ANTENNA SCAN MECHANISM FOR AN INTER SATELLITE LINK OF A CONSTELLATION PROGRAM

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Abstract

For a constellation program, RF Inter Satellite Links between single satellites can support ranging and communication for uploading mission data or telecommands. These data shall be uploaded from one single ground station to the next reachable satellite and transmitted by the Inter Satellite Link to further dedicated satellites. For this function each satellite has to be equipped with 2 **A**ntenna **S**can **M**echanisms (ASM) for data transfer in the K-Band.

The main challenges for the mechanisms are the high speed position change requirement, low mass requirement and the design to cost approach. Furthermore a small envelope to accommodate the 2-axes antenna scan mechanism was provided. The maximum position change of +/- 180° needs to be reached within 3 seconds. All requirements shall be achieved by relying on the heritage design of our downlink antenna pointing mechanism product.

The ASM design approach was based on our 2-axes steerable downlink antenna; however during the definition phase it turned out that some major changes have to be implemented due to mission requirements (high operation speed and long lifetime). Following the design to cost approach most components could be procured from industrial standard but had to be qualified in terms of functionality, performance and life. The following industrial components have been selected:

- Bearings procured from an industrial supplier and modified (cage) by a supporting supplier
- The selected actuator is a standard stepper motor equipped with redundant windings
- The slip ring design was used from a previous project in order to keep the heritage
- Suitable rotary joints for the RF link were provided by a small and flexible company

However at the very beginning of the project some difficulties with the bearings selection and procurement have been identified. Since the most suitable standard catalog bearings were not available in time, we were forced to use alternatives. In parallel due to envelope constrains the re-location of one actuator followed by additional design modifications became necessary. Finally this design modification impacts the bearing loads and consequently the selection of other bearings became necessary.

The paper will show the error propagation generated by lead time issue of a main component and the test results of the final design including friction tests, and first measurements during the life test program.

Driving Requirements

One of the main requirements was to change position within 3 sec up to a maximum angle of 180° in for the azimuth axis and 76 degree in the elevation axis in order to allow a maximum time for communication.

The other challenging requirement was the small envelope for the complete mechanism of 500 mm in height (350mm above the mounting panel and 150mm below) and a diameter of 350 mm.

Design

The design of the ASM is an evolution based on the downlink antennas developed for KOMPSAT and NGSAR. In order to stay within the allowable envelope the ASM is not directly mounted to the outer wall of the S/C but the azimuth stage is sunk into the S/C wall in order to decrease the overall height. Each stage is composed of an actuator with attached planetary gear, a spur gear stage, bearings and a rotary joint for RF signal transmission. Actuator, rotary joint, the slip ring assembly and the main bearings of the azimuth stage are within the S/C while the spur gear stage is just above the S/C panel. The elevation stage (yoke) is mounted

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directly on the azimuth stage carrying the elevation bearings, actuator assembly, rotary joint and antenna. For commonality both actuators are the same. Both stages are using also the same rotary joint to minimize the development effort. The electrical lines are routed via a slip ring assembly towards the elevation stage. During launch the elevation stage is clamped by the launch lock which also blocks the rotation about the azimuth axis.

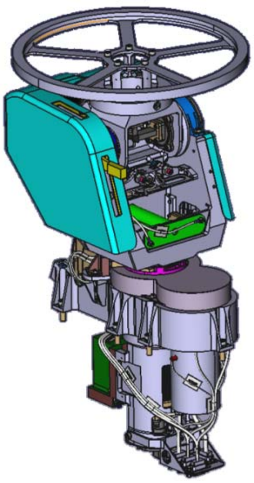

Figure 1: ISL ASM CAD model with antenna dummy

Although a maximum azimuth angle of +/- 180° is required an endless rotation of the azimuth axis was chosen. The decision was based on the heritage design and the required 15 million cycles which routed cables would need to endure. The elevation axis is designed for a rotation between launch position (nadir) and 76° off-nadir. Reference switches on the azimuth axis can be read out during a full rotation while on the elevation axis the reference switch is located in one end positions.

The initial heritage design having the elevation motor located along the yoke structure was abandoned as the bevel gear at the elevation was expected to create issues due to thermal expansion. Further, a large envelope violation occurred which required further configuration changes by moving the actuator closer to the azimuth axis. The movement towards the envelope axis was a benefit for the bearings loads. Due to the necessary relocation of the HRM bracket, the load path of the hold-down mechanism changed with the drawback of reducing the ability to carry a larger portion of the launch loads. Eventually after re-arrangement of motor, antenna and hold-down position respecting the kinematic movement of the parts an optimal solution with minimal envelope violation was found. Previous performed analysis had to be repeated and also the configuration for the life test set-up had to be adjusted as the EM development was performed in parallel.

Components

Actuator stepper motors were chosen which are operated in open loop. This implies that step loss is to be avoided at any circumstance. The trade was to provide sufficient torque margin against a defined maximum angular speed. On top of the stepper motor a planetary gear was mounted which drives the azimuth axis via a spur gear. The following diagram shows the total gear ratio against the max motor speed and motor torque.

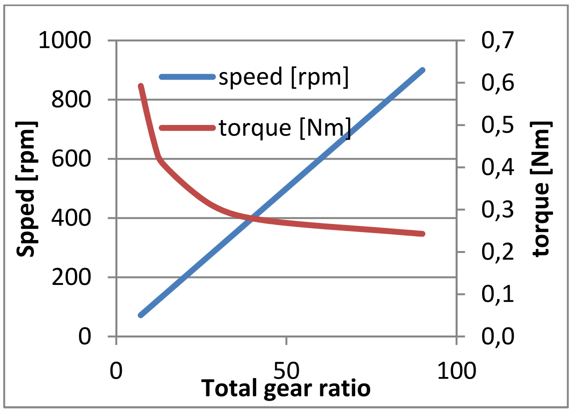

Figure 2: Motor speed & Motor torque vs. gear ratio

As with increasing overall gear ratio, the curve for motor torque becomes asymptotic. The gear ratio was defined based on available standard gears plus the spur in a way of minimizing the envelope and using the heritage gained in the last programs.

For commonality reasons the same actuator was chosen for the elevation axis, except the ratio of the spur gear which had to be adjusted according to the overall kinematic requirements.

The development of the rotary joint based on the experience made on the previous projects was followed consequently. The bearings within the rotary joints were removed and substituted by the already present bearings located on azimuth and elevation axis. The rotary joint of the elevation axis is located within its fixed bearings while the rotary joint of the azimuth axis is attached to the slip ring assembly below the azimuth bearings. The rotor of the rotary joint is attached via a close fit to the rotor of the slip ring assembly attached to the rotating shaft while the stator of the rotary joint is mounted concentric to the housing.

For the slip ring assembly the re-use of an existing design was chosen to minimize delivery time and also to minimize the cost of the development program. The slip-ring can be adapted such that a few more tracks are available in future if needed.

Pointing Accuracy

The pointing accuracy is driven by the resolution of the stepper motor, the backlash within the planetary gear and the backlash of the spur gear. The backlash within the spur gear is reduced via a preloaded spring in the subdivided driving wheel. Further the play within the bearings seats contributes the pointing accuracy. The following calculated pointing accuracy will be achieved.

	Azimuth [deg]	Elevation [deg]
Motor resolution	0.00625	0.00477
Planetary gear	0.056	0.056
Spur gear	0.005	0.005
Bearing contribution	0.186	0.021
total	**0.25**	**0.074**

Figure 3: Pointing Accuracy

Procurement issues

The initial bearings selection needed to be modified due to higher loading. For the azimuth stage the next larger bearings were selected as the heritage bearings were at the limit of their load capability. The lifetime of the mechanism is specified with 12 in-orbit years which leads to use of an oil, which has a low evaporation pressure. Fomblin Z25 was selected also based on the heritage of the former program. Hybrid bearings were selected to avoid an early degradation of the fluid. A cotton phenolic cage was chosen to provide an oil reservoir for the 12 years in orbit. In view of a design to cost approach standard bearings should be used. A large bearing manufacturer provided in its catalog the next size of the bearings and the bearing analysis showed sufficient margin.

The original cage needed to be replaced by a newly manufactured one as the original was impregnated with a preservative agent.

At the point in time to order of these bearings it became obvious that these bearing were never manufactured before and a short term delivery was not possible. Furthermore a minimum bearing quantity needed to be ordered to start production. As the bearings were needed for the characterization and life test as soon as possible, our dealer supported us to find a possible replacement. A similar bearing with smaller ball and a lower load carrying capability were found and ordered after the calculation showed a positive margin. In parallel other companies known in space business were contact but provision of bearings in a short time was not possible. For further projects other suppliers are available providing the required bearings.

A similar issue appears during the procurement of the stepper motors. Late during procurement our supplier informed us about difficulties in the bearing procurement. The required wet lubricated motor bearings could not be delivered on time due to late order and non-standard products. A solution was found to supply these bearings in a reasonable time not shifting the life test schedule too much.

Friction Test Results

The bearing characterization test is an important validation test. As an early breadboard test, this test can provide a good mean to verify the torque budget and the bearing analysis.

For the ISL a bearing characterization test was conducted at seven different velocity levels (1 rpm, 3 rpm, 17 rpm, 25 rpm, 35 rpm, 45 rpm and 100 rpm) and four different temperature levels (-40°C, 0°C, 25°C and 65°C). Each test point was started with a run-in procedure, cycling the bearing clockwise and counter-clockwise for several minutes. Having finished the run-in, the different velocity levels were repeated multiple times, deriving multiple resistive torque measurements.

Figure 4: Bearing Characterization Test Set-up in a TV chamber

Comparing the results of the measurements (see

Figure **5**, it can be verified that the bearing performance is as expected during analysis. Just slight deviations in the resistive torque measurements can be stated between 0 and 65°C, but a high increase for cold temperatures (-40°C) was found. An interesting phenomenon was the higher increase rate of the resistive torque for hot and fast conditions.

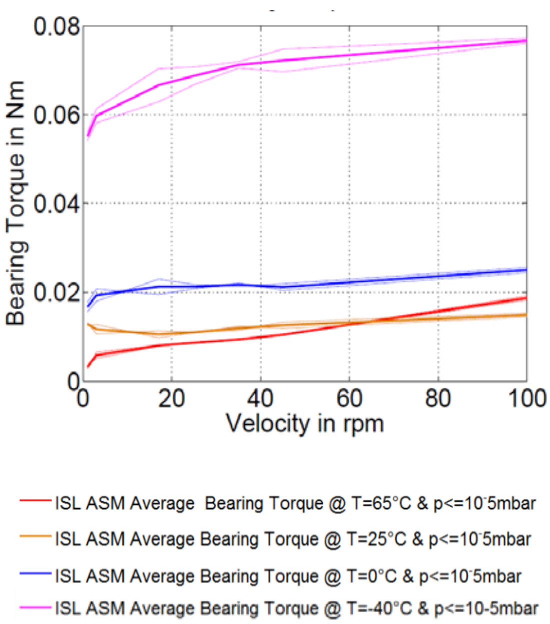

Figure 5: Results of the Bearing Characterization measurements

In summary a good fit between measurement and analysis can be stated. The worst case torque was analyzed with 75Nmm and verified by a measurement of 72Nmm during testing. At lower level of resistive torque a higher, but still acceptable, deviation of 6Nmm between analysis and measurement was found.

This breadboard test gave Airbus DS GmbH the confidence of having a sufficient torque margin and a correct model of the friction behavior of the preloaded bearing configuration.

Life Test

To provide a qualified mechanism at the end of the one and a half year project time and to provide a decision basis for the customer, the initial planned and agreed life test at the end of the qualification campaign was dropped in favour of a life test program, with a separate model, running in parallel with the rest of the campaign. To minimize the cost of the life test the azimuth stage was chosen to be tested only as loading for the azimuth bearings was considered the worst case. The motor and both gears will experience during life testing the largest number of revolution.

Pre loading of life test components

Prior to life testing a correct environmental conditioning is essential for the credibility of the life test itself. This environmental conditioning consists especially of a vibration test and a thermal settling.

The thermal settling itself is in general no issue, and was executed directly before the start of the life test under the defined acceptance temperature level in the TV chamber.

Much more demanding was the task to set up the correct vibration of the Life Test Model.

The elevation stage was simulated using a mass dummy only. Designing a support structure different from the launch lock design required much more time than expected and the change of the initial motor location forced further changes of the life test set-up eating up the schedule margin for the life test further.

At this point in the project cycle the actual component loads during vibration were not known or they have a low confidence level due to the reconfiguration of the elevation axis. At the same time the correct loading with the actual QM loads would be the best way forward in order to have a high credibility of the Life Test. This conflict leads to a very high effort in the structural test prediction with the goal to load the single components as high as possible, but not to overload them. As these results cannot be used directly for the QM later on, a more global approach would have been optimal.

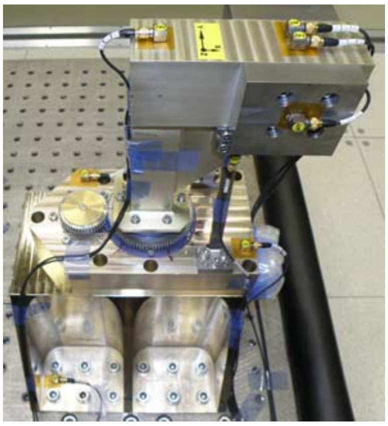

Figure 6: ISL LTM Vibration Test Setup

Eventually it was agreed on ISL project level that the actual input level at COG will be controlled and its effect on the actuator and bearings will be evaluated. Within the prediction for the Life Test Vibration, the final component load level was not known, so that this was a compromise to come as close as possible to the later on loading scheme.

Life test execution

During the accelerated life test the communication time of 27 sec within each 30 second period will be deleted such that the 12 year in-orbit can be simulated within about 7 months. As the motor is continuously driven during the life time test without any interruption, it would heat up much more than during in-orbit operation such that during the life test the motor needs actively to be cooled to its nominal temperature.

The operation profile will be simulated following a set of predefined angles to be repeated during the whole test time while the environment will be changed. The life test shall also verify the capability of the motor bearings, the correct functioning of the planetary gear and the spur gear to survive without significant degradation the required lifetime.

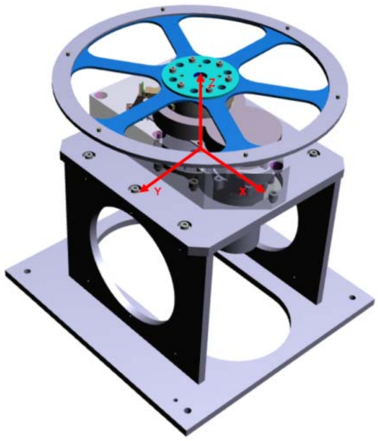

Figure 7: Life test set-up

Future outlook

The EM (EQM) mechanism will be functionally operated and verified following a defined program of different angles derived from an in-orbit simulation of the inter-satellite link. The pointing performance will be verified under nominal and thermal vacuum conditions. Microvibration, RF and EMC tests will complete tests program. The development program shall be finished by end of the year and provide a complete verified demonstrator for future use.

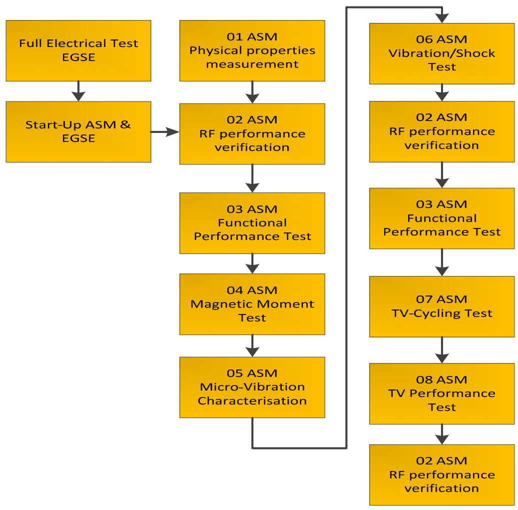

Figure 8: ISL EM test Flow

Lesson learned

- Although standard catalogs bearings used for a design to cost approach, the details do not often fit to the needs in space business. An early verification that the bearings fit to space needs in required.
- Early procurement and detailed definition with suppliers is mandatory to keep the schedule.
- Modeling a dummy life test set-up to meet the load level is not a trivial task while in parallel the development is ongoing. To avoid cost overruns the Life testing should be done with the EM or EQM if a dummy has to be established.