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A COMPARISON OF THE PERFORMANCE OF SOLID AND LIQUID LUBRICANTS IN OSCILLATING SPACECRAFT BALL BEARINGS

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ABSTRACT

The European Space Tribology Laboratory (ESTL) has been engaged in a programme to compare the performance of oscillating ball bearings when lubricated by a number of space lubricants, both liquid and solid. The results have shown that mean torque levels are increased by up to a factor of five above the normal running torque, and that often torque peaks of even greater magnitudes are present at the ends of travel. It is believed that these effects are caused by a build-up of compacted debris in the contact zone, thus reducing the ball/race conformity ratio.

INTRODUCTION

The increasing use of scanner systems on earth observation spacecraft demands reliable and predictable behaviour from oscillating ball bearings. ESTL is increasingly being asked about this aspect of bearing behaviour, both when utilising dry lubrication techniques and liquid lubricants. This paper describes tests performed by ESTL to provide baseline data for comparing these different lubrication techniques. In order to perform this testwork, ESTL has designed and built an in-vacuo test facility which oscillates three pairs of preloaded bearings simultaneously

ESTL TEST FACILITY

A schematic diagram of the rig is shown in Figure 1. The rig incorporates three test stations, allowing different angles of oscillation to be tested concurrently. The test bearings (1) are mounted in a housing at the lower end of the rig. They are preloaded by a pair of belleville washers (2), and the stationary inner shaft is held by the shaft of a Teldix DG1.3 inductive torque transducer (3). The torque transducer is supported by a thin sheet of shim, to allow for small misalignments whilst ensuring torsional rigidity.

The oscillatory motion is induced by a stepper motor (4). Two of the test stations have 25,000 step per revolution microstepping motors fitted, whilst the third has a 400 step per revolution motor. Control is open loop, and the required motion profiles are generated by a PC based indexer control board. The adequacy of the open loop system has been subsequently proved by the post test bearing inspections. The system is very flexible, and relatively easy to programme.

The oscillatory motion is transmitted into the chamber via ferrofluidic rotary feedthroughs (5). The test bearing outer housing is fastened to one end of a main support shaft which has its own housing and bearing system (6). The support bearings were lubricated with KG80 oil. Two high torsional stiffness bellows couplings are used to cater for small misalignments.

MATERIAL COMBINATIONS

To date eight different lubricant/cage combinations have been tested as shown below in Table 1.

TABLE 1

Table of Lubricant/Cage Combinations Tested

	Lubricant	Cage Type
i)	Sputter Coated MoS_2	Duroid 5813
ii)	Ion Plated Lead	Lead Bronze
iii)	Race uncoated	Duroid 5813
iv)	"	Vespel SP3
v)	"	Salox M
vi)	Fomblin Z25	Phenolic
vii)	Braycote 601	Phenolic
viii)	Pennzane SHFX2000	Phenolic

For the coated bearings (i-ii), 0.2-0.5 µm of lubricant film was applied to each race, and in addition the MoS_2 coating was also applied to the balls. For the wet lubricated bearings (vi-viii), the phenolic cages were vacuum impregnated with oil prior to fitting (using Fomblin Z25 in the case of the grease, vii).

TESTED MOTION PROFILE

For each of the cage material and lubricant combinations, measurements were taken of the torque behaviour for a pair of angular contact bearings oscillating over three different angles:

± 0.5° before equilibrium rolling is fully established.

± 5° corresponding to limited rolling.

± 20° large amplitude rolling, but insufficient to cause cage to race material transfer.

Tests were performed over ten million surface passes (2 passes per complete oscillation) under a vacuum of 10^{-5} torr or better. The testing was performed at fairly high rotational speed, which was reduced by a factor of 4 when making torque measurements. This was necessary due to rig torsional natural frequency effects, caused by the relatively low stiffness of the transducer, swamping the real torque signals. Even having restricted the speed, in the case of the ± 20° test it was still necessary for the signal to be electronically low-pass filtered, although this was shown to have no effect on the DC measured levels.

The speed motion profile was trapezoidal with a period of constant speed motion. The chosen motion profile parameters are shown below in Table 2. These parameters were chosen such that the elapsed time for testing at each of the three angles of oscillation would be nominally the same.

Table 2

Motion Profile Parameters

Test Station	1	2	3
Distance	0.99	9.99	40.5 degrees
During Measurements:			
Velocity	0.01	0.1	0.41 revs/sec
Acceleration	0.108	1.08	4.41 revs/sec^2
During Running:			
Velocity	0.04	0.4	1.64 revs/sec
Acceleration	1.337	13.37	54.5 revs/sec^2

The tests were performed at a nominal preload of 60-70 N; unfortunately however, a load-setting problem led to the tests with Duroid cages alone (iii) being performed at higher preloads (100-150 N). All bearings were subjected to a limited run-in prior to testing, with the exception of those coated with MoS_2 (i). These bearings were not run-in in order that there should be no transfer of PTFE from the cages to the races prior to starting the test.

On completion of the tests, the bearings were disassembled and examined optically. Selected components were also examined by scanning electron microscopy (SEM).

BEARING DETAILS

The test bearings were standard 20mm bore profile (conformity 1.14) ED20 ball bearings to ABEC 7 specification manufactured from 52100 steel by SNFA. Further details are shown in Table 3:-

Table 3

ED20 Bearing Size Parameters

Outer Diameter	42 mm
Inner Diameter	20 mm
Bearing Width	12 mm
Ball Size	7.14 mm
Ball Complement	10
Contact Angle	15°

THEORETICAL PERFORMANCE

A number of calculations based on the geometry of the bearings under test can be performed in order to give an idea of the expected torque performance behaviour and the likely scar dimensions. Firstly, for a ball bearing the ball spin frequency per rotation is given by the following equation:-

$$F = [P/(2B)] \times [1-(B/P)^2 \times \cos^2 A]$$

where
F = Ball Spin Frequency
P = Pitch Diameter
B = Ball Diameter
A = Contact Angle

Assuming a ball pitch diameter of 31mm and taking other data from Table 3, the ball spin frequency is 2.06 revs per revolution of the bearing.

For a dry lubrication system relying on lubricant replenishment from the cage, then the theoretically required angle of oscillation will be ± 21.8° before the balls will perform the 90° rotation required for cage material transfer to the raceways.

The lengths of the expected wear scars on the races for the three angles of oscillation tested can also be generated from this ball spin frequency assuming that there is no slip at the ball to race interfaces. The scar length will be given by the following equation:-

$$L = \text{Angle} / 360 \times F \times \pi \times B$$

and the results are tabulated in Table 4:-

Table 4

Scar Length Predictions for Tested Bearings

Oscillation deg	Angle deg	Scar Length mm
± 0.5	1	0.13
± 5	10	1.29
± 20	40	5.14

It is also possible to calculate the expected torque performance and the contact stresses of the test bearings. Calculations have been performed using BAPTISM, the ESTL in-house coding, which has been verified against the results of many bearing tests over the years since its conception. The torques calculated by BAPTISM are those expected for bearings under continuous rotation due to the Coulombic torque contribution.

Table 5 shows the BAPTISM-calculated torque predictions for a pair of ED20 bearings, which is the configuration used in these tests. The table shows the effect on the expected running torque both by increasing the preload and also by reducing the number of balls in contact. The friction level of 0.15 was used as a typical value for lead lubricated bearings (ii).

Table 5

Coulombic Torque Predictions

Preload N	Balls	Friction Coeff.	Torque Nm × 10^{-4}	Mean Hertzian Contact Stress MPa
65	10	0.15	20	679
150	10	0.15	60	890
65	5	0.15	25	850
65	3	0.15	30	1001
65	10	0.2	25	679
65	10	0.05	10	679
65	10	0.5	60	679

In addition the effects of changing friction levels on the bearings can also be ascertained. The value of 0.05 is about the lowest to be reasonably expected and represents a typical value for MoS_2 lubricated bearings (i), whereas 0.2 is the average value for Duroid lubrication alone (iii) and represents the highest expected figure. The Hertzian contact stress figures quoted for each load case are the mean contact stress on the inner race. The Hertzian contact ellipse will be of major axis 0.22mm and minor axis 0.06mm for the standard 65N preloaded pair with ten balls in contact. BAPTISM also predicts that the full rolling torque will not be attained until the angle of oscillation is greater than about ± 2°

As a further exercise BAPTISM has been used to generate a curve of torque versus the conformity ratio of the bearing (raceway diameter ÷ ball diameter) for the nominal test conditions, and this data is shown in Figure 2. It can be seen that this ratio causes a dramatic increase in the expected torque levels as it is reduced.

TEST RESULTS

The material combinations will be split into three groupings to allow the data to be presented in a comparable manner : the dry coated bearings (i-ii); the cage dry-lubricated only bearings (iii-v); and the wet lubricated bearings (vi-viii). Torque levels quoted throughout are those measured for a pair of bearings and are either zero-to-mean or zero-to-peak as quoted. The values have been taken as spot readings at regular intervals on a digital storage oscilloscope, with a hard copy produced on a plotter.

Figures 3-5 relate to the results taken from the sets of bearings oscillated through ±0.5°. These bearings all gave similar outputs which resembled a sine-wave. The coated bearings (i,ii) performed with lower torques than the cage lubricated bearings (iii-v), although the MoS_2 coated bearings had reached torque levels of 100×10^{-4} Nm by the end of the tested 10^7 oscillatory passes. The cage dry-lubricated bearings (iii-v) quickly registered torques of $100\text{-}130 \times 10^{-4}$ Nm. For the oil lubricated bearings, the Fomblin Z25 (vi) showed a rapid increase to 100×10^{-4} Nm before settling back to 80×10^{-4} Nm, whereas the Pennzane lubricated bearings (viii) only showed a gradual increase from 20 up to 40×10^{-4} Nm over the duration of the test. The Braycote 601 grease lubricated bearings (vii) showed a rapid increase over the first million passes to around 60×10^{-4} Nm and then stayed stable for the rest of the test.

The bearings tested at ±5° and ±20° displayed a different torque behaviour, in that they exhibited a square wave profile on start-up which in many cases was modified by a peak on reversal which grew in size during the test. For this reason graphs relating to these angles of oscillation show both a zero-to-mean value for the running zone and a zero-to-peak value relating to the reversal point.

Figures 6-8 relate to the test results taken from the bearings oscillated through ±5°. The MoS_2 coated bearings (i) performed better than the lead (ii) in this instance. The lead mean level increased to $150\text{-}200 \times 10^{-4}$ Nm over the first 3 million passes, whilst the MoS_2 mean level remained low at 20×10^{-4} Nm throughout. Both types suffered a reversal peak torque, $300\text{-}400 \times 10^{-4}$ Nm for the lead and 100×10^{-4} Nm for the MoS_2 by the end of the test. Turning to the cage lubricated bearings (iii-v), the torque of the Duroid caged bearings rapidly rose to 200×10^{-4} Nm and continued to increase to 600×10^{-4} Nm by 6 million oscillatory passes. At the same time a reversal peak level of 1200×10^{-4} Nm was attained and so the test was stopped to protect the torque transducer. The torque of the Vespel caged bearings (iv) also rose quickly to a mean level of 200×10^{-4} Nm for the duration of the test. The peak level on reversal reached a maximum value of nearly 600×10^{-4} Nm at 3 million oscillatory passes, but in this case fell back to 300×10^{-4} Nm by the end of the test. The Salox M caged bearings (v) performed the best in this category and held a mean torque level of 20 × 10^{-4} Nm with a peak of 50-60 × 10^{-4} Nm after an initial short stabilising period. The wet lubricants (vi-viii) performed in a very similar manner throughout this test, with mean torque levels around 20 × 10^{-4} Nm and peak torque levels up to 40 × 10^{-4} Nm.

Figures 9-11 relate to the test results taken from the bearings oscillated through ±20°. The MoS_2 and lead coated bearings (i-ii) performed similarly for over half of the test duration, although the lead bearings were noisier on reversal and ran at higher mean torque levels. By the end of the test however, starting at around 7 million oscillatory passes, the mean torque levels for both types had risen to 100 × 10^{-4} Nm, with peak levels on reversal as high as 200 × 10^{-4} Nm for the lead. The cage dry-lubricated bearings (iii-v) showed no major variations after the initial settling period. The Salox M (v) caged bearings again performed the best of the trio with mean levels of around 50 × 10^{-4} Nm compared with 100 × 10^{-4} Nm for the Vespel (iv) and 150 × 10^{-4} Nm for the Duroid (iii). Again the wet lubricants (vi-viii) performed in a very similar manner throughout this test, with mean torque levels around 15-20 × 10^{-4} Nm and peak torque levels up to 30 × 10^{-4} Nm for the Braycote grease and Pennzane oil (vii,viii). The Fomblin Z25 (vi) recorded higher mean levels, 30 × 10^{-4} Nm, with peak torque levels up to 60 × 10^{-4} Nm during the second half of the test.

POST TEST INSPECTION & DISCUSSION

Inspection of the bearing condition post testing has revealed very obvious contact zones in most cases, especially in the case of the dry lubricants (i-v), which are of sizes in agreement with the predictions in Table 4. In the case of the coated bearings (i,ii) the motion has worn a groove into the lubricant with a build up of debris around the edge. In the case of the cage dry-lubricated bearings (iii-v) compacted zones of material have been generated on the bearing surface during the motion. These details have been confirmed by a small number of Talyrond measurements, and also by removing the debris in the latter case. The wet lubricated bearings also show obvious contact zones of sizes similar to those in the dry lubricated bearings, however the height of these features has not been measured at this time. However it is not believed that any steel bearing surface material wear has occurred in any of these tests.

In a number of cases balls have more than one pair of corresponding contact zone markings indicating that some balls were not in contact at all times. This observation helps to explain the manner in which material can be transferred from the cage to the ball-race interface despite the fact that theoretically the balls do not rotate over a large enough angle.

Figure 12 shows two of the SEM photographs taken of the contact zones post testing. The upper photograph shows the whole of a ±5° contact zone from the MoS_2 test (i). The debris around the edge of the contact zone can be clearly seen. The lower photograph shows the end of a contact zone from the Salox M cage test (v). The end-of-travel debris is visible in the centre, with the contact zone going to the right. To the left is the running-in transfer film. Similar marks have been visible on all the bearings, although not quite so distinct on the wet lubricated bearings (vi-viii).

By reference to Table 5 it is clear that increases in the friction coefficient or the preload setting, or alternatively a reduction in the number of contacting balls within the bearing cannot induce the high levels of torque which have been recorded in these tests. However, changes in the conformity ratio can produce such dramatic changes, as shown in Figure 2. The Talyrond measurements have confirmed that the build-up of debris on both the raceways and the balls is sufficient to close the gap between ball and race, thus allowing such close conformities to be achieved.

CONCLUSIONS

The measurement of torques in oscillating bearings has revealed levels many times higher than would be expected from continuously rotating bearings. Factors of five on mean torque levels are common, and in addition torque peaks on reversal of even higher magnitude have been recorded. This should be taken into account when calculating mechanism drive torque requirements.

It is obvious from the test results that there is no one ideal lubricant technique to cater for all the angles of oscillation, and ESTL will be continuing to investigate this aspect further in the future. It has been shown that it is difficult to explain the torque increases seen in oscillating bearings purely by a change in friction or preload levels or by a reduction in the number of balls in contact, and ESTL therefore proposes that the change in conformance at the contact due to compacted debris build up is the cause of the increased torque levels.

ACKNOWLEDGEMENTS

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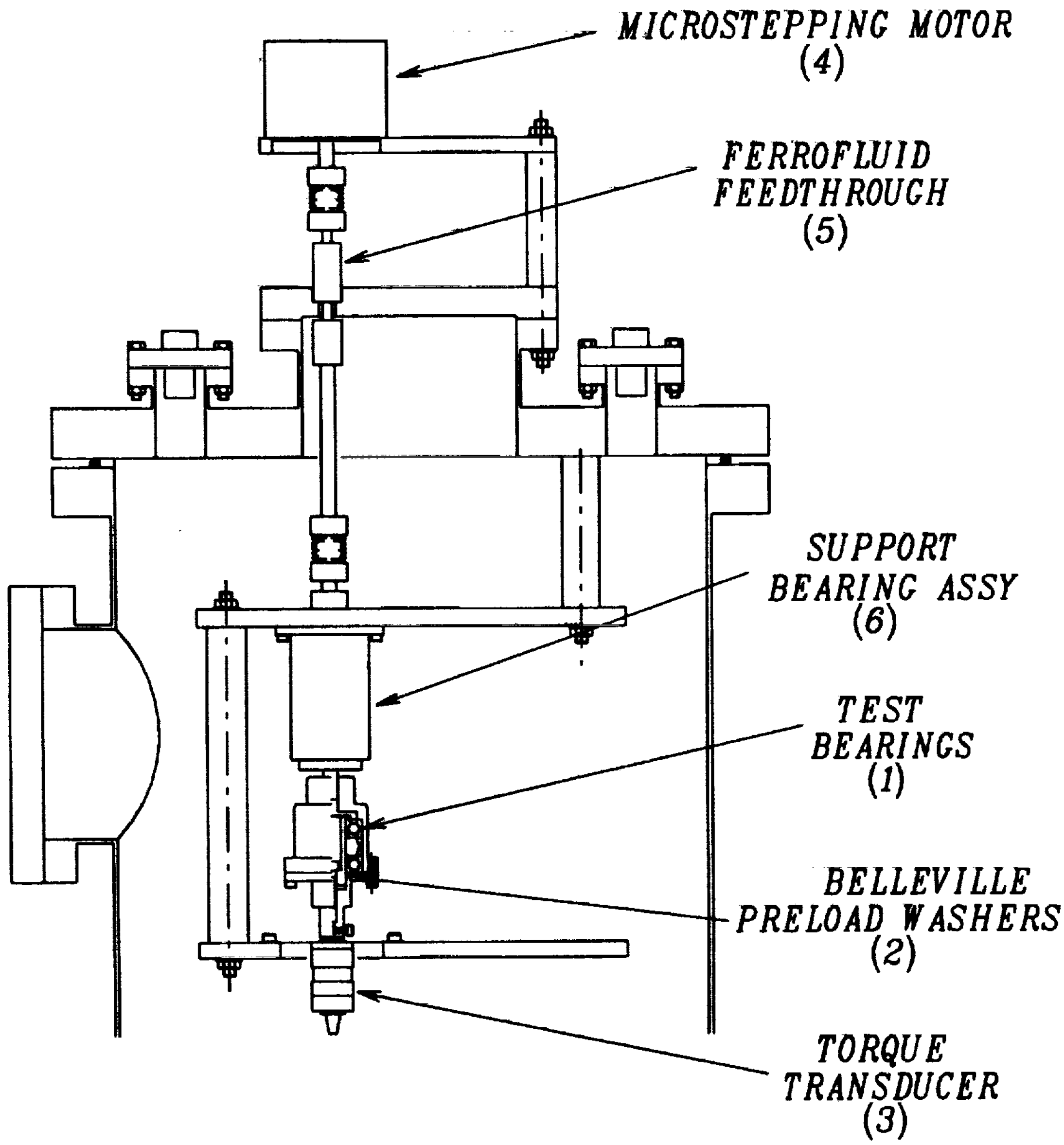
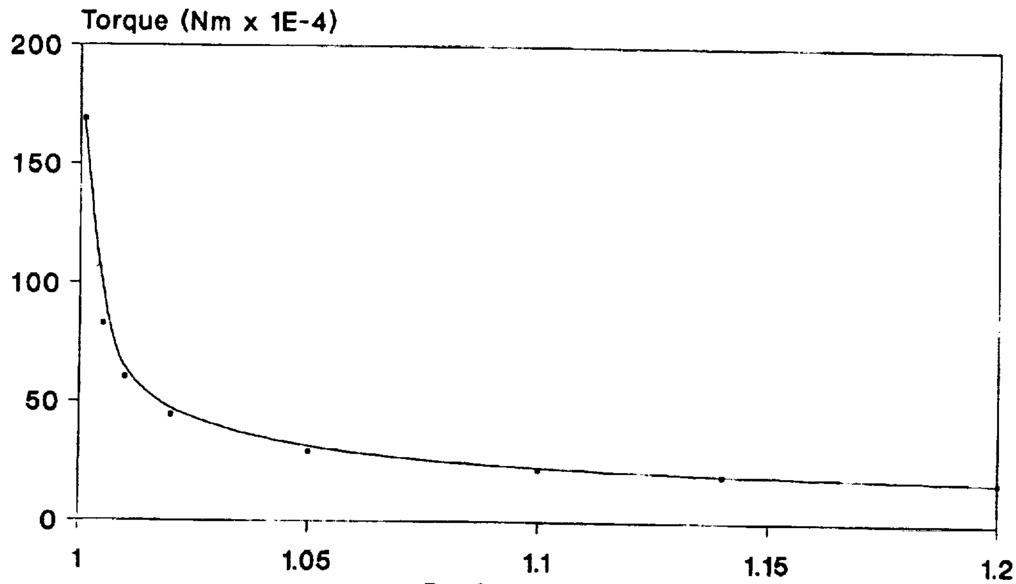


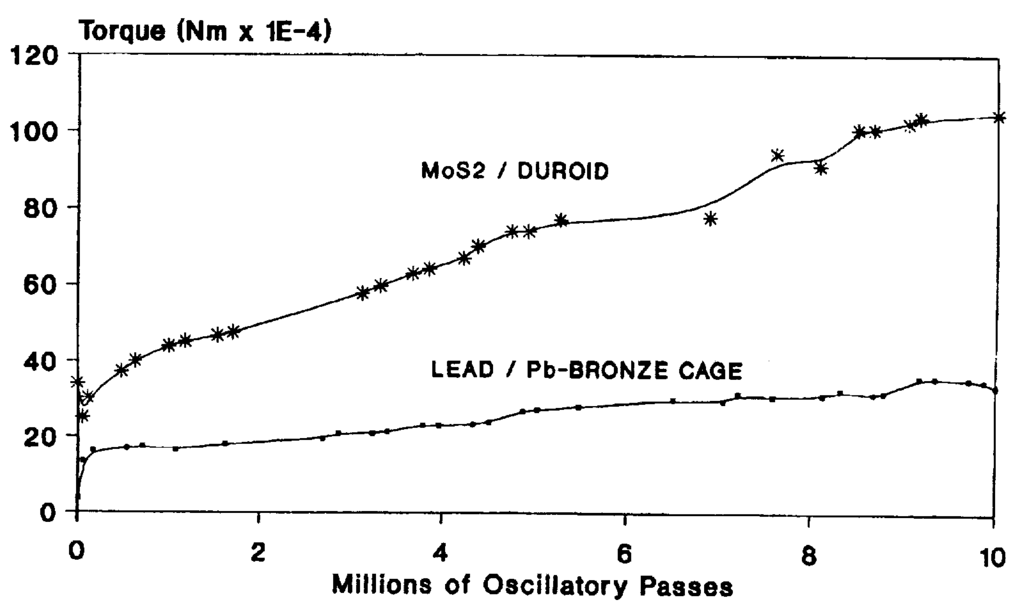
FIGURE 1 OSCILLATING BEARING RIG SCHEMATIC

Figure 2
Calculated Torque versus Conformity for a Pair of Test Bearings



Friction Coefficient 0.15
Preload 65N

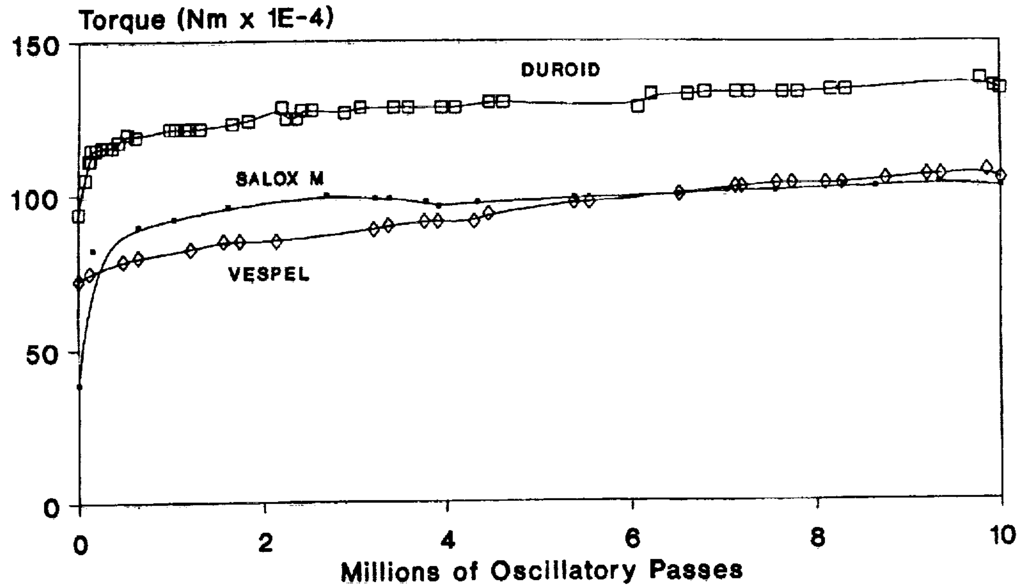
Figure 3
Torque versus Number of Oscillations
Angle of Oscillation +/- 0.5 degrees



MoS2 with Duroid Cage
Preload 65N

Lead with Lead/Bronze Cage
Preload 68N

Figure 4
Torque versus Number of Oscillations
Angle of Oscillation +/- 0.5 degrees

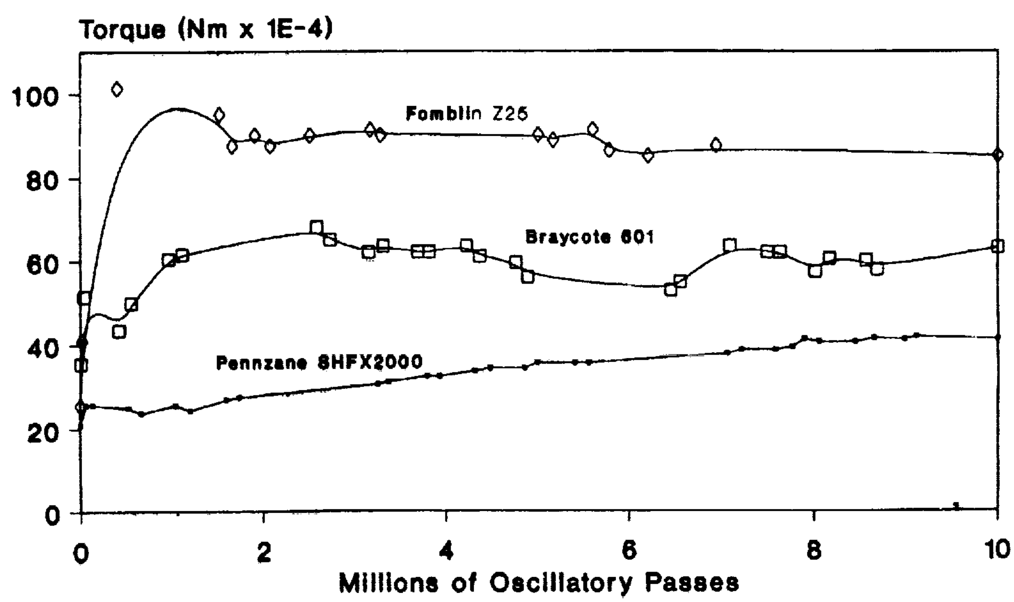


Salox M Cage
Preload 65N

Duroid Cage
Preload 150N

Vespel Cage
Preload 75N

Figure 5
Torque versus Number of Oscillations
Angle of Oscillation +/- 0.5 degrees

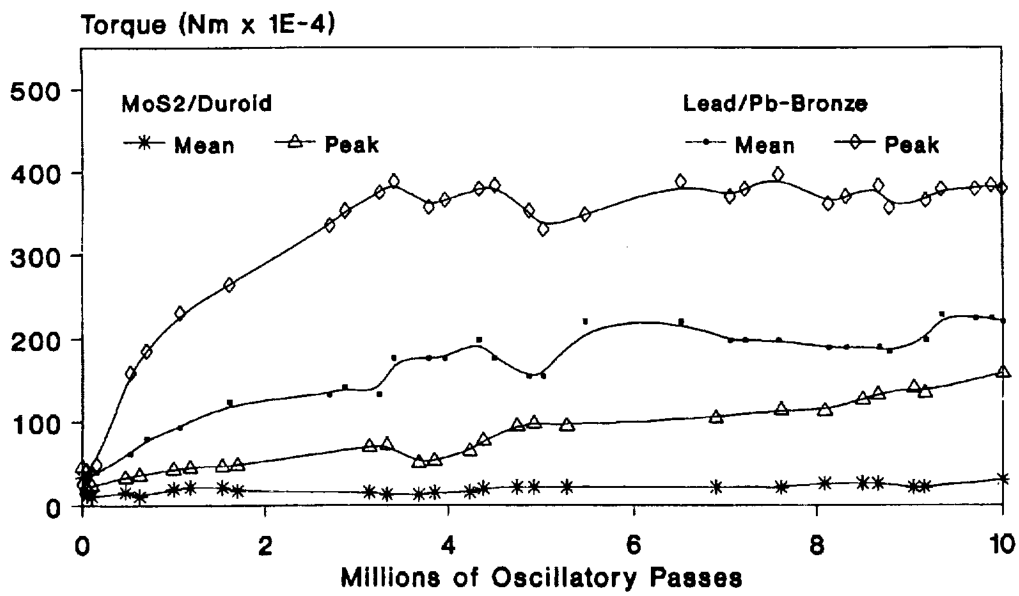


Fomblin Z25
Preload 65N

Braycote 601
Preload 65N

Pennzane SHFX2000
Preload 65N

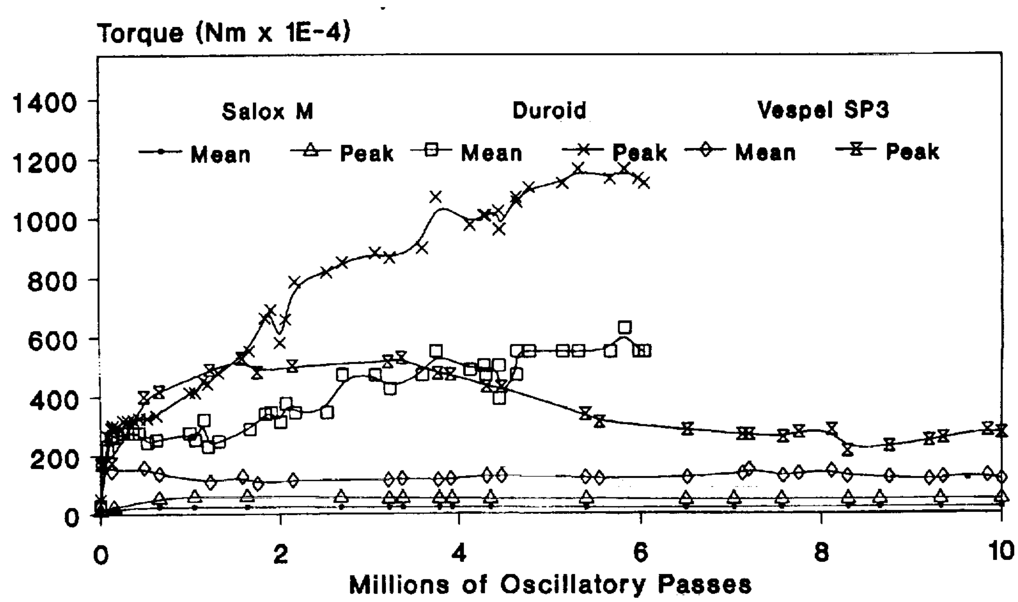
Figure 6
Torque versus Number of Oscillations
Angle of Oscillation +/- 5 degrees



MoS2 with Duroid Cage
Preload 68N

Lead with Lead/Bronze Cage
Preload 62N

Figure 7
Torque versus Number of Oscillations
Angle of Oscillation +/- 5 degrees

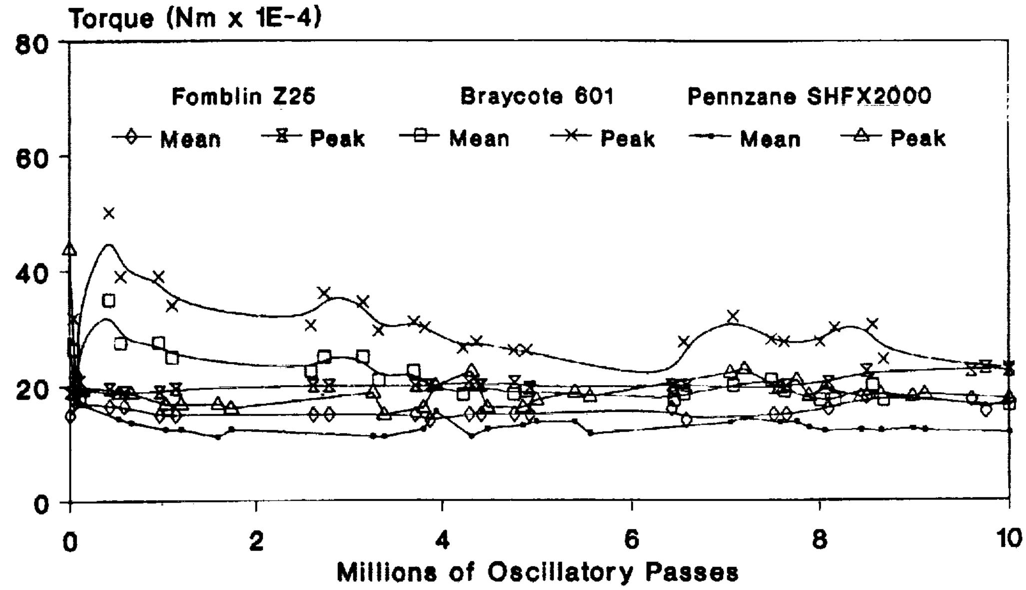


Salox M Cage
Preload 65N

Duroid Cage
Preload 132N

Vespel Cage
Preload 65N

Figure 8
Torque versus Number of Oscillations
Angle of Oscillation +/- 5 degrees

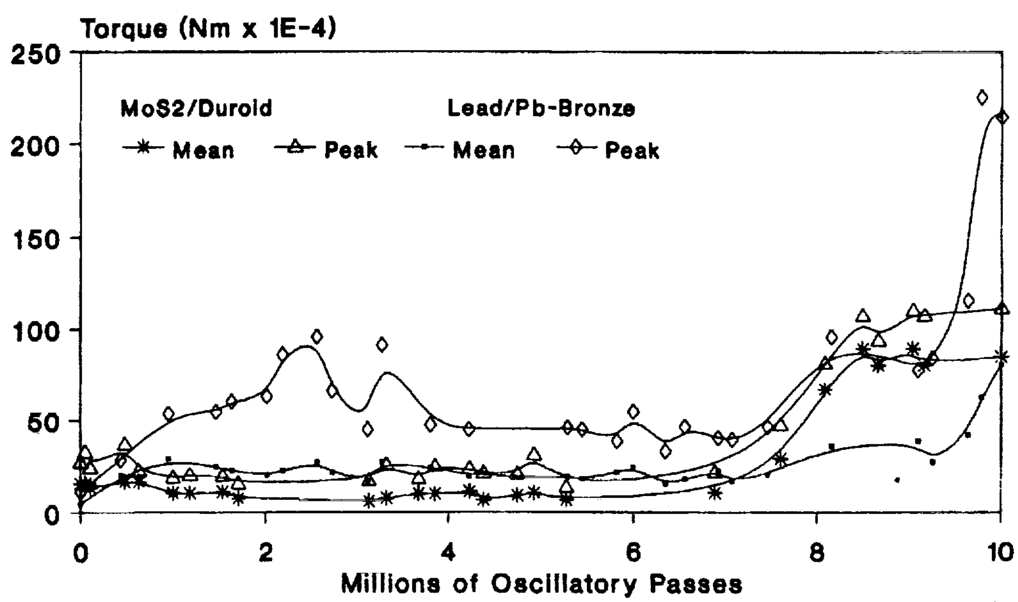


Fomblin Z25
Preload 65N

Braycote 601
Preload 60N

Pennzane SHFX2000
Preload 64N

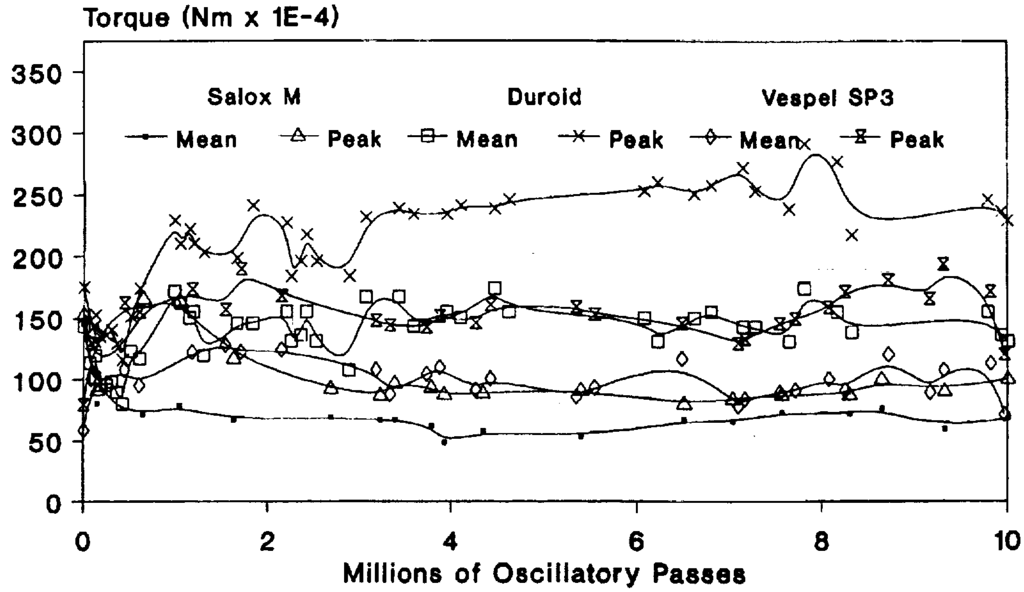
Figure 9
Torque versus Number of Oscillations
Angle of Oscillation +/- 20 degrees



MoS2 with Duroid Cage
Preload 70N

Lead with Lead/Bronze Cage
Preload 75N

Figure 10
Torque versus Number of Oscillations
Angle of Oscillation +/- 20 degrees

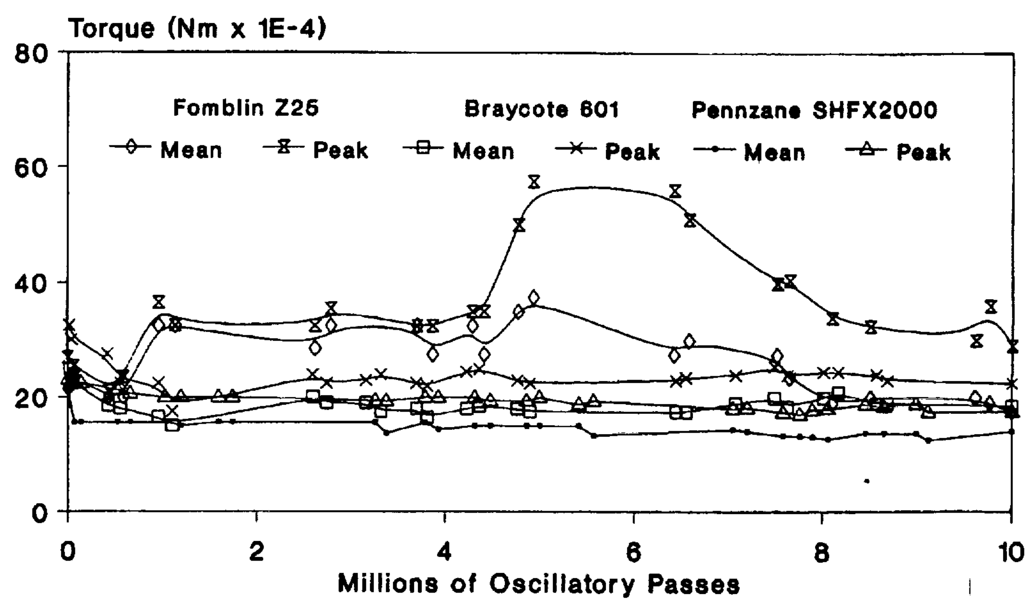


Salox M Cage
Preload 65N

Duroid Cage
Preload 94N

Vespel Cage
Preload 65N

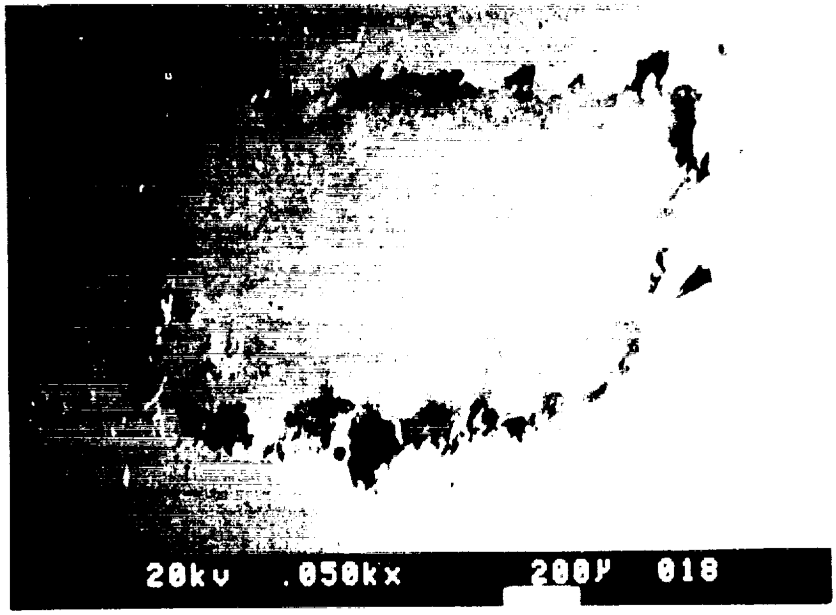
Figure 11
Torque versus Number of Oscillations
Angle of Oscillation +/- 20 degrees



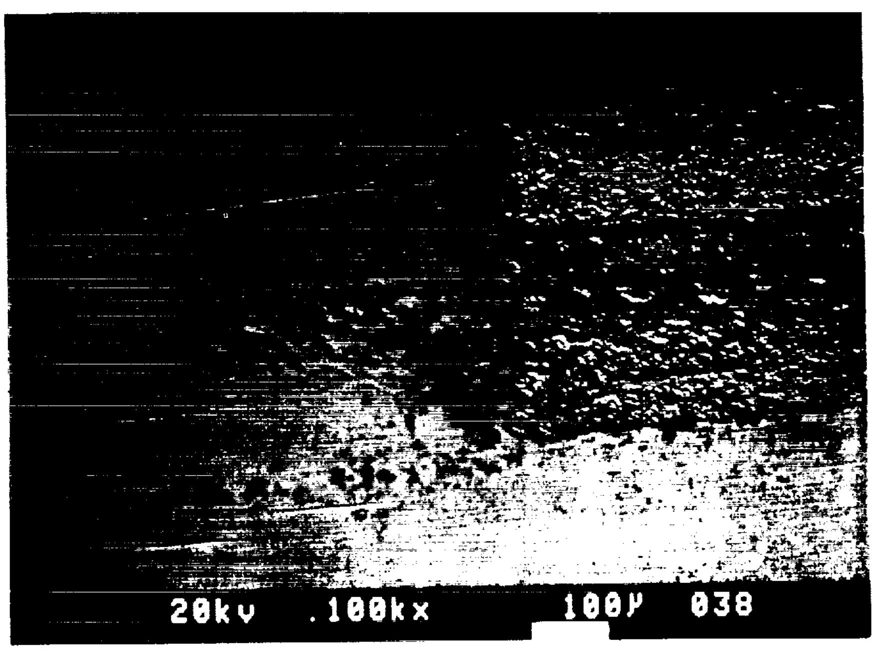
Fomblin Z25
Preload 65N

Braycote 601
Preload 60N

Pennzane SHFX2000
Preload 63N



MoS_ Lubrication (i), ±5° Test
Secondary electron image of inner race contact zone



Salox M Lubrication (v), ±20° Test
Backscattered electron image of inner race contact zone

Figure 12 SEM Photographs of Contact Zones Post Testing