	435.5	100	68.47	2.07	0.1523	1.0E-04
		126	378.90	5.20	0.6690	3.3E-03
G2 SN150 (mineral)	25.7	40	16.99	0.42	0.0945	2.8E-07
		50	21.54	1.5	0.0958	5.7E-05
		63	83.59	2.04	0.2952	1.6E-04
		80	102.55	2.32	0.2852	2.1E-04
		100	151.43	2.63	0.3369	2.7E-04
		126	449.85	5.55	0.7943	4.3E-03
G2 SN500 (mineral)	71.0	63	21.99	0.43	0.0777	1.4E-07
		80	78.77	2.32	0.2191	2.1E-04
		100	132.52	3.04	0.2948	4.9E-04
		126	422.48	5.70	0.7460	4.8E-03
Naphthenic (mineral)	17.9	50	28.82	0.39	0.1282	1.1E-07
		63	48.30	0.66	0.1706	1.5E-06
		80	109.86	2.46	0.3055	2.6E-04
		100	94.24	2.69	0.2097	3.0E-04
		126	430.17	5.50	0.7596	4.2E-03
PETTO (ester)	57.6	63	21.30	0.44	0.0752	1.7E-07
		80	29.66	0.46	0.0825	1.4E-07
		100	60.86	1.93	0.1354	7.9E-05
		126	331.29	5.10	0.5850	3.1E-03
PETC8/C10 (ester)	28.3	50	18.35	0.52	0.0817	6.3E-07
		63	26.45	0.57	0.0934	7.3E-07
		80	68.27	1.93	0.1899	9.9E-05
		100	53.82	1.93	0.1197	7.9E-05
		126	353.72	4.30	0.6246	1.6E-03
TMP C8/C10 (ester)	18.5	40	22.38	0.35	0.1245	7.7E-08
		50	27.25	0.47	0.1213	3.7E-07
		63	25.62	0.58	0.0905	8.0E-07
		80	75.03	2.42	0.2087	2.4E-04

		100	103.82	2.74	0.2310	3.2E-04
		126	353.72	4.20	0.6246	1.4E-03
TMPTO (ester)	42.7	50	15.31	0.4	0.0681	1.4E-07
		63	30.35	0.56	0.1072	6.7E-07
		80	93.53	2.46	0.2601	2.6E-04
		100	82.21	2.54	0.1829	2.4E-04
		126	336.39	4.80	0.5940	2.4E-03
Formulated A	25.5	126	1.08	1.08	0.0019	5.8E-06
		160	1.39	1.21	0.0019	7.2E-06
		200	1.54	1.27	0.0017	6.9E-06
		250	1.62	1.39	0.0014	8.0E-06
		315	1.81	5.50	0.0013	1.7E-03
Formulated B	124.3	63	0.25	0.49	0.0009	3.3E-07
		80	0.39	0.58	0.0011	5.8E-07
		100	0.41	0.59	0.0009	4.6E-07
		126	0.47	0.67	0.0008	6.5E-07
		160	0.54	0.74	0.0008	7.8E-07
		200	0.63	0.74	0.0007	5.7E-07
		250	1.27	1.64	0.0011	1.6E-05
		315	1.56	1.79	0.0011	1.8E-05
		400	1.74	2.04	0.0010	2.4E-05
		500	2.03	2.20	0.0009	2.6E-05
		620	2.36	4.80	0.0008	4.9E-04

[a] Determined based on ASTM D445
[b] Measured using DUCOM imaging device for non-welded ball. In the case of welded ball, the balls were forced to detach and then the scars were measured using vernier caliper

4. Results and discussion

4.1. The characteristics of friction and wear in Four-Ball EP test

The friction and the wear behaviors of a lubricated tribosystem are interrelated, in which wear increases with friction as observed in Fig. 2a and 2b. By substituting the predetermined parameter c (Fig. 2c) into the friction-wear correlation in Eq. (9), the dissipative coefficient of tribosystem U_w (Fig. 2d) can be obtained. Empirically, the change of U_w was in accordance to the shift of the wear region of the triboparts, from a mild wear region to a moderate/severe wear region. At the mild wear region (e.g. 40 kg), the tribosystem was able to dissipate the frictional entropy efficiently without inducing much wear, but the dissipation efficiency decreases with increasing loads. At one point when the wear falls within moderate/severe wear regions, the dissipation efficiency reached the minimum and remained unchanged at higher loads until weld point (e.g. 50-126 kg). It is interesting to note that the value of U_w at the moderate/severe wear region is narrowly within 1.3x10^{13} to 4.7x10^{13} Wm^{-2}K^{-1}, which is much lower compared to 2.4x10^{15} Wm^{-2}K^{-1} at mild wear region. The values indicate that the lubrication efficiency to sustain the triboparts activity at EP condition (within moderate/severe wear region) was about constant and remained unchanged. At mild wear region, the lubrication efficiency changed with operating condition. This was due to the influences of lubricant film thickness (or viscosity effect) and lubrication regimes (e.g. mixed, boundary) associated to the mild wear region. As a result of the change of friction and wear characteristics, the change of lubrication efficiency in this case is expected.

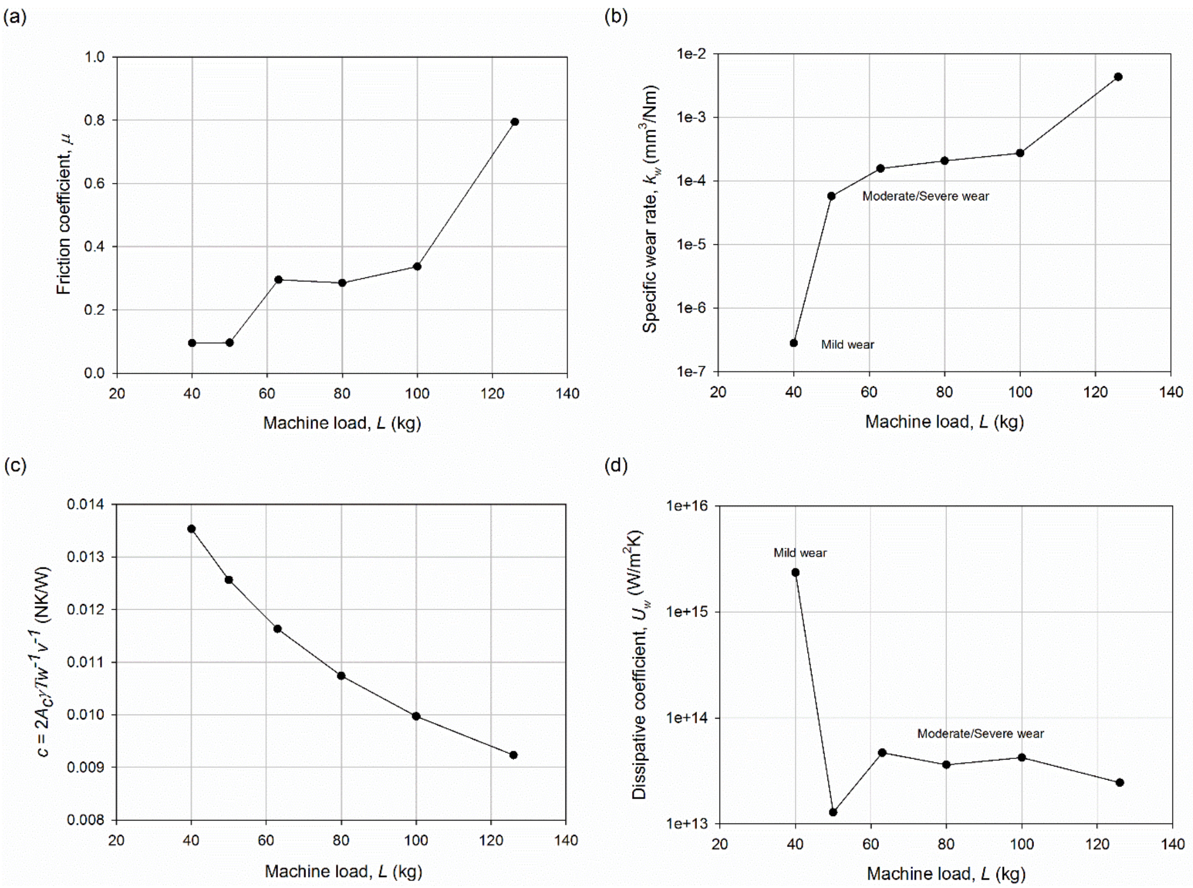

Fig. 2: EP test and modelling results of G2 SN150 mineral oil: (a) wear scar diameter; (b): specific wear rate; (c) characterization parameter of tribosystem; (d) dissipative coefficient

4.2. Predictive probability of pass/fail in EP test

As the dissipative coefficient U_w remained statistically constant throughout the moderate/severe wear region, it can be used to establish a simple friction-wear (μ-k_w) correlation for extreme pressure lubrication. The correlation can be obtained by calibrating Eq. (9) based on the results in Fig. 2. Upon considering the range of the parameter c (0.009-0.013 NKW^{-1}), and the range of U_w ($1.3x10^{13}$ to $4.7x10^{13}$ $Wm^{-2}K^{-1}$) in the moderate/severe wear region, the following μ-k_w relationship for base oil lubrication in EP condition was obtained:

$$117 < \frac{\mu^2}{k_w} < 611 \tag{10}$$

This correlation indicates the friction to wear ratio of base oil lubrication at EP condition. It estimates the range of specific wear rate and hence wear scar diameter of base oil based on a specific friction coefficient, or vice versa. Besides, it serves as a reference to determine whether the lubricant passed or failed the EP test. There is a minimum friction coefficient threshold that the tribosystem must achieve before welding of triboparts happens (or having wear scar diameter greater than 4 mm). Likewise, there is a maximum threshold of friction coefficient when the tribosystem welds. The above thresholds are tabulated in Table 2.

Table 2: The minimum and maximum friction coefficient threshold of base oil lubrication at weld points

Load (kg)	Specific wear rate at weld point (mm^3/Nm) [a]	Minimum friction coefficient threshold to weld, $\mu_{weld.min}$ [b]	Maximum friction coefficient threshold to weld, $\mu_{weld.max}$ [b]
40	3.66E-03	0.6548	1.4964
50	2.93E-03	0.5857	1.3383
63	2.33E-03	0.5217	1.1922
80	1.83E-03	0.4629	1.0579
100	1.46E-03	0.4140	0.9461
126	1.16E-03	0.3688	0.8427
160	9.15E-04	0.3272	0.7477
200	7.32E-04	0.2926	0.6686
250	5.85E-04	0.2616	0.5979
315	4.64E-04	0.2330	0.5324
400	3.65E-04	0.2067	0.4723
500	2.92E-04	0.1847	0.4222
620	2.35E-04	0.1658	0.3789
800	1.82E-04	0.1458	0.3332

[a] Determined based on 4 mm wear scar diameter
[b] Calculated based on Eq. (10)

It is sensible to analyse pass/fail condition of the lubrication based on probabilistic approach, since tribology involves certain uncertainty due to complex physical and chemical mechanisms at different time and length scales. In this work, the procedure to calculate the pass/fail probability of the EP test is illustrated in Fig. 3. When the friction coefficient of a lubrication is below the minimum threshold, the lubrication passes the EP test. If the friction coefficient is within the minimum and maximum thresholds, the lubrication has a probability to fail. In this condition, the associated loads in the EP test is near or at the weld point of the lubrication. Beyond the maximum friction coefficient threshold, the lubrication is expected to have significant wear or causes welding of triboparts, and the associated loads could be beyond the weld point of the lubricant. The probability to fail for the lubrication at specified loading can be calculated as follows:

$$P_{weld} = \frac{\log\left(\mu_{\exp} \middle/ \mu_{weld.\min} \right)}{\log\left(\mu_{weld.\max} \middle/ \mu_{weld.\min} \right)} \times 100\% \tag{11}$$

where μ_{exp}, $\mu_{weld.min}$ and $\mu_{weld.max}$ are the respective tested, minimum threshold and maximum threshold of friction coefficients. The probability P_{weld} spans from 0 to 100% for μ_{exp} within the minimum and the maximum thresholds. When $\mu_{exp} < \mu_{weld.min}$ (50-100 kg loads), the lubrication passed the EP test. At 126 kg load, the results showed that the triboparts is 93% more likely to weld, because its friction coefficient (0.7943) exceeded the minimum threshold and was 7% below the maximum threshold. Therefore, 126 kg can be treated as the estimated weld point for the lubrication.

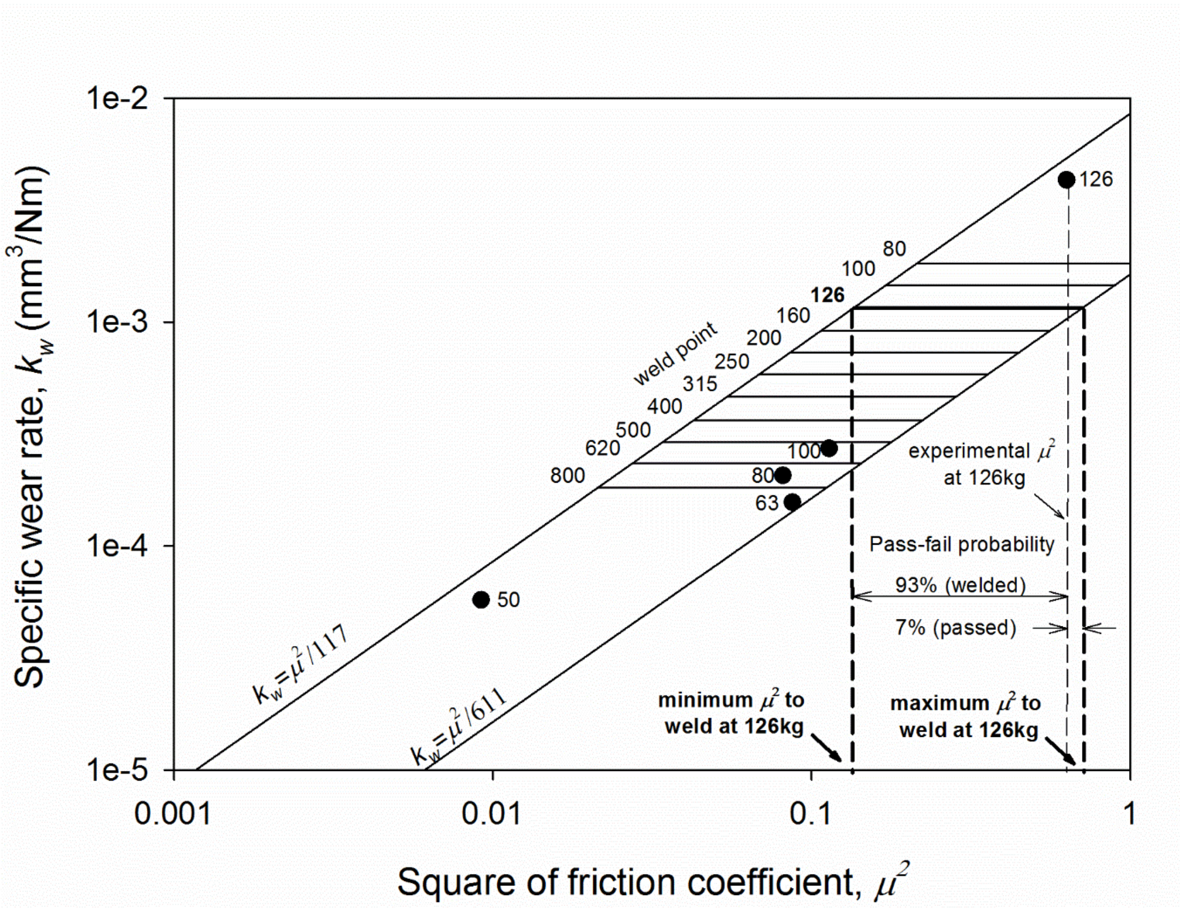

Fig. 3: Prediction of pass/weld of lubrication based on probabilistic approach

4.3. Prediction of EP performances: mineral vs. ester oils

The pass probabilities ($P_{pass} = 1 - P_{weld}$) of the mineral and the ester base oils (Table 1) at 126 kg load were calculated using Eq. (11). The results in Fig. 4 illustrated that the ester base oils (P_{pass} = 36-44%) gave better EP performance than the mineral base oils (P_{pass} = 6-28%), despite having a similar weld point of 126 kg. When comparing P_{pass} with the experimental load-wear index, which is an indicator obtained from ASTM D2783 to denote the capability of the lubrication to withstand extreme pressure condition; both P_{pass} and load-wear index exhibited similar trends, but differed in the degree of change. Only one point (G1 SN500 oil) in the analysis was off the trend. Nevertheless, it is still within the span of the pass probability of the mineral oils. On the other hand, it was found that viscosity has certain impact on EP performance as in the case of G1 BS150 oil. High viscosity G1 BS150 oil (435.5 cP) had at least 40% better EP performance than other G1 base oils (26.4 and 99.7 cP), but still below that of the ester base oils. In this demonstration, this probabilistic approach to predict EP performance should be viewed as a useful estimation tool rather than an absolute deterministic one. The advantage of this approach is that the determination of pass/fail probability needs only 1 experimental friction coefficient data, whereas the load-wear index requires up to 10 experimental data points of wear at various loads.

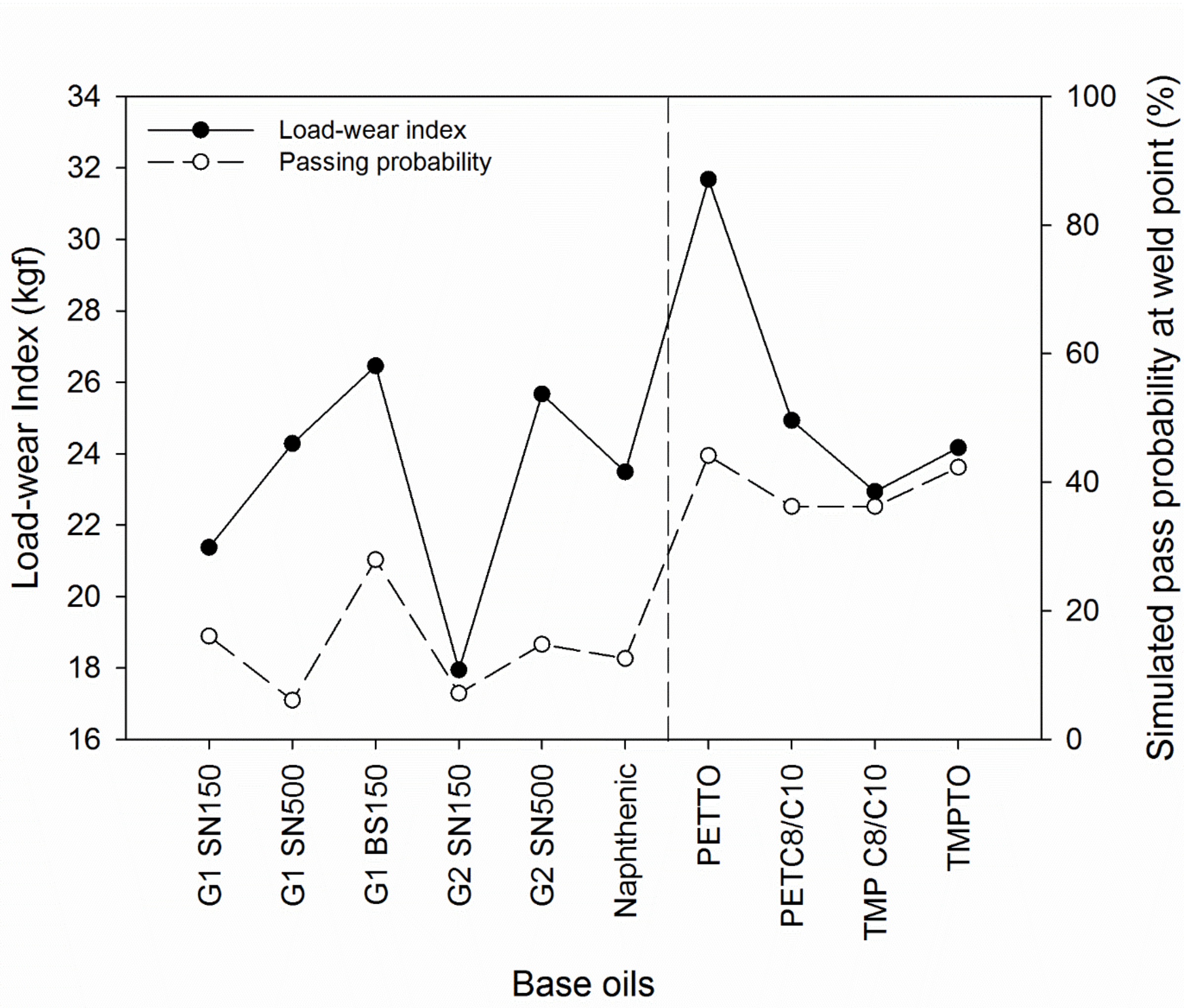

Fig. 4: Comparison of the simulated pass probability at 126 kg weld point (1-P_{weld}) and the experimental load-wear index (ASTM D2783) for mineral and ester base oils

Both the mineral and the ester base oils exhibited similar friction-wear relationship as depicted in Fig. 5. This could be due to the similar entropy dissipation mechanisms and capabilities in both the lubrications. The two lubrications nonetheless exhibited slightly distinct EP performances as discussed previously in Fig. 4. The ester base oils gave slightly lower friction due to the anti-friction capability of the polar ester functional groups [3, 50]. As a result, they had smaller wear scar diameter since k_w is proportional to μ^2. Based on the model results, it seems that friction and wear are not mutually interacting. Rather, friction can be considered as an independent phenomenon due to the complex interaction between lubricant and triboparts;

whereas wear is the dependent consequence, which is the final resolution of the tribosystem to balance out the entropy generation due to friction, whenever the entropy dissipation mechanism by lubricant action is deficient. The lubricant action in this context is not limited to that exhibited by base oils such as heat conduction, self-organization of lubricant molecules and tribofilm formation; but also includes those exhibited by lubricant additivation such as the formation of tribofilm layer due to adsorption or reaction.

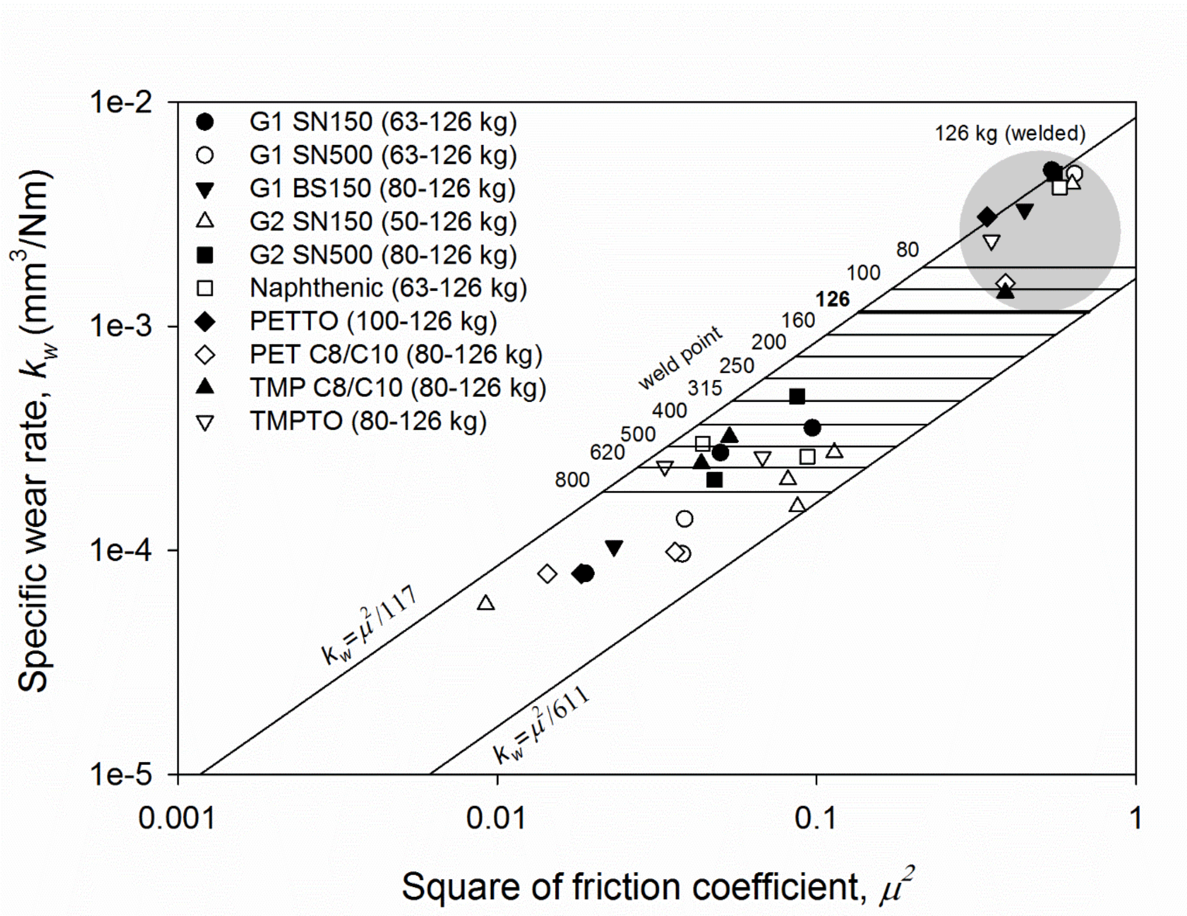

Fig. 5: Friction-wear relationship of mineral and ester base oils in extreme pressure test (ASTM D2783)

4.4. Characterization of tribological behavior of EP additives

Lubricant additive is known to alter the interaction between lubricant and triboparts in certain ways, such as lower friction, wear prevention and so on [51]. From the model perspective, additivation is an additional entropy dissipation mechanism to sustain the activity of the triboparts. The friction-wear relationship of lubrication with additives is expected to be distinct from that of the base oils. The results in Fig. 6 illustrated that the formulated lubricants were able to generate lesser friction and to withstand higher loads as compared to that in the base oils. This observation hypothesized that EP performance and friction are interconnected, such that the lubricant that has higher EP performance (e.g. Formulated B) tends to have smaller friction coefficient in general.

If the tribosystem operates in low friction state, the rate of entropy generation due to friction decreases and the required entropy dissipation rate to sustain the activity also decreases. In this case, U_w of the formulated lubricants (order of 10^8 to 10^{10}) is lesser than that in the case of base oils (order of 10^{13}), indicating that the rate of dissipation of the former is much lesser. It should be noted that the order of magnitude of parameter c in both cases are similar (order of 10^{-3} to 10^{-2}). Because of the lower rate of entropy dissipation, the system can withstand higher loadings before reaching the capability limit of the lubrication to dissipate entropy. The friction-wear correlations in Fig. 6 cannot be regarded as complete or absolute tabulation, without considering the exhaustive list of lubricant base oil, additives and their countless combination. Nevertheless, the correlation is a useful tool to gauge the EP performance for lubricants, considering only few friction-wear data points in EP lubrication (moderate/severe wear region). Generally, a lubrication has better EP performance when its friction-wear data point is nearer to the upper left region of Fig. 6. The position is governed by the dissipative coefficient U_w, which denotes the characteristics of lubricant action in the sliding activity. The above relationships are relevant within the moderate/severe wear region. For mild wear region, model calibration is required to find relevant U_w for further characterizations.

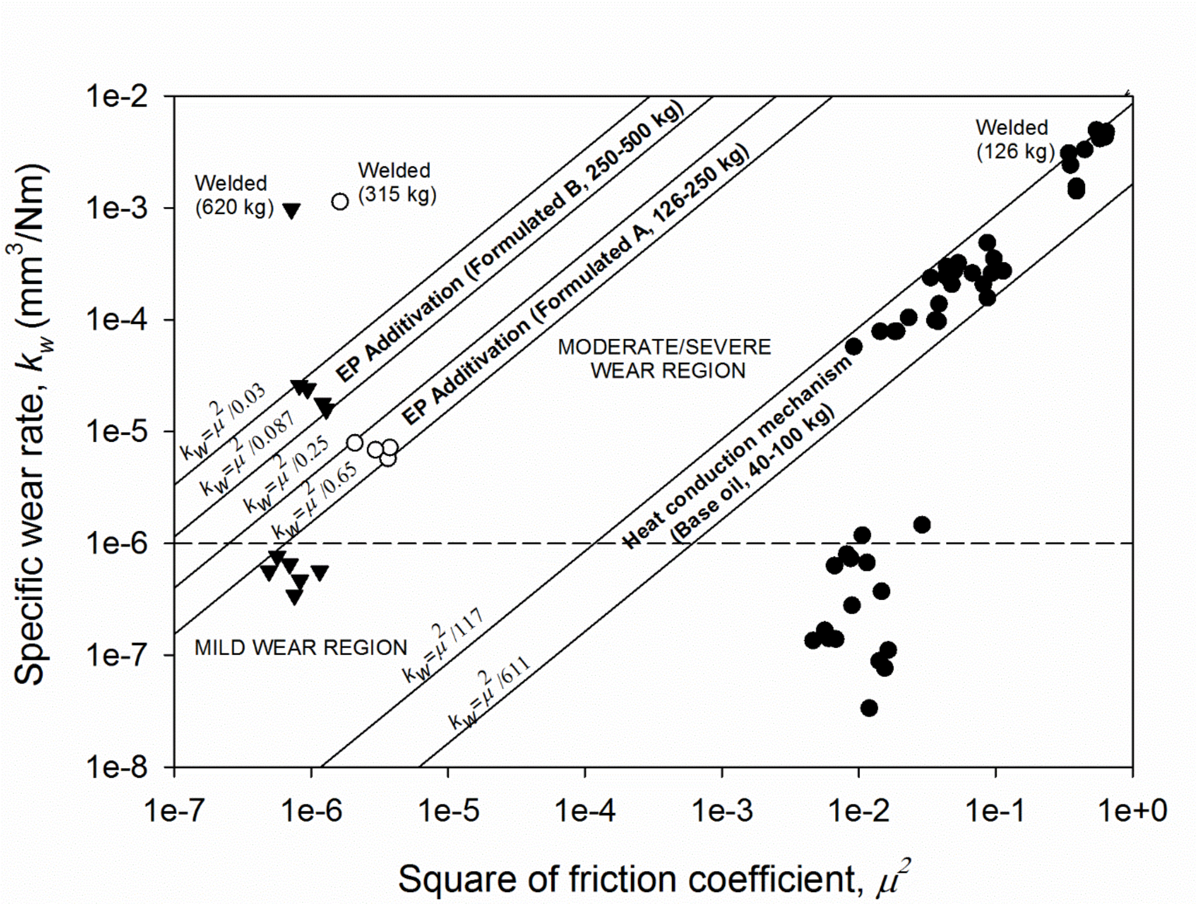

Fig. 6: Friction-wear relationship of base oils and formulated oils in extreme pressure test (ASTM D2783)

5. Conclusion

In this work, for the first time, a single-equation friction-wear model based on entropy balance was developed and validated based on Four-Ball extreme pressure (EP) test (ASTM D2783). The resulting friction-wear correlation is practical and easy to use, since it involves only one parameter namely the dissipative coefficient U_w. This intrinsic parameter describes many aspects of the tribosystem, such as the friction-wear relationship, the lubrication capability to sustain triboparts activity, and the tendency to wear. The applications of U_w demonstrated in this work for quick characterization and estimation of EP performance is feasible without over-reliance on tribotester, and they are closely related to the performance indicator of the standard method. With the fundamentals presented in this work, more study regarding U_w for various formulation or lubricant components can be explored.

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APPENDIX A. Calculation of model parameter in the Four-Ball Extreme Pressure Test

The model parameters in Four-Ball configuration is calculated based on one of the point contacts as in [52, 53]. For a given machine loading, L (kg), few contact parameters can be determined.

The normal load on one bottom ball (kg):

$$L_N = 0.408L \tag{A.1}$$

The contact diameter between the top ball and the bottom ball (mm):

$$d_H = 0.0873L^{1/3} \tag{A.2}$$

The contact area between the top ball and the bottom ball (mm^2):

$$A_c = \pi d_H{}^2 / 4 \tag{A.3}$$

According to ASTM D2783, the rotational speed of the top ball, ω is 1760 rpm. Knowing the distance from the center of the contact surfaces of the underlying balls to the axis of rotation, x is about 3.67 mm according to the machine specification, the sliding speed can be calculated as follow.

The sliding velocity (ms^{-1}):

$$v = 2\pi\omega x / 60000 = 0.676 \text{ ms}^{-1} \tag{A.4}$$

The lubrication temperature is set to room temperature in this modelling study.

In Four-Ball Tribotester, the friction data is reported as friction torque, F_t (kg-mm). To convert it to friction coefficient, the following formula from IP-239 [54] is used.

Friction coefficient:

$$\mu = 0.22248 F_T / L \tag{A.5}$$

The average friction coefficient during the sliding duration of 10 s (as in ASTM D2783) is considered.

In practice, the average wear scar diameter of the bottom balls, d_w (mm) is measured and reported. It can be used to calculate other wear parameter as in [52, 55, 56].

Wear volume of one bottom ball (mm^3):

$$V_w = 1.55x10^{-2} d_w{}^4 - 1.07x10^{-5} L d_w \tag{A.6}$$

Specific wear rate (mm^{3}N^{-1}m^{-1})

$$k_w = V_w / (9.81 L_N v t) \tag{A.7}$$

In this model, finding the ratio of worn surface area to the worn volume is crucial when estimating the change of surface free energy during wear. The surface free energy of AISI 52100 steel ball is about 1.95 J/m^2 (based on Fe as in [57]). The ratio b can be calculated based on the geometry of a spherical cap with sphere radius, r of 6.35 mm (steel ball) and certain height, h. In this manner, the volume of the spherical cap can be seen as the wear volume V_w; the base area of the spherical cap as the worn surface area A_w characterized by the wear scar diameter d_w; and the height h as the depth of the worn volume. The above parameters can be calculated using the following equations.

Volume of the spherical cap (mm^3):

$$V_w = \pi h^2 \left(3r - h\right)/3 \tag{A.8}$$

Base area of the spherical cap (mm^2):

$$A_w = \frac{\pi (d_w)^2}{4} \tag{A.9}$$

The relationship between the depth of worn area, h and wear scar diameter, d_w:

$$h^2 - 2rh + \frac{d_w{}^2}{4} = 0 \tag{A.10}$$

The ratio of worn surface area to the worn volume of a stationary ball (mm^{-1}):

$$b = \frac{\int_{h_0}^{h_1} \frac{A_w}{V_w} dh}{h_1 - h_0} \tag{A.11}$$

As the ratio of worn surface area to the worn volume changes with the wear progression, an average worn area-volume ratio, b to denote the wear characteristics within EP lubrication is needed for model simplicity. The average ratio b value is calculated to be around 18 mm^{-1}, upon integrating A_w/V_w over h within the limits of moderate/severe wear region using Eq. (A.11). In this study, the limits of integration (i.e. h_0 = 0.02 mm and h_1 = 0.32 mm) are calculated based on the typical range of d_w (i.e. 1 mm to 4 mm) using Eq. (A.10). Note that the depth of worn area (h) in practice rarely exceeds 1 mm unless the balls weld.

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Figure Captions

Fig. 1: Schematic diagram of Thermodynamic friction-wear model

Fig. 2: EP test and modelling results of G2 SN150 mineral oil: (a) wear scar diameter; (b): specific wear rate; (c) characterization parameter of tribosystem; (d) dissipative coefficient

Fig. 3: Prediction of pass/weld of lubrication based on probabilistic approach

Fig. 4: Comparison of the simulated pass probability at 126 kg weld point (1-P_{weld}) and the experimental load-wear index (ASTM D2783) for mineral and ester base oils

Fig. 5: Friction-wear relationship of mineral and ester base oils in extreme pressure test (ASTM D2783)

Fig. 6: Friction-wear relationship of base oils and formulated oils in extreme pressure test (ASTM D2783)