ACCELERATED TESTING OF LIQUID LUBRICATED SPACECRAFT MECHANISMS

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ABSTRACT

Where liquid lubricants are employed in spacecraft mechanisms, a fundamental requirement is to demonstrate the integrity of the selected lubricant system under all representative test conditions and durations. This necessitates a life test. As mechanism lifetimes are invariably long, some form of accelerated test procedure is therefore required; the validity of which is a continuing concern. This paper brings together theories describing the performance of liquid lubricants, observations from the literature on (elastohydrodynamic) liquid lubrication and accelerated testing, and ESTL's extensive experience of tribological testing to explain the basis of newly issued guidelines for the performance of accelerated mechanism tests for systems employing liquid lubricants (oils or greases). The guidelines suggest that for oils accelerated tests can be carried out with care, but that for greases, no tribologically valid accelerated test can be carried out. However taking a pragmatic view, in some specific cases, strictly non-valid tests may provide useful information, thus the methodology proposed for oils is extended to greases.

1. INTRODUCTION

The use of vacuum compatible oils and greases is widely and justifiably accepted by the space industry. Yet despite the long and proven heritage of such lubricants in flight, analysis of their function in engineering, that is non-isothermal, rough, elastic contacts remains at the boundary of what is possible with the current computing techniques. Experimentalists also find examination of the precise mechanisms of lubricant replenishment and film formation to be a challenging task. Given this "state of ignorance", it could be argued that the accelerated testing of contacts lubricated by liquid lubricants is fundamentally flawed, since there is no theoretical basis for the extrapolation of accelerated test conditions to the flight application (unlike solid lubricants, operating speed will have a dramatic and in many cases unquantified effect on lubricant rheology, film formation and replenishment).

From a purely tribological viewpoint, a valid accelerated test involving a liquid lubricant requires that the friction forces (and thus forces generated on other components of the system) and fluid film thicknesses (and therefore likelihood of wear or wear rate) within the system under test are representative of the conditions in flight. To ESTL's knowledge, the guidelines recently issued [Ref. 1] provide the first consolidation of the literature on these issues in a form readily accessible to mechanism design and test engineers. This paper represents a condensed version of the guidelines document.

2. BASIC LUBRICATION THEORIES

For fluid film lubricated contacts, there are three major modes of lubrication: **boundary, mixed and fully separating film (elastohydrodynamic /hydrodynamic)**. In the hydrodynamic regime the surfaces are completely separated by a fluid film and wear essentially becomes zero. In the boundary regime no separating lubricant film is generated and asperity interaction occurs. The adverse consequences of this are mitigated by reactive species on the surfaces (e.g. oxides, fatty acids etc.). In the intervening regime, lubrication may be 'mixed' or in non-conforming elements the regime of elasto-hydrodynamic lubrication occurs. The precise definitions of these modes will be considered later.

Classical hydrodynamic lubrication, in which conforming counterfaces are fully separated by a fluid film whose thickness depends only on its viscosity, the convergence of the gap between the counterfaces and their relative speed (as in tilting pad bearings) is rare in spacecraft.

A more common form of hydrodynamic lubrication in spacecraft is that which occurs in non-conforming machine elements, such as gears and rolling element bearings: it is termed elastohydrodynamic lubrication (EHL). In these component types the loads are high and act over relatively small areas giving rise to contact pressures orders of magnitude higher than in plain bearings. Such high pressures effect changes in the oil viscosity and cause elastic deformation of the contacting bodies. These effects combine to allow full fluid film separation. This mode is solved by a combination of Reynolds Equation describing the fluid behaviour and Hertzian theory which describes the elastic behaviour of the materials in contact. Results of extended computer

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solutions give equations for the minimum film thickness h_{min} in various contact types. Note in the examples below the heavy dependence of film thickness on speed but NOT on load:

Point Contacts (Archard and Cowking [Ref. 2])

$$h_{min} = 2.04(\alpha/(1+(2R_x/3R_y)))^{0.74} (\eta_o v)^{0.74} R'^{0.407} (E'/W')^{0.074}$$

(Eq. 1)

where $v = (v_1 + v_2)/2$, the entraining velocity, R' is the relative radius of curvature $(=1/((1/R_x) + (1/R_y)))$, R_x is the relative radius of curvature in the direction of motion, R_y is the equivalent parallel to the rolling axis, E' the reduced elastic modulus and W' is the load of the contact.

Line Contacts (Hamrock and Dowson [Ref. 3])

$$h_{min}/R_x = 2.65U^{0.7} G^{0.54} W^{-0.13} \quad \text{(Eq. 2)}$$

Elliptical Contacts (Hamrock and Dowson [Ref. 3]):

$$h_{min}/R_x = 3.63U^{0.68} G^{0.49} W^{-0.073} (1-e^{-0.68k}) \quad \text{(Eq. 3)}$$

Where
$U = \eta_o v/E'R_x$
$G = E'\alpha$
$W = F/E'R_x^2$
$k = a/b$ (ellipticity parameter)

η_o	Base viscosity of lubricant
E'	Reduced elastic modulus
F	Elliptic integral (= **1.5277+0.6023 ln(R_x/R_y)**)
α	Lubricant pressure-viscosity coefficient
a,b	Hertzian contact semi-minor and semi-major dimensions

The solutions to the elastohydrodynamic lubrication (EHL) problem contain implicit assumptions and idealisations which mean that whilst they provide a good starting point for a design, they cannot be straightforwardly applied to the problem of guaranteeing similarity between accelerated test and flight conditions in real liquid-lubricated contacts. These assumptions are:

<u>Smooth Surfaces in contact</u> Real surfaces are rough and deformable under the influence of the contact loads (prone to elastic or plastic deformations).

<u>Newtonian Lubricants in Use</u> Real lubricants are non-Newtonian in that they have a limited ability to sustain shear stress (above this limit films tend to collapse)..

<u>Isothermal Contacts</u> Viscous/shear heating modifies the temperature of the lubricant so that its properties at inlet to the contact are not those applicable at the bulk environmental temperature. A further complication is also that the properties of the lubricant within the contact itself are affected by heat input due to shearing.

<u>Plentiful Lubricant Supply</u> There is assumed to be a plentiful supply of liquid lubricant present in order to fill the contact zone which is not necessarily the case in real contacts.

Each of these limitations which will be discussed in more detail in the following sections.

3. SURFACE FINISH

For real, rough surfaces in contact of RMS surface roughnesses R_{rms1} and R_{rms2} , as the film thickness decreases, a point is reached where asperity interaction occurs. For any given value of minimum EHL film thickness, a value of the specific film thickness , or λ ratio, can be calculated. The λ ratio is defined as the ratio of the theoretical minimum EHL film thickness calculated for <u>ideally smooth surfaces</u>, $\mathbf{h_{min}}$, to the combined roughness (rms) of the real surfaces, $\mathbf{o_q}$, as follows:

$$\lambda = h_{min}/o_q \quad \text{(Eq. 4)}$$

$$o_q = (R_{rms1}^2 + R_{rms2}^2)^{1/2} \quad \text{(Eq. 5)}$$

The definition of lubrication regimes is usually made relative to measured surface roughnesses of the counterface components. For example, for hydrodynamic lubrication, λ should be greater than 3 (full separation with zero asperity contact). When λ is approximately equal to 3 (some references say 4), an undefined regime of "partial EHL" or Mixed Lubrication occurs in which there is some degree of asperity contact (the normal load is borne partially by the fluid film, partially by the asperities in contact. As λ decreases, the friction will increase until the load is supported entirely by the asperities covered by a thin boundary layer. The interface between mixed and boundary lubrication is diffuse, but is usually considered to be at a point where λ is about 0.8.

In summary, the usual definition of lubrication regimes based on specific film thickness is as follows:

$\lambda > 10$ Hydrodynamic Lubrication
(full film, pressure-viscosity and elasticity unimportant)

$10>\lambda\geq3$ Elastohydrodynamic Lubrication
(full fluid film modified by pressure-viscosity and elasticity)

$0.8<\lambda<3$ Partial EHL/Mixed Lubrication
(increasing asperity contact as λ reduces)

$\lambda\leq0.8$ Boundary Lubrication
(lubrication depends on adhesion of lubricant molecules to counterface surfaces)

The above definitions are frequently used in accelerated testing of liquid lubricated mechanisms on the basis that if two contacts are operated in the same regime, then the conventional wisdom is that the expected tribological behaviour will be equivalent.

Though the lambda ratio, permits convenient definition of the regimes for boundary, mixed and hydrodynamic lubrication, this classical definition of regimes can itself be flawed for the following reasons:

a) the degree of roughness of real surfaces itself modifies the film formation process so that the film thickness is also a function of the surface geometry.

b) the λ ratio is based on a measurement of the undeformed surfaces, the true operating regime cannot be determined from such measurements.

c) the surface roughness of lubricated components in contact changes during test as the counterface surfaces run-in. These changes will themselves modify the operating λ ratio from that assumed on the basis of a single pre-test value.

4. NON-NEWTONIAN LUBRICANT BEHAVIOUR

In heavily loaded contacts, the total shear deformation of the lubricant exceeds the Newtonian limit for the fluid. When this happens, Cann et. al. [Ref. 3] report that the stresses and lubricant flows in two perpendicular directions within the contact will be coupled to each other via the limited shear strength of the oil. In consequence, a pressure gradient in one direction will also influence the flow perpendicular to this (analogous to a dry friction effect in which the direction of the frictional force always counteracts the motion vector).

Most lubricants become glassy solids at pressures between 0.5 and 2 GPa (modest contact pressures), though the onset of this solidification is also a function of temperature. Solidification behaviour may lead to film (side) leakage, and dependent on the leak rate and contact duration, to film collapse. ESTL has contacted the manufacturers of a number of space-compatible oils and greases. However none has been able to provide data on solidification pressure v temperature behaviour of the lubricant, or on the shear strength of the solidified lubricant.

Cann et. al. also state that in contacts where there is pure rolling (no sliding) then the Newtonian oil behaviour may persist above the stress limits indicated (resulting in smoothing of the contact surface roughness and very high λ values). But if sliding exists, this "uses up" some of the strength of the oil in one direction making it easier for local pressure variations to compress oil into asperity valleys permitting asperity collisions (as peaks approach). This is supported by Jacobson [Ref. 4] who found that in pure squeeze and pure rolling, the necessary viscosity required to separate the surfaces avoiding metallic contact is much lower (8.5-14 times for fine surfaces and 32-240 for coarse surfaces in Jacobson's study) than if ANY sliding exists. In this regime, interaction between surface asperity behaviour and lubricant rheology determines lubricant state.

Outside this regime, the behaviour of lubricated asperities within the contact zone depends on surface roughness slope, and wavelength (i.e. the number of asperities in the Hertzian contact), rolling and sliding velocity components, the asperity-contact transit time (time for an asperity to pass through the contact) and mean oil film thickness. Collapse of a lubricant film is a time-dependent phenomenon, also related to asperity wavelength. Shorter wavelengths generate higher rates of collapse than do long wavelength asperities.

In consequence, counterface surface structure and kinematics can drastically change the λ value for zero asperity contact. Cann et al. report that λ values as high as 20 may give occasional metallic contact, and for well run-in surfaces, no metallic contact occurs down to values of λ as low as 0.3. *Based on this observation, the definition of different lubrication regimes above must be re-examined, since it is clear that the un-deformed surface roughness can seriously over- or under-estimate the surface roughness within the loaded contact zone. Smooth surfaces not only need proportionally thinner films for good lubrication, but also the λ ratio for different regimes can be decreased.*

5. NON-ISOTHERMAL CONTACTS

Thermal effects derive from two sources contact inlet zone viscous shear heating (due to a combination of pressure gradient and surface speed effects) and shear heating within the Hertzian contact itself.

The former source has the larger effect on film thickness as it modifies the initial viscosity of the lubricant in the inlet zone from the base viscosity η_o which is assumed usually to be at ambient temperature. Though negligible in very low speed applications, this modification can have a large effect on mean fluid film thickness and friction in the Hertzian contact zone for high speed systems.

In contrast, shear heating within the Hertzian contact itself contributes to friction force, but modifies film thickness relatively little.

A number of studies have investigated these effects and proposed means of accommodating thermal effects into the predicted film thickness. The usual way [Ref. 5] is to define a thermal film thickness correction factor, which is a function of the lubricant thermoviscous constant, thermal conductivity and sliding/rolling speeds of the surfaces.

6. LUBRICANT STARVATION

It is difficult to assess the extent to which a continuous film of liquid lubricant is present in a given real EHL contact. When the entry region to the contact is not sufficiently flooded (i.e. the lubricant meniscus tends toward the Hertzian contact dimension), then pressure generation within the film will be limited (since this region is critical to formation of the film within the parallel region), and the lubricated contact is said to be "starved". Cann et. al. state that EHD film thickness can drop to 50% of its fully flooded value under moderately starved conditions and that for grease lubrication, starvation is frequently observed.

Current understanding of grease and high-speed oil lubrication systems is that there is frequently a degree of starvation present in the contacts. However starvation can also occur in low speed applications with viscous lubricants , for example oils at low temperature [Ref. 6].

In a study of roller bearings, Wilson [Ref. 7] has shown that grease films in roller bearings are initially thicker than their corresponding base oil film (approximately 20-25% thicker, equivalent to a base-oil lubricated bearing with base oil viscosity 30-35% greater than that used) but that with time, the thickness of the grease films reduces to provide a semi-starved film thickness (a factor of approximately 1.7) lower than predicted for the base oil (and correspondingly low torque).

Analyses of starvation are usually based on a need to know the location of the inlet meniscus. However some authors have noted that this location fluctuates with time [Ref. 8], so that analyses which rely on this cannot be generally applicable. However a common assumption which permits progress is the so-called Zero-Reverse Flow (ZRF) or Prandtl-Hopkins inlet boundary condition. Under this assumption, the recirculation zone in the inlet meniscus is effectively eliminated (as there is assumed insufficient lubricant present to permit this). *This condition is understood [Ref. 9] to represent a "convenient and conservative design criterion for practical starved lubrication conditions".*

Dowson [Ref. 10] estimated that <u>for isothermal</u> contacts under the zero reverse flow assumption, the film thickness reduction factors ϕ_{ZRF} (which relate the ratio of minimum starved (under zero reverse flow assumption) to minimum fully flooded (non-starved) fluid film thicknesses) are $\phi_{ZRF}=0.71$ for pure rolling and $\phi_{ZRF}=0.46$ for pure sliding.

If the location of the inlet meniscus is moved further into the converging contact region so that it coincides with the boundary of the Hertzian contact zone itself, then the region of "Parched" lubrication, defined by Kingsbury [Ref. 11] is reached as shown in Figure 1. In this regime, there is in effect no inlet meniscus and the presence of the fluid film is dictated by its rheology (rather than ability to bond to counterfaces as in boundary lubrication). There is insufficient lubricant present to permit a fully-developed fluid film. Kingsbury states that film thickness is defined not by the speed , but by the thickness of the surviving film and rate of inflow and outflow from the film. This is believed to be the regime in which instrument bearings operate and offers the lowest bearing torque for given load and temperature. By implication it may be applied also to inertial devices in spacecraft (e.g. reaction or momentum wheels). In laboratory experiments, it has been observed that instrument bearings run successfully in this regime for several hours with only 80-200nm film deposited. In general, grease lubricated films tend to thin down to a speed-independent non-zero, but stable thickness value - indicating operation in this "parched" regime.

Guangteng et. al. [Ref. 12] showed experimentally that if the quantity of lubricant present was restricted, so that the film was initially semi-starved, film thickness did not increase with speed over the entire speed range (typical EHD behaviour). Instead a threshold speed was reached at which film thickness dropped markedly with the onset of parched lubrication. Above the threshold speed, a stable, but very thin (of order 10-30nm depending on lubricant viscosity) film was formed. *The implications of this observation are that parched lubrication can be reached either by deliberate supply of a very thin lubricant coating on the surfaces of the contact (as in reaction wheels), or by supplying a greater quantity of oil and increasing operational speed or viscosity (for example in an accelerated test, or in* *flight at low temperature). Clearly the onset of parched lubrication cannot be readily predicted, but will have a substantial influence on the results from any accelerated test programme in which the regime is inadvertently entered.*

Bearings operating in the parched regime seem able to operate for long periods with minimal friction and wear. Failures in this regime are said to be due to oxidation or polymerisation of the parched film (which has no possibility of unassisted replenishment), rather than by asperity interaction.

7. SUMMARY PERFORMANCE OF REAL CONTACTS WITH OILS AND GREASES

a) OILS

According to Cann et al., at low specific film thickness the use of the fundamental bulk properties of the fluid (viscosity, viscosity-pressure coefficients) and comparison of the measured roughness (outside the heavily loaded contact) with the theoretically calculated film thickness CANNOT reliably predict contact behaviour. However as specific film thickness increases so that the calculated oil film thickness is large compared to the surface roughness, and if no starvation effects are present, then the smooth surface approximation can then be used to predict film thickness.

In the intermediate regime, in which most spacecraft applications operate, if the lubricated surfaces are not mathematically smooth (and roughness heights are not negligible) compared to mean oil film thickness, then local pressure fluctuations caused by the asperities elastically deform the surfaces and as a result, the lubricated contact is sensitive to roughness structure, direction and oil quantity.

If the contact is starved, normal EHL film thickness calculations over-estimate the film thickness (as they assume fully flooded conditions) so that λ is lower than predicted, but on the other hand, local asperity pressure variations within the heavily loaded contacts will flatten the surfaces and create smoother surfaces (thus improving λ compared to an estimate based on assumed starvation and using the undeformed surface roughness).

For very smooth surfaces, elastic deformation may lead to almost total conformity within the contact and thus to a very high ratio of oil film thickness to roughness - even if the lubricant film is itself very thin.

b) GREASES

There remains a lack of detailed understanding of grease lubrication mechanisms. The nature of the film, the film formation and lubricant supply mechanisms and prediction of performance from bulk fundamental properties are still not fully understood.

In ball bearings, for example, the grease is often channelled by the passage of balls on the raceway. To this extent grease lubrication is a starvation (even parched lubrication) phenomenon, with re-supply dependent on transfer from the meniscus. For a given lubricant there will be a speed/viscosity starvation boundary (as speed increases there is no-longer a possibility of migration to re-supply the contact), above this speed, there will be no possibility of re-supply from the meniscus, and the contact will be fully starved.

Grease films have speed and time-dependent components arising from a residual film of gellant particles combined with a speed-dependent EHD component. Typical of grease-lubricated bearing performance is that a stable, but starved operation regime is reached after the first few minutes The initially thick lubricant film is rapidly depleted by starvation over a period of minutes to a stable, but very thin mean film thickness. The base-oil viscosity alone cannot be used as a basis for calculation of the residual film once starvation has occurred. According to Cann et al. " ***it is impossible to calculate, or use, the λ criterion for any grease-lubricated application***".

8. GUIDELINES FOR OIL-LUBRICATED ACCELERATED TESTING

Based on the literature findings above, ESTL has developed the following methodology for carrying out accelerated tests of spacecraft components and mechanisms lubricated by oils. The methodology is intended to permit similarity of flight worst case (or design) conditions and accelerated test conditions to be demonstrated *as far as practically possible with real engineering components* (but without monitoring film thickness by capacitance or resistance methods since this is impractical at the mechanism level). In order to provide for a conservative test philosophy, the assumption of starved, thermal lubrication is made, this will reduce film thickness predicted from the fully-flooded value, but clearly not so far as for parched lubrication. Another feature of the methodology is that a test cannot be guaranteed at its commencement to generate a valid result. The validity of a test is based on the extent to which the contact surfaces and system preload are modified during test, and this cannot be assessed until test completion. For this reason, it should be noted however that even testing in line with this methodology will not provide a guarantee of a valid test. The steps necessary in order to define an accelerated test using liquid lubrication are:

(i) Calculate Maximum Hertzian Contact Stresses, Contact Dimensions and Locations

On the assumption that contacts (e.g. ball/raceway or gear tooth) can be approximated as Hertzian, work out the Hertzian maximum contact stresses, Hertzian contact dimensions and location of the contact zone under the flight worst case (or design) loads and temperatures. Where multiple contacts are used (for example in a ball bearing) the most pessimistic view should be taken (e.g. the most heavily loaded ball/inner raceway contact). The calculation techniques for Hertzian stresses in different contacts are well documented in specialist texts.

(ii) Measure Surface Roughness of Both Counterfaces

At the contact locations determined in (i) above, measure representative initial (unloaded) rms surface roughnesses of both contacting components (R_{rms1} and R_{rms2}). Calculate the composite surface roughness, o_q according to Eq. 5 above.

(iii) Determine Operating Profiles for the Flight Application

In order to determine the critical points in the flight duty cycle, plot the product of entraining velocity (v) and lubricant dynamic viscosity at the relevant worst case operating temperature (η), $v\eta$ as a function of time. All points of gradient change in this plot are potential critical points to be subjected to analysis. However it may be possible to reduce the number of analysis points.

(iv) Calculate Lubricant Film Thickness Values for Flight Application

Work out a conventional (fully-flooded, isothermal) value of minimum film thickness, h_{min} (using Eq. 1-3 above or similar for the appropriate operating geometry) and corresponding specific film thickness, λ (using Eq. 4 and 5 above) at each of the points identified above for flight operating conditions. The λ value is based on the measured composite surface roughness (o_q) as calculated in (ii) above.

(v) Confirm Lubrication Regimes in Flight Application

Apply empirical corrections available for starvation and thermal effects for the given contact or motion type as exemplified below:

a) Calculate the fully flooded minimum thermal film thickness, h_{minT} based on either Eq. 6 or 7 below (note these are strictly applicable only for line contacts but are frequently used as first approximations for other contact types).

$$\phi_T = h_{minT}/h_{min} = 1/\{1+0.182L^{*0.548}\} \qquad \text{(Eq. 6)}$$

$$\phi_T = 1/\{1+0.1[(1+14.8\delta^{*0.83})L^{*0.64}\} \qquad \text{(Eq. 7)}$$

(due to Wilson and Shen [Ref. 5])

where $\delta^* = (v_1 - v_2)/(v_1 + v_2)$ the slide-roll ratio and $L^* = \beta v^2 \eta_o / k$

b) Calculate the starved thermal dimensionless inlet distance, ψ_{ST} based on the input data and equations from Gohar [Ref. 9] and Dowson [Ref. 10], $\psi_{SI} = 0.764$ for pure rolling ($\phi_{ZRF} = 0.71$) and $\psi_{SI} = 0.312$ ($\phi_{ZRF} = 0.46$) for pure sliding (using Eq. 8 below)

$$\psi_{ST} = \psi_{SI}(h_{min}/h_{minT})^{2/3} \qquad \text{(Eq. 8)}$$

c) Hence calculate the thermal starvation factor ϕ_{ST} or starved thermal minimum film thickness h_{minST}. (from Gohar [Ref. 9])

$$\phi_{ST} = (h_{minST} / h_{minT}) = 1-a^{-c} \qquad \text{(Eq. 9)}$$

Where:

$a = 4.6+1.15L^{*0.6}$

$b = 0.78/(1+0.001L^*)$

$c = (\psi_{ST})^b$

Confirm lubrication regimes using the ratio h_{minST} / o_q (according to the usual definitions given in Section 3 above). If the points of maximum and minimum velocity-viscosity product lie within a single lubrication regime, then only these two points need be assumed critical for future analysis. If however more than one regime is involved, then all points on the plot must be assumed critical to future analysis.

If the solidification pressure for the lubricant is below the maximum Hertzian contact pressure, then it can be assumed that some degree of additional starvation will occur.

(vi) Establish Accelerated Test Conditions

For the lubricant concerned, establish accelerated test conditions. These must achieve:

a) Acceptable test duration

AND

b) Similarity between calculated lubrication regimes for test and flight conditions at all critical points identified in 5. above.

The accelerated conditions are normally achievable by elevating the operating temperature and speed above flight conditions. Note that within any given lubrication regime, the maximum achievable acceleration factor for a given lubricant is defined by the gradient of the lubricant viscosity-temperature characteristic. Higher acceleration factors are achievable using lubricants with steeper viscosity- temperature curves, but clearly these lubricants provide greater variations in performance with temperature under flight conditions (and thus may represent unacceptable lubrication options).

Steel bearings are limited to a maximum useable temperature of around 100-110°C, above which steels begin to temper and cannot be used. In practice this temperature limitation may force acceptance of lower than planned acceleration factor.

Similarity of flight and accelerated test lubricant regimes at all critical points must be confirmed for a potentially valid test procedure.

(vii) Carry Out Accelerated Test

The accelerated test can then be carried out in accordance with the parameters defined above. Frictional forces or torques should be monitored throughout test, as should the temperature local to the contact (or if impractical, at some point on the mechanism body itself). *On completion of the test it is necessary to confirm that the applied preloads remain unchanged and to carry out profilometry on the running contact zones in order to characterise the post-test surfaces.*

(viii) Verify Test Conditions

Following test it is essential to confirm that the nominal lubrication regimes established initially have been maintained throughout. The post-test surface roughness values obtained must be used in order to re-calculate the specific film thicknesses (at the critical operating points identified above) and so to verify lubrication regimes are unchanged compared to initial values.

If the surface roughness of the contacts has been modified during test, then in principle the lubrication regime at one or more critical points may have changed. **In this case the test is rendered invalid**. A typical change would be that an initially rough surface becomes smoother during test, resulting in a change from boundary to hydrodynamic lubrication (and clearly resulting in an invalid test).

9. GUIDELINES FOR GREASE LUBRICATED TESTS

In the above review and discussion, it has been shown that, whereas for oils there is some reasonable correlation between predicted and experimentally measured fluid film thicknesses, for greases, there is no such agreement. In practical grease-lubricated contacts, starvation and parched lubrication are common and there is an additional complication in that neither the role of the grease thickener itself in bearing load, nor the mechanisms of lubricant re-supply in grease-lubricated systems are understood. In the light of these observations, since neither the flight fluid film thickness, nor the value under accelerated test conditions can be reliably predicted for greases, then accelerated testing using greases is not tribologically justifiable. ***Strictly therefore, accelerated mechanism tests using grease lubricants cannot be justified. Such tests are therefore not encouraged by ESTL. The preferred option for grease lubrication is always to simulate flight conditions in representative (non-accelerated) thermal vacuum testing.*** Only in this way can the grease starvation/re-supply effects applicable in flight be correctly represented and the test considered to be a true representation of the flight lubrication situation.

However, in ESTL's view there may be occasions when performance of a flawed accelerated test is preferential to no test at all. For example, we note that a large number of grease lubricated components have been successfully subjected to (flawed) accelerated test campaigns, and yet subsequently provided adequate operational lifetimes in flight. This is likely to be due to the accelerated test providing a greater degree of starvation than occurs in flight. So that, though not representative of the flight conditions, it could be argued that an accelerated test using grease provides a "worst case" test which would verify the integrity of the lubricant system for flight.

Where such compromise is necessary, the methodology outlined above for oils should be adopted with the use of the grease base oil viscosity (in place of the oil viscosity) for fluid film thickness calculations.

10. CONCLUSIONS

No evidence was found in the review that by maintaining both accelerated test and flight contact conditions within the same lubricating regime a valid accelerated test will result. Nevertheless this seems a reasonable approach for oils, given the current "state of ignorance".

Though the assumption is frequently made that full-film lubrication prevails, there is evidence that fluid lubrication is frequently starved or parched, leading to thinner than predicted fluid films. The extent of deviations from predictable behaviour is likely to be greater for greases than for oils. For this reason, accelerated tests with greases are considered strictly to be non-valid.

Given current film thickness prediction abilities, actual measurements are the only safe method of estimating true fluid film thickness during test, but there are practical difficulties in this approach.

In the interests of continued use of liquid-lubricated mechanisms in accelerated testing, future work should focus on obtaining experimental evidence that the approach outlined above is valid.

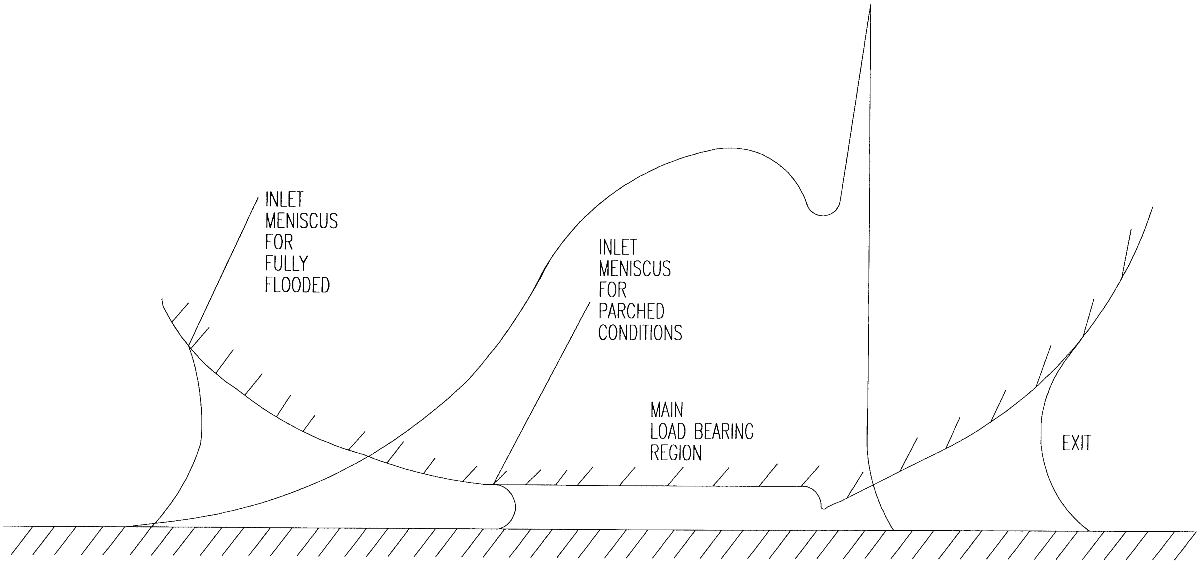

Figure 1 Typical contact showing classical pressure profile for flooded lubrication conditions. As starvation increases, the meniscus moves into the contact region, when parched, the meniscus is within the Hertzian contact zone itself.

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