Accelerated Testing of Tribological Components - Uncertainties and Solutions

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Abstract

A fundamental requirement of any spacecraft mechanism development is to demonstrate the integrity of the selected lubricant system by means of an appropriate life test. As mechanism lifetimes are often long, and development times compressed some form of accelerated test procedure may be required for programmatic reasons though the strict tribological validity of such tests both for fluid-lubricants and for mechanisms employing solid lubricants or self-lubricating bearing is often a point of concern.

This paper discusses the current state of knowledge and limitations regarding accelerated testing, including the influences on lifetime, torque and material wear of the accelerated conditions. The uncertainties of presently available acceleration techniques and limitations of available data and methods are highlighted together with some potential future solutions.

Introduction

The challenges of successful accelerated life testing of spacecraft mechanisms and lubricants have been a recurring consideration for many years (for example [1], [2]). A simple review of the main classes of spacecraft mechanisms, their typical operating speeds and typical un-factored lifetimes is shown in Figure 1 and highlights the desirability of accelerated testing for missions where operational lifetimes (measured in terms of cycles or revs) are "long" but development times may be rather short. However accelerated tests may also be considered even for mechanisms with more modest life requirements for example where confidence pre-tests are needed (prior to qualification), for mechanisms which have a dwell (stationary period) or duty cycle with long stationary periods (e.g. refocus or calibration mechanisms), where a fixed launch date exists (e.g. "missions of opportunity" for comets/planetary exploration), where programmatic concerns dominate or where a compressed development cycle is being taken (e.g. low-cost missions).

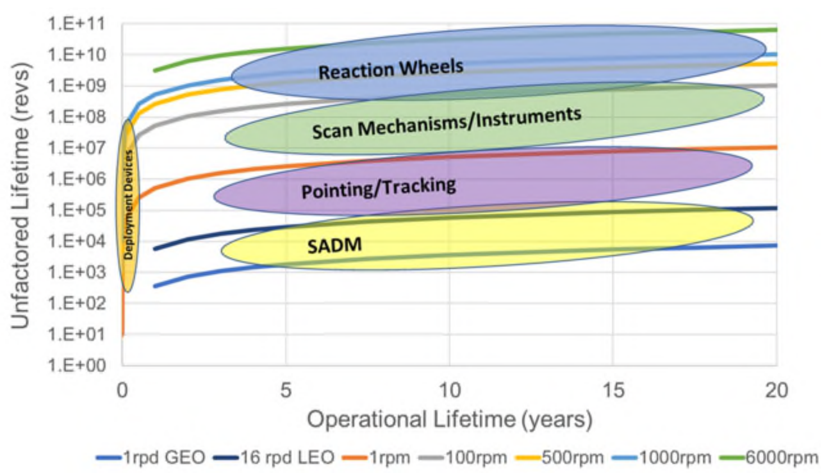

Figure 1. Typical unfactored lifetime requirements in revolutions for different mechanisms

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In 1995 Murray et al [3] highlighted the range of mechanisms for which accelerated testing would be desirable, the technical challenges in developing a valid accelerated test methodology, and a technology roadmap which might lead to greater understanding of the issues. In 1997 ESTL reviewed the topic and in proposing a methodology for a more tribologically rigorous accelerated test for oil or grease lubricated mechanisms [4] also highlighted some of the main challenges and uncertainties which could lead to a misleading test conclusion or invalid test.

More than 20 years on, the authors have experienced many accelerated tests of different types, and this paper seeks to highlight, both by review and presentation of new material, areas where additional understanding has been developed and some remaining technical needs in order to further reduce the risk that an accelerated test method may be adopted which could lead to an invalid or misleading test result – and in the worst case an undertest of life or performance.

Uncertainties of the Usual Approach to Accelerated Test - Fluid Lubricants

Accelerated testing of fluid-lubricated mechanisms is based on the notion that if the same lubrication regime can be maintained between the critical loaded contacts (e.g. ball and raceway) in the test as found in the application at nominal speed then the test will be tribologically valid.

In classical EHL, the lubrication regime is usually determined based on a Stribeck curve approach using an analysis as typically presented by Hamrock and Dowson [4]. Using this approach, a specific film thickness, λ (the ratio of minimum lubricating film thickness to composite surface roughness of the counterfaces) is calculated for the nominal (λ_n) and accelerated conditions (λ_a), the assumption being made that if accelerated test conditions can be established in which $\lambda_a \cong \lambda_n$ the lubrication regimes will be similar and therefore the risk of wear/rate of degradation of the lubricant or other tribological failures might be expected to be similar. By this approach a valid test can in principle be achieved if the increased speed of an accelerated test is compensated, by reduced viscosity achieved by increasing temperature, such that the same specific film thickness and lubrication regime might be obtained in the accelerated test as in the application at nominal operating conditions.

Whilst attractive in its simplicity it was pointed out in [5] that this approach can be inappropriate and flawed for many reasons. Since that time publications by many authors have contributed to the up-to-date status of these concerns for each phenomenon which is reviewed below.

Non-Newtonian Lubricant Behavior

Though the above approach assumes Newtonian lubricants, in fact for many lubricants, including widely used space oils (and their derived greases) there ceases to be a well-defined relationship between shear stress and velocity gradient (or in fact the fluid partially solidifies) at some stress/shear condition. For example, high pressure rheological experiments on the oils 815Z and 2001A [6,7] suggest reversible solidification changes at relatively modest pressures and room temperature as shown in Table 1 (and at lower pressures if temperature is reduced). These figures suggest that at a relatively modest mean Hertzian contact stress >~700 MPa (~1 GPa peak) visco-elastic solidification (rather than the straightforward increase in viscosity with pressure represented by the pressure-viscosity coefficient in the Barus equation) might start to become a factor for these lubricants causing deviations from the expected friction and film thickness behaviors.

Table 1. Onset of Visco-elastic and Elastic-plastic Solid Behavior in Space Oils

Oil	Pressure for Onset of Visco-elastic Solid Behavior	Pressure for Onset of Elastic-plastic Solid Behavior
815Z [6,7]	>~1.1 GPa	>1.6 GPa
2001A [6,7]	>~0.95 GPa	>~1.5 GPa

Whilst such changes are temporary (as the fluid passes though the ball/raceway contact for example) permanent viscosity loss has been reported [8] at mean contact stress > ~2 GPa (~3 GPa peak) for 815Z/Z25. Though a high contact stress for most space mechanism applications, this phenomenon should nevertheless be borne in mind.

Rheological changes as discussed are cited by some authors, notably Vergne, Bair et al (9,10) as the dominant reason for the common failure to correlate experimental and calculated specific film thickness. Indeed, the implication of their work is that the kind of “Quantitative EHL assessment” needed in order to define and demonstrate valid accelerated testing can only be achieved by a fully detailed assessment of the lubricant rheology. To our knowledge such an assessment is not yet available for the most common space lubricants, and the situation for greases seems still more complex – requiring both the rheological properties of the base oil, its flows and thickener behaviors to be well understood.

<u>Batch Variability and Degradation Effects</u>
For fluid lubricants, intra-batch property variations, usually assumed small, are not well understood or quantified in the open literature. Perhaps more significant, but also not well understood at component level, are the effects of degradation/deterioration due to the nature of the fluid lubricant (for example grease bleed (oil separation); evaporation of volatile constituents (e.g. additives which typically have a vapor pressure an order of magnitude higher than the base oil); oxidation or tribo-degradation (viscosity loss or even auto-catalysis)).

<u>Lubricant Availability</u>
The actual quantity of lubricant available at a given contact (e.g. ball/raceway) is dependent on various loss routes which deplete the local supply, primarily migration due to 1g orientation, surface energy driven creep and evaporation. Of these, only the latter is considered calculable, losses due to evaporation being typically estimated using a Langmuir equation approach. However, in general a lubricant may have components of different molecular weights and a range of volatilities. This could render an analysis using the Langmuir equation and published vapor pressure data too simplistic, the true situation being that the rate of mass loss, even for nominally single component fluids without additives is dependent on the thermal history of the lubricant. Indeed, there is evidence [11] that the Langmuir equation may predict higher loss rates than found in practice by approximately an order of magnitude at typical operating temperatures as shown in Figure 2 suggesting a high uncertainty on the quantity of oil to be initially added or remaining within a mechanism at the end of life.

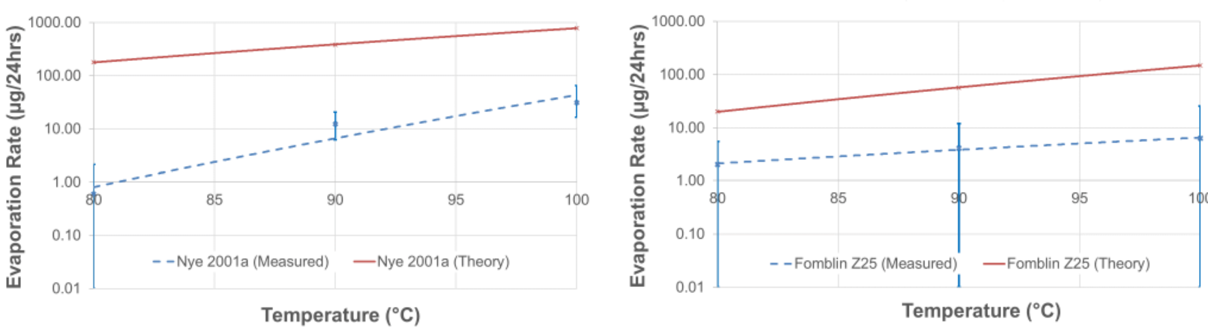

Figure 2. Comparison of experimentally measured evaporative loss rates for Nye 2001A (left) and Fomblin Z25 (right) with the predictions of Langmuir theory [11]

For cotton phenolic cages, which are usually vacuum impregnated over a period of hours to saturate the cotton fabric with the lubricant oil prior to use, there is evidence [12] that even “fully impregnated” cages may absorb a significant mass and a disproportionately high percentage of bearing free oil during subsequent storage as the phenolic matrix itself absorbs oil. The infiltration of oil into the phenolic matrix is a very low rate process, taking months, but as shown in Figure 3 the total mass absorbed can be relatively high and assuming this were to be absorbed entirely from the free oil quantity typically added to a bearing, upon assembly would be a high proportion (rapidly reaching 100%) suggesting that this phenomenon is perhaps an overlooked lubricant loss route which could cause the onset of a bearing with very minimal remaining free oil at the ball/raceway contacts (essentially starved/almost dry).

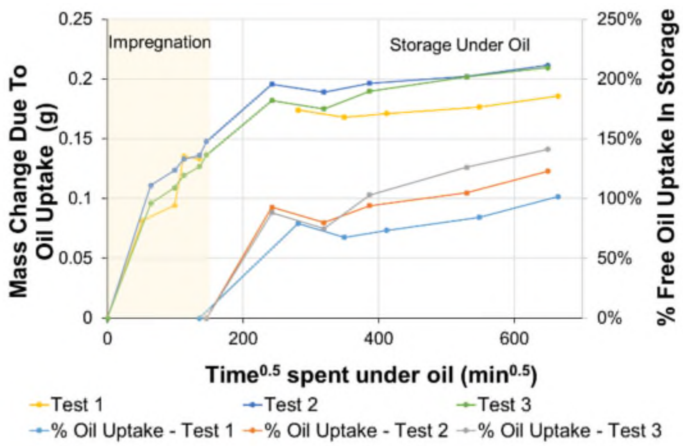

Figure 3. Oil mass uptake by cage and potential impact on proportion of free oil (Basis: 28 µl free oil available, storage period up to 9 months)

<u>Environment</u>

Lubricant flow (including wake effects) and lubricating film thickness behaviors are clearly highly dependent on environmental and local temperatures. Tribometer level tests in both sliding (e.g. Pin-on-Disc tribometer (PoD)) and rolling/pivoting contacts (e.g. Spiral Orbit Tribometer (SOT)) demonstrate that the tribo-life of common space lubricants is environmentally sensitive. SOT tests demonstrate that the assumption of parity of environmental effects between air or nitrogen and vacuum would result in a factor 5-10 over-estimate of the in vacuum lifetime achievable. Whilst the factor may be different in a specific real application, this demonstrates clearly that the potential error from the erroneous assumption of parity can be large.

For example, the lifetimes of the PFPE grease, Braycote 601EF, as measured in the SOT (in terms of normalized lifetime in orbits per rev) and in pure sliding pin-on-disc (PoD) testing (absolute life in revs) in various environments [13] is shown in Figure 4. This data, is reinforced by more recent work for PFPE and MAC oils [14] clearly shows that environment has a considerable impact on both measures of lifetime, implying that life test should be executed in vacuum to avoid under-test.

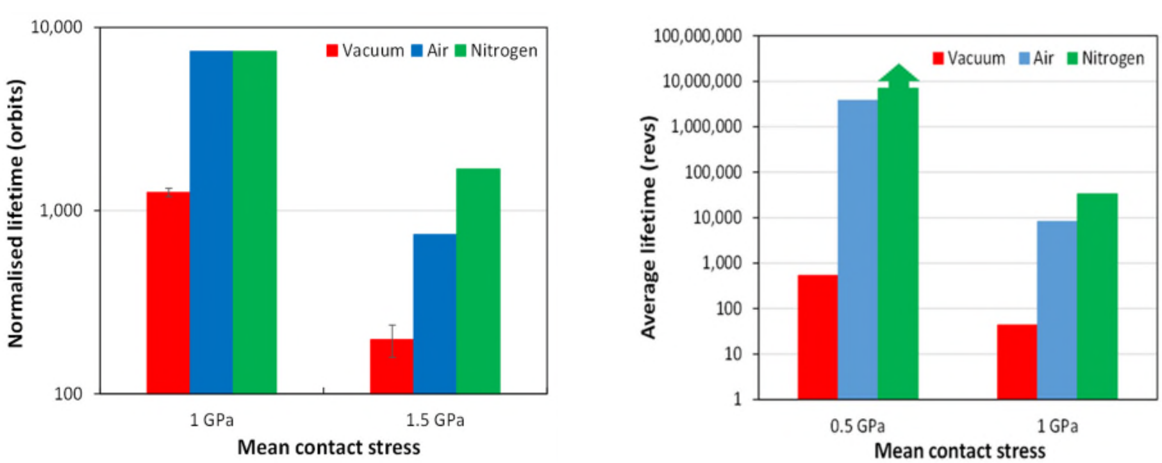

Figure 4. Lifetime of Braycote 601EF in various environments as assessed on SOT (left) and PoD (right) [13]

Incorrect Assessment of Lubricant Regime

Even in the absence of lubricant rheological effects such as solidification or shear thinning there are other challenges in the correct assessment of lubricant regime. The λ–ratio is usually assumed fixed and based on the specified or measured un-deformed ball/raceway surfaces prior to run-in/asperity modification or wear. In reality however, this will likely be incorrectly assessed. A higher λ may be applicable under bearing preload due to contact asperity flattening and λ–ratio may even be variable throughout test, increasing as surfaces run-in or decreasing due to thermally-driven contact angle changes or wear. It may only be possible to assess the impact of the test on the bearing surfaces by sophisticated pre- and post-test profilometry and analysis.

Changes During Operation

For ball bearings, the cage is rarely fully benign and its stability fundamentally impacts both torque performance and wear. For a valid accelerated test, the cage behavior needs to match that of the application at nominal operating conditions. However, changes to ball/cage and cage/land friction and cage geometry (e.g. due to wear) may have a considerable impact on subsequent cage stability or wear. Meaning that at the beginning of bearing life a cage could be stable, but develop instability due to oil absorption or changes in the surface friction or geometry of pockets or lands.

Non-Calculable Performances

Finally, there is a range of "non-calculable", even non-predictable performances, meaning behaviors that are not predictable by analysis, but apparently real and commonly observed phenomena. In this context we might list:

- Evidence for the regimes of starved/parched lubrication – where viscous losses are low, films theoretically of implausibly low thickness, but wear also low [15].
- Differences between fully-flooded and starved films in grease. When fully flooded greases tend to have a thickness higher than predicted based on oil properties alone (by a factor ~ 2), whereas when starved (and indeed there is evidence that greases may "self-starve") the film thickness resulting is ~ 50-70% of that of the base oil under the same speed, load and thermal conditions. (see [16] for example). Indeed, as greases age, both types of behavior may be observable.

Given the above, it was highlighted [5] that in some cases it may not be possible at the start of an accelerated test involving a fluid lubricant to be certain that the test would ultimately be valid, even if the target life was achieved. In the intervening years the proposed "tribologically valid" approach has been little used and given the new knowledge exemplified above now seems to require update since new knowledge renders this kind of accelerated test likely less, rather than more, representative.

Perhaps due to the above uncertainties, accelerated tests of fluid lubricated mechanisms are not common, and it seems that, where used, either the speed and the acceleration factor adopted remain relatively low, or there is an implicit acceptance and engineering judgement that the above effects may be second order or neglectable in the overall performance. Such judgements are not normally supported by experimental data and so the tests themselves remain, in the absence of appropriate experimental data on the fluid lubricant behaviors, to some greater or lesser extent, flawed.

Uncertainties Surrounding Accelerated Test – Solid Lubricants and Self-Lubricating Bearings

It is often assumed that, in contrast to the quite complex situation with oils and greases, the accelerated test of solid lubricated or self-lubricating bearings is much more likely to be valid.

Solid lubricated bearings usually rely on shear of anisotropic lamellar solids (e.g. MoS_2) or low shear strength Face Centered Cubic (FCC) metals (e.g. lead, silver, gold). This "primary lubricant" is supplemented often by a cage material which undergoes the double-transfer mechanism to transfer cage material via the surface of the balls to the raceway, a "secondary lubricant" phenomenon which may become increasingly important later in the bearing lifetime. Since they have no lubricant pre-applied to balls or races, self-lubricating bearings are entirely dependent on this kind of in-situ lubricant double-transfer from the beginning of life

The most common material for MoS_2-lubricated or self-lubricating bearings is PGM-HT (a PTFE glass-fiber, MoS_2 composite) though other transferring polymers and metals may be used also for specific applications or with alternative primary lubricants.

<u>Primary Solid Lubricant – MoS_2</u>

Where the primary solid lubricant is sputtered MoS_2 there is a considerable body of work which documents in general terms the expected performance at tribometer level (for example summarized in [17]) and while life data at component level in absolute terms may be scarce and variable due to different definitions of end-of-life or end-of-test, normalized lifetime plots show the trend to be expected.

In pure sliding, whereas there seems no relationship between the friction coefficient of any individual MoS_2 sample and the lifetime obtained, there is a strong relationship between sliding life and mean contact stress, P_m, in which life is stated [17] to be proportional to $P_m^{-3.8}$, a value also shown in the best-fit line for coatings on a range of substrate materials of different hardness [18] as shown in Figure 5. A similar relationship exists for data from rolling contact experiments using a SOT in which the life of MoS_2 applied to a 52100 steel ball running against a 52100 disc is found proportional to $P_m^{-3.2}$.

A limited number of tests have also been carried out on spur gears [19], and whilst the initial test campaign used low precision, relatively low hardness and non-hunting ratio gears, some tentative relationship of gear lifetime versus mean contract stress from MoS_2-lubricated gears has begun to emerge.

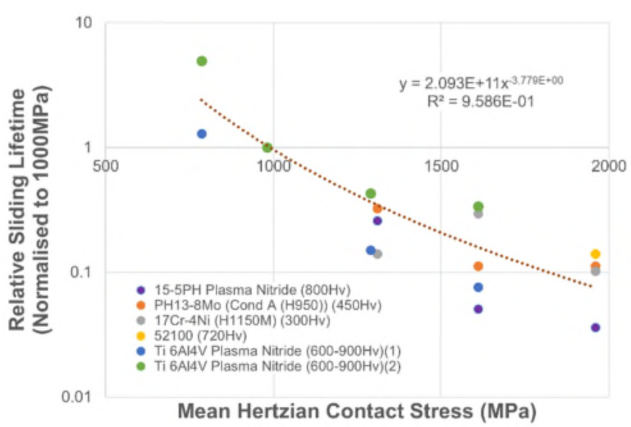

Figure 5. Normalized in vacuum lifetimes of sputtered MoS_2 in PoD test on various substrates [18]

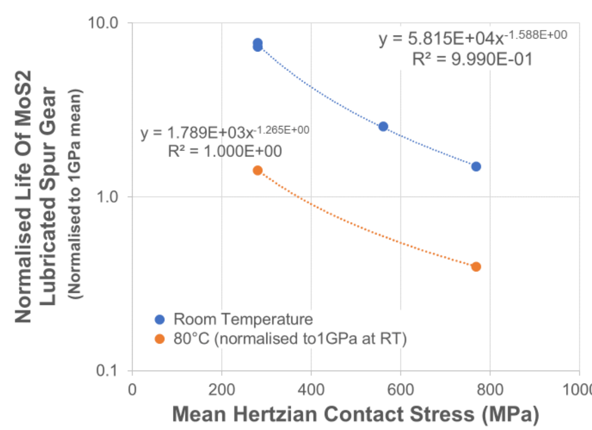

Figure 6. Normalized in vacuum lifetime of sputtered MoS_2 lubricated spur gears [19]

Whilst the spread of results in Figure 5 is due to different material combinations, if any regression curve approach is to be used to estimate life (for example Figure 6 for gear life), or as a basis of accelerated test justification, then the spread of results needs to be minimized. In general. this spread could come from two potential sources, the natural variability of the test (whether tribometer or gear test), or some supposed variability in the sputtered MoS_2 material itself. However, recent work suggests the spread of lifetime obtained from sputtered MoS_2 is broadly similar to that of PFPE or MAC lubricants (when compared in like-for-like test such as the SOT). The typical standard deviation on life of a batch of MoS_2 tested in the SOT is ~15%, a figure which compares well with the variability found in the same test for PFPE and MAC lubricants [20] (measured at 1.5 GPa mean stress and room temperature).

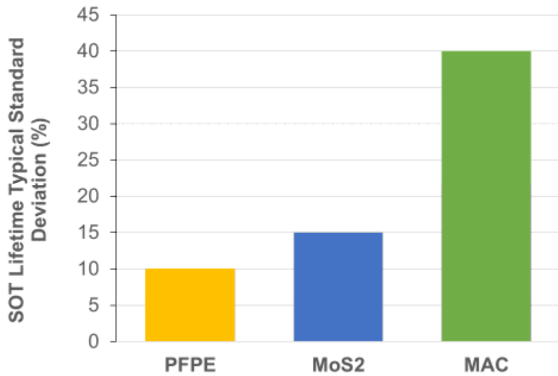

Figure 7. Comparison standard deviation of in-vacuum lifetime of 3 lubricants tested in SOT

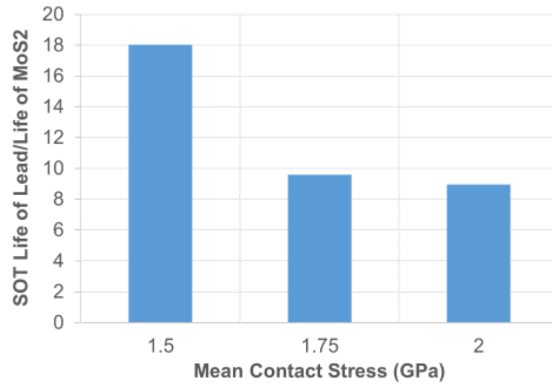

Figure 8. Relative lifetime of sputtered lead as compared to sputtered MoS_2 in vacuum (~1 µm) as function of contact stress

<u>Primary Solid Lubricant – Lead</u>

Since its high ductility permits transfer and re-transfer to different surfaces, like that of other low shear strength metals, the lifetime of lead in rolling contact tribometer tests (and in components) is very long. As it seems to persist as a very thin layer (deposited at around 1-µm thickness but functional even when the remaining film is only ~100 Å) [21]. In the SOT for example (with Pb applied to 52100 steel balls only running against 440C plates) Pb achieves in vacuum lifetimes at least an order of magnitude greater than MoS_2 and appears relatively insensitive to contact stress, however its corresponding friction coefficient is somewhat higher than that of MoS_2 in the same tests.

<u>Primary Solid Lubricant - Summary</u>

In general, primary solid lubricants, whether metallic or lamellar such as MoS_2 can be relatively well characterized by tribometer tests in vacuum, nitrogen and air. For example, in tribometers at least, the relationship between contact stress and life for MoS_2 seems well established, and the dependencies of life on temperature and substrate hardness have also been studied [22, 17]. Nevertheless, the relationship between life and sliding speed, though often stated to be low, is not well documented (although at very low speeds similar to slip velocities in low speed bearings a known friction/speed dependency exists).

<u>Secondary Lubricant – PGM-HT</u>

Regarding the secondary lubricant there are typically two principle accelerated test concerns, firstly that the material itself may display unexpected wear behaviors as a function of speed, stress or local temperature, and secondly that the cage may become unstable at speed, again causing different wear behavior in the accelerated test to the nominal speed application and thus invalidating an accelerated test.

ESTL carried out a program of PoD and SOT test work aimed at improving understanding of the behavior of PGM-HT material. As originally supplied, the material was found to be dimensionally unstable at elevated temperature, a problem subsequently overcome by introduction of a thermal conditioning pre-treatment [23]. In characterizing the tribological behavior of the thermally conditioned material ESTL carried out a number of PoD tests in vacuum [24] which showed that whilst in general higher friction tests correlated to higher wear rate, a wide range of frictional behaviors and wear rates could be expected (Figure 9). Interestingly also the legacy material RT Duroid 5813 (“Duroid”) seemed to have approximately a 4 times higher wear rate than found for the PGM-HT.

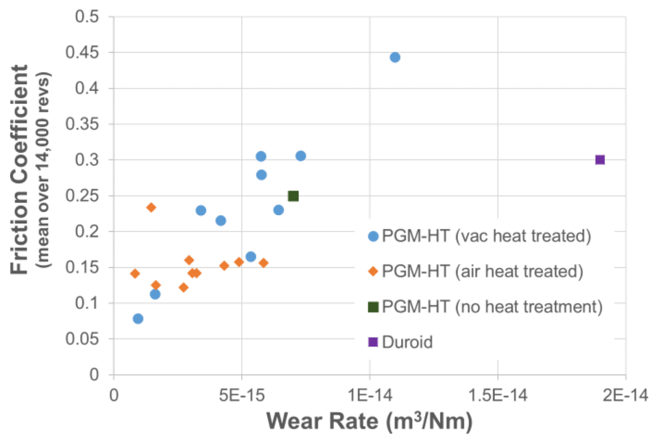

Figure 9. In vacuum PoD friction vs wear rate of PGM-HT and Duroid v Steel [24]
(Mean Hertzian stress 10 -16.2 MPa ,52100 steel disc, Ra < ~0.1 µm, sliding speed 0.01-0.1 m/s)

As a first experiment the SOT was used to characterize the initial formation of a transfer film by the double transfer method described above [25]. In this work a single ball loaded between plates was initially unlubricated but developed a transfer film of the PTFE/glass-fiber/MoS_2 due to its once per revolution impact with the guide plate and in subsequent rolling this material was also transferred from the ball to the flat plates (this was considered as analogous to a cage/land or cage/ball collisions within a ball bearing). Tests were run for only 50,000 ball orbits as a trial, but nevertheless the results added somewhat to understanding.

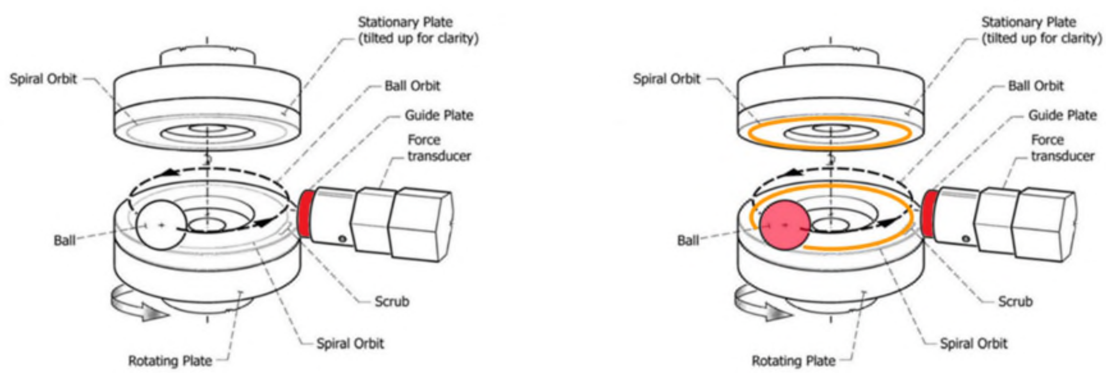

Figure 10. Schematic SOT with polymeric guide plate at start of test (left) and after material transfer from guide via ball surfaces to upper and lower plates (right).

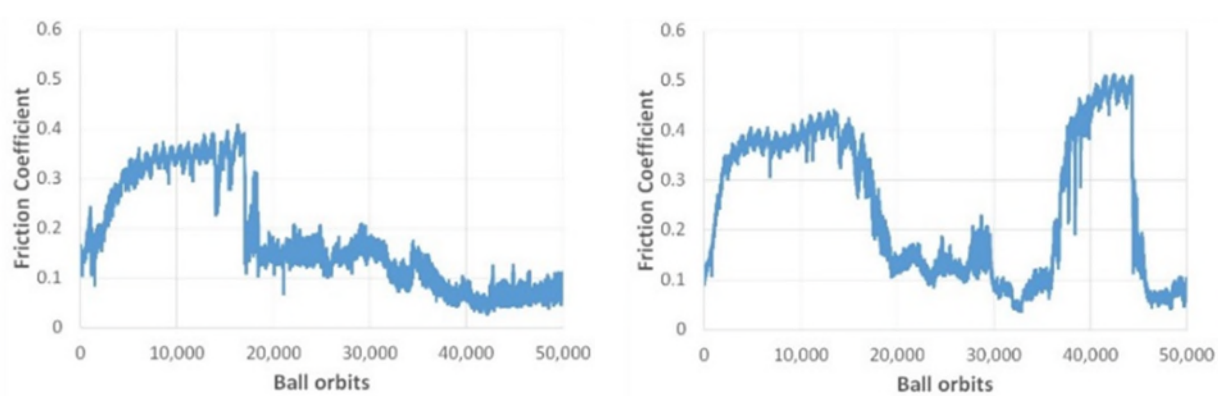

Figure 11. SOT Friction of PGM-HT (left) and Duroid (right)
(In vacuum, mean contact stress 600 MPa) [25]

Tests were carried out in vacuum, moist air and dry nitrogen on three materials: RT Duroid 5813 (legacy material), PGM-HT (current widely used cage material), and C29 [26] a potential future cage material (containing PTFE, carbon nano-fibers and MoS_2). Standard test conditions were 600 MPa mean contact stress (increased to 1000 MPa for high load tests), 50-rpm rotational speed (increased to 100 rpm for high speed tests), 21°C (increased to 50°C for high temperature tests).

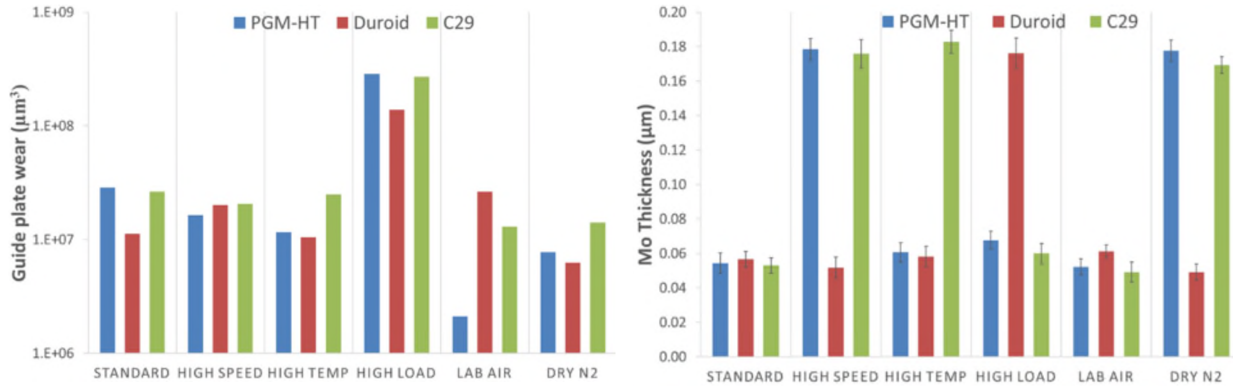

Figure 12. SOT guide plate wear (left) and XRF-measured residual Mo thickness (right) for PGM-HT, Duroid and C29 materials

Results, presented in [27], showed similar characteristics for all materials with variable friction, apparent rapid transitions from a low friction (µ~0.05-0.15) value to a higher value (µ~0.4 or higher). Subsequent measurements of Mo thickness (made by X-ray fluorescence spectrometry (XRF)) showed a two-valued population perhaps also suggesting rapid transitions from one state to another. Subsequent profilometry showed the transfer film to be very thin and discontinuous (with thickness of order 1-3 µm in the “best” cases and some small amount of metallic wear (~1 µm) in the “worst case”.

Given that transfer film formation was shown to be relatively rapid, but also that the film was observed to be quasi-stable with evidence of a high rate of variability of thickness and frictional behavior with time, the equilibrium wear rate of the material in sliding was also investigated. The main concerns were that accelerated test might lead to quite different wear regime, and perhaps difference in transferred material characteristics, especially when thermal effects are also considered. PoD tests were carried out in vacuum under load and speed conditions considered appropriate for the simulation of cage/ball and cage/land contacts for low-speed, nominal (~160 rpm) and accelerated bearing test conditions.

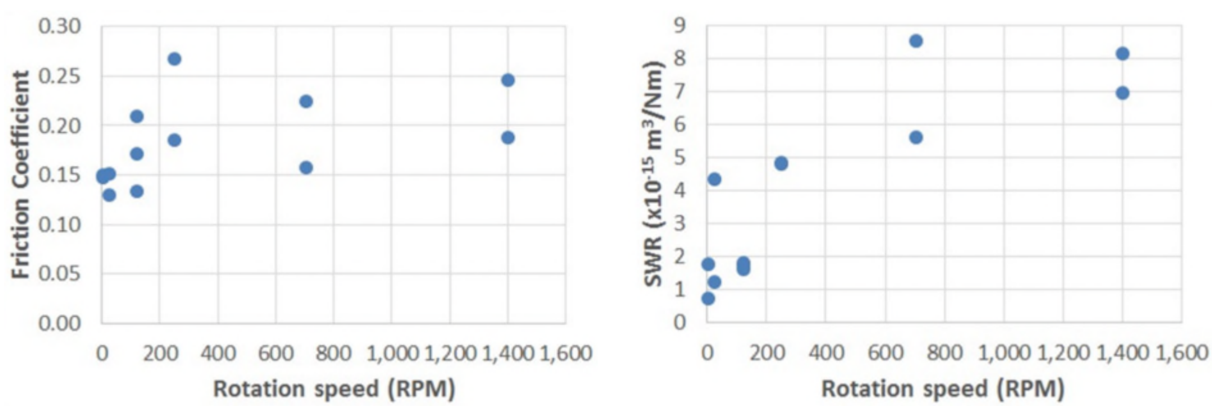

Figure 13. PoD Friction and Specific Wear Rate for PGM-HT v steel in vacuum as function of rotational speed (4 N, ~17 MPa mean)

As can be seen, for pin-on-disc tests there is quite a wide spread of friction and wear behavior and although mean friction seems approximately constant with speed, there may be some small increase in equilibrium wear rate with speed (but also a relatively high variability between tests).

When considering the impact of temperature (at temperatures typical of an accelerated test 21°C and 50°C), again the changes in tribological behavior were modest.

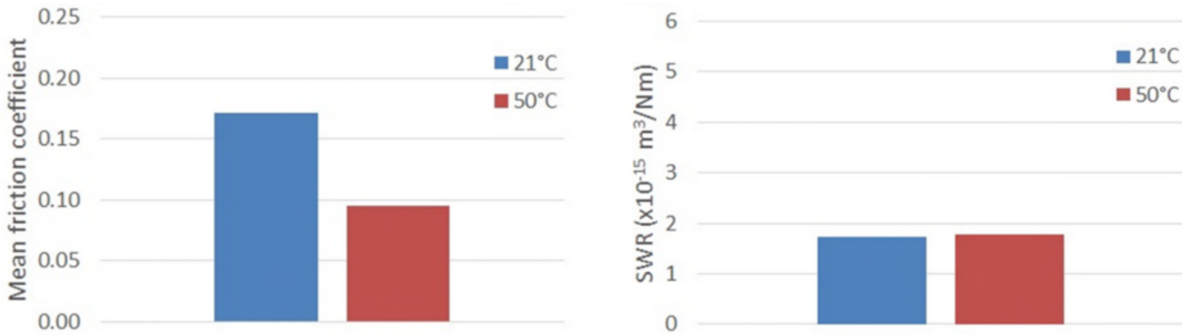

Figure 14. Effect of temperature on in vacuum friction coefficient and specific wear rate of PGM-HT v steel

When considering the impact of load, though there is clearly also some spread, results show that while the mean friction is not greatly impacted by increased load, the wear rate does increase with increased load, a result which substantiates the practice of accelerating test by increasing both load and speed when using this material.

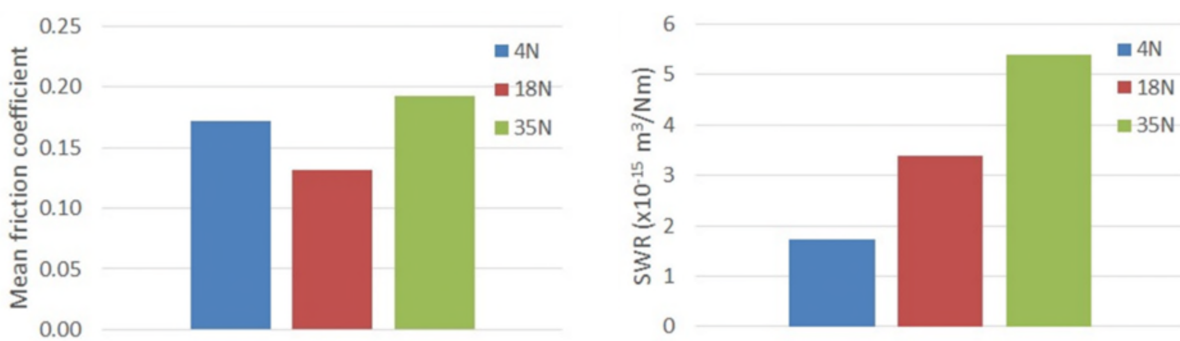

Figure 15. Effect of load on in vacuum friction and specific wear rate of PGM-HT v steel

Whilst it is acknowledged that small changes in friction can impact cage stability, the above data suggests that provided the bearing cage remains stable throughout test then acceleration of a test by increasing speed, load or both could be appropriate. This is because in general the friction and wear behaviors for PGM-HT have a quite wide spread and the effects of speed on wear, even over a quite wide speed range are approximately “within the noise” for this material.

The detection of cage instability then becomes a key enabler for the accelerated test envisaged. A recent test at ESTL [28] highlighted that a cage can be stable, quasi-stable or unstable at various times depending on complex factors (friction coefficients, instantaneous geometry, mass distribution, speed etc). Whilst validation of cage numerical models requires a detailed understanding of these phenomena, it may be argued that the success of an accelerated test needs only to demonstrate the absence of instability in the accelerated test if none is found at nominal speed operation.

The onset of cage instability will be detectable in the frequency domain (by the presence of the cage whirl frequency and other easily detectable frequency domain changes) or in the time domain by assessing the dependency of torque on speed (if unstable the mean torque may show a dependency on rotational speed-squared since torque becomes more directly related to cage energy rather than load-dependent (Coulombic) ball/raceway friction (essentially fixed independent of speed).

In principle therefore, the validation of an accelerated test of a solid or self-lubricating bearing seems achievable, on the basis of tribometer tests of primary and secondary lubricant behavior and the monitoring of cage stability throughout the test. However more proof, in the form of a substantial number of bearing tests of the validity of this kind of approach would be needed and the existing evidence for this is now explored.

For MoS_2 Lubricated Bearings

ESTL's Space Tribology Handbook [17] summarizes data for MoS_2 bearing lifetime from [29]. Whilst strictly relating to the legacy PTFE/Glass-fiber MoS_2 material RT Duroid 5813 (rather than PGM-HT), in this program the end-of-life of the MoS_2 film was defined by the onset of a mean torque equivalent to a friction coefficient of 0.3 between ball and raceway. This data, shown in Figure 16, therefore relates to the MoS_2 "film lifetime" within the bearing and not specifically to the cage material performance.

For Self-lubricating Bearings

Data taken from [30] has been used as survey of the lifetime of self-lubricating bearings using PTFE/glass-fiber/MoS_2 cages. It should be noted that the data from this survey contains both tests which completed end-of-life (according to some undefined criterion) and those for which the end of test may have been due to completion of the test target revolutions (with or without failure). Furthermore, this data relates to a range of different bearing sizes and types as well as to the legacy Duroid material and to the currently used PGM-HT.

If these two populations are combined, a power-law trend-line seems to provide an appropriate fit to the data. The fit could likely be improved if the data were expressed in terms of ball passes (rather than revolutions), nevertheless, this seems a reasonable "first approximation" guide to lifetime to be expected.

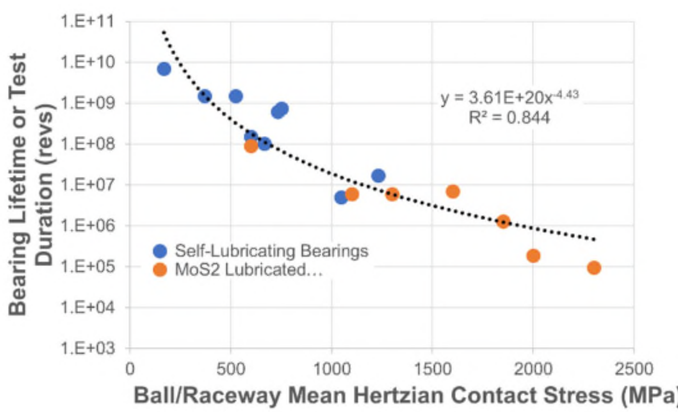

Figure 16. In-vacuum bearing life or test duration v ball/raceway Hertzian contact stress for MoS_2 films and self-lubricating bearings

In summary, even if the behavior of the lubricant is known at tribometer level, the extrapolation of such test results to bearing level is difficult due to differences in the definition of end-of-life torque for different applications. A moderately well-defined relationship exists between bearing life and contact stress as shown above. This kind of relationship has been used in the past to substantiate highly accelerated tests based on increasing speed AND load for low cost and mission of opportunity applications. The observation that products developed from an original mission of opportunity qualification carried out under accelerated load and speed testing have demonstrably met and exceeded life requirements in flight, with outlier units completing lifetime >3 billion revs [31] suggests the approach may not only have been pragmatic for a schedule requiring full development of spacecraft from contract to flight in ~27 months, but entirely appropriate.

In summary we could state that whilst at component level the relationship between contact stress and life is beginning to become clearer, to reduce the spread of results (in the above curve experimental results may be a factor 10 higher or lower than predicted by the curve) more controlled tests are needed.

New Analysis and Measurement Techniques

Given the number of complexities and considerations applicable to the accelerated testing of fluid and solid lubricated bearings, it is desirable to classify these considerations as likely to be of first or second order significance such that in test definition first order effects would be accurately addressed, whereas compromises could perhaps be made concerning full representativity for second order effects without invalidating the whole test. In this respect, the use of quite sophisticated test techniques at component level might inform mechanism/instrument level accelerated tests (in which inevitably the quality of data which can be obtained may be relatively low).

In order to develop further the understanding of fluid, solid or self-lubricating lubricated bearings an enhanced Advanced Bearing Test Rig (ABTR) facility has been developed (which traces its conceptual heritage to the work of Ward [32]. This new bearing test rig enables simultaneous direct measurement of lubricant film thickness, torque, preload, inner/outer ring electrical resistance and local temperature. The bearing system used is compliantly preloaded (allowing compliance to be a test variable) and the lubricant film thickness is determined by measurement of shaft motion from an on-axis high sensitivity capacitance displacement transducer.

Measurements derived from this kind of setup include:

- Lubricating film thickness or torque evolution vs. speed or test duration (in time or frequency domain)
- Improved bearing level understanding of solid-lubricated bearing performance – e.g. PVD film life, onset of transfer-film dominated behavior, effect of run-in, speed or environment on subsequent lubricant film behavior
- Effect of preload compliance on lubricant performance
- Cage stability (presence of cage whirl frequency in the FFT of torque)

First data, presented below, relates to a Type 7004 bearing with preload 48 N (~820/707 MPa peak Hertz stress at inner and outer raceway contacts), compliantly preloaded using a spring (compliance 1.7E-4 m/N). All testing is in vacuum (<1E-5 mbar). In Figure 17 we show the mean torque and film thickness evolution for a bearing lubricated only by transfer film formation from its PGM-HT cage (i.e. a self-lubricating bearing without MoS_2 applied to balls or races initially). From this it can be seen that after an initially high peak, some millions of revolutions are required for the torque and film thickness to become quasi-stable. If we follow the evolution of this behavior further into the operational life of the bearing, we see that there are periods of high and low torque which seem to be accompanied by increased PGM-HT film thickness. Statistically the two parameters film thickness and torque are closely and positively correlated with a coefficient of 0.83. The peak film thickness reached during run-in is around 2.5 µm (which correlates to the 1-3 µm transfer films found in the SOT trials above), however the mean film thickness seems to be around 0.5 µm when the bearing is stationary. It is also notable that the mean torque observed is a little higher than calculated, even assuming a friction coefficient, μ=0.4. This may be due to the relatively low preload used (hence perhaps not all balls are fully loaded as assumed in the bearing model), or to local changes in ball/raceway conformity due to the presence of the transfer film.

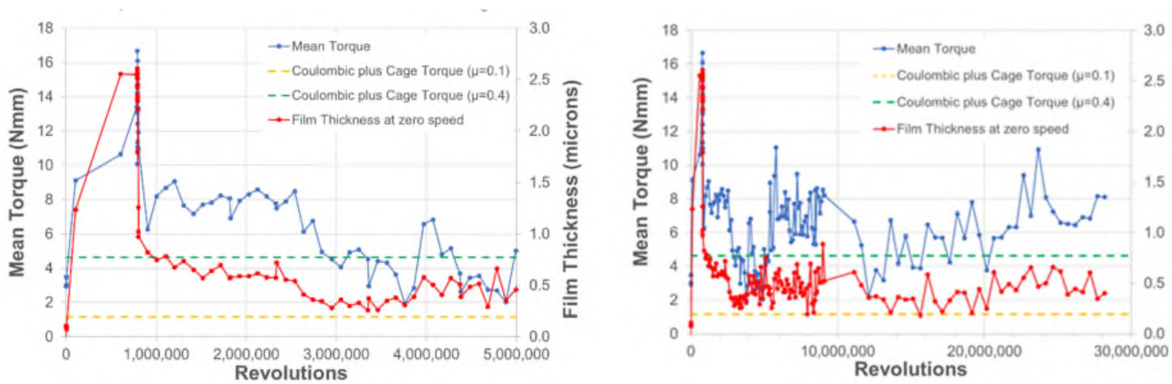

Figure 17. Type 7004 Bearing mean torque and PGM-HT film thickness vs. revolutions in vacuum

In future work, the facility will be used to characterize not only self-lubricating bearings, but also solid and fluid lubricated bearings using oils and greases. For appropriately characterized fluid lubricants, it is hoped that the lubricating film thickness measured will be correlated to predictions. For solid and self-lubricating bearings, it is hoped that the speed effects on cage wear and transfer film formation (including perhaps its viscoelasticity), will be understood so as to substantiate or not the accelerated test approach with increased speed and/or load.

Discussion

Accelerated testing of oil or grease lubricated bearings by increasing speed alone or speed and temperature remains risky (high probability of a misleading result). In addition to the main uncertainties regarding the rheological behavior of the fluid (especially at the high pressures within the contacts) and its flow, especially where grease is used, the local quantity of lubricant available seems difficult to predict. The conventional Langmuir theory seems to be predict higher evaporation rates for space oils than are measured in practice, yet fully saturated cages seem capable of absorbing a high proportion of the free oils within a bearing within their phenolic matrix. These competing factors and others mentioned above result in relatively high uncertainty concerning the lubrication regime in any space bearing either under nominal or accelerated test conditions. However, a series of carefully controlled tests in which the film thickness is monitored throughout test using the ABTR could improve understanding and even remove some of the more significant test uncertainties.

For solid and self-lubricating bearings, since the friction and wear behavior of both lubricant, transfer film and cage material are relatively well known and correlated to contact stress, then the possibility of accelerated test by speed, load or both seems feasible and indeed this method has already been used with some success. Nevertheless, the data newly available from the ABTR will allow the substantiation of claims concerning the validity of accelerated test methods and a ranking of the concerns listed above as of primary or secondary significance. This facility will also provide information of dynamic and static film thickness and may prove or dis-prove the assertions that behavior is substantially speed independent.

Conclusions

Despite considerable developments in understanding and characterization of fluid lubricants in recent years, accelerated test of fluid lubricated bearings remains highly likely to produce misleading results for the reasons discussed.

Life tests not carried out under an appropriately representative environment must be avoided as they will inevitably result in a tribological under-test. For example, results from Braycote 601EF suggest that tribo-lifetime of the grease performance is 1-3 orders of magnitude slower in air or nitrogen than in vacuum and similar results are documented for oils (and of course for solid lubricants).

Accelerated tests of solid or self-lubricating bearings are already supported by considerable tribometer data as summarized here. However, at bearing level, life data is much more scarce and there is a much larger spread of results. Better controlled and instrumented tests, such as those proposed for the ABTR are needed in order to reduce the spread so that existing regressions may be refined into still more credible and useful life estimation tools.

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