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An early method for the technical diagnosis of pin-on-disk tribometers by reference friction measurements in EHL conditions

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Abstract

Reference tests are widely used to calibrate scientific instruments but are potential candidate to establish technical diagnosis procedures for scientific instruments too. A reference test is currently lacking in tribology. Though, it would allow users to check that tribometers are properly working and may also form a yardstick for cross-laboratory comparative studies. In this paper two easy-to-use reference testing procedures for the diagnosis of commercial pin-on-disc tribometers are established resorting to a special no-wear test setup in EHL conditions. Several tests were carried out with two tribometers and two commercial oils, and both standardized testing modes were investigated: unidirectional-rotating and linear-reciprocating mode. The friction curves from more than 350 tests were analyzed to generate meaningful statistics supporting the robustness of these procedures. The test setup proved to be suitable for the task since the summary of the results showed an excellent repeatability of friction curves concerning appearance and average values

Highlights

- A novel reference test procedure is proposed for the pin-on-disc tribometers technical diagnosis
- The EHL condition is exploited to obtain stable and highly repeatable friction curves
- A user-friendly lubricated test setup is put in place in order to fit industrial applications
- Measurements are performed with 2 tribometers in 2 laboratories, then analyzed and compared

Keywords

Friction; Pin-on-disk; Diagnostics; Tribometer; Reference test; EHL

Abbreviations and Nomenclature

AAV	Average of the average coefficient of friction values
ASt.DV	Mean of the standard deviation values of coefficient of friction
CoF	Coefficient of friction
E'	Composite elastic modulus
E _b	Ball elastic modulus
E _d	Disc elastic modulus
EHL	Elastohydrodynamic lubrication
F _N	Normal load
F _T	Tangential friction force
G	Dimensionless material parameter for EHL equations
G _V	Dimensionless Hamrock's viscosity parameter
G _E	Dimensionless Hamrock's elastic parameter
HVo	High viscosity oil
k	Ellipticity factor
Λ	Lambda (or roughness) factor
L	Material parameter (Moes); $L = G \cdot (2U)^{0.25}$
M	Load parameter (Moes); $M = W / (2U)^{0.75}$
MVo	Middle viscosity oil
R'	Composite curvature radius of mating surfaces
R _b	Ball radius
R _d	Disc radius

52	SRR	Slide-to-roll ratio
53	St.DAV	Standard deviation of the average coefficient of friction values
54	U	Dimensionless speed parameter for EHL equations
55	\bar{U}	Entrainment speed
56	v_b	Ball linear speed
57	v_d	Disc linear speed
58	VI	Viscosity index
59	α	Pressure-viscosity coefficient
60	α_T	Temperature-viscosity coefficient
61	η_0	Dynamic pressure in atmospheric conditions;
62	ν_b	Ball Poisson ratio
63	ν_d	Disc Poisson ratio
64	W	Dimensionless load parameter for EHL equations
65		
66		

67 1. Introduction

68 It is well known that the coefficient of friction (CoF) of a tribological pair is far from being only a
69 characteristic property of the materials involved into the contact[1] since it also depends on many other
70 parameters: speed, load, temperature, humidity, wear, size/scale...to name but a few [2]. In such a scenario, the
71 test rig itself is also expected to affect the estimation of the coefficient of friction, as a result of its mechanical
72 layout, its own dynamic characteristic and the specific test set-up and contact geometry. Therefore, no
73 experimental value of the coefficient of friction can ever be stated as “representative” or “correct” in absolute
74 terms. Precisely for these reasons the DIN 50322 standard accepts, for example, that friction results from “model
75 tests” (i.e. typical simplified laboratory tests, including pin-on-disc) may be very different compared to the
76 results from “field tests” (i.e. tests in actual operating conditions) even if similar materials or components are
77 involved. The ASTM G99 standard, the referral standard for pin-on-disc method, also warns that there is no
78 ensurance the tests will predict the behavior of a given material in actual application under conditions differing
79 from those in the test.

80 In the field of tribology results are usually scattered and it is not obvious to find consistent results under the
81 same testing conditions, especially in dry testing conditions where the materials wearing-out process introduces
82 uncertainties. The results of the Interlaboratory tests included in the ASTM G99 standard are themselves rather
83 scattered, unsuited to represent a reference because of wear. Moreover, very often average values and
84 coefficients of variations for CoF are shown without even disclosing actual friction curves, which can vary a
85 lot when repeating merely the same test although average values are similar. Nonetheless, should one be able
86 to measure consistent friction curves (let us call it an “usual” result) under the same conditions, over time and
87 across different tribo-testing machines of the same kind, such conditions may form, at least, a relative reference.
88 By extension, a reference test may be established and a criterion to assess if one particular tribometer is
89 potentially affected by technical problems can be derived by comparison of the specific outcome with the one
90 usually expected. This approach conforms indeed to ISO 13372:2012 which defines the attitude of technical
91 diagnosis to collect data and information (i.e. condition monitoring [3]) to detect problems and deviations from
92 normal conditions.

93 The aim of this experimental investigation is therefore to develop a reference procedure for pin-on-disk
94 tribometers based on reference tests featuring a stable and repeatable characteristic coefficient of friction. This

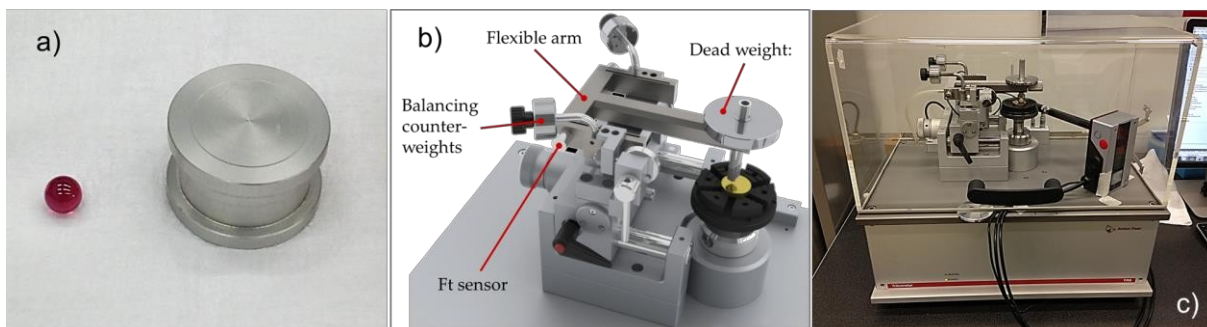
95 procedure would include one reference test intended for unidirectional rotating mode and one for linear
96 reciprocating mode. In order to achieve this challenging result, this paper explores a special no-wear test setup
97 in elastohydrodynamic lubricated conditions and its peculiar friction performances. The authors wish to
98 demonstrate the reliability of the chosen method which is then proposed as a monitoring and diagnostic tool for
99 tribometers, able to make an assessment on either software / hardware issues or issues in terms of calibration.
100 The method qualifies as Structural Health Monitoring (SHM) [3] applied to a very special mechanical system,
101 i.e. a scientific instrument, where friction force itself acts as the monitoring parameter for which a reference
102 value is available. In pin-on-disk tribometers the CoF is calculated following the classical definition: $\mu = F_T / F_N$
103 where the quantity that is measured is indeed the tangential friction force F_T acting at the contact area, and F_N
104 is a known normal force.

105 In the last 60 years a lot of scientific works have investigated the elastohydrodynamic lubrication (herein referred
106 to as EHL or EHD) whose theoretical foundations were laid by Grubin [4], Dowson and Higginson [5] and
107 Hamrock and Dowson [6]. EHL is a contact mode typical of lubricated non-conformal contacts. It is the typical
108 contact condition found in machine elements interacting under low geometrical conformity, where loads act
109 over relatively small contact areas, such as the point contacts of ball bearings and the line contacts of roller
110 bearings and of gear teeth. EHD phenomena also occur in some low elastic modulus contacts, such as lip seals
111 [7]. Among them, a multitude of papers have investigated the coefficient of friction in EHL point contacts.
112 However, all these works focused either on several aspects of lubrication modes or the
113 development/optimization of numerical models for specific experimental cases or specific applications. In
114 almost everyone a ball-on-disc test rig designed on purpose was used. Test rigs for interferometric film thickness
115 measurements are typically used when the aim is to correlate friction with lubrication regimes and plot the
116 Stribeck curve. For example, friction values measured in this way were reported in the works by Zhang et al.
117 [8],[9], Fu et al. [10], [11] and Ciulli et al. [12], Carli et al. [13], Gonsel et al. [14], Nishikawa et al. [15] who
118 investigated all lubricated conditions with testing parameters rather similar to those in this paper. Hansen et al.
119 [16][17], Bjorling et al. [18]. Vengudusamy et al. [19] and Guegan et al. [20] reported the effects of roughness
120 on traction coefficient (CoF) by mapping transitions of lubrication regimes. Hansen et al. supported also
121 interferometric observations with electric contact resistance (ECR) technique. Also, Nishikawa et al. [21] and
122 Han et al. [22] investigated friction phenomena in reciprocating sliding motion with a test rig of the same kind.
123 Ball-on-disc traction test rigs are often resorted to when the evolution of traction coefficient is studied in
124 response to the variation of the slide-to-roll ratio. With regard to this, it is worth to cite the work by Vegudusamy
125 et al. [23] and Angel et al. [24]. Schwing-Reib-Verschleiss (SRV) tribometers are sometimes used too, for
126 example in the same papers by Vegudusamy et al. [23] and Han et al. [22]. Only some authors in the scientific
127 literature dealt with the friction behavior in lubricated conditions with pin-on-disk tribometers, like Anderson
128 [25] and Podgornik [26] and Grützmacher [27]. Anderson performed lubricated tests with a classical pin-on-
129 disk tribometer, he used a quite particular experimental set-up where water was used as lubricant and large
130 amount of wear was unavoidable. Grützmacher tested a steel-steel friction pair in ball-on-disc lubricated
131 conditions with testing parameters (rotational speed, track radius and load) very similar to those used for this
132 study. However, he used much lower viscosity oils and focused on the transition from fully-flooded to mixed
133 lubrication because of centrifugal forces at varying track radii. Muller and Ostermayer [28] used a High Load
134 Tribometer (HLT), which basically consists of a pin on disk set-up, to study the problem of starvation in
135 hydrodynamic lubrication of conformal contacts. Bai et al. [29] measured traction by means of a multi-purpose

136 tribometer in linear reciprocating pin-on-disk configuration. Kovalchenko et al. [30] measured a number of
137 Stribeck curves with a pin-on-disc investigating the hydrodynamic lubricated contact with a flat pin though.
138 To the best of the authors' knowledge, no other author has ever attempted to introduce a reference test or
139 reference procedure in the tribology field to verify the testing apparatus itself, neither exploiting the EHD
140 lubrication or another contact condition as a mean (rather than an end).
141 For the purpose of robust reliable statistics, the results from two pin-on-disk tribometers located in two different
142 laboratories and in different environmental conditions were compared in this paper.
143 An empirical approach was basically followed in this study. No in-depth examination of the contact mechanics,
144 and no experimental lubricant film thickness measurements are provided here as it goes beyond the scope of
145 this paper. Yet, application of the available EHL equations is briefly presented in the next sections. This is just
146 to support experimental evidence by checking that the predicted lubricant film thickness with the chosen testing
147 parameters is in line with a no-wear EHL regime, leastwise.
148 The results presented hereafter are to be intended as preliminary results which need further verifications and a
149 wider statistical base for acceptance as standard procedure.

150 2. Materials and Methods

151 The present experimental campaign was made possible thanks to the collaboration of the Department of
152 Mechanical and Aerospace Engineering Laboratory (DIMEAS) at Politecnico di Torino (Torino, IT) and Anton
153 Paar TriTec (Corcelles-Cormondèche, CH). A total of two different Anton Paar pin-on-disk tribometers were
154 used: a TRB tribometer and a TRB³ tribometer of next generation. Both the instruments are compliant with the
155 ASTM G99 and ASTM G133 standards. Manufacturer's technical specifications of the two instruments are
156 presented in Appendix A and their functional scheme is shown in Figure 1b.



157
158 **Figure 1.** (a) Material pair for tests; (b) functional scheme of the Anton Paar pin-on-disk tribometers; (c) temperature
159 and humidity sensor fitted inside the testing chamber
160

161
162 A lubricated contact is formed between a ruby ball (Saphirwerk AG, Brügg, CH) and a microscope round cover
163 slip made of D263M[®] borosilicate glass (Schott AG, Mainz, DE). Such a thin glass slip cannot be installed
164 directly into the spindle clamping device as it is too fragile. To overcome this critical issue, the glass slip was
165 glued on the top of an aluminum sample-holder (Figure 1a). Several samples were used, and the gluing of the

166 cover slip was handcrafted using fast-setting glue. The ball was mounted into a pin-shaped ball-holder thus pure
 167 sliding occurred at the interface (with a SRR ^[1] equal to 2).
 168

169

Table I. Properties of the liquid lubricants and the tribological pair

OILS	Base stocks	Density [kg/dm ³]	Viscosity [cSt] @ 30°C	α ^[1] [GPa ⁻¹]	V.I. ^[2]
Anton Paar Testing Oil MV (MVo)	Semi-synthetic oil (Mineral oil and PAO base stocks)	0.861	122.2	30.5	101
Anton Paar Testing Oil HV (HVo)	Fully synthetic oil (PAO, Poly-1-decene, Polybutene base stocks)	0.839	740.3	34.8	123
MATERIAL PAIR	Elastic modulus [GPa]	Poisson ratio	Radius [mm]	Rq [μ m]	Thickness [mm]
Ruby sphere	390	0.22	3	0.006	-
Glass cover slip	73	0.208	17	0.020	0.1

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¹ Pressure-viscosity characteristics are calculated with So and Klaus' analytical model [31].

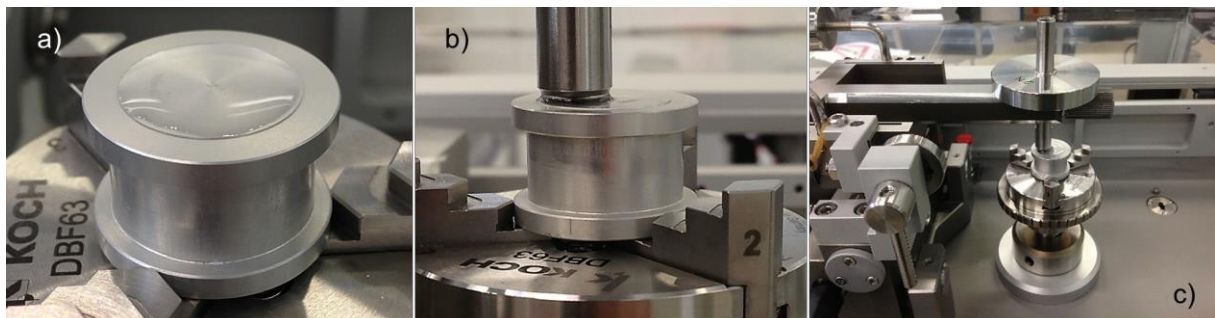
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² Reference is made to the ASTM D2270 standard

172 Two types of commercial liquid lubricants were employed: a middle-viscosity and a high viscosity oil (herein
 173 referred to as MVo and HVo). Both are products of Anton Paar TriTec SA and their main properties at the testing
 174 temperature are listed in Table I, together with the material properties of the sample pair. Both the oils are
 175 thermally stable, hydrophobic, resistant to atmospheric agents and characterized by a relatively high viscosity
 176 index (V.I.). Being commercial products, these lubricants come with their technical sheet certified by a
 177 meteorological laboratory. Tables of viscosity and other physical properties for the two oils are presented in
 178 Appendix B as provided in technical sheets. Viscosity values in Table I are interpolated via the popular Walters'
 179 formula considering the working temperature of 29°C.

180 The most widely used experimental techniques for testing lubricated contacts resort to either oil bath or
 181 continuous active or passive oil supply flow. In this study a little amount of oil was added on the top of the glass
 182 sample with a syringe before starting each test, as much to completely cover the surface region where the
 183 interaction between the solid surfaces takes place (Figure 2a).

184



¹ Slide-to-roll ratio defined as: $SRR = 2 \cdot (u_d - u_b) / (u_d + u_b)$, where 'd' stands for "disk" and 'b' for "ball"

185 **Figure 2.** (a) lubricant applied on the sample surface before a test; (b) zoomed view of the lubricated contact; (c) test
186 set-up for rotating mode tests.

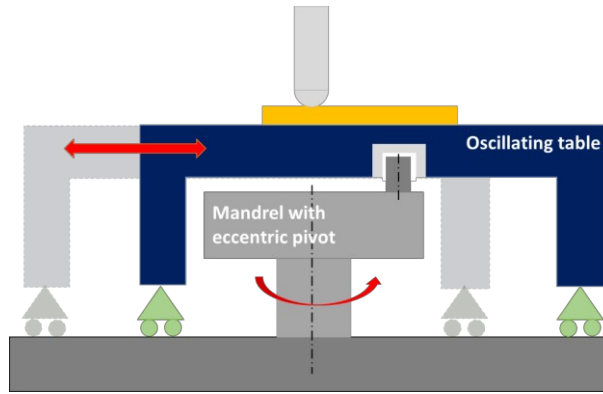
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188 Bai et al. [29] reported successful full-film lubrication of sample surfaces by a very similar method without the
189 system running into starved lubrication. More extreme techniques to supply lubricant to the contact region are
190 also reported in the scientific literature. As an example, Li et al. [32] proved that lubrication by oil droplets
191 supply is an effective way to form a continuous lubrication film at the contact region and avoid lubricant waste.
192 Figure 2c shows the complete experimental set-up just before running a unidirectional rotating test and Figure
193 2b is a zoomed view of the lubricant meniscus during a test run.

194 The contact load was applied placing dead-weights on the measuring arm coaxially to the ball-holder. The
195 tribometer vertical load range is 0.25N to 60N (see Appendix A). The MVo was selected for tests with relatively
196 low vertical load, so as to explore the lower part of the instruments load range; the HVo was used for tests with
197 higher vertical load so as to explore the higher part of the load range of the instruments. Preliminary tests
198 allowed to identify the most favorable testing parameters. As to rotating tests, 1N and 2N loads have been tried
199 out at 100rpm with the MVo. Tests under 1N and 2N yielded very similar results in terms of average CoF, but
200 more regular curves showed up in the case of 2N load. Possible reasons for a higher regularity with higher
201 normal load are system vibrations which are more negligible with higher loads and the lower influence of
202 roughness as a result of a slightly greater contact area. As to the HVo, the desired load parameter was 60N at
203 first, i.e. the upper bound of the instruments load range. However, after preliminary tests under load of 30N to
204 60N, coupled with speeds from 100rpm to 200rpm, none of the loads higher than 30N guaranteed complete
205 separation between solid surfaces and the lubricating film ran systematically into failure with the glass slip
206 breakup. On the contrary, test runs under 30N load and 150rpm yielded enough repeatability to give rise to a
207 reference condition.

208 Some of the empirical formulae currently available in the scientific literature were applied to predict the
209 lubricant film thickness and a full-film lubrication state or neary-full-film lubrication is predicted with all the
210 testing conditions and lubricants. Par. 5 gives more details.

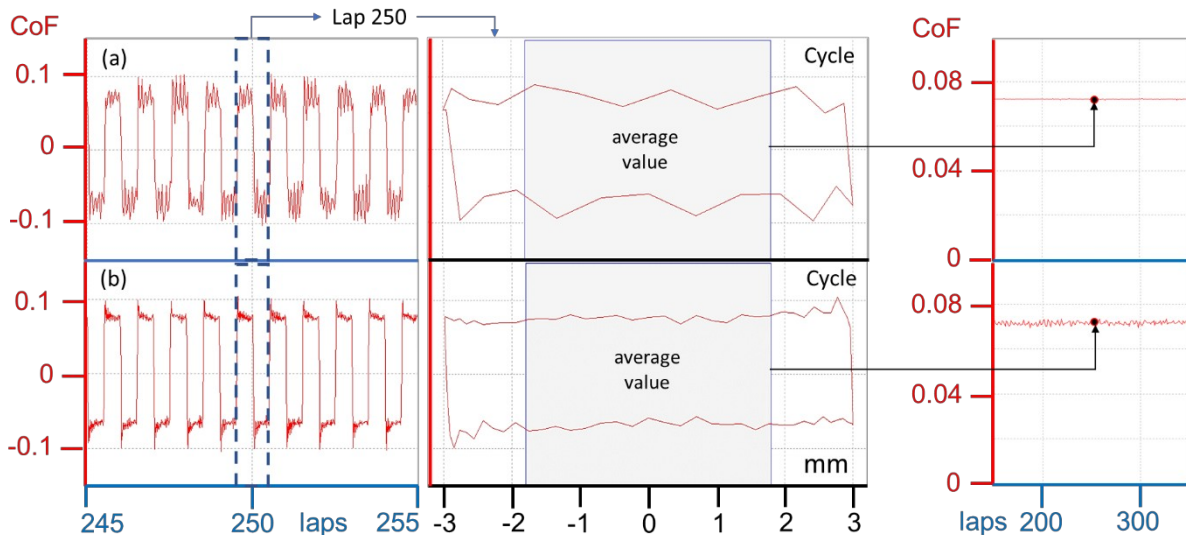
211 For linear reciprocating tests, load was kept at the same value as for the corresponding rotating tests, whereas
212 the linear oscillation frequency of the oscillating plate varied in the preliminary phase. Frequency of 3Hz, 2Hz
213 and 1Hz were tried out and it turned out that the higher the oscillation frequency the more uneven is the friction
214 signal. This effect is not totally clear and could be linked to either augmented film thickness dynamics (e.g film
215 thickness fluctuations due to dynamic reactions in response to motion reversal) or stronger vibrations spreading
216 the instrument frame and affecting the very sensitive LVDT sensors output. The former hypothesis is in line
217 with results by Nishikawa and Kaneta [15],[21] for reciprocating EHL point contact. The latter hypothesis is
218 justified by the fact that Anton Paar provides an adapter kit to perform linear reciprocating tests that transforms
219 the standard rotary configurations into the linear motion configuration (Figure 3). It consists of a sliding plate
220 driven by a dedicated mandrel with an eccentric pivot. This adapter kit introduces two main modifications to
221 the instrument mechanical layout: the mass subjected to sinusoidal acceleration is larger, and the driveline
222 suffers some more backlash in couplings between moving parts (e.g. clearance at the eccentric-plate coupling).
223 Mechanical vibrations are likely to grow rapidly with speed as a result, because of stronger shocks across the
224 driveline.



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Figure 3. Anton Paar adapter kit to perform linear reciprocating tests.

Figure 4a and 4b shows the comparison among the extreme values: 1Hz (Figure 2b) is the frequency that gives rise to the smoothest friction curve and the best cycle shape (at equal sampling rate, always fixed at 80Hz).



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Figure 4. Raw friction curve (on the left), example of cycle shape (in the middle) and equivalent friction curve (on the right) with (a) 3Hz and (b) 1Hz reciprocating sliding frequency.

233 To allow for comparison with rotating tests, raw friction curves from linear tests (left side of Fig. 4) were
234 analyzed by a cycle-by-cycle averaging technique based on which a single representative average value is
235 extracted from each cycle. The area framed by a blue solid square in Fig. 4 delimits the measuring points of
236 each cycle involved into the averaging process, about 3/5 of a cycle. Only the points placed within the central
237 portion of each cycle were taken into account, because variations of speed are minimum there and the
238 hydrodynamic effect is maximum. In doing so, an equivalent friction curve can be plotted collecting the
239 representative values from each cycle as curve points (see the right-hand side of Fig. 4). This technique allows
240 to represent the output of linear tests through equivalent curves having similar trend as rotary friction curves
241 and comparable average values, so that the former can be compared to the latter. The reader should be able to
242 verify this same graphical technique was exploited in other experimental works dealing with linear
243 reciprocating contacts, like Bai et al.[29].

244 Table II provides the testing parameters for the 4 measurement conditions selected in the end; each condition
245 was tested separately. Information about the maximum contact pressure is provided in this table for the

246 individual testing conditions as well. In lubricated point contacts the contact pressure value can be estimated
 247 approximately according to the Hertzian elastic contact model. According to Hamrock et al. [33] the fully
 248 developed EHL condition, in the sense of the piezoviscous-elastic behavior, usually originates when the
 249 maximum contact pressure exceeds 0.5 GPa with common industrial oils. Contact pressure values listed in
 250 Table II are inside the above range; no further verifications were carried out, nor possible because of the test
 251 rig layout (interferometry was impracticable) and the materials tested (ECR analysis was impracticable).

252 **Table II.** Testing conditions

	Load [N]	Spindle Speed [rpm]	Frequency [Hz]	Duration [cycles]	Track radius [mm]	Stroke ^[1] [mm]	Lubricant quantity [μ L]	Contact pressure p_{Hz} (max) [GPa]
MVo (rotating mode)	2	100	-	1000	4 to 7	-	60	0.562
MVo (linear mode)	2	-	1	500	-	6	40	0.562
HVo (rotating mode)	30	150	-	1000	5 to 6	-	100	1.386
HVo (linear mode)	30	-	1	500	-	6	60	1.386

253 ¹ The stroke amplitude is here intended as the segmented length traced by the ball on the disk, i.e one half of the
 254 peak-to-peak distance over a cycle.

256 In this study minor changes in the entrainment speed were accepted and several track radii were sequentially
 257 set in the range from 4 to 7mm for the MVo and from 5 to 6mm for the HVo. Rotating and linear reciprocating
 258 modes were tested separately as this latter requires a specific module to be installed on Anton Paar tribometers.
 259 Every condition was repeated several times with both the tribometers, as specified below in Table III.

260 Much attention was paid to deeply clean and degrease the entire equipment with chemical pure Acetone and
 261 Isopropyl alcohol (IPA) at the end of each test run. Samples and tools were handled with latex gloves and dried
 262 with lint-free tissues. Optical microscopy allowed to check the integrity of the glass surface, the absence of
 263 wear on the ball and the effective removal of any trace of used lubricant and dust before and after every test
 264 run. A temperature and humidity sensor were fitted inside the testing chamber (see Figure 1c) to follow the
 265 evolution of the environmental parameters and make sure that each repetition is performed in consistent climatic
 266 conditions. The average temperature inside the testing chamber was in the range from 20 to 29°C and moisture
 267 content from 30 to 95%. The control of the test rig and data acquisition was both performed with the dedicated
 268 Anton Paar InstrumX[®] Software.

269 The whole experimental survey took several weeks to be completed and each tribometer has been periodically
 270 recalibrated, following the user manual advice.

271 **3. Results**

272 Several repetitions of each testing conditions were carried out: each set covered at least 32 test runs, and many
 273 more in most of the cases. Two average tangential force levels were obtained: 0.145N for tests with the MVo
 274 and 2.139N for tests with the HVo.

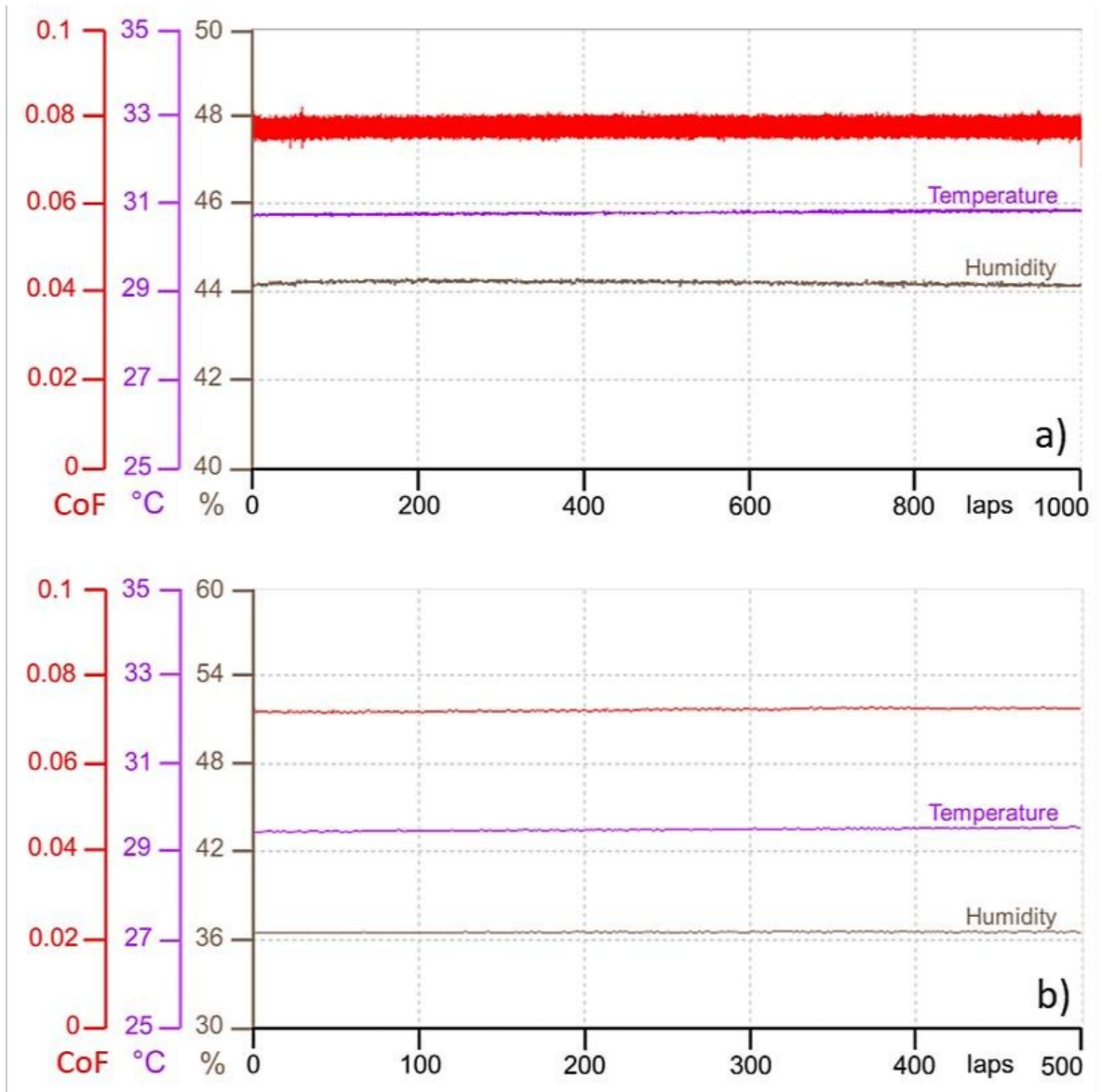
275 Table III gives an overview on the results corresponding to the conditions listed in Table II with the two
 276 tribometers. Figure 5-8 show the output of one friction test belonging to each set of tests listed in Table III.
 277 Since each set of tests includes 32 to 58 tests, it is impracticable to present graphically such an amount of data
 278 in an aggregated manner. Only one friction curve out of 32 to 58 is then displayed to represent the entire set
 279 which it belongs to. Temperature and humidity curves are showed as well, when available as plottable data. . It
 280 is here recalled that, in what follows, displayed curves of rotary test are raw data; those of linear reciprocating
 281 tests are not raw data. The latter were previously analyzed through a cycle-resolved averaging technique, as
 282 already discusses in Sec. 3, in order to depict equivalent friction curves comparable to rotating tests curves.

283 **Table III.** Overview of the results of coefficient of friction for each set of repetitions

Instrument	Oil type	Number of samples	Number of repetitions	AAV	St.DAV	ASt.DV	Laboratory
TRB³	MVo (rotating)	3	46	0.0757	0.0014	0.0017	Anton Paar TriTec laboratory, Corcèlles (CH)
	MVo (linear)	1	32	0.0702	0.0016	0.0003	
	HVo (rotating)	3	42	0.0758	0.0011	0.0022	
	HVo (linear)	4	37	0.0766	0.0008	0.0002	
TRB	MVo (rotating)	1	38	0.0731	0.0022	0.0035	Politecnico di Torino, DIMEAS laboratory, Torino (IT)
	MVo (linear)	2	58	0.0715	0.0006	0.0002	
	HVo (rotating)	3	42	0.0713	0.0011	0.0027	
	HVo (linear)	3	54	0.0767	0.0005	0.0002	

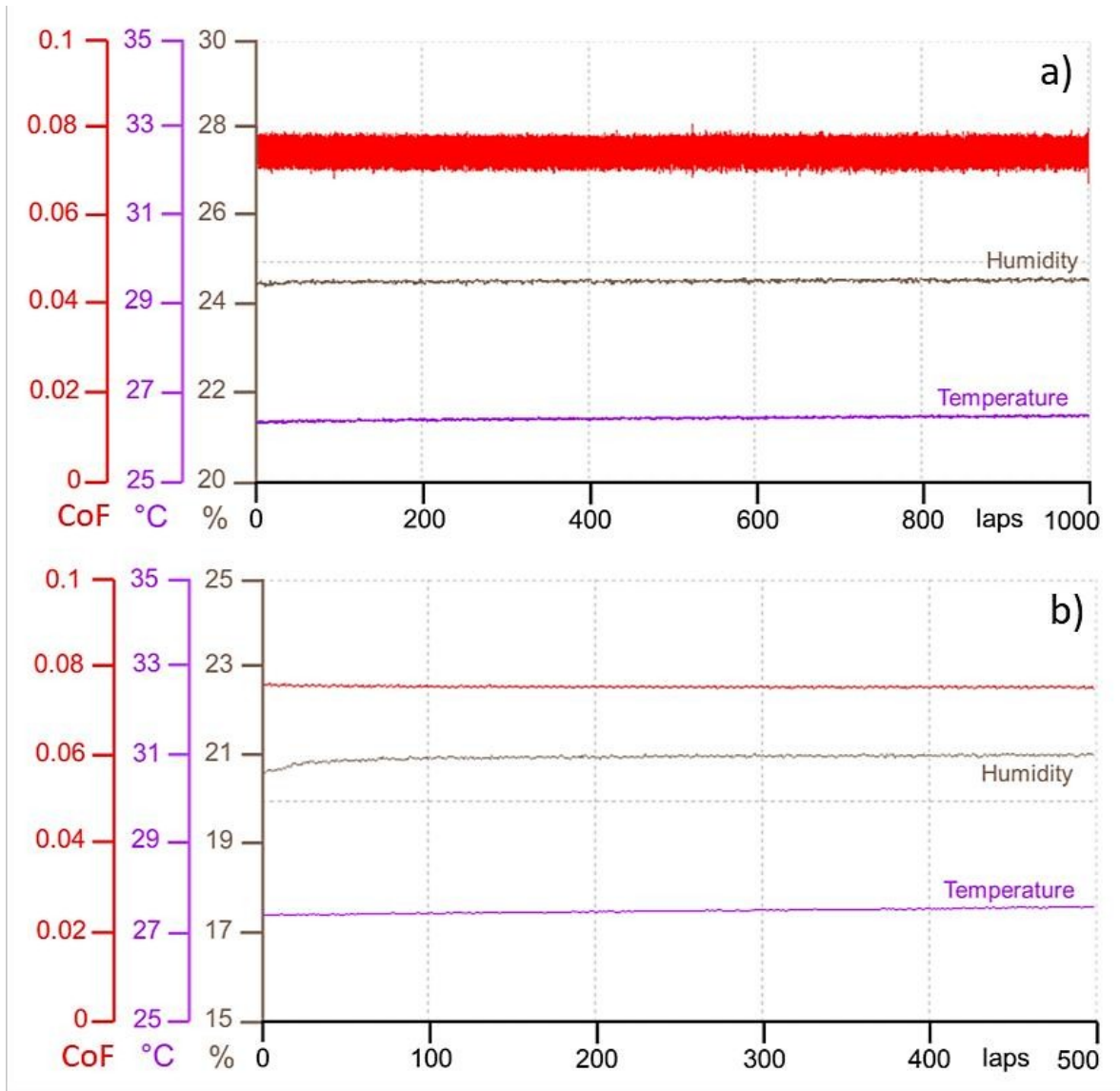
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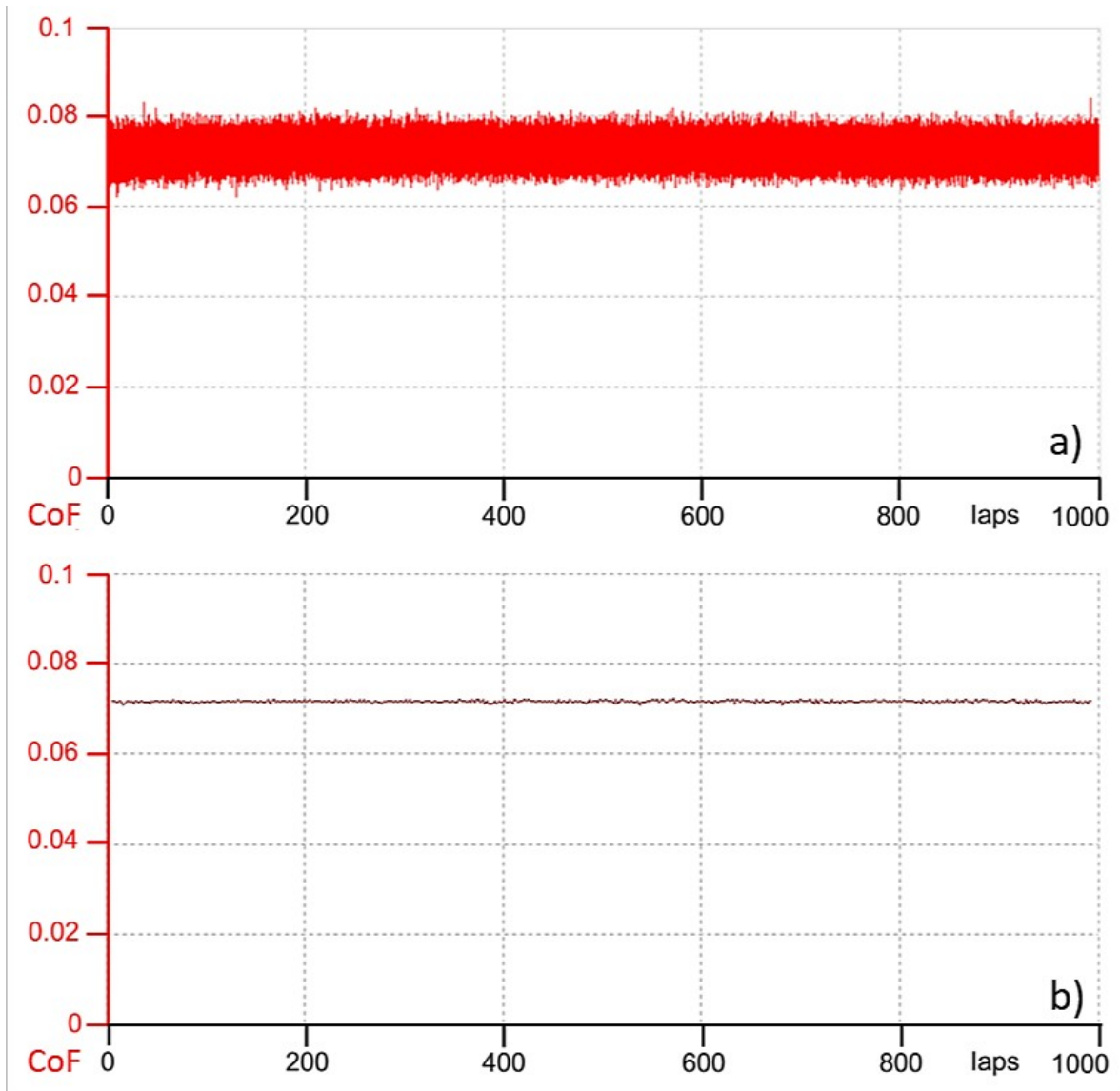
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Figure 5. Example of a CoF curve obtained with the TRB³ and the MVo(2N load) in (a) rotating and (b) linear reciprocating mode.



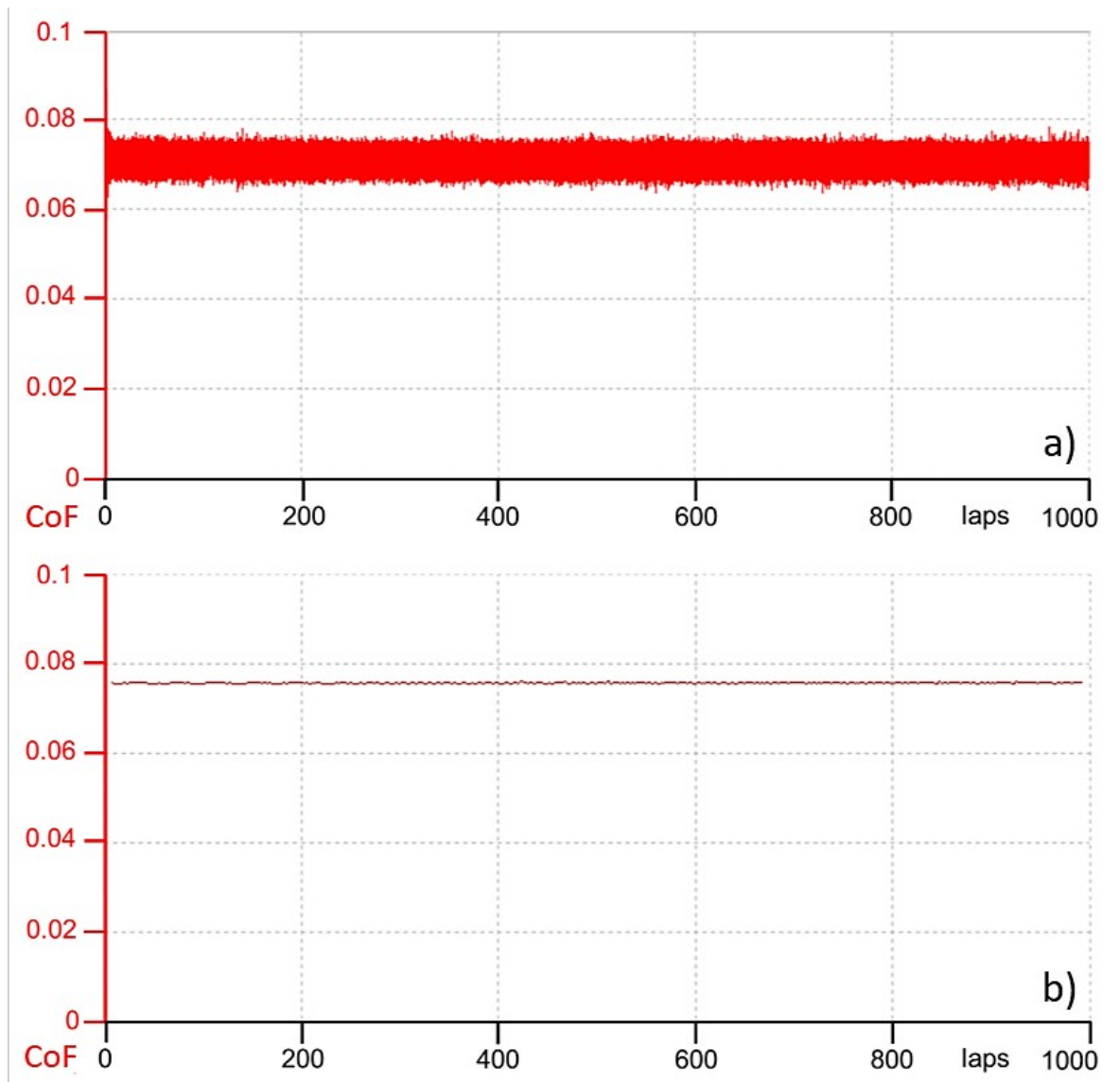
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Figure 6. Example of a CoF curve obtained with the TRB³ and the HV_o (30N load) in (a) rotating and (b) linear reciprocating mode.



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Figure 7. Example of a CoF curve obtained with the TRB and the MVo (2N load) in (a) rotating and (b) linear reciprocating mode.



295
 296 **Figure 8.** Example of a CoF curve obtained with the TRB and the HVo (30N load) in (a) rotating and (b) linear
 297 reciprocating mode.

298 Fig. 5 to 8 combined with values in Table III prove that all the test runs in all conditions provided
 299 exceptionally stable and repeatable results in terms of both friction curves shape and average CoF values. This
 300 is also evidence of the fact that the risk of starvation effects (oil loss out of the contact zone) is sufficiently low
 301 despite the lack of oil bath. A relevant concern among others is related to centrifugation of the lubricant outside
 302 the contact. Grützmacher et al. [27] reported a near no-wear tribological condition in pure sliding ball-on-disc
 303 tests at 6mm track radius and 0.08m/s sliding speed with a Castrol PAO30 (30cSt viscosity) oil. They attributed
 304 this result to little enough centrifugal forces to have a nearly zero lubricant film shrinking and concluded that
 305 higher viscosity results in a less pronounced influence of the centrifugal forces on the lubrication regime. In the
 306 present study, friction tests featured much higher viscosity oils and lower rotational speed (thus, centrifugal
 307 forces), so that centrifugal effect should be negligible with no impact on the lubricant film build-up and stability
 308 at the contact interface. Moreover, Grützmacher et al. used not-additivated oils for which effect of varying
 309 adhesive properties on retaining the lubricant in the contact can be neglected. On contrary, commercially fully

310 formulated oils have additives which strengthen the solid-liquid adhesion properties, thus reducing further the
311 risk of centrifugal oil leakages.

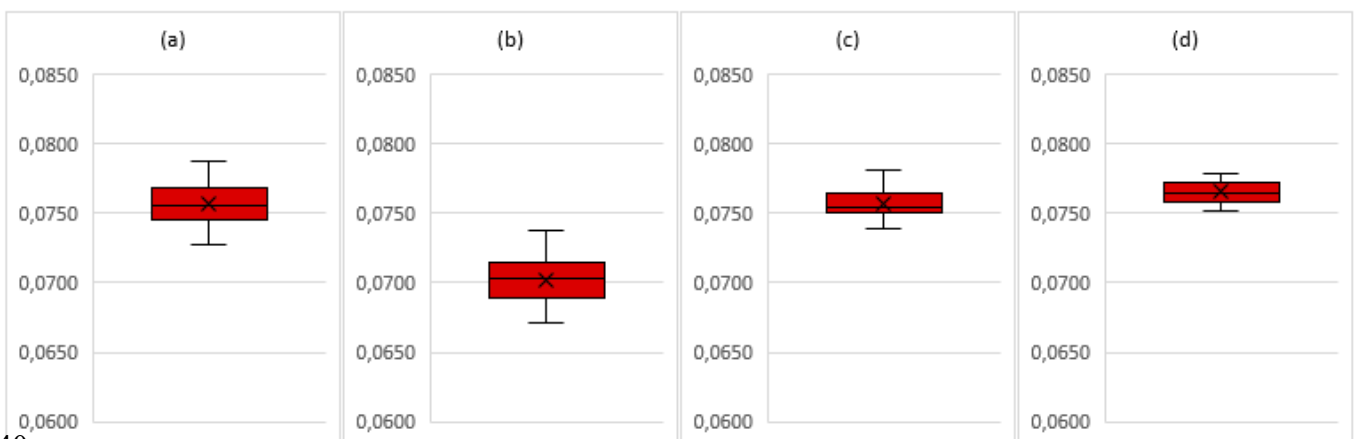
312 EHL friction tests appeared to be very sensitive to disturbances (shocks and vibrations) coming from the
313 environment, even at low intensity. Example of external disturbances experienced during the experimental
314 campaign are people walking by the machine, noise and vibrations coming from rooms nearby, accidental hits
315 on the table, etc...therefore care should be taken to avoid these kinds of disturbances. The cleanliness of the
316 surfaces played an important role too. While carrying out preliminary tests it also happened that some of these
317 tests produced very unstable friction curves whose mean value and noise were very different than usual ones,
318 despite obvious external disturbances were lacking. This kind of erratic behavior was later proved to be
319 imputable to imperfect cleaning, in particular dust and fibers passing through the thin oil film.

320 Although ASTM G99 [34] (the sole standard for pin-on-disc tribological tests) recommends using all the data
321 from each set of measurements, including outliers, no test where either the influence of external disturbance or
322 cleanliness issues were obvious has been taken into account in this study, as they are not representative of the
323 instrument working state. The authors of this paper would like to underline that the approach taken in this
324 investigation is not in contrast with the standard anyway. Outliers are values that deviate from the average “by
325 accident”, i.e. linked to unavoidable accidental errors associated to the phenomenon under study. In this study,
326 discarded tests are the result of systematic and well identified external causes (anyway impossible to totally
327 overcome). From this perspective, the discarded results are not even attributable to the phenomenon under
328 study, so they are neither outliers.

329 Such high sensitivity of tests has pros and cons in SHM techniques. A very good detection capability of little
330 damages/problems is expected, by the price of high sensitivity to some healthy changes of operational and
331 environmental conditions too [3].

332 The box-plots in Figure 9 to 12 show the scattering of the mean values within each set of repetitions presented
333 in Table III. The values of the CoF obtained in this work belong to the range from 0.01 to 0.1 which is very
334 typical for EHL with industrial oils, as confirmed by Stachowiak and Batchelor [35]. Besides, the whole set of
335 average values obtained with both the MVo and the HVo fall inside the sub-range from 0.065 and 0.080.
336 Equivalent friction curves from linear tests feature an apparent lower curve noise and lower scattering because
337 they portray data which have already undergone some processing; for this reason, their scattering is naturally
338 lower.

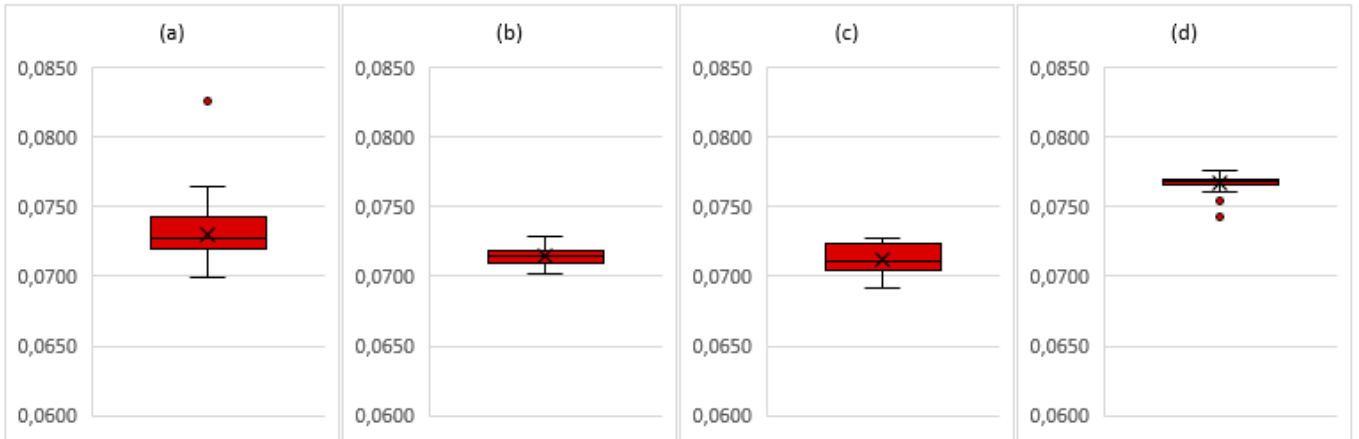
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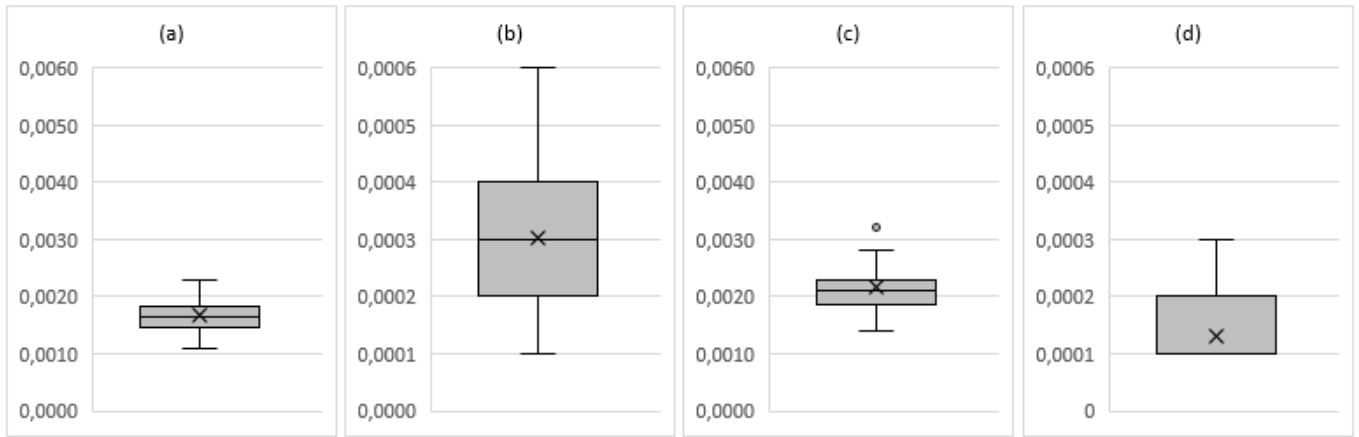
Figure 9. Box-plots of the mean CoF values for each set of repetitions on the TRB tribometer: (a) MVo rotating tests; (b) MVo linear tests; (c) HVo rotating tests; (d) HVo linear tests.



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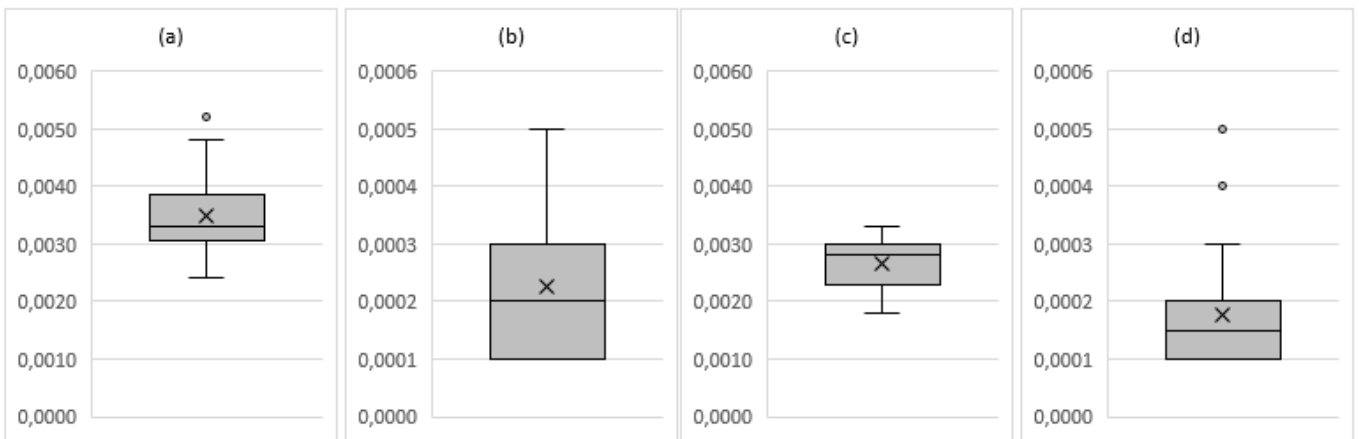
Figure 10. Box-plots of the mean CoF values for each set of repetitions on the TRB tribometer: (a) MVo rotating tests; (b) MVo linear tests; (c) HVo rotating tests; (d) HVo linear tests.



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Figure 11. Box-plots of the CoF standard deviation values for each set of repetitions on the TRB tribometer: (a) MVo rotating tests; (b) MVo linear tests; (c) HVo rotating tests; (d) HVo linear tests.



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Figure 12. Box-plots of the CoF standard deviation values for each set of repetitions on the TRB tribometer: (a) MVo rotating tests; (b) MVo linear tests; (c) HVo rotating tests; (d) HVo linear tests.

352 It is commonly known that the environmental state, i.e. temperature and moisture content, influences what
353 happens at the tribological interface. This is true especially in dry contacts, but also lubricated contacts with

354 little amount of lubricant may be affected . Temperature is a strong influencing parameter, large variations
 355 would definitely influence the lubricant rheological behaviour performance and thus friction measurements.
 356 Conversely, the moisture content variations could turn out to be less important. The main concern about
 357 humidity is the risk that water droplets condensate on the oil surface; if the little amount of lubricant is not able
 358 to completely insulate the contact region from the environment, the lubricant might become a water-oil mixture
 359 and load bearing capacity might be affected. In order to be sure that humidity has no influence on friction curves
 360 despite the little amount of oil, some additional sets of tests were carried out into an environment close to the
 361 dewpoint. These comparative tests (whose results are omitted) revealed that the contact zone is effectively
 362 insulated from the environment and procedures apply regardless of the ambient humidity.
 363 The effect of temperature was not directly addressed in this study because the instruments were operated in
 364 thermally stable environments. As a general rule, temperature variations should be kept within a well-defined
 365 narrow range for the sake of reference measurements. However, scientific instruments like tribometers are
 366 usually operated in laboratory rooms where temperature is controlled well enough (20°C to 30°C typically) to
 367 neglect temperature effects.

368 4. Discussion

369 4.1. Statistical analysis of results

370 For the sake of statistical analysis, each test has been accounted for through two statistical parameters which
 371 describe the friction curve appearance: the average ($\bar{\mu}_{CoF}$) and standard deviation ($CoF_{St.Dev.}$) of the
 372 experimental CoF points of test. The overall statistics has been compiled by collecting these two parameters
 373 from each test to calculate three representative quantities for each series of tests:

- 374 - The average of the test average values $\bar{\mu}_{CoF}$ (herein referred to as AAV);
- 375 - The standard deviation of the test average values $\bar{\mu}_{CoF}$ (herein referred to as St.DAV);
- 376 - The average of the test standard deviation values $CoF_{St.Dev.}$ (herein referred to as ASt.DV).

377 Section “Discussion” will return to these statistical quantities to reveal the tribometer-to-tribometer variability
 378 and the test-to-test variability for each condition.

379 Based on values listed in Table III, two types of variability of the results within each set of repetitions are
 380 discussed below: tribometer-to-tribometer variability and test-to-test variability.

381 Details on the tribometer-to-tribometer variability of the CoF mean value can be deducted comparing the AAVs
 382 in similar conditions by different instruments. Percentage-point differences are listed in Table IV and it is quite
 383 evident that repeatability on different instruments is very good.

384 **Table IV.** Tribometer-to-tribometer variability

Representative quantity	MVo – rotating	MVo – linear	HVo – rotating	HVo – linear
AAV _{TRB} - AAV _{TRB3} [%]	3,5%	2%	6%	0.2%

385
 386 The St.DAV has to be taken into account to evaluate the test-to-test variability on the same tribometer in equal
 387 testing condition, see Table V. Values of the St.DAVs ranging from 0.65% to 3% of the corresponding AAVs
 388 point out that the scattering is low for each testing condition, even though more than one sample are used

Table V. Test-to-test variability

Tribometer	Representative quantity	MVo – rotating	MVo – linear	HVo – rotating	HVo – linear
TRB	St.DAV / AAV [%]	1.85%	2.23%	1.45%	1.04%
TRB ³		3.00%	0.84%	1.54%	0.65%

390

391 . Much of this scattering is believed related to the gluing of the glass disco onto the stub and to the position
 392 change of the two counterweights necessary to equilibrate the arm own weight. Since the entire experimental
 393 campaign was carried out over several weeks, the counterweights position changed repeatedly, and a slightly
 394 different position of the counterweights means a slightly different effective load applied on the contact. The
 395 CoF is very low, thus even minimal variations are found to affect somewhat the test mean value. For the same
 396 reason, a contribution coming from the tolerance of the arm balancing must be accounted for the difference
 397 between the AAV obtained with various instruments.

398 The ASt.DV measures the average friction curves noise over a series of tests in same conditions. Table III shows
 399 that a similar noise (i.e. the “thickness” of the friction curve itself) characterizes the curves measured in the
 400 same test condition by the same instrument. A tiny noise variability may appear if the same test is repeated by
 401 different instruments (tribometer-to-tribometer variability) and may also appear using different samples, but
 402 measurements remain always consistent with one another. The thickness of friction curves seems to be an
 403 instrument own characteristic, but further verification on many more tribometers will be necessary to validate
 404 this conclusion. Linear tests have an ASt.DV that is one order of magnitude smaller than the corresponding
 405 value of rotating tests with the same oil. Such a discrepancy is nevertheless more apparent than real: being linear
 406 tests analyzed with a cycle-by-cycle averaging technique their ASt.DV suffer, in a way, double averaging.

407 4.2. Reference testing procedure

408 It becomes clear that each test condition owns a characteristic mean value and, to some extent, a characteristic
 409 standard deviation value of the friction coefficient; both disclose little to no affecteion by the instrument used
 410 to run the test. This value is herein referred to as the ‘Characteristic AAV’ and calculated by further averaging
 411 the corresponding AAVs by the two instruments (see Table III). The characteristic AAV is reported in Table
 412 VI along with the other global statistical quantities calculated similarly.

413 **Table VI.** Global values

Lubricant	Test configuration	Characteristic AAV	Global St.DAV	Global ASt.DV
MVo (2N)	Rotating	0.0744	0.0018	0.0026
	Linear reciprocating	0.0708	0.0011	0.0003
HVo (30N)	Rotating	0.0735	0.0011	0.0024
	Linear reciprocating	0.0767	0.0006	0.0002

414

415 On the basis of these data, reference ranges of acceptability for the CoF are proposed for each testing condition
 416 and the corresponding limiting values are displayed in Table VII. Reference ranges of acceptability involve
 417 both the CoF test average and its test standard deviation. These ranges of acceptability for the CoF average are
 418 determined as follows:

$$419 \quad \left\{ \begin{array}{l} (Characteristic\ AAV + 3 \cdot Global\ St.\ DAV) < \bar{CoF} < (Characteristic\ AAV - 3 \cdot Global\ St.\ DAV) \\ CoF_{St.Dev.} < Max(St.Dev) \end{array} \right.$$

420 where the Max(St.Dev.) is equal to the maximum standard deviation value recorded during each series of tests
 421 in this study.

422

423 Any user disposes of two parameters to assess the proper functioning of its own tribometer: the average CoF
 424 value of the reference test must remain into the prescribed range and the standard deviation value of the same
 425 test should not exceed the prescribed value.

426 These ranges of acceptability are proposed as a yardstick through which assessing whether a tribometer is
 427 properly working with enough confidence. Ranges of acceptability are necessary to account for imperfect
 428 reproducibility of test conditions and unavoidable production differences among the instruments, as already
 429 addressed above in Par. 4.1. Whenever one of the two parameters do not meet the requirements, no further
 430 specific instructions are provided by such procedure in the present release anyway, except that something is
 431 going wrong with the tribometer. Features extraction from the monitoring parameter signal would be necessary
 432 indeed to obtain complete information on potential damages of the instrument [3]; e.g. through a case study
 433 based on known damages/malfunctions one could provide the atlas of correlations between the occurrence of
 434 some damages/malfunctions and the output of the reference test. This further step has not been included yet in
 435 this work. Nonetheless, this procedure is already able to perform a useful technical diagnosis. If the average
 436 value is out of range and the test is performed correctly (no effect of contamination or dust, etc...) then a problem
 437 of calibration of the instrument could be there, or a sensor failure or even the arm manufacturing fault (the
 438 measuring arm stiffness is calibrated). On contrary, if the standard deviation is out of range, it could suggest
 439 that abnormal vibrations due to either the spindle bearings, or the electric motor, or some other poorly fixed
 440 component is arising; or even that an electrical disturbance problem due to electronic cards may distort the
 441 measured CoF signal.

442 **Table VII.** Ranges of acceptability

Lubricant	Test configuration	Acceptable range for the CoF average value		Acceptable CoF standard deviation	
		Lower bound	Upper bound	Maximum value	
MVo	Rotating	0.069	0.080	TRB ³ :	0.0026
				TRB:	0.0052
	Linear reciprocating	0.068	0.074	TRB ³ :	0.0006
				TRB:	0.0005
HV0	Rotating	0.070	0.077	TRB ³ :	0.0032
				TRB:	0.0033
				TRB ³ :	0.0004

443

444 **5 Checking the EHL contact condition**

445 The pin-on-disk set-up does not allow any direct film thickness measurements. In this specific case,
446 interferometric measurements were impracticable because of the test rig layout and ECR analysis was
447 impracticable because the tested materials are electrical insulating. However, there exist indirect evidence
448 suggesting that the ball and the sample were separated by a continuous film of lubricant in all the tested
449 conditions. Notably, no signs of wear on the ball and sample were visible under the optical microscope, and
450 friction curves were very regular and smooth with a low CoF value.

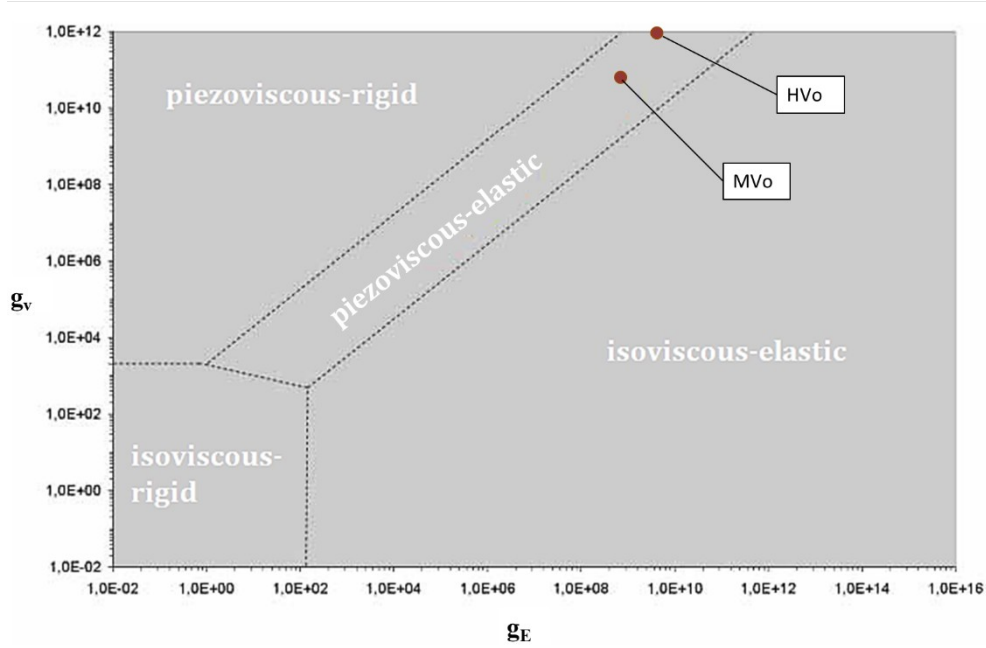
451 The precise determination of the lubrication regime is not the primary objective of this paper. The friction
452 coefficient is the star in this paper indeed, not the lubricant film thickness, and its repeatability is the core feature
453 of the proposed test condition. With that in mind this section has been introduced just in support of the above
454 experimental evidence, so as to strengthen the plausibility of the no-wear regime (as microscope inspection
455 suggests) and focus the correlation between lack of visible wear and CoF repeatability in tribological testing.
456 Some of the currently available empirical formulae to predict the lubricant film thickness were applied blindly,
457 according to the task they have been developed for. The film thickness and the corresponding value of the
458 roughness factor $\Lambda^{[2]}$ (or lambda ratio) is computed for the tests in rotating mode only. The lambda ratio has no
459 unified definition: someone defines it as the ratio of the central (or average) film thickness to the composite
460 roughness, like Tallian et al. [36] and Poon et al. [37], others refer to minimum film thickness to be more
461 conservative, like Stachowiack et al. [33] and Hamrock et al. [35], others do even make it clear, for instance
462 Bair and Winer [38]. For the sake of precaution, the minimum film thickness at the exit region of the contact is
463 taken into account here.

464 The output of three empirical formulae are compared: the popular equation by Hamrock and Dowson [6], one
465 model developed specifically for EHD sliding contacts by Wilson and Shew [39] and the general equation
466 proposed by Masjedi and Khonsari [40]. Strictly speaking the film thickness theory by Hamrock and Dowson
467 was originally developed for nominal pure rolling contacts; though some authors in the scientific literature state
468 that it may also apply to slide-roll conditions [41],[42]. Wheeler et al. [42] represented on a M-L plane the
469 validity range of several empirical equations by a number of authors, M' and 'L' being the Moes' parameters
470 [43]. According to this map, Masjedi and Khonsari formula better describes the testing conditions explored in
471 this study. The corresponding Moes parameters are L=9.47, M=1219.9 for the MVo and L=18.7, M=2377.2 for
472 the HVo tests. The reader can verify that these values identify two points laying into the Masjedi and Khonsari
473 equation domain into the diagram reported in [42].

474 Table VIII summarizes the values of the physical properties and geometrical parameters having a role in the
475 film thickness equations. Non-dimensional parameters 'U', 'W' and 'G' are computed according to Hamrock
476 and Dowson and 'C' is the thermal-sliding correction factor introduced by Wilson and Shew [39]. The 'C'

² The roughness factor is classically defined as: $\Lambda = h_{min} / \sqrt{R_{q,1}^2 + R_{q,2}^2}$ where h_{min} is the minimum lubricant film thickness and $R_{q,1}$, $R_{q,2}$ are the root-mean-square values of the mating surfaces roughness.

477 factor takes into account the inlet shear heating effect and is basically used as pre-multiplying factor for the
 478 Hamrock and Dowson equation. The ‘C’ factor is function of the SRR and the thermal loading parameter ‘ L_t ’
 479 which takes into account the temperature-viscosity effect and the thermal conductivity of the lubricant [39].
 480 Figure 13 represents the Hamrock-Dowson’s chart of the EHD operating conditions for lubricated rolling-
 481 sliding point contacts [33]. It ensures that the lubricated interface is subjected to the piezoviscous-elastic
 482 behavior with both the MVo and the HVo, which is the range of applicability for the above formulae. G_v and
 483 G_E are the dimensionless Hamrock’s viscosity and elasticity parameters.



484
 485 **Figure 13.** Hamrock-Dowson’s chart of EHD operating conditions highlighting the points corresponding to the tests
 486 with the MVo and HVo.

487
 488 **Table VIII.** Summary of the physical and geometrical parameters

Common parameters		Specific parameters	MVo		HVo	
			Rotating	Linear	Rotating	Linear
R_d [m]	0.003	v_d [m/s]	0.0628	0.0189	0.0942	
R_b [m]	∞	v_b [m/s]	0.0000	0.0000	0.0000	0.0000
k ^[1]	1	\bar{U} [m/s] ^[1]	0.0314	0.0094	0.0471	
R' [m] ^[1]	0.0015	F_N [N]	2	2	30	30
E' [GPa] ^[1]	129	η_0 [cP]	105.2		621.1	
SRR	2	α [GPa ⁻¹]	30.5		34.8	
		α_T [K ⁻¹] ^[2]	3.66		2.74	
		h [Wm ⁻¹ K ⁻¹] ^[3]	0.18		0.31	

489 ¹ Entrainment speed calculated according to Hamrock et al. [44].

490 ² Here assumed equal to the “ASTM slope” [34] from Walther formula.

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³ Thermal conductivity for the HVo has been calculated according to the formula proposed by Larsson and Andersson [45] for PAO. Being the chemical composition of the MVo confidential, a typical value for light oils has been taken into account for the calculations.

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Table IX lists the forecast provided by each equation. Since sliding speed was handled as an almost free parameter, which varies in the narrow range corresponding to the track radius variations defined in Table II, the forecasts in Table IX were calculated for the mid-range linear speed, i.e. at 5.5mm track radius.

Table IX. Predicted values of the roughness factor (rotating mode only)

Oil Type	Equation	ρ_{Hz} (max) [GPa]	U	W	G	C	L_t	Calculated h_{min} [nm]	Λ
MVo	Hamrock and Dowson [6]					-	-	17.6	1.22
	Wilson and Shew [39]	0.564	$1.72 \cdot 10^{-11}$	$6.92 \cdot 10^{-6}$	$3.91 \cdot 10^3$	0.93	0.00	10.7	0.75
	Masjedi and Khonsari [40]					-	-	17.3	1.20
HVo	Hamrock and Dowson [6]					-	-	68.0	4.71
	Wilson and Shew [39]	1.391	$1.52 \cdot 10^{-10}$	$1.04 \cdot 10^{-4}$	$4.47 \cdot 10^3$	0.81	0.01	43.3	3.00
	Masjedi and Khonsari [40]					-	-	69.8	4.84

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Hamrock and Dowson equation and Masjedi and Khonsari equation give similar outputs. Both predict Λ close to unity for the MVo and greater than 4 for the HVo. By the way, the Hamrock and Dowson formula for minimum film thickness is known to be conservative as predicts film thickness slightly lower than measured [33]. Wilson and Shew's formula is the worst-case scenario as it considers the diffused shear thinning effect due to sliding. This model allows for shear stress to produce heat into the nanometric lubricant film; heat is responsible for the loss of viscosity and load-bearing capacity with reduces the supporting lubricating film, as a consequence. According to this model, if L_t is lower than 0.1 then thermal effects are usually said to be negligible [39] and the lubricant behaviour should not differ very much from that in pure rolling conditions. Nonetheless, this approach likely reveals too conservative for the scenario under consideration. In this work the spindle speed was kept as low as possible primarily to avoid any contribution from the tribometer dynamics. This contributed also to keeping L_t at low values, i.e. thermal effects should be limited, especially when viscosity is low. Despite the L_t factor takes a value much lower than 0.1 a non-negligible 10% to 20% reduction of the Lambda ration appears in all the cases (see 'C' factor values in Table IX). Identification of alfa and k in L_t may have played a role since no experimental values are available for these quantities (see Table VIII). In the opinion of the authors, however, although a 10% or 20% is not a negligible difference, the the lubrication regime foreseen by Wilson and Shew's formula does not differ significantly compared to the other formula. The predicted conditions for the MVo are always a limiting full-film lubrication (or an extremely smooth mixed lubrication) with Λ about 1, no matter what the equation used; similarly, complete full film lubrication is always expected with the HVo since Λ is always equal or greater than 3, no matter what the equation used.

520 The lambda ratio is a very easy-to-use parameter to correlate the film thickness to lubrication regimes and
521 friction. Its use among engineers and researchers is persistent, evidence that a vast need for such a simple tool
522 there exists for estimating the state of lubrication in industrial problems such as in machine components design.
523 However, it has many limitations and some authors in the scientific literature have agreed over the years that
524 establishing a relation between the film thickness and the initial (nominal) surface roughness is not sufficient
525 to foresee precisely the lubrication state and the related level of friction [46], [47], [48]. In the past years
526 various authors have plotted experimental Stribeck curves (or Stribeck-like curves) against the roughness factor
527 proposing a number of threshold values for Λ setting the onset of mixed and full-film lubrication; among the
528 others it is worth to cite Tallian et al. [36], Poon et al. [37] and Bair and Winer [38]. This is mainly due to the
529 fact that roughness related effects in EHL problems are scale dependent and operating conditions dependent
530 (see [48], [46] and [35]), meaning that limiting values of Λ , if any, are themselves virtually function of
531 roughness, lubricant properties and operating conditions.

532 For this reason, setting Λ limiting values of general validity is hardly feasible. Generally speaking, mixed
533 lubrication is observed when $1 < \Lambda < 5$ and full-film EHL when $\Lambda > 3$ [33] (the reader should notice that the ranges
534 overlap). In the range $1 < \Lambda < 3$ mixed lubrication often comes down to some glazing of the surface; asperities
535 flatten out either elastically or plastically under extreme pressure and effective body separation exists even if
536 estimated lubricating quality still suggested vast contact interference [16] (micro-EHL' occurs [35]). This
537 explains why a lot of machine elements operate with little apparent damages close to $\Lambda=1$ where surface
538 distresses should prevail [35]. This same motivation could explain why no wear appears in tests with the MVo
539 despite the estimated lubricant film is of the same order of magnitude of roughness.

540 Moreover, E' in Table VIII accounts for the Young modulus of ruby (sphere) and bulk glass (disc) only. Any
541 contribution to the local deformation coming from the glue beneath the disc has been overlooked; glues have
542 elastic moduli of few GPa. Being the glass disk very thin, the equivalent surface compliance of the flat counter
543 body could be higher than what is considered in calculations. Lastly, the roughness factor has been calculated
544 with respect to the minimum film thickness value, which is conservative. The minimum film thickness
545 characterizes the two lateral necking structures in the contact outlet region, representing a very tiny portion of
546 the whole contact though. Even though rubbing of the bodies were assumed, asperities interaction would be
547 restricted to this tiny portion of the contact. Most of the contact is supported in the central flat region in fact,
548 where lubricant thickness is expected to be larger (1 to 3 times larger than minimum film thickness, at least
549 under isothermal Newtonian lubrication conditions [49]).

550 To the best of the authors knowledge, no empirical equations have been developed to predict the film thickness
551 of sliding EHD point contacts in linear reciprocating mode. Two attempts were made by Petrousevitch et al.
552 [50] and Hook [51], but for nominal line contact only (e.g. cylinder-cylinder contact). Previsions from formulae
553 for stationary conditions cannot be fully trusted as they are not able to properly describe the interface condition
554 along the whole stroke. They might roughly represent what happens at the stroke mid-point, where the kinematic
555 approaches stationary conditions, but they would predict a null film thickness at both the stroke ends where
556 speed is nominally zero. According to Nishikawa et al. [21] fluid-dynamic effects assure non-null film thickness
557 even under motion direction reversal.

558 6. Conclusions

559 In this paper two industrial reference testing conditions were developed on two Anton Paar pin-on-disc
560 tribometers for two testing mode: unidirectional rotating and linear reciprocating mode. Two commercial oils
561 were compared, a middle viscosity oil (MVo) and a high viscosity oil (HVo). A special lubricated contact with
562 no lubricant bath was tested where a little amount of lubricating oil was added on the top of the sample before
563 each test. Despite the absence of any oil bath, the lubricant surface tension has proven high enough to retain the
564 oil in the contact area, preventing the transition from full-film lubrication to starved lubrication with sufficient
565 confidence. This set-up was selected as it perfectly meets some key features required for reference tests suitable
566 to industrial applications: it minimizes cleaning issues and the related waste of time; it lowers the risk of
567 lubricant contamination and avoids material waste. The result is a process easy to prepare and carry out, cheap
568 and potentially fit for the application on any tribometer whatsoever.

569 As to the oil quality, the HVo look a priori the best choice in terms of performance within the scope of this
570 application, as its higher pressure-viscosity coefficient ensures a stronger supporting effect and larger calculated
571 lubricant film. Available formulae for film thickness allowed indeed to estimate the lubrication regime for the
572 testing conditions in rotating mode to a first approximation; based on available data, full-film lubrication is to
573 be expected for the HVo and a very gentle mixed lubrication for the MVo where separation of the rubbing
574 bodies still exists. Nevertheless, the use of the MVo has still many advantages. Since perfect cleaning is of
575 paramount importance, the degreasing procedure after each test is particularly laborious even if small amounts
576 of the HVo are used. Acetone and common gasoline, which are normally aggressive enough to dissolve the
577 MVo, usually fail and specific strong spray solvents must be used in the interest of saving time. Therefore, the
578 MVo is still preferable for verification tests at low contact load, due to the ease of applying and removing the
579 lubricant, keeping the equipment clean and, consequently, lessening the environmental impact.

580 All the test runs provided exceptionally stable and repeatable results, as discussed earlier in this paper, but an
581 extensive investigation on a number of tribometers is necessary in the future anyway to understand to what
582 extent these values are actually reliable. Moreover, all these significant results were attained with no strict
583 requirements either in terms of speed, which is the most influencing parameter over the EHL regimes, or in
584 terms of geometrical repeatability of samples. Therefore, the proposed method may apply to the technical
585 diagnosis of tribometers by letting users make, at least, a preliminary assessment on the proper functioning of
586 the instrument itself.

587 This paper is intended as the first step of a more comprehensive investigation which would need many more
588 pin-on-disk tribometers of a number of manufacturers. A case study on the effectiveness of this procedure in
589 highlighting an existing malfunction would be meaningful, e.g. by reproducing known typical troubles and
590 malfunctions then recording the output of the same reference test.

591 If these further observations would disclose promising results, the outlined procedures could be proposed to
592 become part of an international standard and adopted by those industrial laboratories which require regular
593 examinations of their tribometers. It may be also an effective verification method for inter-laboratory cross
594 studies, before which a common sample should be tested to make sure the instruments all provide consistent
595 measurements.

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600 **Appendix A – Instruments Specifications**

601 Instruments specifications are presented in Table A1 in compliance with manufacturers technical specifications.

602

603 **Table A1.** Technical specification of the two Anton Paar Tribometers

Machine specifications	TRB³	TRB
Normal load (dead weight/s)	0.25N to 60 N	0.25 to 60 N
Friction force	up to 20 N	up to 20 N
Friction force resolution	0.06 mN	0.06 mN
Rotation speed	0.2 - 2000 rpm	0.3 – 1500 rpm
Rotation speed resolution	0.0001 rpm	0.0001 rpm
Linear frequency	0.01 - 10 Hz	0.01 - 10 Hz
Linear stroke	up to 60 mm	up to 60 mm
Linear stroke resolution	2 mm	2 mm
Sample diameter	up to Ø 56 mm	up to Ø 60 mm
Radial position (radius)	up to 40 mm	up to 40 mm
Radial position resolution	0.05 mm	0.05 mm
Angular position resolution	0.1°	0.1°
Relative Humidity	0% to 99% (integrated)	15% to 99% (external)
Relative humidity resolution	0.01%	0.8%
Temperature	-45° to 125°C (integrated)	-100° to 200°C (external)
Temperature sensor resolution	0.015 °C	0.1 °C

604

605 **Appendix B – Lubricants technical data**

606 Table B1 provides the data about viscosity and other physical properties for the two lubricating oils used in this
 607 experimental work. Data correspond to certified values provided in technical sheets by certified laboratory.

608 **Table B1.** Physical properties of lubricating oils

OILS	Property	Test method	Value
Middle Viscosity oil (MVo)	Density @15 °C [kg/dm ³]	ASTM D1480	0.861-0.862
	Viscosity [cSt] @40°C	ASTM D445	57

	Viscosity [cSt] @100°C		7.6
	Viscosity Index	ASTM D2270	101
	Flash Point [°C]	ASTM D92	225
	Pour Point [°C]	ASTM D97	-15
High Viscosity oil (HVo)	Density @25 °C [kg/dm ³]	ASTM D1480	0.844
	Density @40 °C [kg/dm ³]		0.835
	Density @100 °C [kg/dm ³]		0.799
	Viscosity [cSt] @25°C	ASTM D2162	1008.0
	Viscosity [cSt] @40°C		421.0
	Viscosity [cSt] @100°C		41.4
	Viscosity Index	ASTM D2270	123
	Flash Point [°C]	ASTM D92	>93

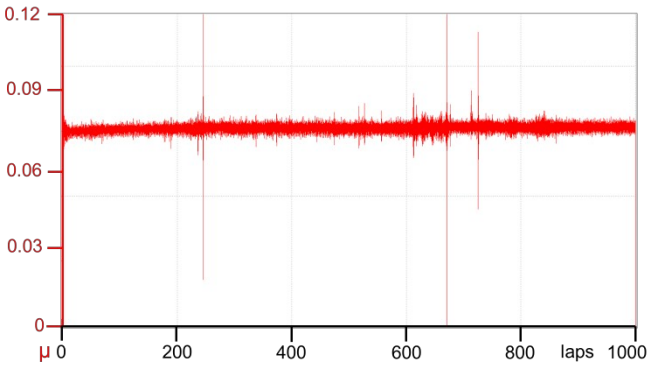
609

610 **Appendix C – Atlas of rejected tests**

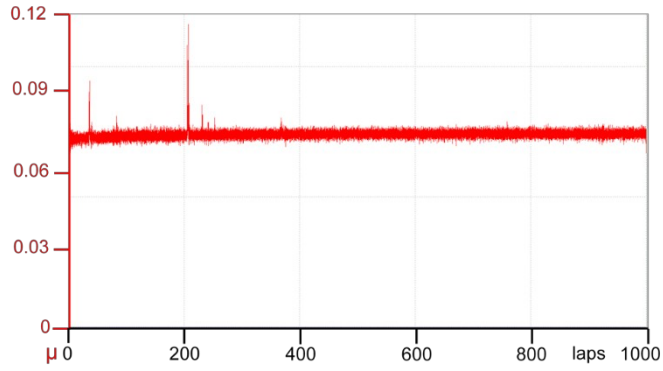
611 Table C1 collects some examples of discarded tests in unidirectional rotating mode related to identified causes.

612 Table C2 is the corresponding table for linear reciprocating tests.

613 **Table C1.** Atlas of rejected tests in unidirectional rotating mode tests

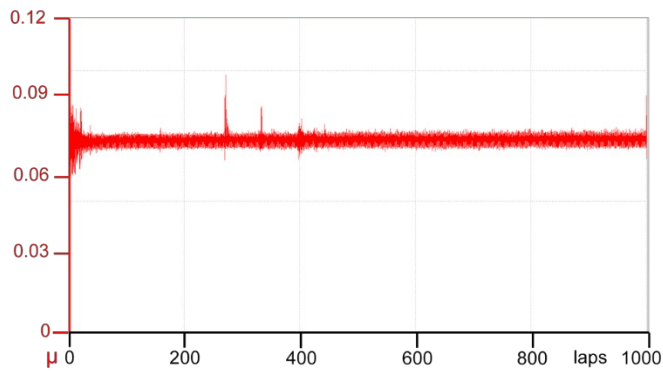
Category	Curve appearance	Possible disturbances sources
Better to discard		Minor vibration impact e.g. slight touch on the monitor or table

Better to
discard



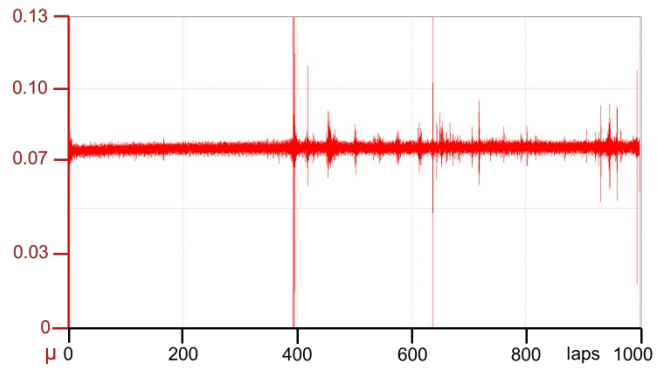
Small fibers in
the oil.
Dust on sample
surface or in the
oil;

Better to
discard



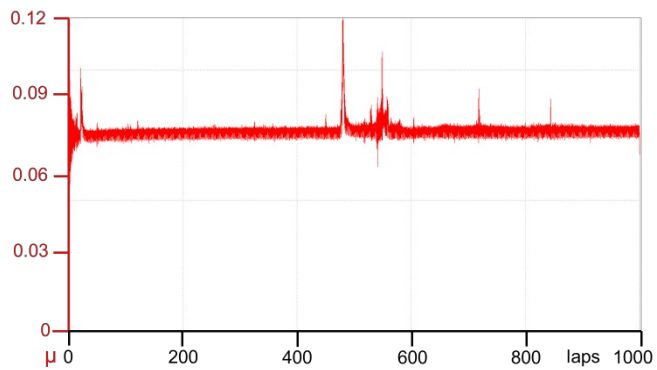
Small fibers in
the oil.
Dust on sample
surface or in the
oil;

Discard

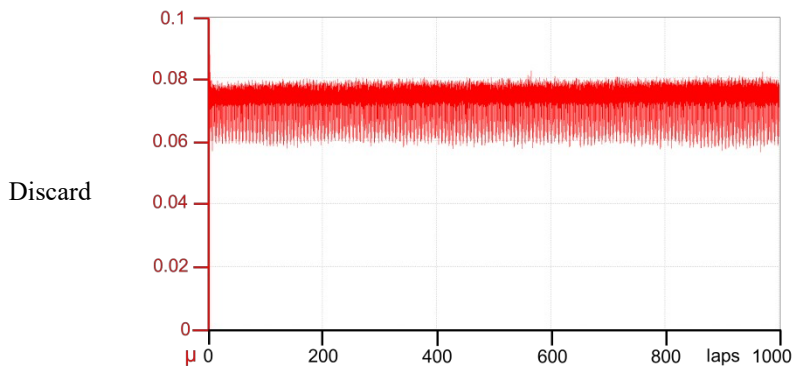
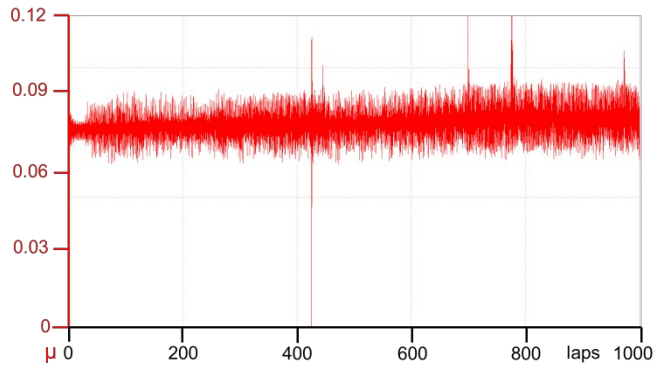
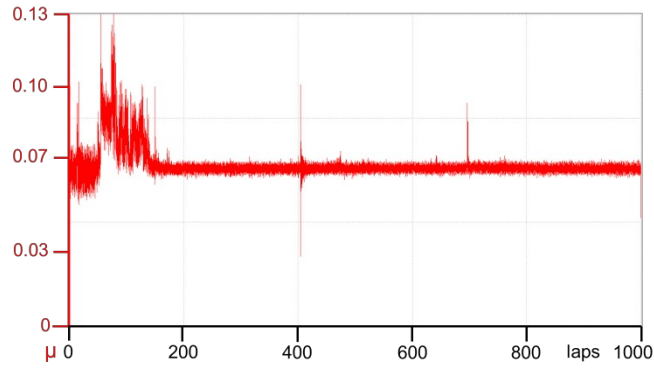


Major vibration
impact e.g.
passage of a
heavy forklift
nearby the
instrument

Discard



Trace of oil left
on the surfaces.
Large fibers laid
on the surfaces;



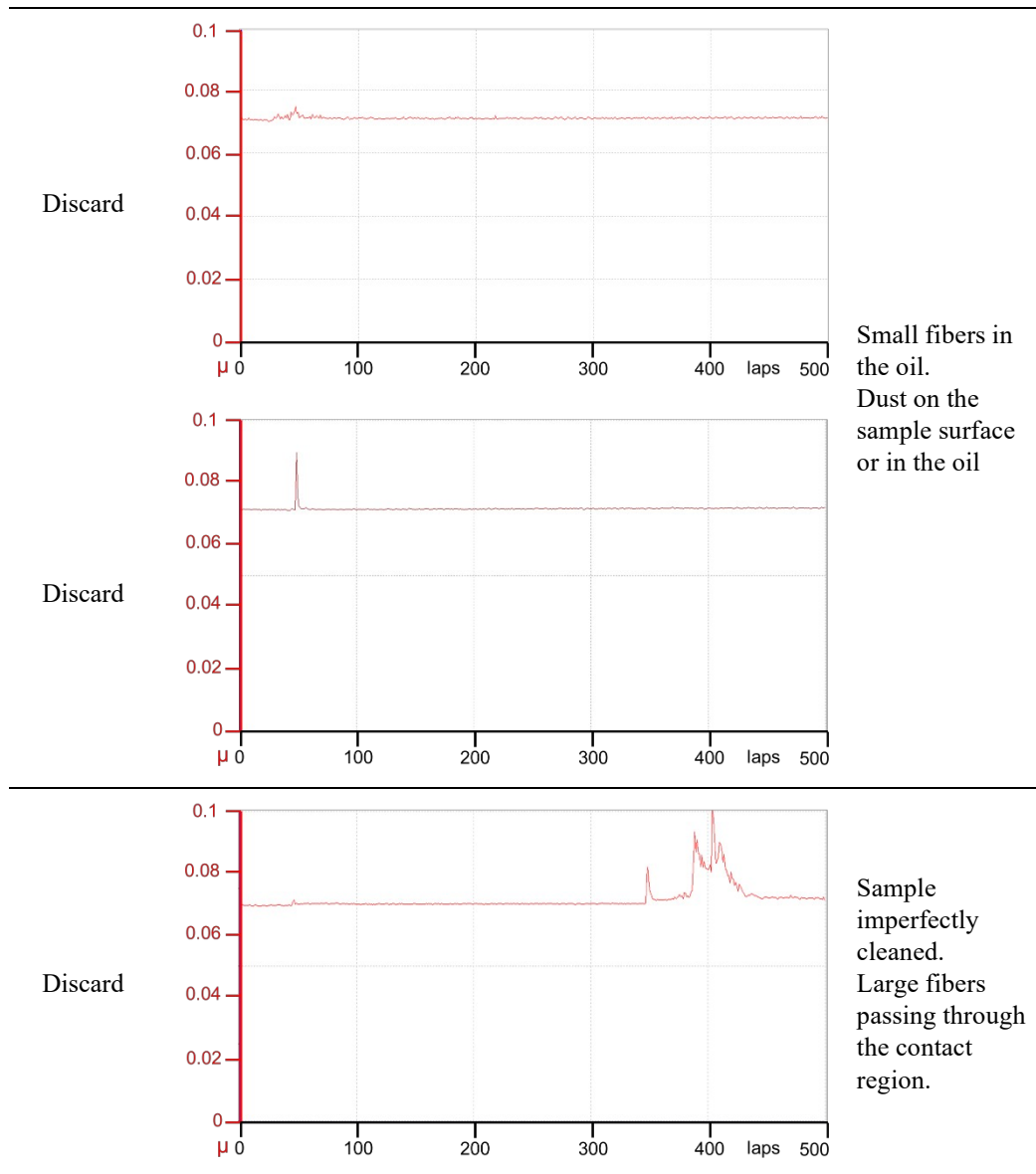
Discard

Severe damage or scratch on the sample surface

614
615
616

Table C1. Atlas of rejected tests in linear reciprocating mode tests.

Category	Curve appearance	Possible disturbance sources
Better to discard		<p>Small fibers in the oil. Dust on sample surface or in the oil</p>



617

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620 **References**

- 621 [1] P.J. Blau, Friction Science and Technology: From Concepts to Applications, II Ed, CRC Press, Boca Raton, 2008.
- 622 [2] P.J. Blau, Experimental Aspects of friction research on the macroscale, In: Fundamentals of Tribology and Bridging
- 623 the Gap Between the Macro- and Micro/Nanoscales (ed. B. Bushan), Springer-Science, Dordrecht, 2000, pp. 261-
- 624 278.
- 625 [3] H. Czichos, Technical Diagnostics: Principles, Methods, and Applications, NCSLI Measure 9(2) (2014) 32-40.
- 626 [4] A.Grubin, L. Vinogradova, Investigation of the contact of machine components, Central Scientific Research
- 627 Institute for Technology and Mechanical Engineering, Russia, 1949.
- 628 [5] D. Dowson, H. Higginson, Elastohydrodynamic lubrication, I Ed, Pergamon Press, England, 1966.

- 629 [6] B. Hamrock, D. Dowson, Isothermal Elastohydrodynamic Lubrication of Point Contacts - Part I, ASME Journal of
630 Lubrication Technology 98(2) (1976) 223-228.
- 631 [7] B. Bushan, Principle and Applications of tribology, II Ed., John Wiley & Sons, New York, 2013.
- 632 [8] X. Zhang, S. Kanapathipillai, T. Wu, Z. Peng, Friction behavior and friction mechanisms of rolling-sliding contact
633 in mixed EHL, Tribology International 114 (2017) 201-207.
- 634 [9] X. Zhang, Z. Li, J. Wang, Friction prediction of rolling-sliding contact in mixed EHL, Measurement 100 (2017)
635 262-269.
- 636 [10] Z. Fu, F. Guo, P. Wong, Friction–speed characteristics of elastohydrodynamically lubricated contacts with
637 anomalous film shapes, Journal of Engineering Tribology 226 (2012) 81-87.
- 638 [11] Z. Fu, P. Wong, F. Guo, Effect of Interfacial Properties on EHL Under Pure Sliding Conditions, Tribology Letters
639 49, 31-38 (2013).
- 640 [12] E. Ciulli, K. Stadler, T. Draexl, The influence of the slide-to-roll ratio on the friction coefficient and film thickness
641 of EHD point contacts under steady state and transient conditions, Tribology International 42 (2009) 526-534.
- 642 [13] M. Carli, K.J. Sharif, E. Ciulli, Thermal point contact EHL analysis of rolling/sliding contacts with experimental
643 comparison showing anomalous film shapes, Tribology International 42 (2009) 17–525
- 644 [14] S. Günsel, S. Korcek, M. Smeeth, The Elastohydrodynamic Friction and Film Forming Properties of Lubricant Base
645 Oils, Tribology Transactions 42:3 (1999) 559-569
- 646 [15] H. Nishikawa, M. Kaneta, Stribeck friction curve in point EHL contacts, Tribology Online 1 (2006) 1-4
- 647 [16] J. Hansen, M. Björling, R. Larsson, Topography transformations due to running-in of rolling-sliding non-conformal
648 contacts, Tribology International 144 (2020) 106-126.
- 649 [17] J. Hansen, M. Björling, R. Larsson, Mapping of the lubrication regimes in rough surface EHL contacts, Tribology
650 International 131 (2019) 637–651.
- 651 [18] M. Björling, R. Larsson, P. Marklund, E. Kassfeldt, Elastohydrodynamic lubrication friction mapping – the
652 influence of lubricant, roughness, speed, and slide-to-roll ratio, Proc. IMechE Part J: J. Engineering Tribology 225
653 (2015) 671-681.
- 654 [19] B. Vengudusamy, C. Enekes, R. Spallek, EHD friction properties of ISO VG 320 gear oils with smooth and rough
655 surfaces, Friction 8(1) (2020) 164–181.
- 656 [20] J. Guegan, A. Kadiric, A. Gabelli, H. Spikes, The Relationship Between Friction and Film Thickness in EHD Point
657 Contacts in the Presence of Longitudinal Roughness, Tribology Letters (2016) 64:33.
- 658 [21] H. Nishikawa, K. Handa, M. Kaneta, Behavior of EHL films in reciprocating motion, JSME International Journal 38
659 (1995) 558-567.
- 660 [22] Y. Han, J. Wang, W. Li, R. Ma, X. Jin, Oil film variation and surface damage in the process of reciprocation
661 oscillation transformation, Tribology International 140 (2019) 105-128.

- 662 [23] B. Vengudusamy, A. Grafl, F. Novotny-Farkas, Comparison of frictional properties of gear oils in boundary and
663 mixed lubricated rolling–sliding and pure sliding contacts, *Tribology International* 62 (2013) 100-109.
- 664 [24] V. Anghel, R.P. Glovnea, H.A. Spikes, Friction and Film-Forming Behaviour of Five Traction Fluids, *Journal of*
665 *Synthetic Lubrication* 21 (2004) 13-32.
- 666 [25] P. Anderson, Water-lubricated pin-on-disc tests with ceramics, *Wear* 154 (1992) 37-47.
- 667 [26] B. Podgornik, J. Vizintin, S. Jacobson, S. Hogmark, Tribological behaviour of WCyC coatings operating under
668 different lubrication regimes, *Surface and Coatings Technology* 177–178 (2004) 558–565.
- 669 [27] P.A. Grützmacher, A. Rosenkranz, S. Rammacher, C. Gachot, F. Mücklich, The influence of centrifugal forces on
670 friction and wear in rotational sliding, *Tribology International* 116 (2017) 256–263.
- 671 [28] M. Muller, G. Ostermayer, Measurements of partially lubricated contacts on different scales, *PAMM Proc. Appl.*
672 *Math. Mech* 17 (2017) 629-630.
- 673 [29] L.Bai, Y. Meng, Z. A. Khan, V. Zhang, The Synergetic Effects of Surface Texturing and MoDDP Additive Applied
674 to Ball- on- Disk Friction Subject to Both Flooded and Starved Lubrication Conditions, *Tribology Letters* (2017)
675 65:163
- 676 [30] A. Kovalchenko, O. Ajayi, A. Erdemir, The effect of laser surface texturing on transitions in lubrication regimes
677 during unidirectional sliding contact, *Tribology International* 38 (2005) 219-225.
- 678 [31] B.Y.C. So, E.E. Klaus, Viscosity-Pressure Correlation of Liquids, *ASLE Transactions* 23 (1980) 409-421.
- 679 [32] S.Y. Li, F. Guo, X.M. Li, C.L. Liu, Elastohydrodynamic Lubrication with Oil Droplets, *Tribology Letters* (2017)
680 65:162.
- 681 [33] B.J, Hamrock, S.R. Schmid, B.O. Jacobson, *Fundamental of Fluid Film Lubrication*, II ed., Marcel Dekker, New
682 York, 2004.
- 683 [34] ASTM G99-17, “Standard Test Method for Wear Testing with a Pin-on-Disk Apparatus”, ASTM International,
684 West Conshohocken, PA, 2017
- 685 [35] G. Stachowiak, A. Batchelor, *Engineering Tribology*, III ed., Butterworth-Heinemann, Oxford, 2005.
- 686 [36] T.E. Tallian, Y. P. Chiu , D. F. Huttenlocher, J. A. Kamenshine, L. B. Sibley, N. E. Sindlinger, Lubricant Films in
687 Rolling Contact of Rough Surfaces, *ASLE Transactions* 7 (1964) 109-126.
- 688 [37] S.Y. Poon, B.Eng., and D. J. Haines, frictional behaviour of lubricated Rolling-contact elements, *Proc Instn Mech*
689 *Engrs* 181 (1966-1967) 363-389
- 690 [38] S. Bair, W. O. Winer, Regimes of Traction in Concentrated Contact Lubrication, *ASME Transactions* 104 (1982)
691 382-386
- 692 [39] W. Wilson, S. Sheu, Effect of Inlet Shear Heating Due to Sliding on Elastohydrodynamic Film Thickness, *ASME*
693 *Journal of Lubrication Technology* 105 (1983) 187-188.
- 694 [40] M. Masjedi, M.M. Khonsari, On the effect of surface roughness in point-contact EHL: Formulas for film thickness
695 andasperity load. *Tribol Int* 82(A) (2015) 228–244.

- 696 [41] S. Odi-Owei, B.J. Roylance, Traction Behavior in Sliding ehl contacts, *Journal of Lubrication Technology* 100
697 (1978) 115-120.
- 698 [42] J. Wheeler, P. Vergne, N. Fillot, D. Philippon, On the relevance of analytical film thickness EHD equations for
699 isothermal point contacts: Qualitative or quantitative predictions?, *Friction* 4 (2016) 369-379.
- 700 [43] H. Moes, Communication. In *Proceedings of the Symposium on Elastohydrodynamic Lubrication*, London, 1965:
701 pp. 244–245.
- 702 [44] B. Hamrock, *Ball bearing lubrication: The elastohydrodynamics of elliptical contacts*, John Wiley & Sons, 1981.
- 703 [45] R. Larsson, O. Andersson, Lubricant thermal conductivity and heat capacity under high pressure, *Proc. Instn Mech*
704 *Engrs* 214 Part J (2000) 337-342.
- 705 [46] Y. Zhang, N. Biboulet, C.H. Venner, A.A. Lubrecht, Prediction of the Stribeck curve under full-film
706 Elastohydrodynamic Lubrication, (in press).
- 707 [47] P. Cann, E. Ioannides, B. Jacobsen, A. Lubrecht, The lambda ratio – a critical re-examination, *Wear* 175 (1994)
708 177-188.
- 709 [48] I. Krupka, P. Sperka, M. Hartl, Effect of surface roughness on lubricant film breakdown and transition from EHL to
710 mixed lubrication, *Tribology International* 100 (2016) 116-125.
- 711 [49] P. Sperka, I. Krupka, M. Hartl, Analytical Formula for the Ratio of Central to Minimum Film Thickness in a
712 Circular EHL Contact, *Lubricants* (2018) 6-80.
- 713 [50] A. Petrousevitch D. Kodnir, R. Salukvadze, D. Bakashvile, The investigation of oil film thickness in lubricated ball-
714 race rolling contact, *Wear* 19 (1972) 369-389.
- 715 [51] C.J. Hooke, The minimum film thickness in lubricated line contacts during a reversal of entrainment-general
716 solution and the development of a design chart, *Proc Instn Mech Engrs* 208 (1994) 53-64.
- 717