DEVELOPMENT OF DRY LUBRICATED HARMONIC DRIVES FOR SPACE APPLICATIONS ("harmLES")

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1. ABSTRACT

Today, Harmonic Drive® gears are used in several space flight mechanisms as they provide advantages like zero backlash, a high gear stiffness and a high transmission accuracy. In most cases those gears are used in grease lubricated condition, whereas this is always linked to the risk of outgassing and limits significantly the operational temperature.
In order to increase the temperature range, trials to apply solid lubricants to Harmonic Drive® gears, as commonly used for e. g. bearings, were performed. Based on these trials it was found that the gears can be operated even at -269°C. Anyhow, although being used in various cryogenic applications, the reachable lifetime is comparably short. Hence the EU – funded project harmLES was started in 2011 in order to increase the accessible lifetime by developing a new Harmonic Drive® gear type. This activity is based on an integrated approach covering gear design, materials and coating.

2. STATE OF THE ART

1.1. SURVEY OF COATINGS AND CONCEPT

One major part of development is an appropriate solid lubricant coating. So, as first part of the project, a selection on candidates of solid lubricant coatings was done. For space application the most commonly used solid lubricant coatings are based on MOS2 done by PVD. They have wide heritage in bearings. Besides MoS2, WS2 is almost similar in structure, but no commercial supplier is known, except from Dicronite. This type of coating however is different in it's simple process but offers only a very thin coating leading to limited life times. MoS2-coatings are also offered with dopants. There are some commercial suppliers offering such coatings like TEER (UK) and HOHMAN (USA). TECNALIA a Spanish research entity offers a composites coating based on MoS2 and WC, which has shown already flight heritage in "SPACELAB" [2]. As general alternative, DLC-based coatings might be of interest. Their major concern is low friction under vacuum. Research in the early 2000 have raised a concept on DLC-coatings offering low friction in vacuum based on very high hydrogen contents. Results were obtained within an ESA-project and were published at ESMATS. However, that research has been ceased and no commercialisation was achieved, hence it is not available anymore. A review on currently offered DLC based coatings revealed only a few promising candidates. Three of them were procured and tested in pin-on-disc tests under vacuum and proved to have no promising low friction under vacuum. Moreover, a second restriction was identified: the roughness of the toothing in harmonic gears was assessed as being very high for application of DLC (which is a hard coating working best under lowest roughness).

Hence, from this first iteration, the most promising concept for solid lubricant coatings was concluded to be MoS2 as pure or as composite coating.

1.2. TRIBOLOGICAL TESTING - STRATEGY

As mentioned MoS2 coatings are well accepted for bearing applications. It is also well accepted to use as first test method on material level so-called pin-on-disc tribometers (PoD). Typically the coating is applied to the discs, and commercial bearing balls are used as counterpart. Depending on application in most common tests, unidirectional motion is used.

On the other hand, the motion in the toothing of a Harmonic Drive® gears is not unidirectional but reciprocating with small amplitudes, in range of 0,1- 0,5 mm.

During the project "harmLES" on material level a twofold strategy was selected: Pin-on-disc tests were performed to firstly validate the lubrication effect and to get data for comparison to data existing from other publications typically based on PoD-testing. Secondly, a so-called fretting test device developed by AAC was used to enable validation of the coatings towards small oscillating motion (0,1mm). It also enabled accelerated testing as 10 Mio revolution can be performed in only 3 days.

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1.3. FRETTING AND MOS_2

AAC developed a test method for investigation of cold welding under fretting. It uses a pin-on-disc-configuration similar to PoD-tribometers, but in an oscillating movement of only 50µm amplitude which referred to as “fretting”. The aim was to assess materials and coatings for their ability to prevent cold welding (especially under upper stage during launch when strong vibrations may lead to fretting when the spacecraft is already exposed to space.) Based on the device a method was developed, agreed by ESA and published as STM paper [3]. In short, testing is done under fretting motion of 200Hz with an amplitude of 50µm. the loads are selected to achieve a maximum Hertzian pressure of 60% of elastic limit at begin of the test. Several studies on cold welding on typical space material combinations and coatings were performed. A summary is published in [3], but data is also available online [4].

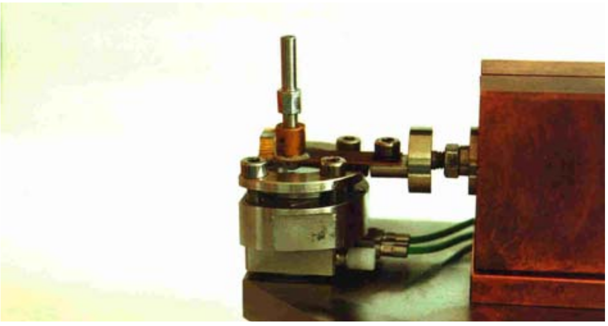

Figure 1: Fretting device: Detail showing the fixation of pin (upper rod) and disc (mounted directly on a force transducer).

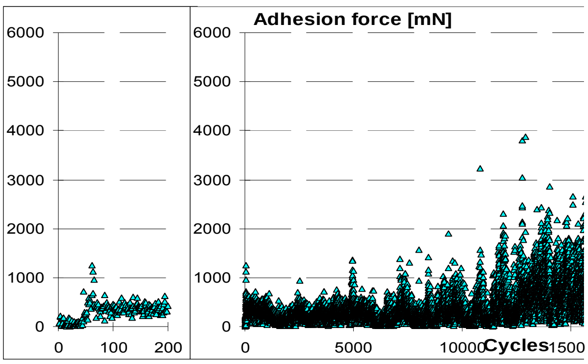

Figure 2: Adhesion force as function of cycles: Detail on the left shows that lubrication of pure MoS2 is lost after 50 cycles = 100.00 revolutions (1 cycle = 2000revs).

As the motion of teeth in a harmonic gear is more close to that fretting motion than to usual pin-on-disc tribometers (minimum are several millimetres of stroke), the review on candidate coatings was extended to data from these studies. Although there is no data on the steels and roughness needed in this project, data on quite similar combination was found: SS17-7PH coated with standard MoS2 with a slightly lower roughness of Ra~0,1-0,2µm was found. They show that MoS2-coatings work under these conditions, but their lifetime was less than 100.000 revolutions which is much less than known for bearing applications (range of millions) [STM-279]. SEM-Analysis after the full test shows that the MoS2 coating is completely worn and severe adhesive wear is found.

3. COATING DEVELOPMENT

2.1 PROCESS AND TRIBOMETER TESTS

Following the tribological experience on MoS2 coatings towards fretting, the main focus was put on composites coatings based on process by TECNALIA. A CemeCon CC800/8 magnetron sputtering PVD unit was used for the deposition of the alloyed MoSx-WC films. The MoSx-WC coating (~1.9 µm thick) was applied onto different steels: AISI 440C (Ra≤ 0.025 µm and Ra=0.2 µm), CRONIDUR®30, SS17-4PH and SS17-4PH with pre-treatment. The microhardness tests on these coatings showed hardness values under a maximum load of 10 mN between 4.1 to 4.9 GPa, showing a small variation. These small differences in coating hardness among the different substrates could be explained by the different substrates hardness and roughness.

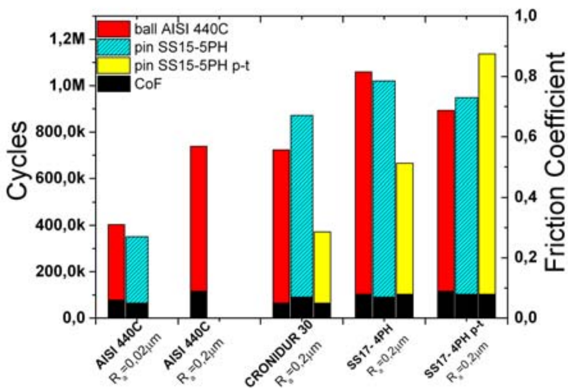

Figure 3: Endurance and average CoF of the MoSx-WC coating measured on the different pin-on-disc tests. Coating reference: MoSx-WC (25988, ~1.9 µm). Substrates: AISI 440C (polished); 440C, CRONIDUR®30, SS17-4PH and SS17-4PH with pre-treatment (Ra= 0.2 µm). Pin/ball: AISI 440C, SS15-5PH, SS15-5PH with pre-treatment (polished).

A MT2000 Tribometer (Pin-on-Disc) was used to carry out wear and friction tests at controlled humidity (10%), applying a load of 5N to several steels ball/pin (AISI 440C, SS15-5PH and SS15-PH with pre-treatment) of 6 mm dia. (Ra ≤ 0.025 µm) at a constant sliding speed of 0.5 m/s. The test criterion was to stop the test when friction coefficient was higher than 0.4 due to delamination or fast depletion of the film and its lubricating protection. The endurance life of the films and the average CoF are shown in *Figure 3* for the different pin-on-disc tests.

Main conclusions from these tests are:

- Low friction (CoF ≤ 0.09) and long endurance is observed in all the tests.
- Surfaces with small roughness (Ra= 0.2 µm) show longer endurance levels than the polished surface (440C disc, Ra<0.02 µm). The endurance increases between 2 and 3 times when testing discs with non-polished surfaces of Ra= 0.2 µm. This is in agreement with the tribological behaviour of soft solid lubricants (MoS2 based coatings, ~1 µm thick) that shows the maximum endurance on surfaces with roughnesses between Ra= 0.1 to 0.2 µm. Endurance levels on polished surfaces are lower and when increasing the roughness over 0.3 µm there is a dramatic decrease of the endurance.
- The highest endurance when testing against AISI 440C balls is observed against the coating deposited onto the SS17-4PH disc. Other contacts (excluding the polished AISI 440C disc) show less difference on endurance between them.
- The highest endurance when testing against the SS15-5PH pin is observed against the coating deposited onto the SS17-4PH disc. In any case, the other contacts (excluding the polished AISI 440C disc) show slightly lower endurance levels than this one.
- The highest endurance when testing against the SS15-5PH with pre-treatment pin is observed against the coating deposited onto the SS17-4PH with pre-treatment disc. Unlike the other testing pins (AISI 440C and SS15-5PH), the influence of the substrate is very strong on the endurance of the coating when testing against the SS15-5PH with pre-treatment pins.
- When considering the three contacting materials without any pre-treatment (CRONIDUR®30, SS17-4PH and SS15-5PH), the coating onto CRONIDUR®30 and SS17-4PH disc show the highest endurance when tested against the SS15-5PH pin.
- When testing against the SS17-4PH pre-treated disc all the test show high endurance values. The highest endurance value in the entire tests is obtained when having a pre-treatment in both contacting material (the SS17-4PH with pre-treatment disc + MoSx-WC against the SS15-5PH with pre-treatment pin).
- If only one steel should be pre-treated in the contact SS17-4PH against SS15-5PH, it is better to apply the pre-treatment onto the SS17-4PH substrate. But, this should be confirmed by testing SS15-5PH pre-treated discs against SS17-4PH pins.

2.1 FRETTING TESTS

In a first fretting testing sequence, MoS2-based coatings were applied onto steel discs of type and roughness similar to the harmonic gears (in general the used steels like SS17-4PH are softer than bearing steels and the roughness is much higher compare to bearings, Ra varies from 0,2 to 0,4µm). In first fretting tests against uncoated pins made of SS15-5PH at Ra~0,6µm, it was found that pure MoS2-coatings show a proper solid lubrication and an initial phase without adhesion, but their lifetime was found to be low, only 350.000 revs.

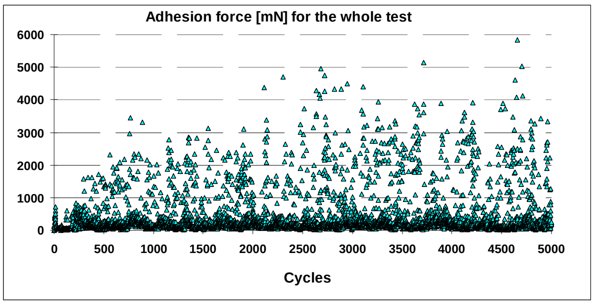

Figure 4: Adhesion as function of cycles (1cycle=2000revs) between SS17-4PH with pure MOS2 coating in contact to blank PH15-5PH pin

At this stage also the machining of the harmonic drives could be optimised, enabling now a roughness of approx Ra~0,2µm. As main candidate, a composite coating based MoSx/WC developed by TECNALIA and flown on SPACELAB, was tested on reduced roughness. Adhesion was found but being lower than for uncoated steel combination. Increasing thickness of the coating from ~1,8 to ~2,8µm did further reduce the adhesion and increase life time to ~7 mio revs.

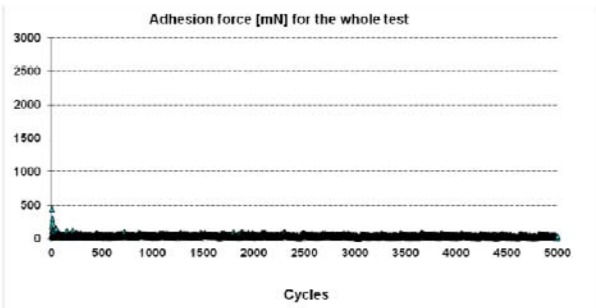

Figure 5: Adhesion as function of cycles between SS17-4PH with pre-treatment and MoSx/WC coating in contact to PH15-5PH pin (1cycle =2000revs)

Finally, a pre-treatment was done on the steel before coating. This revealed the best performance in fretting: test was stopped after 10mio revs without adhesion /

cold welding. This means that the solid lubrication mechanism was still working. Post-Investigation by SEM showed a proper tribo- layer consisting of MoSx/WC.

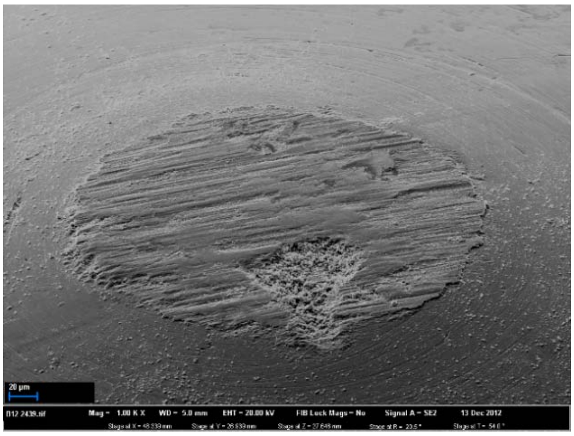

Figure 6: Contact zone of pin SS15-5PH after test to MoSx/WC coating showing a well developed tribo-layer

4. DEVELOPMENT STATUS ON GEAR LEVEL

Within the harmLES project, the first year was mainly focussed on principal investigations based on Pin on Disc (PoD) testing. The results of these studies were used as input for the re - design of the Harmonic Drive® gear, which is basically performed in year two and three of the project, which will end beginning June 2014. Therefore the re – design of the gear is not closed yet. Anyhow, in the meantime detailed analysis concerning potential starting points for the optimisation of the gear have been executed. The results were transferred to a development approach, including several systematic steps for the implementation of potential improvements. Today, the manufacturing of the first test gears has been finished and initial testing on gear level has started. The results of the analysis, the development approach and the first test results shall be presented herein.

5. REQUIREMENTS

The Harmonic - Drive® gear today is in its geometry optimised towards grease lubrication for terrestrial applications. So far the approach for the use in space was to copy the gear design, whereas materials and lubricants are substituted by space feasible products. This works very well for numerous grease lubricated space applications. Anyhow, driven by the duplicated design (e.g. tooth shape) the gears are often oversized with respect to their mechanical properties, whereas with respect to the special tribological demands the design might be improved. Therefore, the possibility to reduce mechanical properties leaves the opportunity for an optimisation of the gear design especially towards the needs of dry lubrication. Anyhow, as target for the development the main gear characteristics of the dry lubricated gear shall be kept close to those of standard gears, whereas only the basic properties shall be mentioned herein.

The transmission accuracy shall remain below 60arcsecs as specified for terrestrial gears. It is the same for the repeatability which is specified with a maximum of 3arcsecs, which is in line with the standard. The minimum gear stiffness is foreseen to be at a level of approx. $1.1*10^4$ Nm/rad which is close to the stiffness of a HFUC gear size 20. Beside quantitative properties, a main qualitative feature of the Harmonic - Drive® gear is seen in the zero backlash, which is a decisive advantage compared to other gear types. It is anticipated to keep this property although the development becomes even more demanding, as the zero backlash introduces an internal gear preload. This results in a contact stress within the tribological contacts, even if no external load is applied to the gear.
With respect to operational demands, the lifetime is planned to be at a level of 17,000 output revolutions at an output torque of 4Nm. Additionally the achievable lifetime at higher loads (e.g. 16Nm) will be considered. The operating temperature will be between -200°C and +150°C.

6. ANALYSIS AND SIMULATION

The re - design of the gear is guided by two main aspects, which are on the one hand various test results with dry – lubricated gears gained at Harmonic Drive AG in advance to the harmLES - project. On the other hand the necessity is seen to gain a deeper understanding on wear mechanisms based on detailed analysis of the tribological contacts.
Driven by a request for a special cryogenic application, in 2009 an internal test sequence with dry lubricated Harmonic Drive® gears was performed. As result, the toothing and the contact between Flexspline (FS) and Wave Generator bearing (WGB) outer ring were identified to be the life limiting tribological contacts for coated gears. Additionally, the tests suggested the assumption of a strong dependency between the gears lifetime on the internal pre – load. Therefore the basic ideas for the gear re – design are to reduce the contact stresses and the sliding path within the above mentioned contacts.

As starting point for the gear development a CobaltLine – 20 – 100 gear was chosen. The CobaltLine gear type is the advancement of the HFUC series providing higher torque capacity and longer lifetime. As this configuration provides as well advantages for dry lubrication, it was decided to benchmark with this gear type.

In order to confirm the development approach the benchmarking gear configuration has been subjected to detailed analysis concerning the maximum Hertzian contact stress and the sliding path within the mentioned tribological contacts. Additionally, driven by the need to develop a backlash free gear, the assumed dependency between contacts stress and internal gear preload was investigated.

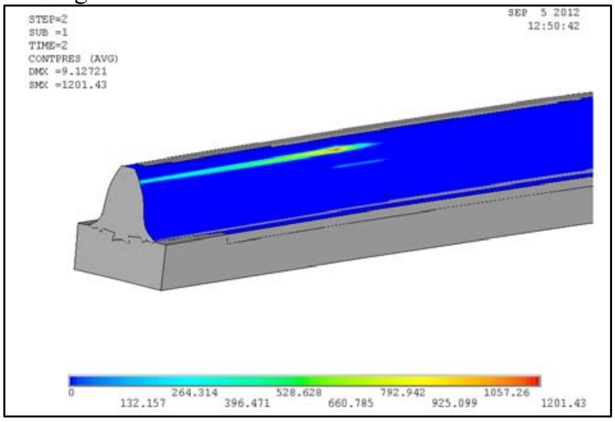

Figure 7: FE Submodel of FS tooth

The inputs for the determination of the peak Hertzian contact stress were derived via the solution of the full FE-Model of the Harmonic Drive® Gear. This was followed by the resolution of respective submodels for each tribological contact, providing the contact forces. As example, Figure 7 shows the submodel of the FS tooth. With the knowledge of contact forces and shape of the contact area, the Hertzian contact stress could be derived. The results of the calculations at an applied output torque of 24 Nm are shown in **Table 1**. As mentioned, the dependency on the internal gear preload (herein assigned as light, medium and hard) is given as well.

Internal gear preload	light	medium	hard
Maximum Hertzian contact stress [MPa]			
Within toothing	1300	2000	2300
Between FS and WGB	2500	2100	2200

Table 1: Results of calculation of peak Hertzian contact stresses

For both contacts a stress level was identified for the benchmarking gear type, which is above the loads typically applied to coatings used for vacuum applications [2]. This result confirms on the one hand the experiences made within tests so far. On the other hand it underlines the necessity to adopt the gear design in terms of contacts stresses towards the needs of dry lubrication.
Another finding is that the contact stress especially within the toothing is strongly dependant on the internal gear preload, highlighting that for the new design special attention should be paid to this dependency.

The sliding path within the toothing is derived via a Matlab® tool allowing a 3D simulation of the tooth engagement. An example is shown in Figure 8, whereas the blue outlines represent one Circular Spline (CS) tooth. Each outline stands for one cross section over the length of the tooth. The red outlines correspond to the FS which is depicted fully engaged into the CS.

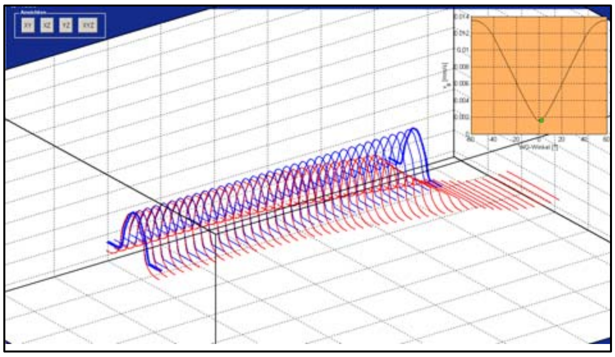
Figure 8: Tool for simulation of tooth engagement

Based on this tool, the sliding path for the benchmarking gears was determined, giving a result of 0.10mm per tooth engagement. With a first optimisation of the tooth profile, followed by an optimisation of the gear kinematic this value could be decreased significantly.

7. DEVELOPMENT APPROACH

The gear development itself is seen under two aspects, which is on the one hand the development of an appropriate coating, on the other hand the geometric optimisation of the gear. Following the schematic shown in Figure 9 the two aspects are followed in parallel paths. The starting point for the activities is a benchmarking gear which is the commonly used CobaltLine – 20 – 100 gear designed for grease lubrication.
The geometric adaptations will be performed in two subsequent steps, whereas in a first step the internal gear preload will be reduced, keeping the commonly used tooth profile. In a second step the optimised tooth profile will be tested with respect to its influence on the gear lifetime.
The development of the coating on gear level is mainly linked to the results of lab testing described in section 2 and will as well be performed in two different steps. In a first configuration the impact of the coating thickness will be investigated. In a second step the influence of a heat treatment increasing the substrates load bearing capacity will be studied.With that approach the impact of each single change on the gears lifetime is evaluable and especially with respect to the geometric changes of the gear it allows to understand general mechanisms that can be transferred to further gear sizes and ratios. At the end, the findings are evaluated and transferred to a final gear configuration.

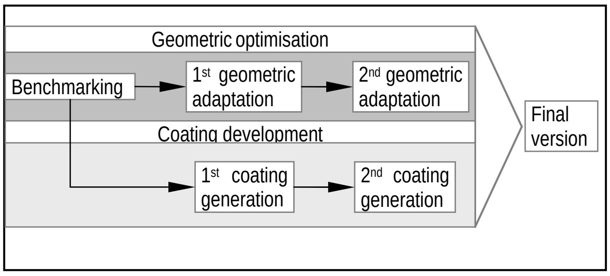

Figure 9: Schematic depiction of development approach

8. GEAR MATERIALS AND LUBRICATION

As materials for FS and CS the commonly used 15-5PH cond. H1075 and 17-4PH cond. H1150 were selected, as previous PoD testing did not show the necessity to deviate from this standard. As WGB a hybrid bearing was chosen, made of X30CrMoN 15 1 races and Si3N4 balls, as the combination of ceramic and steel promises tribological advantages for dry lubrication. The retainer is made of Tecasint® 1391, which is a MoS2 containing PI material that has been identified as potential European sourced retainer material for dry lubricated bearings [1].

With respect to the coating, based on previous PoD testing (ref to chapter 2), the critical tribological contacts (which is the toothing and the contact between FS and WGB outer ring) were coated with MoSx/WC provided by Tecnalia, as this seems the most promising selection. The bearing itself is coated with WS_2 (Dicronite®DL5), which should act as initial lubrication providing low friction during running in, until MoS_2 from the bearing cage is transferred to balls and races. A picture of a coated component set is shown in Figure 10.

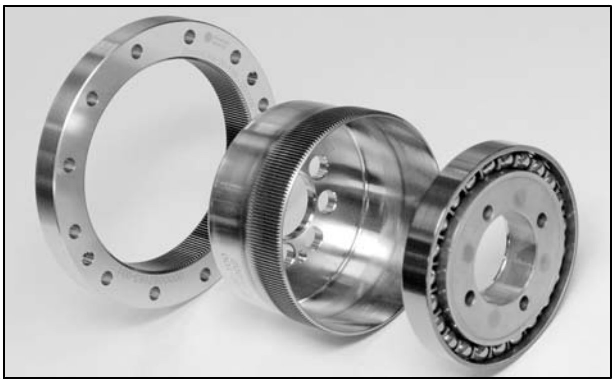

Figure 10: Coated CobaltLine - 20 - 100 component set

9. GEAR TESTING

The vacuum endurance tests were performed at AAC on the “Salotte 1” test rig. The test rig allows the integration of a vacuum suitable testbox with an integrated test gear which is supplied by Harmonic Drive AG. The tests themselves are performed under the following conditions:

Temperature:	24°C (thermal vacuum testing for final gear version)
Pressure:	$<10^{-5}$mBar
Input speed:	250 min-1
Output torque:	4Nm / 16Nm
Load profile:	constant
Rotational direction:	cw (view on output shaft)
Orientation test gear:	vertical (Wave Generator up)

The health monitoring is performed via the efficiency that is continuously recorded for the operating direction. Additionally after distinct intervals the efficiency is measured for both directions. As failure criterion, a decline in the gears efficiency by 20% was fixed.

Following the integration of the test gear into the testbox, the gears are characterised at HDAG prior to the test. The characterisation is repeated after vacuum endurance test. Results of the pre test characterisation are given in Table 2. It shall be mentioned, that the given numbers are mean values of the respective gear configuration, whereas for Benchmarking and the 1st coating generation each three gears have been considered. For the 1st geometric adaptation overall 4 gears were measured.

It can be found that the stiffness K1 is in a similar range for Benchmarking gears and for gears of the 1st coating generation, which was expected as both gears are of the same type. A decrease in the gear stiffness is seen for the 1st geometric adaptation which is linked to the implemented geometric changes. This lowers on the one hand the contact stresses within the tribological contacts, effects on the other hand on the gear stiffness. A wind up curve of each gear configuration is shown in *Figure 11*.

Configuration	Stiffness K1 [Nm/rad]	Transmission accuracy [arcsec]	NLST [Nmm]
Benchmarking	$2.1*10^4$	39.8	46
1st coating generation	$1.9*10^4$	40.9	32
1st geometric adaptation	$1.2*10^4$	31.5	31

Table 2: Results pre test characterisation in coated condition

Following the characterisation, vacuum endurance testing was executed. The test program started with the benchmarking gear configuration, which was followed by the gears with the 1st coating generation, where 2 gears were tested. Finally, the configuration with the 1st geometric adaptation was tested, whereas herein 2 test results are presented.

Figure 12 shows the course of the efficiency for the benchmarking gear and the two gears with the 1st coating generation versus the number of output revolutions.

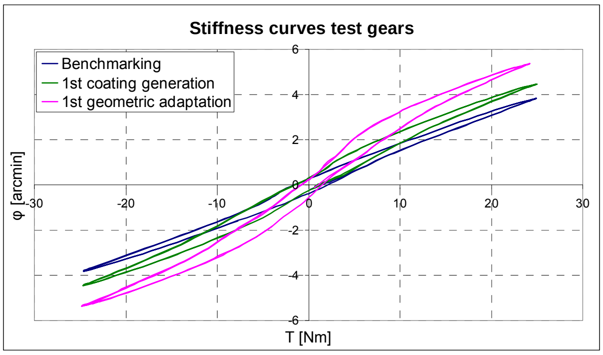

Figure 11: Stiffness curves gears for Benchmarking, 1st coating generation and 1st geometric adaptation

It can be seen that for all gears after a short period (approx. 100 output revolutions) of high efficiency a rapid decline occurs. In order to derive if the applied load of 16Nm is too high, the load was reduced to 4Nm for a test gear, which results in a lower efficiency of approx. 45% in comparison to 66% before. Anyhow, even a reduction in the output torque did not lead to a significant lifetime improvement, compared to the set target of 17,000 revolutions.

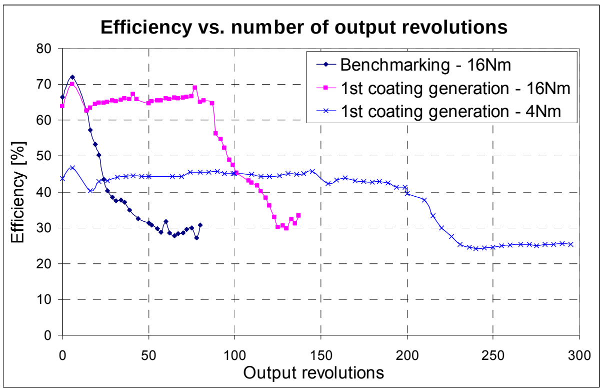

Figure 12: Course of efficiency for testing of first dry lubricated HD gears

The post test characterisation of the gears did not show significant changes neither for the transmission accuracy nor for the gear stiffness. The only parameter that changed was the NLST that increased approx. by the factor of 2, which is probably linked to the decrease in the efficiency. The visual inspection of the components shows the onset of wear within the toothing (ref. to Figure 13) for all three gears. The optical impression was confirmed by EDX analysis, unveiling that the coating was pointedly worn away.

These initial experiments were followed by tests with the gear with 1st geometric adaptation. As already mentioned, the main target of the implemented optimisation was to lower the contact stresses to an acceptable level. Anyhow, performing a first test at a load level of 16Nm lead to comparable results as for the benchmarking gear; after a short period of operation (~100 OPR) the efficiency dropped and the test was stopped. A significant difference to the benchmarking was found during the visual inspection – evaluating by optical means, the coating still seems to be present and no deformation of the teeth could be established. The investigation by SEM principally confirmed the optical evaluation. The contact zone was still entirely covered with coating, but it was also found that at the border of the contact zone a flaking – off of the coating could be observed.

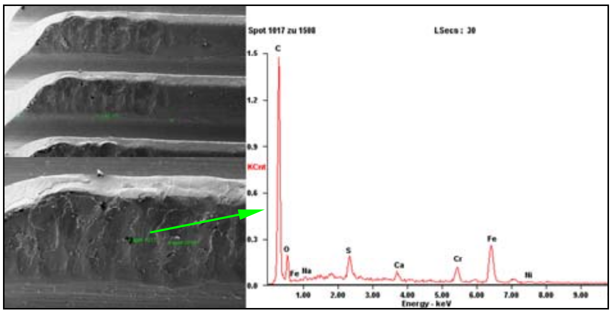

Figure 13: Loaded tooth flank of FS after test. Top left and bottom left: SEM image of FS tooth, right: EDX spectra within contact zone

Taking the vertical orientation of the test gear within the test rig into account and evaluating the picture of wear leads to the suggestion that the decline in the gears efficiency is caused by coating particles that moved into the WGB, impeding the smooth running of the bearing. In order to evaluate, in how far the implemented geometric change could improve the situation within the toothing, it was decided to cover the WGB for the subsequent tests, knowing that further improvement of the coating is still necessary.

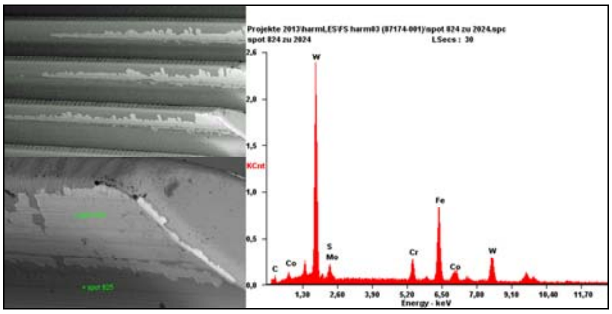

Figure 14: Loaded tooth flank of FS after test. Top left and bottom left: SEM image of FS tooth, right: EDX spectra within contact zone

The next test was again performed with a reduced load of 4Nm, hence the initial efficiency is at a level of 35%. The course of the efficiency is shown in *Figure 15*. With the covered WGB the rapid initial decline in efficiency as observed in all former tests, does not occur. After approximately 500 OPR a slight decrease in the efficiency can be seen lasting approximately until

1000 OPR. This is followed by a constant period until 4200 OPR. After further 450 revolutions the test was finally stopped.

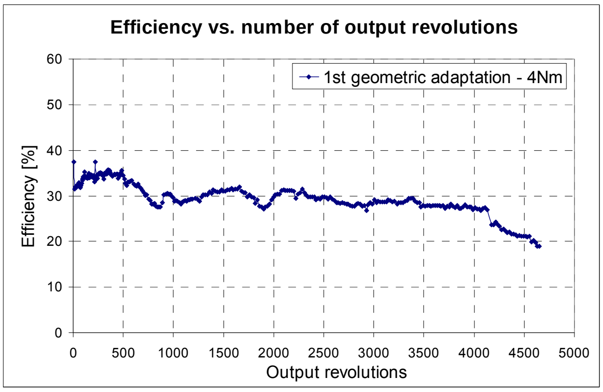

Figure 15: Course of efficiency of dry lubricated Harmonic Drive® gears with 1st geometric adaptation.

The results of the post test characterization did show significant changes as well for the gear stiffness, which has decreased by a factor of two, as for the transmission accuracy which deteriorated to 115 arcsec. Both values are indicating that wear has occurred within the gear. A preliminary inspection of the gear components shows the onset of wear within the toothing, whereas especially on the loaded tooth flank the coating seems to be worn away. Detailed investigations concerning the wear condition of the respective gear are currently ongoing

10. SUMMARY

The results of the initial tests are in line with the simulations that have shown that the contact stresses within the unchanged gear are too high for dry lubrication. Even an increase in the coating thickness as performed for the 1st coating generation could not improve the situation. Furthermore it was shown that a reduction of the external load did only lead to a minor elongation of the gears life, suggesting the assumption that the internal preload is mainly responsible for the short operational period. This underlines the necessity of the implementation of geometric changes to optimize the gear for dry lubrication.

Within a first development step, geometric changes were implemented in order to decrease the contact stress within the gear. Performing a first test it was found, that after comparable operation time as for the unchanged gears, the coating within the toothing was still present, illustrating that the implemented geometric change is beneficial. Being able to ascribe the efficiency decline to coating particles enhancing the smooth running of the WGB, measures were implemented to hinder the particles to find the way into the bearing. The operation time of the gear within the subsequent test could than be increased by a factor of at least 20 or to more than up to 4000 output revolutions. This clearly shows the benefit of the implemented changes.

11. OUTLOOK

Until today, the first steps towards an optimised gear design for a dry lubricated Harmonic Drive® gear have been performed. Especially the geometric adaptation clearly lead to an advancement of the gears lifetime. Anyhow, with respect to the targeted lifetime improvements are still necessary.

Based on the first test results on gear level it was found that a further improvement of the coating, to avoid the observed flaking – off, is needed. This will be performed based on further PoD testing. Additionally it was found that in general the gears efficiency is fairly low which was ascribed to high friction within the WGB – obviously the Dicronite®DL5 coating did not work satisfying. It was therefore decided to change this coating to MoS_2 for the bearing races.

According to the development approach (ref. to Figure 9), two further gear configurations are to be tested before fixation of the final gear version. The intended changes are derived on the one hand based on results from PoD – testing, where the application of a heat treatment to increase the substrates load bearing capacity, lead to an improvement in the coatings lifetime of at least a factor of 10. On the other hand, with the implementation of a new tooth profile, based on calculations a significant reduction of the sliding path within the toothing is possible. Those changes are seen to be crucial for the achievement of the defined requirements.

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