LIFE PREDICTION OF FLUID LUBRICATED SPACE BEARINGS: A CASE STUDY

Massimo Palladino[1], Johan Murer[2], Sebastien Didierjean[2] , Lionel Gaillard[1]

[1]ESA-ESTEC, Keplerlaan 1, 2200 AG Noordwijk ZH (The Netherlands), Email: Massimo.Palladino@esa.int

[2]KDA, Kongsberg Defence and Aerospace (Norway), Email: Johan.A.Murer@Kongsberg.com

ABSTRACT

The literature is rather limited for what concerns prediction methods for life test performance of fluid lubricated bearings for space applications. Moreover, even if test data is available, it is most of the time associated to some specific tests with conditions difficult to be extrapolated to other applications.

The case study presented in this paper questions the performance of Braycote601EF and MaplubSH100b in the European Space Agency (ESA) Sentinel-3 project Solar Array Drive Mechanism (SADM) bearings. The main activity dealt with the analytical determination of an estimation of the life performance of the fluid lubricated bearings. It shows the difficulties to take into account the difference of preload and size of the bearing (number of balls) when trying to extrapolate previous test results.

Two approaches for lifetime assessment are discussed based on:

- Available test results database
- ESTL Spiral Orbit Tribometer results

Test results have been compared with the analytical evaluations. Test results and comparisons with the model are presented in this paper and show in most of the cases a good correlation.
It then pushes in favour of the build-up of a life test results database to strengthen the presented approaches.

1. INTRODUCTION

Even if solid lubrication appears in most cases very well adapted to SADM lubrication, mainly because of its low influence of temperature, non-evaporation and the fact that, life tests can be highly accelerated (resulting in cost savings), fluid lubrication is still interesting because of its good performance in air for ground tests, low sensitivity to damage in vibration and generally longer life. Secondly, fluid lubricants have a much more solid and wide spread heritage.

When fluid lubricant is selected as lubricant baseline in a project, two main questions need to be answered. The first one concerns the performance of the fluid lubricant with respect to the required number of bearing revolutions and lifetime duration in space with respect to evaporation. On this topic, some models based on the extrapolation of bearing tests performed at ESTL exist and are widely applied. However, these models appear to be very conservative, giving sometimes order of magnitudes of difference with actual test results. One of the main reasons for this is related to the fact that the reference bearing tested at ESTL were under starving conditions. Furthermore, the extrapolation from a bearing in starving conditions to another type of bearing filled with a given quantity of lubricant is not obvious. The second question is about ensuring that the bearing stays within the nominal lubrication regime when accelerating the life test. Some models to predict the maximum speed before leaving the actual lubrication regime have been developed mainly by ESTL.
In the present paper, some analytical calculations have been done to answer the two first questions. Then tests have been done to correlate with the analytical models results. All these steps are presented in the following.

2. MODELLING CONSIDERATIONS

2.1. Life analysis of fluid lubricated bearings

First step: Evaporation analysis

When a life assessment of a bearing has to be performed, it is important to take into account the lubricant losses due to the temperature profile and the mission duration. For the Sentinel-3 case presented

'14th European Space Mechanisms & Tribology Symposium – ESMATS 2011'
Constance, Germany, 28–30 September 2011

here, it has been considered that 12.5 years will be spent at a maximum temperature of 85 deg C. Models to assess this topic have been elaborated by ESTL. They are based on the Langmuyr equation. Taking into account the presence of a labyrinth seal, the following formula is used (ref. [1]):

$$Q = 0.0436 \frac{Pd\pi b\left(\frac{M}{T}\right)^{0.5}}{1 + 0.375\frac{L}{b}} 10^{-3} \qquad (1)$$

In this formula, Q represents the mass loss per second (in kg per second), T is the maximum temperature in Kelvin, M is the molecular weight of the lubricant (in grams per mole) and d, b and L are respectively the "flattened" labyrinth seal diameter, depth and length. P is the lubricant vapour pressure. By then multiplying this mass loss per the mission duration, the total mass loss is found. It is good practice to consider a semi-sealed system by having a zero length virtual labyrinth seal. The total amount of lubricant loss is then removed from the initial amount of fluid lubricant part of the grease lubricant to evaluate the quantity of lubricant that will be considered for the lubricant consumption/ degradation analysis. For our case, the lubricant losses are 0.08 mg for Braycote 601EF and 0.11 mg for MaplubSH100b by considering the actual labyrinth seal dimensions. This shows that Maplub tends to evaporate more than the Braycote.

Second step: Consumption/ Degradation analysis

For our case, a free volume of about 100 cubic millimetres is allowable to be filled with grease. Therefore, taking into account the molar mass and the fact that the grease fills about 5 % to 10 % of the free volume, there will be, as measured, 8 mg of Braycote601EF and 8 mg of MaplubSH100b. After taking out the lubricant losses due to evaporation, the total amount of lubricant is 7.92 mg of Braycote601EF and 7.89 mg of MaplubSH100b at the beginning of the consumption/ degradation phase.
The consumption/ degradation model is based on the Spiral Orbit Tribometer (SOT) tests results obtained at ESTL (ref.[2]). This type of test consists in having a ball between two plates (one fixed and the other one rotating) in such a way that the motion of the ball in an angular contact bearing is well simulated. Indeed the ball can roll and slide at the contrary of a pin-on-disc test where only sliding motions are performed.

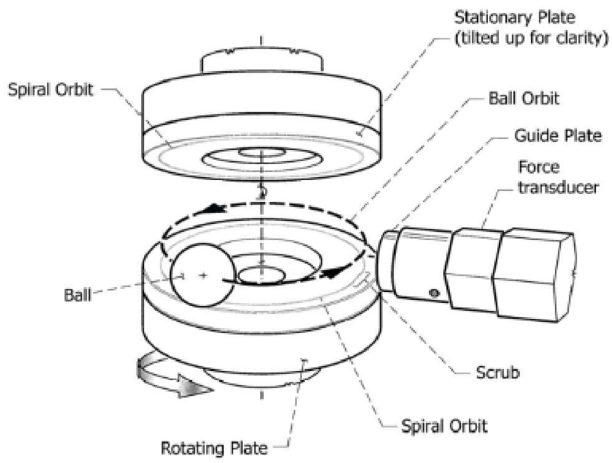

Figure 1: Spiral Orbit Tribometer test

In this SOT test, 50 micrograms of lubricant are used and the number of orbits the ball does until failure occurs (huge increase in the friction) is recorded. The contact stress in the test set-up is 1.5 GPa. The SOT test is performed under vacuum. For the analysed lubricants, the SOT test has provided the following results (ref.[2]):

- Braycote601EF: 200 orbits/microgram of lubricant
- MaplubSH100b:1200 orbits/microgram of lubricant

The consumption/ degradation analysis consists in the application of the following methodology:

SOT tests have demonstrated that the life of fluid lubricants is sensitive to contact stress. An empirical relationship between lubricant life (obtained by SOT) and contact stress is:

$$Life = K\exp\left(-3.35 p_m\right) \qquad (2)$$

Life is the number of ball orbits until failure; K is a constant and p_m is the mean contact stress.

By knowing the mean contact stress for the Sentinel-3 application (800 MPa), it is therefore possible to evaluate the number of ball orbits per microgram of lubricant. In this case, the number of ball orbits per microgram of lubricant is:

- Braycote601EF: 2087 orbits/microgram of lubricant
- MaplubSH100b: 12520 orbits/microgram of lubricant

The number of ball passes for the Sentinel-3 bearing is an important parameter to evaluate the consumption/ degradation of the lubricant:

$$Nbp = \frac{N_b}{2}\left(1-\frac{b_d}{p_d}\cos(\theta)\right) \qquad (3)$$

Nbp is the number of ball passes, N_b is the number of balls in the bearing, b_d is the ball diameter and p_d is the pitch diameter.

For the Sentinel-3 case, the number of ball passes is 3.85. Knowing the number of ball passes and the fact that the SOT test set-up is such that for one ball orbit the rotating plate does two revolutions, it is possible to estimate the number of bearing revolutions until failure of the lubricant. This is done with the following formula:

$$Nrevs = \frac{2Life}{Nbp}quantity \qquad (4)$$

Nrevs is the estimated number of bearing revolutions until failure, Life is the number of ball orbits per microgram of lubricant found in the SOT, Nbp is the number of ball passes in the bearing and "quantity" is the amount of lubricant after having taken out the evaporated amount of lubricant.
The following results are found:

- Braycote601EF: 8.6 million revolutions
- MaplubSH100b: 51.3 million revolutions

The model presented here has been originally used to predict the allowable number of revolutions at motor bearing level for the ESA GALILEO IOV SADM. The model predicted 7.2 million revolutions (mainly due to a much higher grease quantity available in the bearing (141 mg of Braycote601EF) despite higher preload and contact stress: 1.7 GPa). The model has shown a good correlation with the test results (ref.[2]). This is the reason why this model has been reused for Sentinel-3.

Alternative degradation analysis based on bearing tests done at ESTL

Alternatively, an analysis based on bearing tests done at ESTL (ref.[1]) has been performed for the Braycote601EF. The methodology is explained hereafter:

The starting point is that a ED20 angular contact bearing (back-to-back pair) with 40 N preload and 637 MPa contact stress, filled with 1% of free volume has demonstrated 10 million revolutions lifetime with Braycote601EF. The number of ball passes for this bearing is 6. Therefore, the total number of ball passes is 60 million. By applying the equation (2), it is possible to estimate the theoretically achievable total number of ball passes in a bearing with 800 MPa contact stress (Sentinel-3 bearing). This gives a total number of ball passes of about 34 million. It is known that for the Sentinel-3 bearing, the number of ball passes is 3.85. Therefore, the expected lifetime is about 9 million bearing revolutions. It is interesting to notice that this approach gives, in the end, similar results than the approach presented in the previous paragraph. This gives confidence in the prediction model.

2.2. Accelerated life test analysis

The analysis has indicated the following results:

- The Braycote601EF, initially considered as the baseline lubricant, will not meet the Sentinel-3 lifetime requirements of 140 million bearing revolutions
- The MaplubSH100b will have a rotation lifetime performance of at least 10 times the Braycote601EF, but will still not necessarily meet the requirements.

Due to these negative analytical/SOT predictions, it was decided that a specific bearing test would be needed to prove that a fluid lubricant could still be qualified for Sentinel3.

The actuator manufacturer, based on previous experience, was confident that a fluid lubricated solution (like Braycote601EF) would be compliant with the required lifetime

Analytical developments

In this paragraph, some analytical developments are presented to evaluate the maximum possible speed during the accelerated life test of the Sentinel-3 bearings.
The formulas for this assessment have been derived by ESTL and are given in ref. [4].
The dimensionless minimum film thickness is evaluated with the following formula:

$$H_{\min} = 3.63\frac{U^{0.68}G^{0.49}\left(1-e^{-0.68k}\right)}{W^{0.073}} \qquad (5)$$

U, G, W and k are dimensionless parameters mainly dependant on the bearing geometry, the bearing speed, the bearing contact stress, the bearing materials and the lubricant viscosity. In the analysis, the base oil viscosity is considered (base oils are Braycote815Z for

Braycote601EF grease and Nye2001a for MaplubSH100b grease).
The thermal effect based on viscous shear heating (local heating) will result in a reduction of the minimum film thickness. This effect is taken into account in the model.
The starvation effect consists in a decrease of the pressure in the film when the lubricant meniscus tends towards the Hertzian contact dimension. This effect results in a decrease of the minimum film thickness. This effect is taken into account as well.
Knowing the surface roughness, the specific film thickness λ can be computed. It is considered that the following lubrication regimes can be defined:

$\lambda < 1$: boundary lubrication regime; $\lambda > 4$: hydrodynamic regime; $1 < \lambda < 4$: mixed regime

By analysis, the following limitations have been found:

* At -20 deg C and nominal speed, with Braycote601EF, the lubrication regime is mixed (specific film thickness between 1 and 4). In order to accelerate as much as possible, the cold case was agreed to be tested at 20 deg C. By analysis, the mixed regime will end at 20 deg C at a speed of 1320 RPM. It was therefore agreed to test at a maximum speed of 1000 RPM at 20 deg C to simulate the cold case.

* At 85 deg C and nominal speed, with Braycote601EF, the lubrication regime is boundary (specific film thickness below 1). By analysis, the boundary regime ends at a speed of 770 RPM. It was agreed to accelerate the test up to max 500 RPM.

The analysis estimated the test duration at about 105 days for Braycote.

* At -20 deg C and nominal speed, with MaplubSH100b, the lubrication regime is mixed. In order to accelerate as much as possible, it was agreed also here to test the cold case at 20 deg C. By analysis, the mixed regime will end at a speed of 2383 RPM. It was therefore agreed to test at a maximum speed of 2000 RPM at 20 deg C to simulate the cold case.

* At 85 deg C and nominal speed, with MaplubSH100b, the lubrication regime is boundary. By analysis, the boundary regime ends at a speed of 4400 RPM. It was therefore agreed to accelerate the test up to a speed of 2000 RPM.

The analysis estimated the test duration at about 31days for Maplub

Experimental Stribeck curves for correlation

When doing the analysis to evaluate the speed to be used during an accelerated life test, it is always good practice to also perform a measurement of the Stribeck curves by increasing the speed up to a maximum speed (to cover all the speed range of interest) and then coming down to the minimum speed. The following figure shows the measurements obtained at 20 deg C (simulating the cold case) for the Nye2001a (base oil of the SH100b grease):

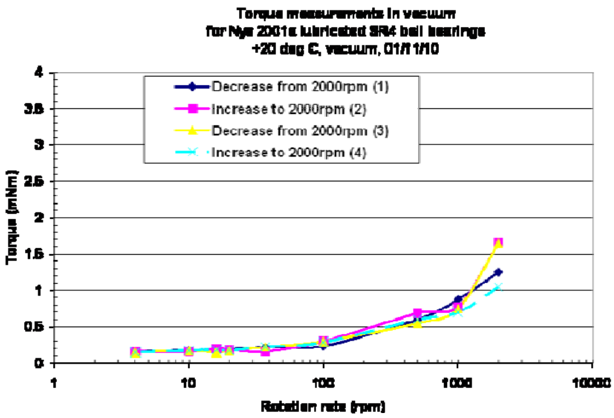

Figure 1: Stribeck curves measured at 20 deg C (cold case)

The Stribeck curve measured at minimum temperature (-20 deg C) is given hereafter:

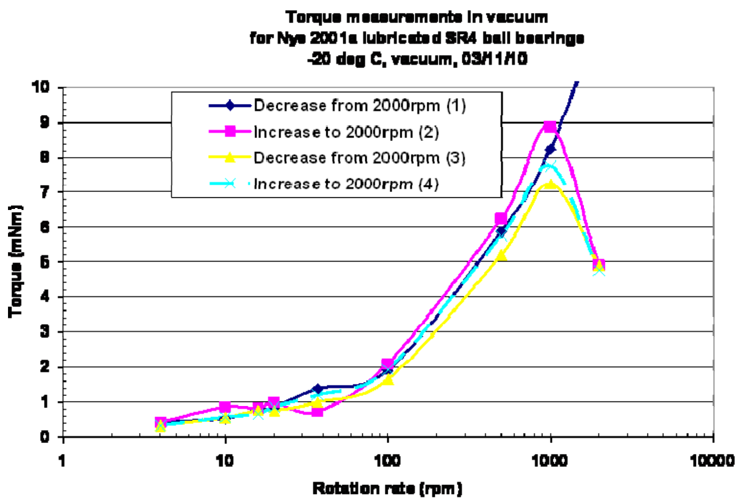

Figure 2: Stribeck curves measured at -20 deg C (minimum temperature)

The nominal speed is 16.5 RPM. By comparing the torque at -20 deg C and 16.5 RPM (Figure 2) with the torque at 20 deg C and 2000 RPM (Figure 1), it is deduced that a test at 2000 RPM and 20 deg C is representative since the torque is similar to the torque at nominal speed and -20 deg C. This validates the speed, to be used during the accelerated life test, found by analysis.

Generally, similar results have been found for Braycote601EF and MaplubSH100b at 20 deg C as it is shown on the following figure:

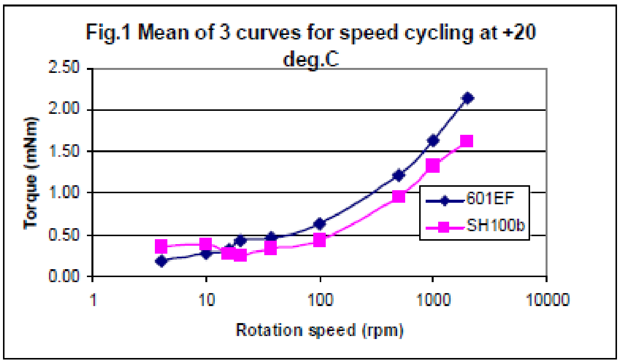

Figure 3: Stribeck curves measured at 20 deg C for Braycote and Maplub

To investigate the hot case, the following Stribeck curve has been measured at 85 deg C:

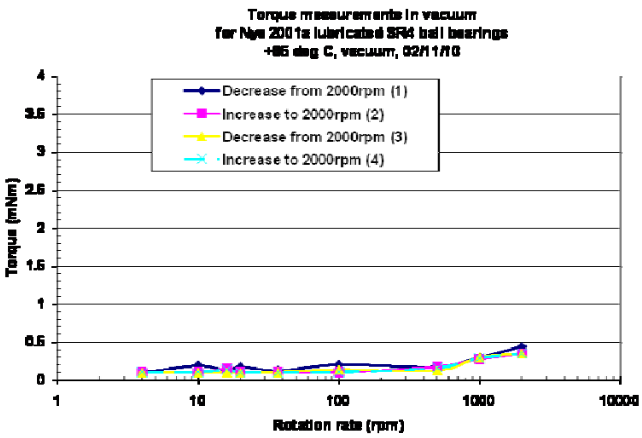

Figure 4: Stribeck curves measured at 85 deg C (maximum temperature)

The obtained Stribeck curve shows that between the nominal speed (16.5 RPM) and the proposed speed (2000 RPM), the torque is rather constant, therefore the proposed speed is acceptable since it will not change the lubrication regime.

From the Stribeck curves measurements of the greases (base oil + additives) Braycote601EF and MaplubSH100b at 85 deg C, it has been seen that, while the Braycote601EF has a similar behaviour than the Nye2001a, apart of the fact that the test speed has to be lower than for the Nye2001a, the MaplubSH100b shows a high torque at very low speed. This is illustrated on the next figure:

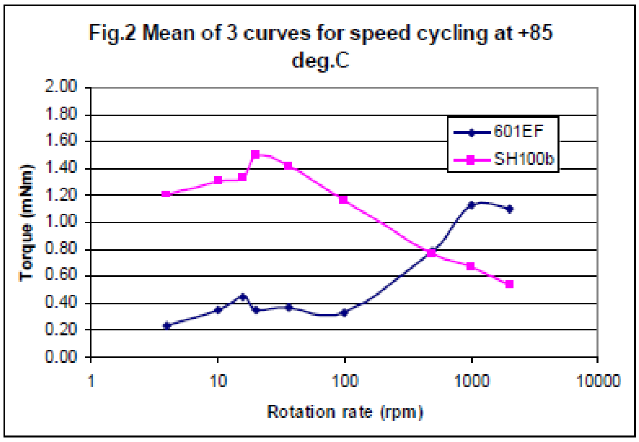

Figure 5: Stribeck curves measured at 85 deg C for Braycote and Maplub

Indeed, while the Braycote601EF shows an increase in the torque between 16.5 RPM (nominal speed) and 500 RPM (proposed speed for the accelerated life test), the MaplubSH100b shows a decrease in the torque between 16.5 RPM and 2000 RPM (proposed speed for the accelerated life test). Since the model is based on the base oil (Nye2001a for the MaplubSH100b grease) and that good correlation is found between the model and the Nye2001a Stribeck curve, it has been deduced that this effect is most probably coming from the PTFE particles added to the grease base oil. The fact that the bearing is more filled with Maplub than Braycote (indeed the same 8 mg mass of Maplub and Braycote has been put in the bearing, while the molar mass is much lower for Maplub) can only increase the effect of the PTFE particles. This implies however that it is very important to consider in the torque margins assessment, the torque measured for the grease at the nominal speed.

3. TEST RESULTS ON BEARINGS

Bearings filled with 8 mg of Braycote601EF and 8 mg of MaplubSH100b have been used for an accelerated life test done at ESTL. The tests have been done in vacuum with period of high temperature (85 deg C) and low temperature (20 deg C) with also some periods at the minimum temperature (-20 deg C). It has been considered that 75 % of the lifetime is spent at high temperature and 25 % of the lifetime is spent at low temperature. The torque has been recorded during the tests. The torque traces are shown hereafter:

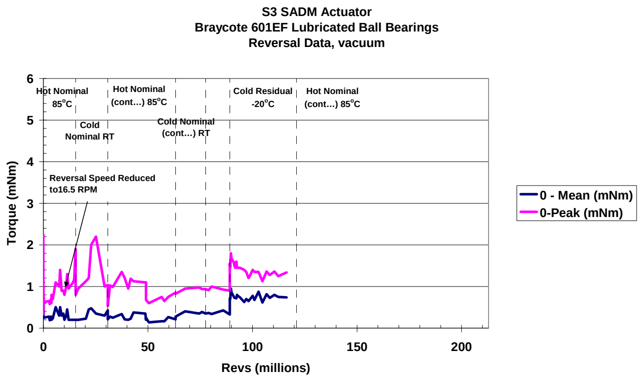

Figure 6: Accelerated life test results for Braycote601EF (the red curve is the peak torque while the blue curve represents the mean torque)

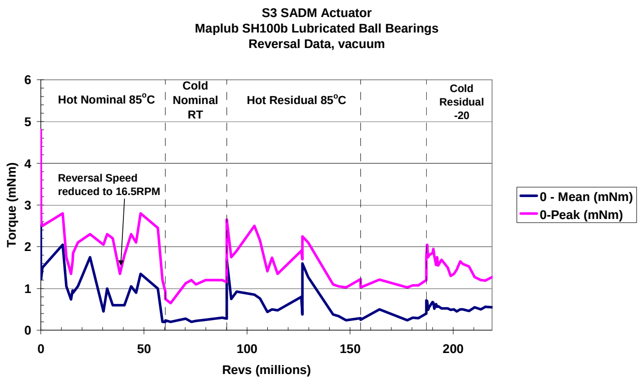

Figure 7: Accelerated life test results for MaplubSH100b

From the test results, the following information is found:

- Braycote601EF bearing has accumulated 130 million revolutions
- MaplubSH100b bearing has accumulated 210 million revolutions
- For both lubricants, the torque traces appear rather stable, indicating no traces of breakdown of the lubricant
- Since the required lifetime is 140 million revolutions, MaplubSH100b has been chosen as the baseline lubricant

4. TENTATIVE OF CORRELATION BETWEEN LUBRICANT DEGRADATION ANALYSIS AND TEST RESULTS

It has been shown that the consumption/degradation analysis based on the SOT and the degradation analysis based on bearings tests done at ESTL give similar results in terms of lifetime (8.6 million revolutions for the first method and 9 million revolutions for the second method, concerning Braycote601EF). It is however clear that the proposed analysis and extrapolation of ESTLs bearing test are conservative with respect to the specific test results obtained for Sentinel3.

A possible explanation is that the lubrication regime in the accelerated test for both cold and hot case is not in reality representative compared to the analysis and SOT test extrapolation. For example, it is known (ref.[1]) that for Braycote601EF lubricated bearings, a duration of about 1000 millions revolutions is possible at cold temperature mainly due to the fact that the bearing is operated in mixed and outside the boundary lubrication regime. It is possible that the accelerated test at hot case is still running too fast and is in reality more into the mixed regime than low end boundary. Considering that the SOT and the bearing test data used for the lubricant degradation analysis is relative to low end boundary lubrication regime only, it could explain the difference in life time.

Also, the mixed regime in each cold case can possibly replenish the lubricant and renew the lubricant film improving the conditions for the follow on hot case. Thus, the bearing life is not purely an accumulation of revs in boundary regime.

A second explanation can be the very small starved amount of lubricant applied in the SOT possibly giving a non linear wear rate, which when extrapolated into the larger lubrication amount present in the ball bearing test, will give a much longer life.

However, it is clear from this discussion that it is needed to have a much more complete bearing life test database at hot temperature (boundary regime) and cold temperature (mixed regime) to have enough inputs for the proposed lubricant degradation/ consumption models.

5. CONCLUSIONS

The main conclusions of the work presented in this paper are:

- There is in general a good correlation between the analytical models to predict the accelerated life test conditions (acceleration factor depending on temperature) and the test results

- Lubricant degradation/consumption models based on Spiral Orbit Tribometer and on a general bearing test by ESTL gi ve similar results
- The lubricant degradation/consumption models for e xtrapolation to other specific cases like the Sentinel-3 program appear very conservative when compared with the actual test results obtained.
- A wider database of bearings life tests with different lubricants and at different temperatures and/or lubrication regimes is considered necessary to p redict more accurately the lifetime performance of fluid lubricated bearings

6. REFERENCES

[1] E.Roberts, *Space Tribology Handbook,* ESTL (2007).

[2] M.Buttery, *A Spiral Orbit Tribometer Assessment of Space Greases,* ESA Mechanisms Final Presentation Days (2010)

[3] P.J.Schungel, *Clarification of S33 QM Lubrication Issue*, Ruag (2010).

[4] S.D.Lewis, *Guidelines for Accelerated Testing of Liquid Lubricated Mechanisms, ESTL* (1997).