Scanning System Development and Associated Bearing Cage Instability Issue

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Abstract

The Scan equipment used to control the MicroWave Radiometer Imager (MWRI) of the polar orbiting FY3 meteorological satellite was developed and delivered to the customer by Astrium Satellites GmbH. The instrument detects and monitors meteorological and biosphere environmental anomalies.

The key elements of the design and the most important design requirements and design features are described hereunder.

Special attention is paid to the bearing cage anomaly observed on the Scan Compensation Mechanism during the test campaign and to the subsequent root cause investigation performed.

Introduction

The scanning equipment consists of a Scan Drive Mechanism (SDM) rotating the instrument package, of a physically independent Scan Compensation Mechanism (SCM) used to eliminate disturbance moments induced by the SDM and of a closed loop Scan Control Electronics (SCE) controlling and synchronising the two mechanisms.

The heavy MWRI instrument package mounted on top of the Scan Drive Mechanism requests conical antenna scanning. This requirement leads to a continuously rotating mechanism of high scan rate stability.

One EQM and two FM sets of scanning equipment were developed, qualified and delivered to the customer.

During the mechanism test campaign a randomly occurring bearing noise was observed in the SCM. Apart from the mechanism design, the paper explains the way up to identification and elimination of the bearing noise issue under the given technical and programmatic constraints and provides the lessons learnt from this experience.

Key Equipment Design and Performance Requirements

The mass of the instrument package mounted on top of the SDM is 60 kg at an instrument Moment of Inertia of 2.7 kg-m^2. The instrument has to be accelerated within 180 s to the nominal spin rate.

The nominal spin rate is 1.7 s/rev. In order to achieve the required radiometer performance, the spin rate shall stay stable within an error bandwidth of ±0.34 ms compared to the nominal value. In order to achieve this stability, closed loop control is implemented.

For compensation of the momentum induced by the SDM into the S/C, a Scan Compensation Mechanism is provided. The SCM is physically independent from the SDM and it is designed as a separate mechanical unit mounted in line with the SDM rotation axis to the instrument structure. Both units, the SCM and the SDM are controlled by the digital controller implemented in the FPGA of the Scan Control Electronics (SCE).

The SCM rotation rate is directly synchronised to the SDM scan rate via the controller. The SDM scan rate is multiplied by a factor of 16, which leads to a rotation speed on the Scan Compensation Mechanism of about 565 rpm.

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The chosen SCM speed ratio is the result of a trade-off between bearing life margin and flywheel inertia at minimum resulting flywheel mass (and at the given maximum allowable Flywheel diameter of 400 mm).

The equipment life is 3 years in orbit; this yields nominally 890 million revs on the SCM over life without margin. The operational temperature for the mechanisms is -30 to +50 deg C.

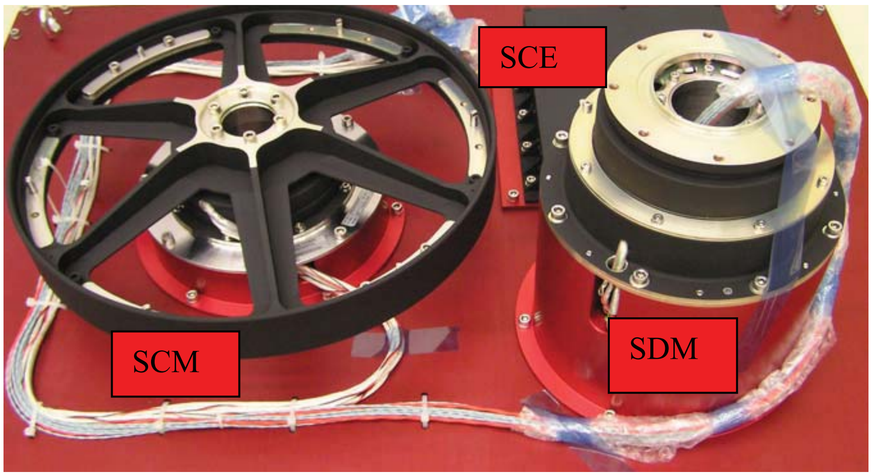

Figure 1. Scan Drive Equipment mounted to the Transportation Jig.

Scan Drive Mechanism Design

The SDM consists of a drive module using a set of inclined ball bearings, of a redundant brushless DC Motor and of a redundant optical encoder for closed loop velocity control. On the rear end of the drive Module, a slipring unit for power and signal transfer from/to the rotating Radiometer Instrument package is attached. The Harness is guided through the hollow drive shaft to the slipring rotor and from there via the slipring brushes to the slipring stator.

The I/F to the rotating instrument mounted on top of the mechanism is formed by a launch off-load device. The launch off-load device consists of axially pre-loaded flexible metallic bellows and of an accurately shaped conical Interface. The combination of both elements provides instrument alignment accuracy and torsion stiffness during mission as well. During launch the conical I/F is lifted off from the mechanism I/F, so to de-couple the instrument load path from the mechanism. After release of the Instrument HRMs on S/C side, the instrument is pulled by the preloaded SDM bellows back into the conical I/F in order to allow transfer of the drive torque from the SDM to the instrument at accurate alignment of the instrument rotation axes.

The Rotor shaft is mounted via two pairs of hard preloaded angular contact ball bearings in back to back arrangement into the SDM housing. The bearings are manufactured from Stainless Steel 1.4125 (440 C), the cages are manufactured from Phenolic Resin, vacuum impregnated with oil.

The motor is of the brushless DC type with completely cold redundant independent Stators. Its maximum operation torque is 2.3 Nm.

The redundant optical Encoders provide absolute position data with 16-bit resolution. Its output data are used for motor commutation on the one hand and for high accuracy motor velocity control on the other.

The Slipring is composed of 19 solid gold tracks and redundant gold alloy brushes to comply with the expected 3 years orbit life corresponding to about 55 million revs.

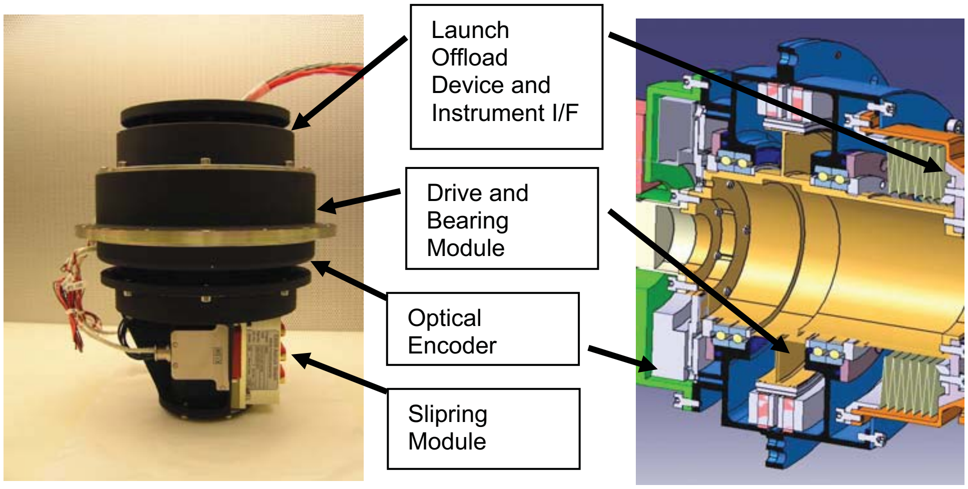

Figure 2. SDM Design

Scan Compensation Mechanism (SCM) Design

The function of the SCM is to compensate for the momentum induced by the rotating instrument into the S/C. The SCM consists of a drive module using a brushless DC Motor with redundant motor stators. The SCM motor is similar to the SDM, however it is optimised for the higher operation speed. Its maximum operation torque is 0.9 Nm. For rotation velocity feedback 3 Hall Sensors are included in each motor stator. To the motor output shaft a flywheel unit is attached. Trim masses for fine-tuning of static imbalance are mounted to the Flywheel.

Concerning the bearing configuration it has to be noted that the bearings and its lubrication concept were kept identical to the SDM in order to gain synergy by similarity during the mechanism development and bearing procurement process.

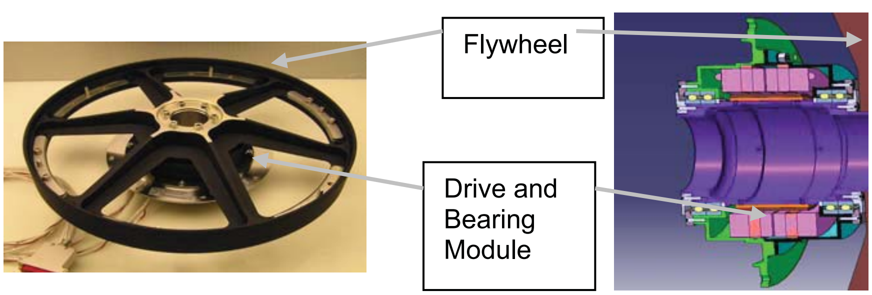

Figure 3. SCM Design

Scan Control Electronics

The Scan Control Electronics (SCE) is built up as a completely cold redundant unit, each supporting the main respectively redundant motor windings of each mechanism and receiving feedback from the redundant high-resolution encoders respectively from the Hall sensors. The controller is implemented into the FPGA.

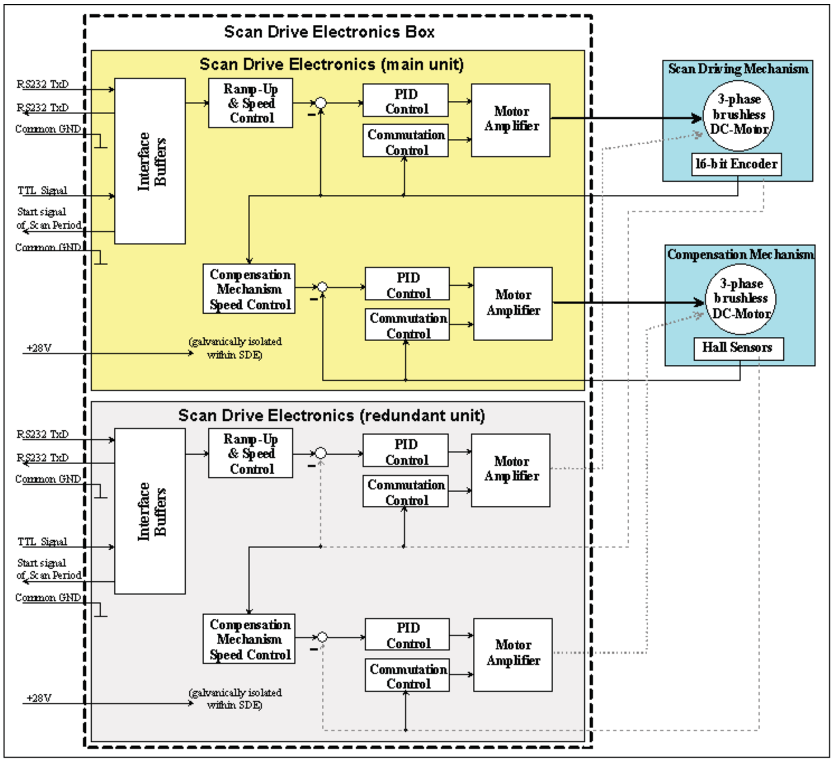

Figure 4. Scan Control Electronics Block Diagram

Mechanism Performance Test Summary

Apart from the standard verification approach applied to space mechanisms, two main critical functional performance data had to be verified. One of them was the rotation period of 1.7 s/rev at a stability of ±0.34 millisecond; the other one was the maximum residual momentum of <0.1 Nms induced into the S/C by the Scan equipment during start-up and operation.

Figure 5 shows the equipment performance test setup. In Figure 6 the typical scan rate deviation from the nominal value over a time of 2000 seconds is shown, while in

Figure **7** the residual torque profile over time is depicted. The scan rate deviation was measured by reading out the high-resolution encoder signal; the residual momentum was measured via a Kistler torque test device.

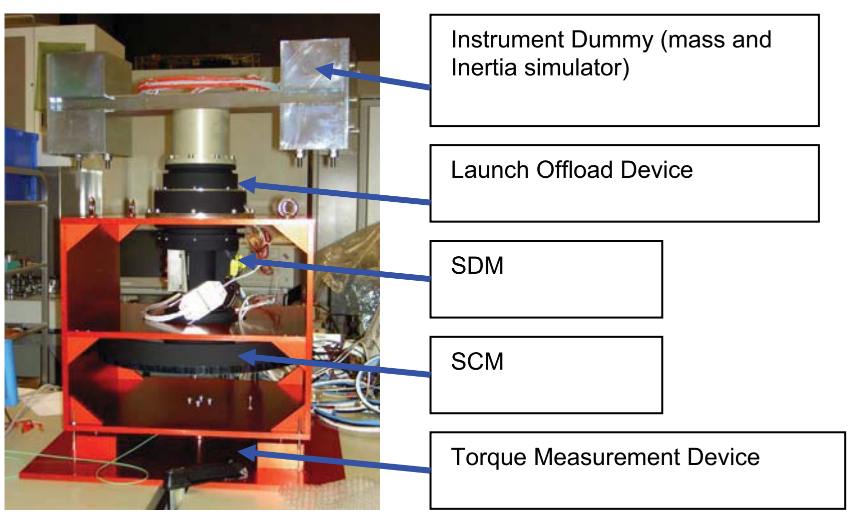

Figure 5. Functional Performance Test Set-up

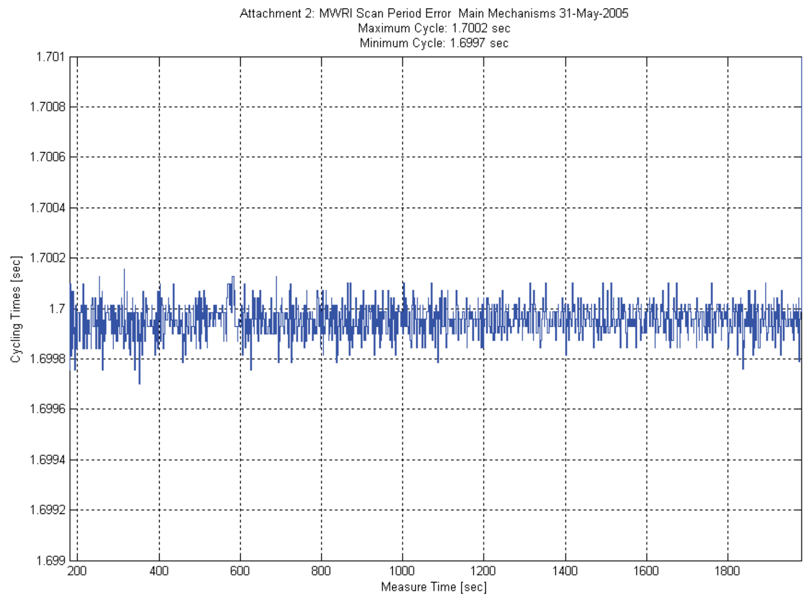

Figure 6. Typical deviation from nominal Scan Rate over time

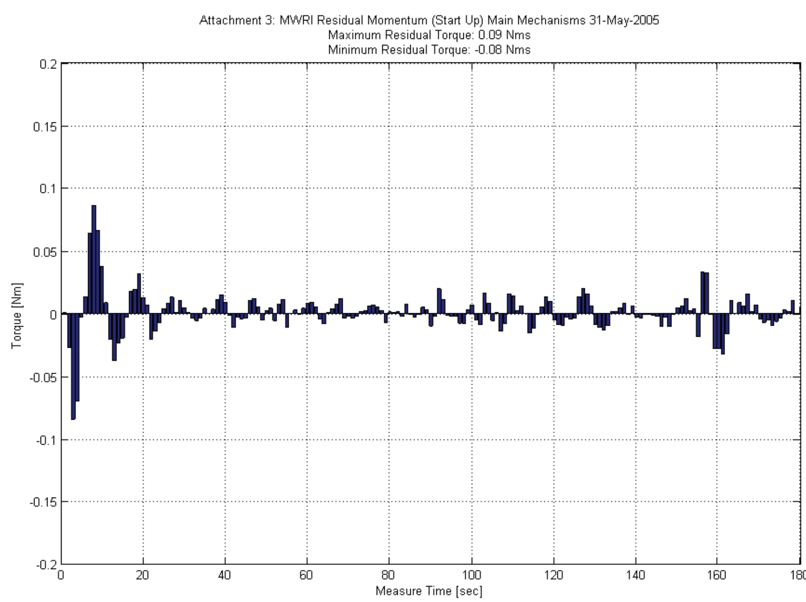

Figure 7. Typical induced torque profile during instrument start-up (180 seconds)

Bearing Unit Characteristics

Both, the Drive Mechanism rotating at slow angular velocity and the Scan Compensation Mechanism rotating at higher velocity, make use of an identical bearing configuration. The only major difference between the SCM bearing and the SDM bearing is defined by the higher bearing pre-load applied to the SDM bearing as a consequence of the higher operational forces acting to the SDM due to a potential later instrument CoG shift out of the nominal rotation axis.

By design, TiC-coated balls were preferred, however due to procurement issues originated by the transfer of the TiC coating process from Europe to the US, the availability of TiC-coated balls was limited at the time of bearing procurement. The finally delivered TiC-coated balls were mounted into the bearings by the bearing supplier; however, scratches of unknown origin were identified during bearing inspection on the balls. This led to the implementation of the back-up bearing design using steel balls and to a subsequent change of the lubrication system. In the following table (Table 1) the bearing characteristics are listed.

Table 1: Bearing Characteristics

Bearing Characteristics	**SDM Bearing**	**Differences for SCM**
Bearing Type	Thin Ring	
Arrangement	Back to Back	
Material	440 C	
Cage Material	Phenolic Resin	
Ball material	TiC-coated steel balls (was changed later to steel only)	
Vacuum impregnated	Z 25 (was changed later to MAP grease and Nye oil)	
Cage guidance	Inner Race riding	
Number of Bearings	2 pairs, back to back	
Outer Diameter	98.425 mm	
Inner Diameter	77.788 mm	
Height of Bearing Pair	15.874 mm	
Ball diameter	4.762 mm	
Number of balls	42	42 (original number, reduced later)
Contact angle	27 deg	
Preload	750 N	400 N
Max Contact Stress	1200 MPa	1000 Mpa
Axial Stiffness	250 N/µm	330 N/µm
Radial Stiffness	450 N/µm	550 N/µm
Angular Stiffness	330 Nm/mrad	410 Nm/mrad
Running Torque	<0.08 Nm	<0.04 Nm
Calculated Life	>1.18E+10 revs	>18.6E+8 revs
Rotation velocity	35.3 revs /min	564.7 revs/min

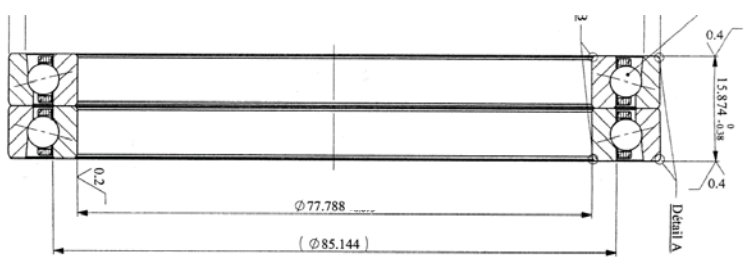

Figure 8: Bearing Dimensions

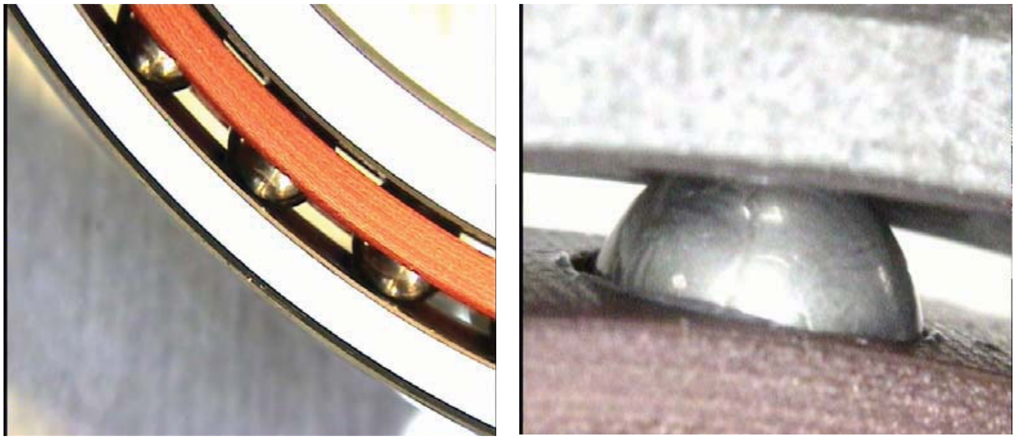

Figure 9. SCM Bearing Details

Observed Bearing Noise Issue on the SCM

In order to gain design and procurement synergy (considering the long lead times of ball bearings) within the extremely tight development schedule of 10 month for the EQM and of 12 month for the first FM equipment, an identical bearing concept was chosen for both mechanisms. It was confirmed by analysis, that at the given rotation speed of about 565 rpm on the SCM (performing momentum compensation) compared to 35 rpm on the SDM (supporting the MWRI instrument), the bearing life margin at the given load conditions was still very good.

Dynamic cage instability effects at the comparably low maximum rotation speed of about 9.5 revs/s were not expected. This latter assumption was supported by references where similar bearings had been used in Helicopter Subsystems at much higher speed.

After integration of the EQM equipment, functional testing was started. After some hours of perfect operation a sharp scratching noise was observed, obviously coming from the bearing unit of the fast running Scan Compensation Mechanism. The good mechanism performance was not affected by the phenomenon; the performance values were stable and as good as before.

The noise was not reproducible and it occurred only from time to time. After start up periods of noise free operation were observed, followed by periods of sharp randomly occurring noise. During mechanism speed up to the nominal velocity, the issue was typically observed at rotation velocities above 500 revs / minute, while during deceleration the effect was observed down to about 100 rev/min.

Consequently the mechanism was dismantled and all bearing seats and bearing Interfaces were checked again in order to identify any in-correct integration, tolerance or cleanliness issues and to verify the lubrication and cage condition. All tolerances were found as expected and no misalignment, debris, bad lubrication, bearing degradation or faulty integration could be detected.

The mechanism was re-integrated an operated again. All friction values were as expected, the operation was smooth. However, after some hours the noise came back. Again it was found that the issue did not jeopardise in any way the mechanism performance or power consumption, however the remaining general lifetime concern was quite severe.

During one subsequent performance test, a quite hard noise anomaly was detected, which suddenly disappeared. From this time onwards, the phenomenon was never detected any more throughout all the qualification test campaigns, neither after vibration, nor during hot or cold operation. The unit operated perfectly.

Therefore the origin of the observed temporary anomaly could not be clearly identified at this stage on the EQM unit. Consequently the H/W was delivered to the customer for system testing activities according to the contract and still within the required lead time of 10 month.

Three month after delivery, an alert was received from the customer, who had observed again noise in the mechanism.

After receiving the mechanism back on site, the origin of the anomaly could be investigated in detail. At that time the FM1 mechanism was already integrated and functionally tested without any issue. However after an extended storage time of some days prior to environmental acceptance testing, also on FM1 the noise issue could be observed, however in a less severe manner than on the EQM. Therefore it was clear from now that a systematic effect was occurring.

Investigations and Results

Due to the missing characterisation possibility of all potentially influencing parameters in the specific application, a comprehensive empiric investigation was started in order to identify the major drivers for the anomaly.

By design, the SCM integration concept was kept simple. This design feature was identified as a key advantage during all following H/W investigations, since it was possible to dismantle and re-integrate the mechanism at least once a day for check of function and identification of the success of a corrective measure performed.

Basically four different causes for the anomaly were considered:

1) Bearing mounting conditions (in the mechanism)
2) Bearing internal design and load conditions
3) Cage design
4) Lubrication concept

The most important modifications performed on mechanism and bearing level are listed in the following table (Table 2) and the observed effects on the noise anomaly are described:

Table 2: Modifications performed on the EQM Mechanism

Potentially Influencing Parameter	**Modification**	**Observed Effect on noise reduction**	**Comment**
1) Bearing mounting conditions			
Potential Bearing integration / tolerance issue	Dismantling of mechanism and detailed inspection	No improvement possibility	All bearing seats and mounting interfaces were in nominal condition
Potential alignment issue due to two pairs of ball bearings in back to back arrangement.	Dismounting of one bearing pair, operation with only one pair of bearings	No improvement observed	to avoid static over-determination
2) Bearing internal design			
Too low bearing preload	Increase of pre-load compared to the nominal value	No improvement observed	

Potentially Influencing Parameter	Modification	Observed Effect on noise reduction	Comment
Too high bearing preload	decrease of pre-load to about 300 N	Cage noise was eliminated during ambient pre-testing, however this was not confirmed after vibration. It was observed after vibration that the bearing outer rings tended to rotate in the bearing seat w.r.t. each other at reduced bearing preload	After Vibration test at reduced pre-load, bearing noise was observed again.
Ball diameter variation out of nominal range with the effect of “slippage” of individual balls	New balls used	No improvement possibility	The ball diameter variations were in the nominal range. A new set of bearing balls was mounted, no noise reduction observed
Degradation of races due to potentially wrong ball to race contact line	Inspection / oil filtration	No improvement possibility	The races were inspected at supplier premises by microscope and were found to be all nominal
Degradation of ball surfaces	Inspection	No improvement possibility	The surface conditions of the balls were nominal
Potential inner / outer race circularity out of tolerance	measurement of race circularity	No improvement possibility	The circularity tolerances were measured. The values were within the specified toerance.
3) Cage design			
Too weak cages (large cage diameter at low cross section)	Manufacturing of stiffer cages with higher cross section	The noise production became worse after implementation of stiffer cages	The stiffer cages did not solve the issue. The observed noise became even worse. The original cage cross section was maintained
Cage guidance. The cage design and cage tolerances were reconsidered and different inner and outer race riding cages were alternatively manufactured.	Change of cage design from inner to outer ring riding configuration, Change of cage tolerances.	No improvement observed	Nominally the bearing cages were inner ring riding, they were changed to an outer ring-riding configuration. Since a positive effect on the bearing behavior was not identified, the inner riding configuration was maintained as the baseline.
Number of Cage pockets	The even number of pocket cages (42) was changed to an odd number of pockets (39)	No improvement observed	According to theory high velocity bearings gain cage stability if an odd number of balls is used. The number of 39 was maintained throughout the tests though it did not yield any improvement w.r.t. observed cage behavior.
4) Lubrication			
Lack of lubricant	Re-lubrication of the bearing	Noise disappears for some hours only, no long term improvement observed	Fomblin Z-25, Vacuum impregnated cages

Potentially Influencing Parameter	Modification	Observed Effect on noise reduction	Comment
Lack of lubricant	Extended non-operation time (storage time)	Noise issue was more severe after extended mechanism storage time (some days)	
Type of Lubricant	Change from Z-25 to Nye oil and MAP grease	No improvement observed	However: General improvement of lubrication performance over life could be achieved by the changed lubrication concept in combination with steel balls. Potential chemical degradation of Z 25 in combination with the steel balls and steel races could be avoided.

While the above measures 1) and 4) of Table 2 could be performed on Astrium site, the measures 2) and 3) directly related to the bearing configuration, had to be implemented by the bearing supplier but verified on mechanism level. It was found that this process was extremely time consuming and inefficient since it was very difficult to intervene regularly into the supplier standard manufacturing process for implementing any new parameter modification. Furthermore the bearing supplier himself was depending on lower tier suppliers (e.g. for cage manufacturing) so additional delays were resulting as a consequence.

Therefore it was decided to perform all necessary further bearing investigations and modifications directly at Astrium Satellites within the mechanism team. This approach allowed complete in-house control of the further investigation process without any un-necessary schedule delay. For this purpose all required bearing dismantling and re-integration tools were manufactured and the bearing dismantling and mounting process was defined in order to allow complete handling and control of all necessary further bearing modifications, independent from the bearing supplier. After dismantling the EQM bearings and cages on Astrium site, all parts were inspected and no degradation or abnormal condition of parts or lubricant was observed.

After implementation and check of the above measures it was quite clear that the anomaly was generated by the bearing cages as a consequence of occasionally occurring cage instability effects at the given rotation speed. However the mechanism of action causing the instability effects with subsequent noise production was not really understood.

Only three findings influencing the noise production remained:

> The first important observation was that under low bearing pre-load the outer bearing races of a bearing pair tended to slightly rotate w.r.t. each other. This led to the suspicion that during operation unexpected forces produced by the cage are acting to the outer race.
>
> The second observation was that at increase of cage stiffness (increase of cage cross section) the noise effect got worse.
>
> In addition it was observed, that after an extended storage time (e.g. weekend) the noise issue was more severe than under continuous rotation. This was also in line with the fact that after delivery of the EQM to the customer (long storage time resulting from transportation, unpacking, start-up activities) the noise issue was observed again.

All three above observations lead to the conclusion that un-expected forces in the bearing were acting with the result of subsequent noise production.

The observation made during the first EQM tests namely that after occurrence of a sudden hard noise in the bearing the noise issue disappeared completely for all remaining test campaigns, could now be explained by a forced rotation of an outer bearing ring of a bearing pair as a consequence of a transient jamming force produced by the cage. After rotation of the bearing ring to its new position, the cage stability might have been improved. This theory would also comply to the fact that for a stiffer cage the noise issue got worse (higher forces produced by the cage).

The key question was now to find the origin and influencing parameters for the transient forces obviously produced by the cage with the consequence of generating the observed bearing noise. The only possible explanation had to be searched in the cage configuration itself.

A special tool was manufactured to allow inspection of the cage circularity and the EQM cage dimensions were found to be within the “as designed” tolerances. Also the pocket dimensions and clearance to the inner race was in tolerance (chosen such that at cold operation the shrinking cage could not jam on the inner race due to different thermal expansion between race and cage).

The conclusion drawn from all the previous investigations was that the bearing cages might be deflected or pushed out of their nominal position from time to time thus generating high local friction forces and subsequent bearing noise. After relaxation of the individual force peak, and nominal stable cage rotation, the noise was not observed for a certain time. This theory was supported by the fact that the noise issue could be deleted for a certain time after bearing re-lubrication leading to lower friction and consequently also to lower built up cage forces.

As a consequence, a method had to be identified allowing to avoid the generation of stresses between bearing cage and balls or to damp the dynamic motion of the cage in a way suitable to avoid noise generation. As a result a quite un-conventional idea was realized. Three bearing balls of each bearing were dismounted from the positions 120 deg apart, i.e. three of the 39 cage pockets were left without a ball. This measure was performed after finding a positive Hertzian stress margin for a 36-ball configuration.

The effect was significant. The noise issue was not observed any longer and even during temperature test at qualification temperatures the bearing run smoothly and completely nominal. Also during a subsequent life test lasting for 4 months at nominal speed (corresponding to 100 million bearing revs), no further noise issue was observed.

Though the effect was not completely understood, it could be concluded that the bearing issue could be solved by changing the dynamic behavior of the bearing cage. This was achieved by taking out dedicated balls from three positions of the cage pockets. Figure 10 illustrates schematically the measures taken. The baseline Cage Configuration using 39 Ball pockets in an un-symmetric configuration with uneven pocket spacing was maintained, however 3 balls were taken out of this configuration (every 13th ball) to come to the final 36-ball configuration.

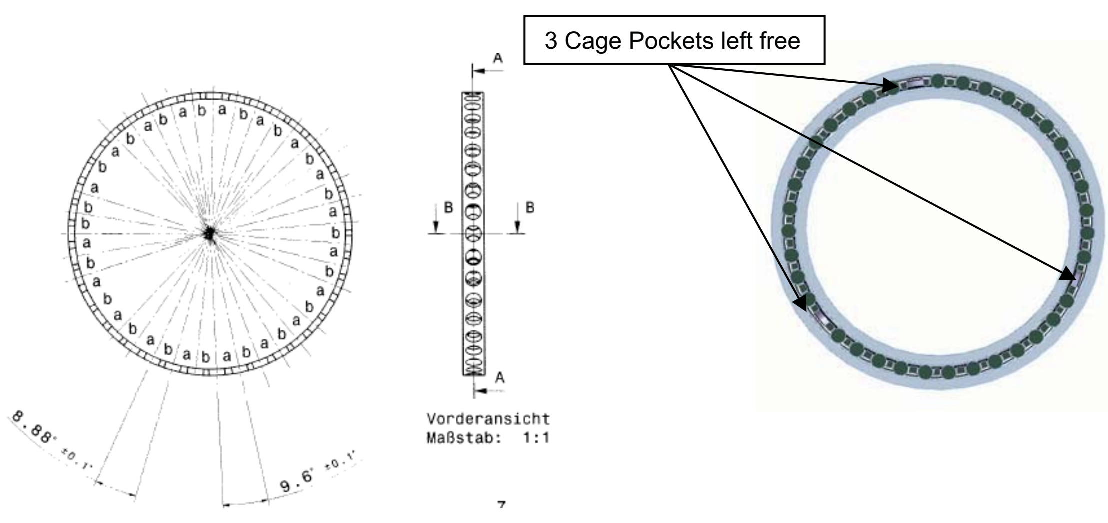

Figure 10: Bearing cage configuration and final bearing configuration using 36 balls only

As a consequence the identified measure was also introduced into the FM mechanism however here the observed noise issue was still observed in a reduced form. This led to the conclusion that the positive effect obviously caused by the reduction of balls in the EQM mechanism was compensated by another effect present in the FM bearings.

The remaining suspicion was, that the cage configuration as used in the EQM bearings was different from the FM cage configuration. This theory was fed by the fact that EQM and FM cages were manufactured from different batches of phenolic resin.

Since the existence of the cage instability issue was obviously also related to the environmental conditions e.g. to the mechanism storage time, time of operation, hot/cold operation and humidity, the theory was supported, that the poor geometric stability of the Phenolic Resin cages was one of the potential root causes for the noise issue.

Phenolic Resin is known to be quite sensitive to manufacturing processes, storage conditions and production batch and the resulting quality of a bearing cage is even depending on the location of the used material within the manufactured piece of raw material.

A non-circular shape of the bearing cage might have caused a “Hula Hoop” effect at the given rotation velocity resulting in abnormal forces between cage and balls with subsequent noise production.

Though at supplier premises the actual cage dimensions were checked on a 3D measurement machine, the confidence in these measurements was quite low, since the weak thin walled cages were expected to deflect under the contact load of the mechanical measurement device.

The FM1 mechanism disassembled and the bearings were completely stripped on Astrium Site. The measuring gauge already available from the EQM bearing inspection process was used to perform proper measurement of circularity and diameter of the cages in steps of 1/100 mm on all FM bearing cages.

It was found that the cages used in the FM mechanism were out of tolerance by up to 0.3 mm in diameter, compared to the EQM cages that were of good circularity.

As a consequence the selection of cages for the FM mechanism was taking place in order to identify the cages of maximum diameter conformity. These selected cages were then locally overworked by hand in order to generate a good circular shape.

After vacuum re-impregnation, the re-worked cages were integrated into the bearings again and the subsequent mechanism test at ambient conditions yielded prefect results. The noise issue on the FM1 was completely eliminated at the operational temperature extremes. Only at low survival temperature, which was -8 deg below worst-case operation temperature, a slight noise production was still observed. This was probably caused by cage dimensions changing with temperature from the circular shape at ambient conditions to any non-circular asymmetric shape at low temperature extremes in combination with the higher viscosity of the lubricant under this condition.

This finding supported the theory of the high influence of non-perfectly shaped cages at the given bearing size and rotation speed.

Since the thermal deflection over temperature is reproducible, the noise effect disappeared again completely after heating up the bearing from the survival temperature to the low operational temperature and from there up to ambient.

By using the identified 36-ball configuration supported by selection and optimisation of the available spare cages for good circularity, noise free performance over the whole required temperature range could be realized also on the both FM mechanisms.

Potential Causes for the Observed Instability Effects

Though a final conclusion on the root cause for the observed cage instability effect could not be drawn and the influencing effects could not be quantified or proven by an analytical model so far, the observations made on the H/W supports at least the following assumptions:

The cage instability is mainly caused by the dynamic forces resulting from a non-circular shape of the Phenolic resin cages (potentially also influenced by local density differences in the material).

A good oil film between the balls and the cage helps to avoid cage instability.

A weaker (more flexible) cage yields better dynamic stability than a more rigid gage does. In case instability occurs, the rigid cage is expected to produce higher transient forces between balls and cage pockets than a weaker (more flexible) cage does.

Taking out three balls of the bearing in a way as described above contributed to cage stability. This effect is potentially caused by a reduction of the lateral cage run-out or by reduction of dynamic cage forces produced by transient non-linear ball to ball and ball to race contacts. However the observed effect cannot be completely explained and proven without having established a very detailed dynamic bearing model that would have to be correlated with H/W test results.

Technical Conclusions and Lessons Learnt

Do not trust information gained from "similarity" of bearings. Bearing behavior is defined by a variety of parameters (size, material, cage dimension and tolerances, cage guidance, ball to cage clearance, ball dimensions, pre-load, mounting environment, environmental conditions, lubrication, life, operation velocity, operation profile, etc.) and no application is identical to another one. Check carefully these conditions before believing in "similarity".

Thin ring bearings of the given size (diameter in the 100-mm range) and cage dimensions, operated at elevated speed (above the 500-rpm range) and at the given boundary lubrication conditions seem to be sensitive to cage instability effects, especially if the cage is not perfectly shaped.

Improper (non-circular) cage shape can be expected especially if Phenolic Resin cages are used, since here the potentially in-homogenous local material characteristics tend to cause un-symmetric internal stress levels after manufacturing and at exposure to temperature extremes. Also local density variations in the material might contribute to resulting eccentric forces produced at high-speed operation.

In applications requiring bearings of elevated rotation speed and at comparably large dimension, the use of Phenolic Resin cages is not recommended by the above reasons. Substitutes shall be considered at the beginning of the design process (e.g. Vespel, PEEK or similar).

Though the effect was not completely understood, it was found that residual dynamic effects due to cage un-symmetry were damped out and mitigated after reducing the number of bearing balls from 39 to 36 by leaving every 13th cage pocket free. Using only bearing cages with an odd number of balls (39) unsymmetrical spaced, did not cause any improvement compared to an even ball number of 42.

In order to reduce the project risk, advantage should be taken from a simple mechanism design principle allowing easy dismantling of the mechanism in case of bearing issues. This implies that e.g. bearing press fits should be avoided if possible in order to support an easy and safe bearing integration and dismantling process.

Programmatic Conclusions and Lessons Learned

The observation was made that bearing suppliers produce very good standard bearing quality in view of dimensions of races and balls. Therefore a preferred option to mitigate technical and programmatic risks is to procure standard high quality bearings from experienced bearing suppliers, if no specifically designed bearings are definitively required for the application. The lead times for standard off the shelf bearings are often very short compared to specifically designed space bearings.

Even if a bearing supplier offers dedicated space bearings, the main production process will normally focus towards manufacturing for industrial or military application. Therefore the support in case of bearing issues observed after bearing delivery is often limited and the implementation of any necessary changes might cause significant implications within the supplier standard manufacturing process.

In addition, the mechanism developer should be able to dismantle and re-integrate the procured bearings on site for later inspection, lubrication, pre-load or cage modification according to in-house procedures in order to mitigate the remaining technical and schedule risk.

By following the process as described below, extreme advantages were already gained in follow on projects w.r.t. schedule, technical risk and costs: The procured standard bearings are dismantled after delivery by the mechanism developer. The cages are designed according to the needs and produced from the material best suited for the application. The bearings are then adjusted to the requested bearing preload by grinding the race shoulders to the required dimension and lubrication is performed according to the needs. Then the bearings are integrated on site and inspection as well as early life test under representative worst case conditions is performed.

High speed bearings should be tested in any case in a representative bearing test set up at worst case operational conditions before implementing it into the space mechanism design. This becomes possible within the normal (tight) development time frame, if the procurement approach as described above is realized.