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RESEARCH

Comparison of Grease and Solid Lubrication of Rolling Bearings Under Small-Stroke Reciprocation for Space Applications

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A B S T R A C T

Rotary actuator is a device responsible for the precise rotation of sensitive instruments such as antennas. Requirements for its precision and stiffness are extremely high. Actuators used in the space environment have to work for long period, under extreme condition and without possibility of repair or replacement. One of the main components responsible for its proper function are bearings. They precisely support shafts, and usually operate in a very small angular range with many repetitions. Selection of the proper bearing design and also the lubrication of the bearing can have significant effect on the performance and life of the entire satellite. Purpose of this study was to select and test suitable lubricant to support operation of such bearing. Sets of bearings lubricated by Rheolube 2000 and MoS2 PVD film were tested in thermal vacuum chamber under small angle oscillatory motion with the same start and end position for every cycle. Torque and bearing noise was evaluated during the tests and bearings were inspected. Rheolube 2000 was selected as the best lubricant for the described conditions because of the acceptable torque, lower bearing noise and easier manipulation during assembly and testing. However, both options proved to be applicable.

1. INTRODUCTION

Communication satellites has been used on various Earth's orbits since 1970' and are getting into more importance ever since. Modern life on earth is now dependent on the network of thousands of different satellites on different orbits. However, all of the satellites has one common feature – they have to send data and communicate with ground control in general. Thus Communication is very sensitive to pointing accuracy of antennas and error in the process can lead to the degradation of communication bandwidth or ultimately to the loss of the ability to control the satellite. Devices responsible for the positioning are called rotary actuators. Schematic illustration of the rotary actuator developed in current project is in Fig. 1.

Rotary actuator (RA) is attached by one end to the satellite's body and is holding the antenna on the other side. Relative rotary motion is ensured

between both ends by step motor. Motion takes place with high demand for precision usually in order of thousandths of degrees, and high stiffness. RAs can be assembled in series of two or three RA units to enable 3D positioning in the way such as robotic arm. Entire assembly is generally referred as APM (Antenna pointing mechanism). Examples and detailed description of an APM's can be found in [1,2].

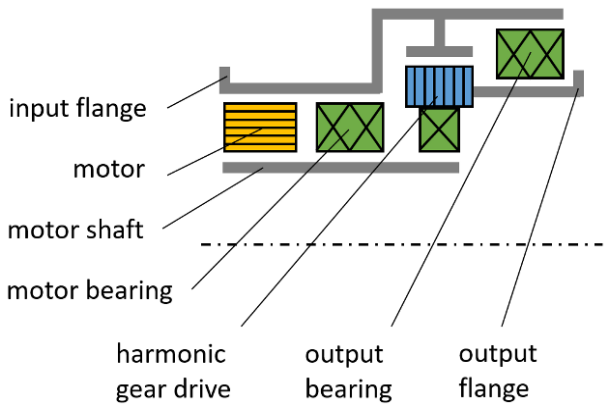


Fig. 1. Schematic illustration of the rotary actuator.

One of the main issues following the design process of the rotary actuators is the selection and testing of the bearings and its lubrication. Bearings have to provide stiffness, allow smooth rotation of the rotor and have to be able to work in very small angles, mean while supporting precise pointing movement of antenna. Typical required life-span of RA for the antennas in the harsh space environment, without possibility for maintenance or replacement is about a decade.

Most common type of the bearing used in space applications is pair of angular contact ball bearings with specified preload as there is no actual weight load due to the lack of gravity. Lubricant ensuring lifespan, energy-efficient and smooth rotation can be either solid or liquid type.

Solid lubrication is mainly represented by sputtered thin film of MoS2. There is extensive heritage of its applications, although with known downsides. Main benefit is extremely low friction in vacuum, however there is very limited lifetime once run in humid air because wear rate rises significantly under water vapour exposure. Therefore, in-air testing of the mechanism is almost impossible. Another known issue is generation of the wear particles which can be in the form of free lumps and which are obstacles for the rolling element. This is known to increase bearing torque noise which is extremely unwanted as it can influence the precision of the positioning [3].

Liquid lubricant can be in the form of oil or grease, where base oil for the grease is usually the same as the one used in pure oil lubrication. Problem with liquid lubrication in general is very low ambient pressure increasing evaporation rate and the large range of the operating temperatures. Specific conditions exclude most of the lubricants used on ground. Only two usable groups of oils are multiply alkylated cyclopentanes (MACs) and perfluoropolyethers (PFPE). Group of PFPE oils has the lowest vapour pressure which provides longer lifetime, but has known issue with chemical decompositions. Autocatalytic degradation of lubricant's ethers occurs when the lubricant is in contact with chemically active surface of the bearing steel [4,5]. Longer lifetime due to absence of fluor chains is found in MACs. However, disadvantage of the MACs is higher vapour pressure causing faster evaporation and lower useful temperature range due to the lower viscosity index.

Selection of the proper lubricant has to be tailored for specific application. High speed bearings such as the one used in reaction wheels [6] or gimbals systems [7] usually rely solely on the oils because of the benefits of the EHD lubrication regime. Prolonged mission duration or extreme temperatures usually determines usage of solid lubrication. Example of such extreme is bearing in James web space telescope where temperature requirement is down to 30 K [8].

More detailed information about general differences between individual lubricants, their tests and comparison and space environment in general can be found in [9,10]. Extensively detailed explanation of space lubrication is provided in the Space Tribology Handbook [4].

Main goal of this paper is to provide results from the component level testing of the output bearing (Fig. 1). Bearings are used in the newly developed RA and later in APM assembly. Purpose of the testing is to select the best lubricant and to verify function of the bearing in relevant conditions for its defined lifetime. The

most challenging for lubricant is to protect the bearing for many repetitions of small-angle oscillation with the same start and end position. One candidate from the both liquid and solid group is selected and tested in the real bearing under relevant conditions.

2. MATERIAL AND METHODS

Chapter is divided into part describing bearing and lubricant selection followed by operating and testing conditions, experimental apparatus and test procedure. Only general description of the lubricants and bearing geometry is presented as detailed information is generally not publicly available.

2.1 Operating ant testing conditions

Operating conditions selected for the bearings were derived from requirements for the complete RA. The RA is intended for operation at Low Earth Orbit (LEO), where the temperature and pressure limits are as listed in **Table 1**. Testing temperature follows the operating temperature in **Table 1**. Pressure is however different as achieving UHV (ultra-high vacuum) is technically complicated and puts high demands on the used materials and testing chamber. Only HV (high vacuum) was used for the test. Requirement for all bearing tests was set to 1x10-3 mbar (0.1 Pa). Pressure difference has no significant effect on the lubrication process and also the water vapour content is already really low – only around 15ppm, if ambient humidity in clean room is below 65 % RH. Only difference to be expected is vaporization of the lubricant from the influence of the vapour pressure of the lubricant. However, test was not planned for such long period to get reliable data from the evaporation loss. Therefore, lower vacuum was used. All tests were done in cleanroom by ISO 7 and in the thermal vacuum chamber (TVAC).

Table 1. Test conditions.

Parameter	Limit value(s)
Operating temperature	-40 / +90 °C
Non-op temperature	-50 / +100 °C
Pressure Pa	0.1 Pa
Test room cleanliness	ISO 7 by ISO 14644-1

2.2 Bearing and lubricant

Bearings were manufactured by external company with extensive heritage in space applications. Used bearing type is a ball angular contact super-duplex with split inner ring – DB configuration. Bearing parameters are listed in Table **2**.

Table 2. Bearing parameters.

Parameter	Value (s)
Outer diameter	89 mm
Inner Diameter	71 mm
Width	16 mm
Ball diameter	4.7 mm
Ring and ball material	AISI 440C
Ball complement	2x42
Tolerance class	ABEC 7T
Rings hardness	58 HRC
Race roughness (Ra)	<120 nm
Preload range defined by manufacturer	300 - 350 N
Contact angle	25°
Maximal Hertzian pressure (calculated)	0.9 GPa

Table 3. Rheolube 2000 parameters.

Parameter	Value (s)
Temperature range	-45 to 125 °C
Thickener	Sodium soap
Base oil	Cyclopentane
Kinematic viscosity of the base oil (-40 / 40 / 100 °C)	72 000 / 110 / 15 cSt
Viscosity index	137
NLGI Grade	2
Density	0.89 g/cm^3
Vapour pressure	2.4 x 10^{-8} Pa

Bearings were tested in two options – with liquid and solid lubrication to compare the two options. The best candidate from each group was selected based on the literature review. Three bearings were delivered with 1 g of Rheolube 2000 grease fill. Rheolube 2000 is sodium soap grease based on Nye 2001 MAC base oil. Available information provided by manufacturer are in Table 3. Lubricant was supplied into bearings by the manufacturer of the bearings. Bearing cages are manufactured from the phenolic-cotton and their impregnation was done also by the manufacturer. Oil used for the vacuum impregnation is the same as the base oil in grease used in the bearings.

Grease represents a liquid lubrication version of tested bearing throughout the paper.

Solid version of the lubrication for another 3 bearings from the same batch, was provided by European space tribology laboratory (ESTL). The bearings were shipped to ESTL in the form as they were obtained from manufacturer with no modifications. ESTL has cleaned and checked the bearings upon delivery. Clean bearings were coated with thin layer of the molybdenum disulphide (MoS_2) by standard PVD process established for space bearing coating. Thickness of the layer was about 1 μm and was produced on the balls and raceways on the rings. External company also redesigned the original cages and changed its material. It enables to extend the life of the bearing by providing additional material by wear process. Material for the cage was PGM-HT, which is self-lubricating composite of PTFE, glass fibres and MoS_2. Coefficient of the thermal expansion for the material is higher than for the phenolic cages. Therefore, cages were designed to be aligned on the inner ring in the lowest temperature and on the outer ring for the highest temperature.

2.3 Test apparatus

Testing apparatus was designed specifically for the testing of the output bearings in TVAC. All components and material are compliant with the vacuum level and temperatures. Schematic illustration of the test rig can be seen in **Fig. 2**.

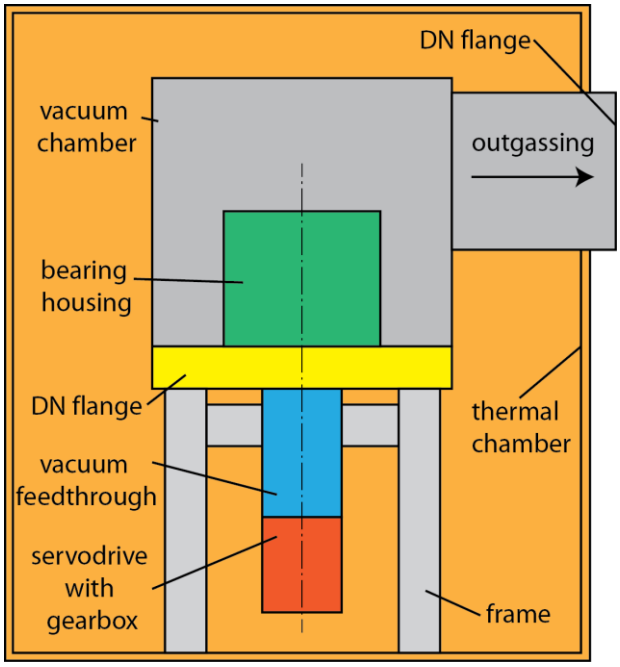


Fig. 2. Schematic illustration of the TVAC test rig.

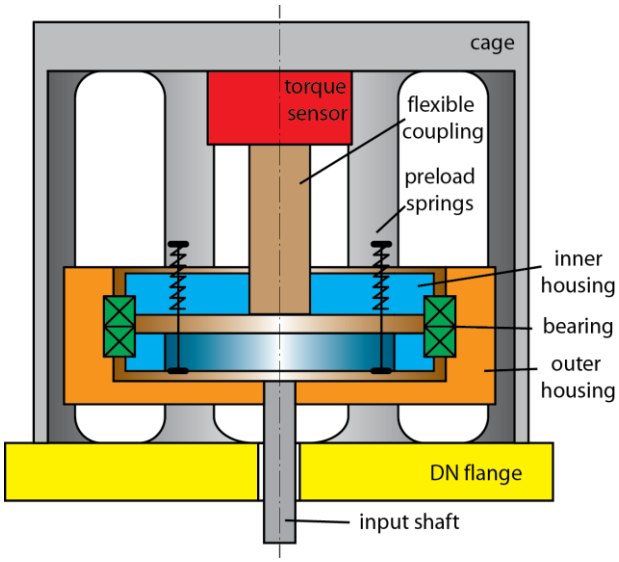


Fig. 3. Detailed illustration of the green part from Fig. 32 – bearing housing.

Concept where entire vacuum chamber is placed in the thermal chamber was used. Servo drive used to rotate bearing was attached from the bottom of the horizontally placed flange and rotary motion from the motor was transferred into vacuum chamber via magnetorheological feedthrough. Bearing can be seen schematically in **Fig. 3**. Outer ring of the bearing was rotated and inner ring was stationary. Reaction torque was measured by static torque sensor attached to the inner ring. Bearing inner ring which is split was preloaded by 3 springs (soft-preload) with total force about 360 N. Temperature of the inner ring was measured through the entire experiment by set of thermocouples.

2.4 Test procedure and cycle

Testing procedure was selected based on the worst case scenario for the bearing operation and predicted lifetime 10 years. RA works very often for the fine positioning of the antenna which consist of the movement within very small angles. Large angles are also used, but much less frequently. Therefore, small angle oscillation, with same start/end position, which is worst case from the point of lubrication, was selected to be tested. Specific angle 4° was selected to ensure, that there is no overlap between individual ball tracks on the rings – each ball travels within designated area. Total number of 4° cycles was selected to be 1 million and every step consists of 4° motion back and forth. Therefore, number of cycles is 500 000 with 1 000 000 overrollings through the track on the rings. Frequency of the

oscillations was at relatively low frequency of 1.2 Hz. It was calculated based on the planned RA rotation speed and lubrication condition. Higher frequency would increase speed and reach different lubrication regime for the bearing's contacts such as EHL. It implies, that a fluid film was not build-up during the movement, and bearing operates in boundary or mixed regime as it is expected in the RA. Testing was separated into cold and hot part. Total of 200 000 cycles was done in low temperature (-40 °C) and remaining set of 300 000 cycles in high temperature (+90 °C). Parameters are listed in Table 4. Bearings were always run-in under same load, same pressure, ambient temperature, but lower speed for 750 complete revolutions. No cleaning or re-lubrication was done after running-in.

Table 4. Testing parameters.

Testing parameter	Value
Running in (360° revolution)	750
Oscillation angle	4°
Total number of cycles (100%)	500 000
In low temperature (40%)	200 000
In high temperature (60%)	300 000
Oscillation frequency	1.2 Hz

Rotation was stopped after completion of the running-in and whole TVAC was cooled down to -40 °C. Then the sequence of 200 000 cycles was done and rotation was stopped again. Remaining 300 000 cycles was done after heating up to +90 °C. Bearing was taken out after test and disassembled for inspection of the lubricant and surfaces.

2.5 Recording of the results

Reading of the running torque was done during entire experiment but not all of the data were recorded. Total duration of the experiment was between 7 – 10 days for each bearing and sampling frequency of the torque data were 500 Hz. Size of the data would be enormous if all would be saved. Therefore, two levels of recording were established. In so-called long-time recording average, min and max value of torque for one cycle was stored. Maximum, minimum and mean torque from the past oscillation was evaluated and saved together with timestamp and temperatures. Size of the data file was significantly smaller with this approach and the most important information are still kept. Another so called short-time recording was done aside. Recording at full frequency (500Hz) was saved every 20 minutes and duration of the recording was 6 seconds. This recording of several cycles was included to allow later detailed analysis of the motion.

3. RESULTS

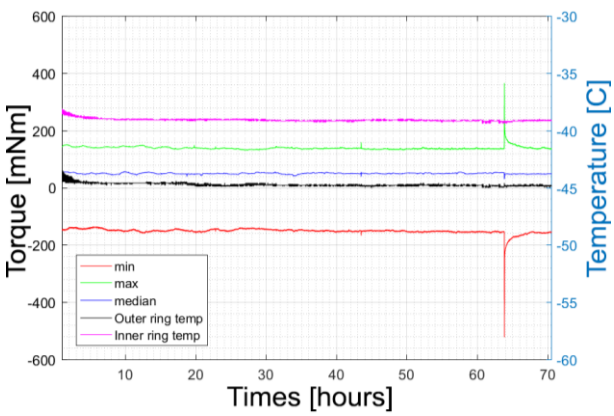
Results presented in this chapter are divided by the lubricant type to Grease and MoS_2. Three bearings were tested with each lubricant type, but results from one representative are shown as there were no important differences. There is long and short data record for each lubricant type as was described in previous chapter (2.5).

3.1 Grease lubrication

Bearings were tested with fill around 1g of the Rheolube 2000 and were equipped with phenolic cages. Running-in was done for all bearing as described before.

3.2 Grease - long record

Results from the low temperature test are shown in **Fig. 4**a. Sharp peaks seen at 64 hours are caused by the pausing of the motion because of the maintenance of the cooling system. It can be seen that peak values of the torque are reaching to 0.5 Nm. However, such high values can be seen only when motion is stopped for longer period, nevertheless it is real value that has to be counted with. Usual maximum can be seen around 180 mNm with nominal values around 30 mNm. Total duration of this part of experiment was slightly over 70 hours.



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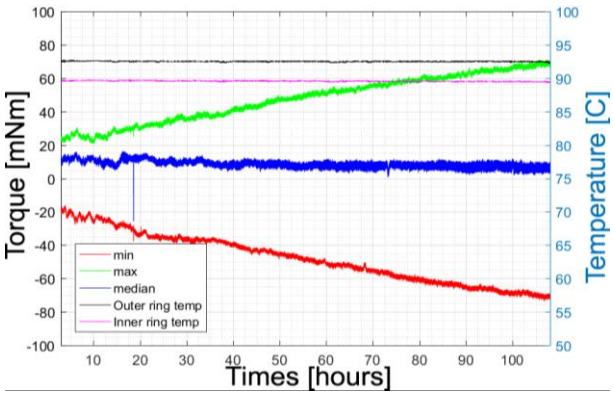
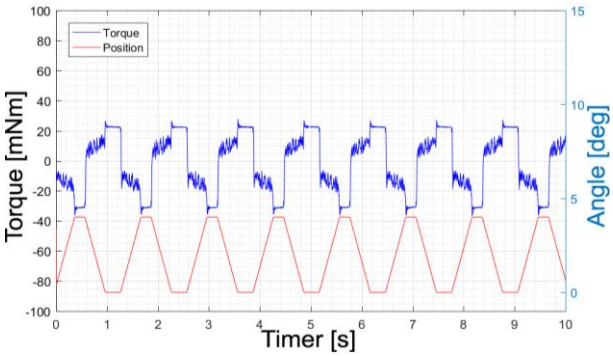


b)

Fig. 4. a) Rheolube lubricated bearing torque in cold, and b) hot temperature over entire test.



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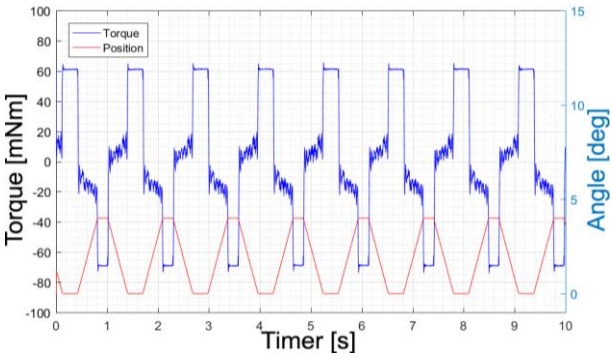


b)

Fig. 5. a) Fast record from the start, and b) end of the experiment in Fig. 4b.

Torque evolution from the high temperature can be seen in **Fig. 4**b. Most visible aspect in the plot in **Fig. 4**b is gradual magnitude increase of maximum and minimum torque with cycles. Absolute maximum value started at around 20 mNm and increased towards 70 mNm in the end of the experiment at 110 hours. However, it looks that only small part of motion is increased, as median is remaining almost without change at around 10 mNm. Behaviour can be seen in detail in **Fig. 5**. Torque remained basically without change during the motion, but spikes take place at the end of the motion and it seems as the bearing is left with tension after motion as torque remained high even during static no-motion time. This behaviour is interesting, however, not crucially important for the operation of the bearing. Even the maximal values observed at the end of the experiments did not reached the half of the values seen in the cold temperatures. Therefore, it was investigated no further. However, there is hypothesis established in the discussion chapter.

3.3 Grease - short record

One of the critical moments was increased friction torque during the first motion after cooling down. It was expected that the lubricant flows into the track during ambient temperature stage, and becomes an obstacle, hard to overcome, when its viscosity significantly increases after cooling down. Results of the first cycles after cooling can be seen Fig. 6, where absolute maximum value of the torque is around 0.3 Nm. Torque was afterwards decreasing gradually when motion was established for the next cycles and lubricant was pushed sideways by rolling motion. Similar behaviour can be caused also by the lubricant filling the gap between cage and rings or/and balls.

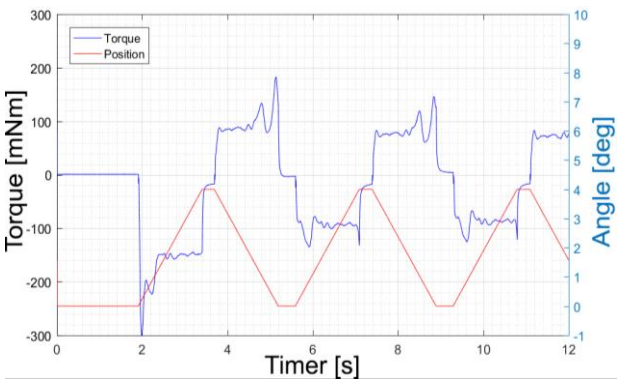


Fig. 6. First cycle after cooling down at static position.

Another so called diagnostic type of test was done during the experiment. Influence of the rotational speed to the frictional torque was conducted for low and high temperatures. Also standard deviation (SD) of the motion part without final peak was evaluated during individual cycle. SD parameter was used as the evaluation of the bearing noise or the fluctuation of the moment during the steady motion. This should be kept low since it is one of the key parameter influencing final precision and resolution in pointing applications.

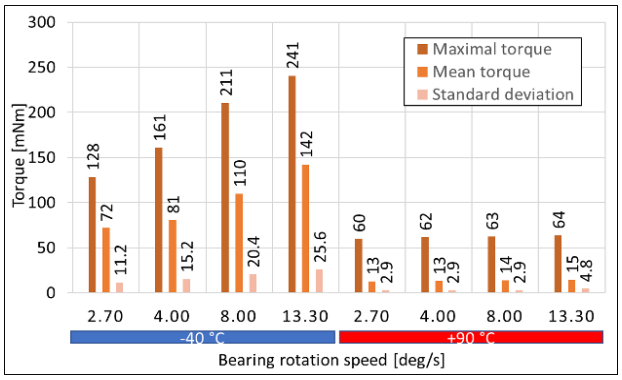


Fig. 7. Diagnostic results for grease lubricated bearing at different rotational speed.

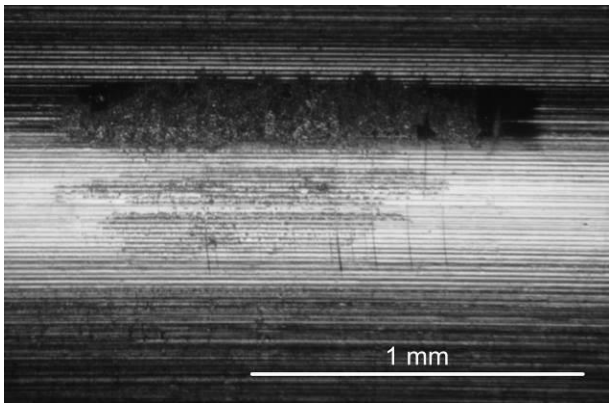
Results from the diagnostic cycles for various speeds at two different temperatures are shown in Fig. 7. Maximum and nominal torque can be seen for each speed and temperature together with SD. Values presented in Fig. 7 were obtained at the end of the experiment, where bearing was already slightly worn. The values are slightly higher than ones found at the start of the experiment. However, highest values found throughout the test are considered to be most important, so only values found at the end of the test are shown. There was evidently higher torque in the low temperatures compared to the high temperatures. Also, there was an influence of speed on torque at low temperatures, where viscous flow is one of the major resistance sources. Resistance due to viscous behaviour was increasing with increasing speed as one would expect. This increase was not visible at high temperatures where frictional torque comes from contacts which are operated in a regime where there is no significant effect of speed. It can be either boundary lubrication regime or even high pressure EHL lubrication where speed increase does not lead to significant increase of friction.

3.4 Grease - bearing inspection and surface analysis

Bearing was disassembled after the experiment and inspected. Firstly, visual inspection was done to check the state of the lubricant and presence of the particles. No particles were found and colour of the lubricant was still pure white - without change. Lubricant condition can be seen in the **Fig. 8**.

Further inspection was done with optical microscope and optical profilometer used to capture surface texture. Newly created solid tribolayer was found in several places where there was a running track between rings and balls. It can be seen in Fig. 9a.

Fig. 8. Condition of the lubricant after the experiment – can be seen mainly on the cage.



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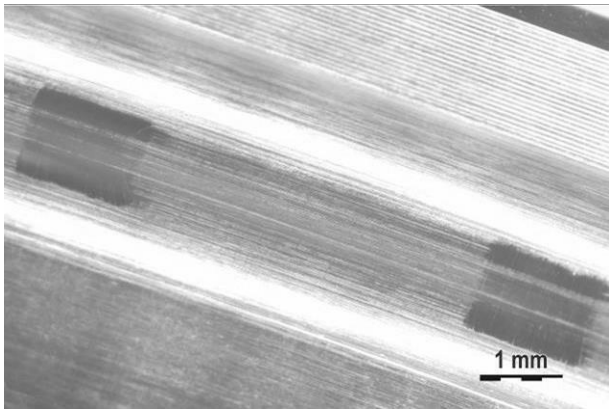


b)

Fig. 9. a) Solid film on the inner ring raceway, and b) change of the texture in the contact areas.

Full length of the layer was estimated to be 1.4 mm. This roughly corresponds to the length of the rolling track of the individual contact in the bearing. There was the same layer in smaller magnification showing of the texture change on the bearing inner ring in Fig. 9b - darker rectangles.

Layer deposited on the surface was very thin and hard to remove or even scratch. Three dimensional map is shown in **Fig. 10** and height

was evaluated to be around 1 μm above surrounding surface. Height of the feature was roughly equal to the original turning texture. Chemical composition of the layer is not known.

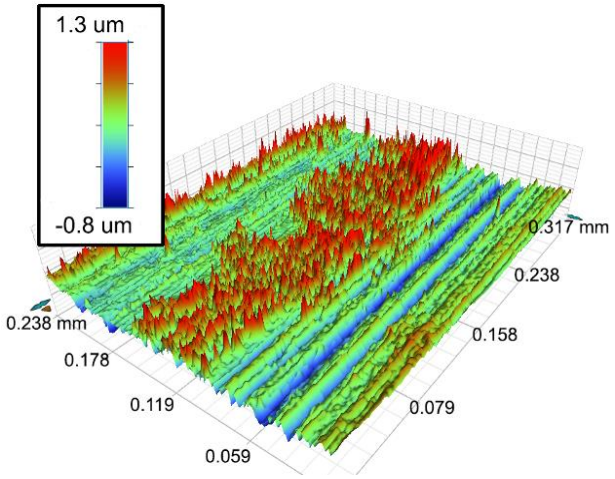


Fig. 10. Surface map of the spot from Fig. 9a.

Bearing cage was also examined, but no signs of wear or surface changes were found, so images are not presented here.

3.5 Solid lubricant

Solid film of MoS_2 was coated on all contacting surfaces for the same bearings as it was used in the previous chapter (Grease lubricant). Original supplied phenolic cages were replaced by PGM-HT, which is a material compatible to MoS_2. Test procedure was kept mostly the same, with one exception of running in.

Running in was done according to the instruction of the coating supplier. Bearing was loaded with specified preload and rotated continuously in the vacuum environment under ambient temperature. Fast rotation by ±1080° with speed 30 RPM was done several times in row. Slow rotation with speed 6 RPM was done after that to measure the torque. Bearing torque was very high from the start (around 150 mNm), but it was decreasing with every next repetition. Limit for the torque was set to 60 mNm. It was expected that torque will decrease to this value eventually. Bearing had to be run by described procedure for several thousands of revolutions before torque was found under the limit. Bearing was taken out from the test rig and flushed with isopropyl alcohol (IPA) to cleanse all free particles loosen from the film.

3.6 Solid - long record

Results from the high temperature test are shown in **Fig. 11**a. It can be seen that torque was gradually increasing over entire 90 hours of the experiment, but tended to stop the increase and even up at the values under 200 mNm. Mean value was not changing significantly, only noise increase could be observed from around 50 hours (blue curve).

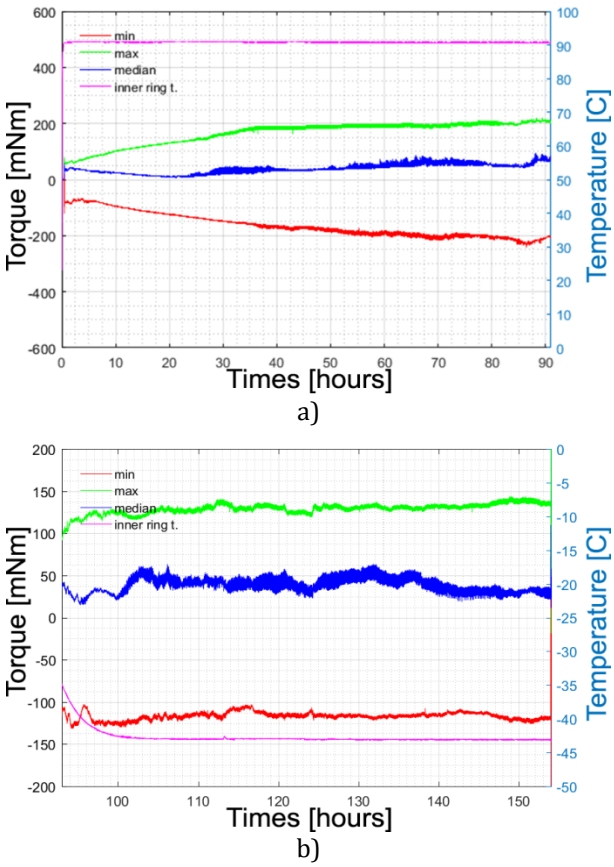


Fig. 11. a) MoS2 lubricated bearing torque in high, and b) low temperature.

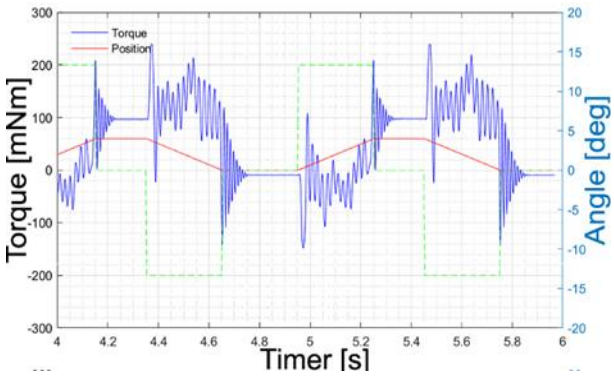
Experiment continued after cooling down of the entire test rig and results for low temperature can be seen in **Fig. 11**b. Torque did not change any more, but scatter of the mean values increased even more. It can be seen that absolute values of the torque were not influenced by the temperature significantly. Maximum values even for low temperatures were under 150 mNm, which is lower than grease lubrication in low temperature. Higher values of the torque observed during low temperatures can be however influenced by the permanent changes in the bearing done during previous high temperature test rather than by temperature itself. Variations in the mean value

can suggest progressing changes in the MoS_2 layer such as releasing new free particles and relocation of existing ones. It can also lead to overall increase in the running torque.

3.7 Solid - short record

In short recording, depicted in Fig. 12a, it was found, that maximum and minimum torque was occurring mainly at the start and end point of the ball's track. Peaks can be observed at the position corresponding to 0° and 4°. Interesting fact is, that peaks at these positions were observed even when motion was changed – for example between +2° and +6°. Peaks can be seen in Fig. 12b at the 4° position, which was original end point. It was suggested that peaks related to the position can be caused by debris build-up at that part of the rings. It was found, that torque peaks could be smoothed or decreased by so called regeneration cycles, where the bearing is rotated continuously for several full rotations.

Regeneration cycle was done with three full revolutions of the bearing and the same test as in Fig. 12a was repeated subsequently. Result can be seen in Fig. 12c where the same transition through 0° or 4° position does not show such peaks in the torque and even noise is lower.



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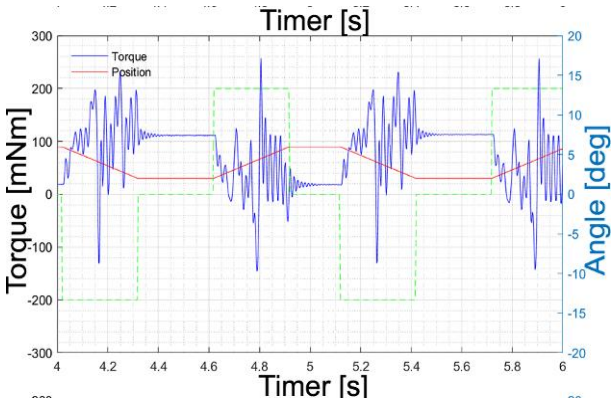

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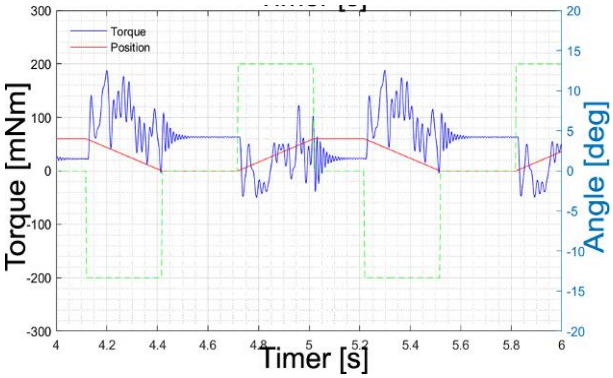

c)

Fig. 12. a) Torque through the oscillation cycle; b) change of the oscillation position; and c) torque after regeneration cycle.

There seems to be positive offset of the zero value equal to roughly 50 mNm for torque data in Fig. 12. Torque should be roughly symmetrical. This is most probably caused by improper sensor calibration as seen often during experiment. Relative changes in the torque are correct.

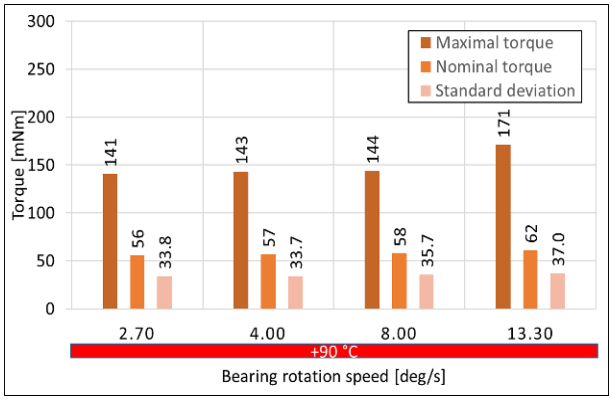

Fig. 13. Diagnostic results for MoS2 lubricated bearing for different rotational speed at the end of the entire experiment.

Influence of the speed and temperature was also evaluated for MoS_2 lubrication in the same way, as in liquid lubrication. However, the influence of the temperature was found negligible, so it is not shown in the plot. Only results in 90 °C are plotted in **Fig. 13**. No relationship with the speed can be seen, which is an expected behaviour for solid lubrication. Results of the SD for new bearings are no presented here, but values in the **Fig. 13** are roughly 3-4 times higher than ones found at the start of the experiment. SD parameter was observed to be between 7-13 mNm for the freshly run-in bearing. Therefore, increase of the torque noise represented by SD parameter is quite severe.

3.8 Solid - bearing inspection and surface analysis

Inspection was done after the experiment in the same way as with liquid lubricant. Bearing was disassembled and visually checked for presence of free particles, that would suggest excessive wear. However, only few particles were found. It was within expected amount and in much smaller scale compared to running-in. Bearing was cleaned by IPA and soft tissues and inspected. Image of the inner ring can be seen in **Fig. 14** – no wear was observed at this scale.

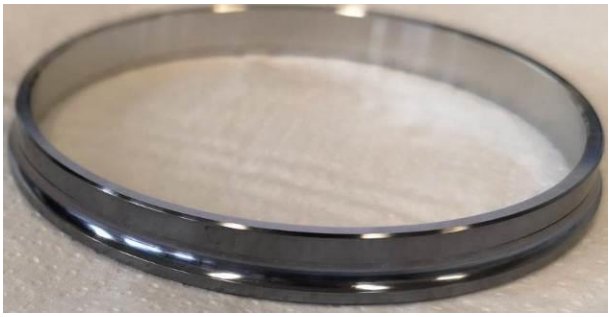

Fig. 14. Inner ring with MoS_2 coating after the experiment.

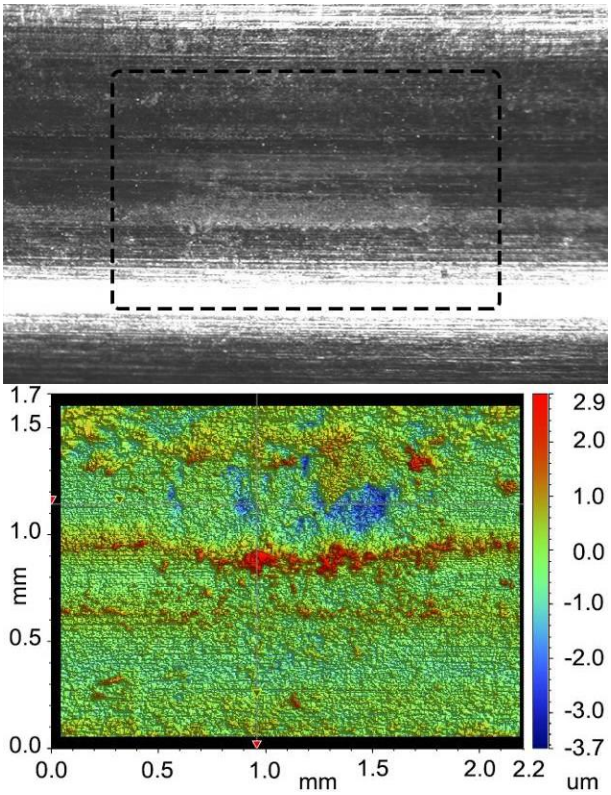

Fig. 15. Detailed topography of the inner ring (MoS2) with transferred film.

Inspection by use of the microscope and optical profilometer found small area in rolling track and wear patterns, that are clearly visible in top of **Fig. 15**. This patterns were present at places with spacing according to the spacing of the balls in the bearing. Therefore, it can be expected that they were created during experiment. Its topography can be seen in bottom of **Fig. 15**. MoS_2 lubricant acts as soft thin film protecting the surface against wear. It can be redistributed during the lubricating process as it can be observed in **Fig. 15**, where the redistribution of MoS_2 particles forms holes (blue areas) and build-ups (red areas). Lubricant is simply transferred from one area to another and there was seen no global loss of MoS_2 coating due to wear mechanisms when comparing area in the former ball-track to places outside the track. However, downside of the redistribution process is increased noise in the bearing operation which was observed during later stages of the experiment. No additional wear that would suggest damage of the bearing steel was found. All features sizes were found equal or lower to the thickness of the MoS_2 coating.

4. DISCUSSION

Increase of the torque during both tests (liquid and solid lubricant) is the main aspect observed during the experiments. Although, that trend is similar, it is not identical. Different explanation for each lubricant option is proposed.

4.1 Grease lubrication

Most probable hypothesis as seen by the authors is degradation of the small volume of the grease present within the rolling tracks. Velocity profile with speed increase and decrease provides wide range of operating conditions as seen in general in Stribeck's curve. However, boundary lubrication regime usually found at lowest velocity is not visible from the start of the current experiment. It was shown that grease thickener can provide higher film thickness and lower friction compared to pure oil lubrication [11]. However, the thickener ability is limited by its life. Thickener fibres are constantly sheared during repeated cycles, degraded and are losing its properties [12]. Limited volume of the lubricant is degraded more rapidly without supply of the new lubricant caused by small reciprocating motion. That could explain steady increase of the torque visible during the

acceleration and deceleration parts of each cycle – it is slow transition towards boundary lubrication followed by higher contact friction.

Topography features observed during post-test inspection are most probably connected to the natural lubricant behaviour and are not to be concerned with. Deposits found in grease lubricated bearings originates likely from EP additives in the grease base oil which would only proof their function. Function of EP additives was explained by Mistry et al. [13] or by Johnson and Hils [14].

4.2 Solid lubrication

Solid film of MoS_2 does not contain anything like EP additives, but film material itself is soft and can be transferred easily which is known behaviour and reason for the use of the PGM-HT material for ball cages. Transfer of MoS_2 was described for example by Suzuki and Prat [15].

Steady increase of the torque during the test with MoS_2 coating is surely of different origin than in grease lubricated bearings. Most probable reason is slowly increasing volume of redistributed MoS_2 particles across the rolling surfaces. Surfaces are getting more "bumpy" which leads to increasing bearing noise as presented by SD parameter. Increase of the MoS_2 film thickness at some places can cause additional rolling resistance and thus overall torque resistance of bearing. Similar behaviour can be observed during running-in. However, there is full rotation of the bearing and mostly just releasing of the free particles rather than relocation. Short 4° movements during major part of the test is probably too short to provide re-lubrication from the cage. However, number of cycles is probably small enough to be supported only by MoS_2 layer on surfaces. It is not easy to separate the effects at this point.

Second, probably combined with previous explanation, can be gradual change of the ball-race conformity resulting in increase of the rolling resistance. Conformity of the raceways can directly influence the bearing performance and life – see [16,17] for details. Free particles and lubricant can conglomerate at the edges of the rolling track and change the effective radius of the raceways. This can eventually sufficiently alter the geometry and cause increase in the torque. Evidence supporting hypothesis is observed change of the topography and fact that regeneration cycles implemented in the testing procedure partially decreased the peaks in the torque gradually building up. Also, peaks were observed only at the peripheral positions during the motion. Peaks occurrence remained connected with the specific positions of the bearing after build-up and even change of the oscillation position does not change occurrence of the peaks as can be observed in **Fig. 12**.

Important with such short stroke is also realization that there is no possible resupplying of the lubricant from the cage as in case of full revolution and primary ring-ball contacts are reliant strictly on the lubricant present in the contact area with low to none resupply. However, cages should contain lubricant anyway (impregnation or particles in structure) for the reason of lubrication ball-cage contact.

5. SUMMARY AND CONCLUSION

Study was focused on the testing two sets of bearings for specific use in short oscillatory rotation found in RA. Lubricants were selected specifically for slow short repeated motion and represents suitable choice from both categories – liquid and solid lubricants. MoS_2 lubricant was picked as representative of solid lubricants and Rheolube 2000 as representative of liquid lubricants. Selection was done based on available literature.

Short 4° cycles were done with 500 000 repetitions at roughly 1 Hz frequency in TVAC. Part of the test was done at -40 °C, and the other part was done at +90 °C. Temperatures represents extremes of described application of RA at LEO. Main observed parameter is torque and torque noise, but also state of the lubricant and bearings were assessed at the end of each test.

Main result is that none of the bearings actually failed or was damaged, and both options could be used. All results indicate only minor changes in the topography and increase of the torque. All tests were accompanied by steady increase of the bearing torque and noise. However, because torque values didn't exceed 200 Nmm for slow cycles at speed 2.7 mm/s, they were assessed as acceptable for the mechanism. Based on the

surface and lubricant condition it is expected that bearing could sustain the operation for significantly longer period than tested.

Rheolube 2000 was selected as final candidate for the lubrication in RA bearings. Main reasons are:

- Lower bearing noise (important for the precision).
- Overall low torque (except for low temperatures, where MoS_2 outperforms).
- More convenient for testing (MoS_2 has limitation for in-air operation).
- Lower price.

All downsides of the selected lubrication option were considered and mitigated or accepted.

- Evaporation (volume of grease has to be calculated for the lifetime).
- Design demands for labyrinth seals (strongly recommended to decrease evaporation).
- Temperature limitations (Exceeding of the temperature range can lead to severe loss of lubricant by evaporation or exceeding of the torque limits).
- Necessary impregnation of the cages to decrease oil loss.
- Use of creep barriers

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