Selection, preparation and lubrication of middle size ball bearings for infrared instruments

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Abstract

Infrared instruments need to be cooled to minimise thermal radiation. All these instruments are fitted with various optical components, which in order to be remotely selectable are mounted on wheels or turn tables. Both, the recent development in infrared detector technology and the construction of the new generation of large telescope lead to a growing request for large size infrared instruments. The past generation of instruments was generally working with a 20 to 30mm optical beam. In order to fully explore the performance of the detector with the modern telescopes we need instruments, which can accommodate an optical beam up to 150mm diameter. This very large increase in the size of the optical components lead us to experiment with larger size bearings (typically with bore diametre of 100 mm and more) .

Introduction

The European Southern Observatory is currently building one of the largest telescopes in the World: The Very Large Telescope (VLT). One of the first instrument to equip this telescope is an infrared imager spectrometer (ISAAC). This instrument includes 13 functions among which the grating unit is the most demanding one. This unit which carries 2 gratings back to back will carry a total rotating mass of about 8000g. The following table summarises the most important parameters of the technical specification:

Table 1. Technical requirements

1. Angular stability (while the instrument is rotated)	< 2 arc sec
2. Angular positioning accuracy	< 3 arc sec
3. Angular run out (wobbling movement of the axis)	< 3 arc sec
4. Operating temperature	< 70 K
5. Operating pressure	< 1E-6 mbars
6. Cooling time (from room temperature to 80K)	< 24 hours

As the VLT is going to be installed on a remote site in the Acatama desert in Chile , these performances have to be guaranteed over a period of 10 years with minimum human intervention.

Choice and adaptation of the ball bearing

As the requirements show clearly, the unit needs to position and support the optical elements accurately. During observation the telescope follows the movement of the stars and the instrument which is directly bolted on it follows its movement too. All components of the instrument are subject to permanent rotation of the gravity vector. A first assumption would lead to consider a very rigid bearing as only solution to meet the point 2 of the technical requirements. The rigidity of a bearing depends directly on its internal preload (preloading of the balls onto the races) . The

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experience gained with optical instruments operating at room temperature shows that it is not reasonable to increase the internal preload such that the bearing can meet this specification. An excessive preload will directly lead to an increase of the running torque and eventually to a prematurate damage of the races . A better approach is to balance the rotating mass on both sides of the bearing plane. If this operation is done very carefully by addition and fine tuning of a counter-weight, angular stability of a fraction of an arc second can be reached even with a very low internal pre-load.
The second main driving parameter in the choice of the bearing is its thermal conductivity. Ideally we would like to cool the rotating mass through the bearing in order to avoid either a slip cooling finger which would lead to a higher drag or a flexible thermal connection which would forbid a continuous rotation of the unit. The thermal conductivity of a bearing depends mainly on the following parameters:
- Number, diameter and material of the balls
- Nature of the ball/race contact
- Contact pressure of the balls on the races

Three different types of pre-loaded bearing have been tested: a single four point bearing, a pre-loaded pair of thin ring bearing and a pair of angular contact ball bearing. Further tests have been carried out on this last bearing type in order to improve the thermal conductance using robin and tungsten carbide (WC) balls. A last attempt to improve the thermal behaviour has been done using gold coating on the races.

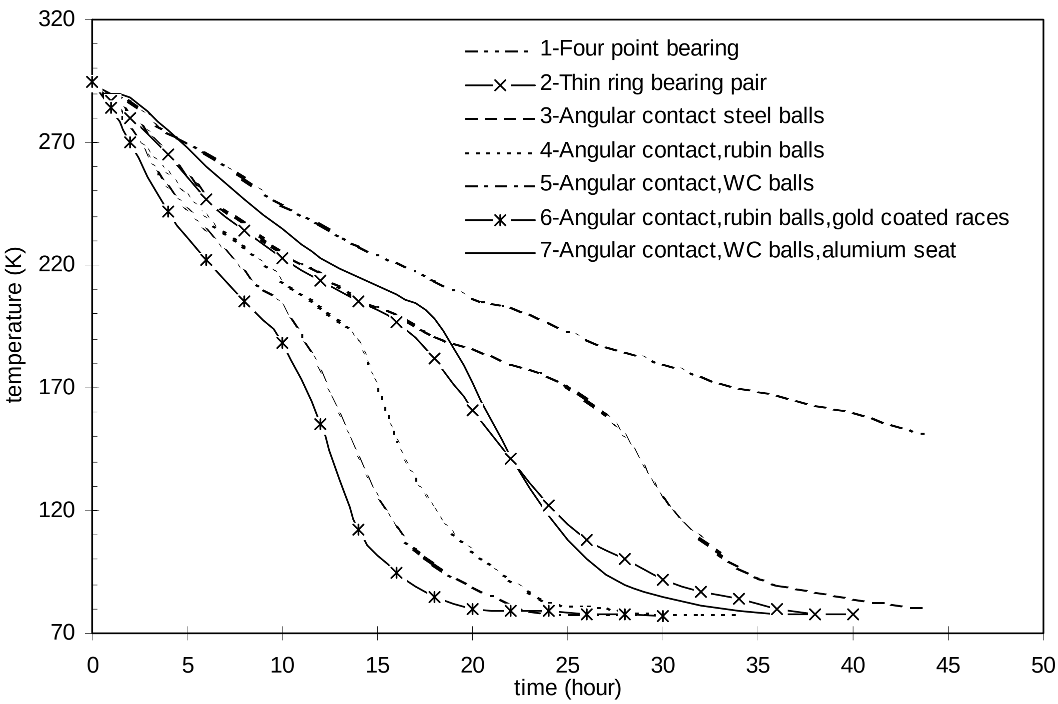

Figure 1. Cooling of a 8000g aluminium mass through various ball bearings

The poor thermal performance of the four-point bearing (curve 1) can easily be explained by the low number of balls it includes. The thin ring bearing (curve 2) which uses some 200 balls has a disappointingly low conductivity. This is mainly due to the fact that only a restricted number of the balls is in contact, and this is the case even when the bearing is mounted on a seat with high geometric quality. Using a material with high thermal conductivity for the balls has a direct impact on the conductance of the bearing. Moreover the use of balls from different material than the races has the second advantage of preventing any cold welding which would be

favorised by the high vacuum environment. The lack of experience and the high fragility of the robin lead us to abandon this material even if the results are very promising. The material and the coating of the races are very important. However the mounting of the bearing also plays an important role in the thermal behaviour. In order to ensure the mechanical performance the inner ring of the bearing is to be mounted with a press fit on a seat machined in a geometrical quality 5. The best thermal performance is obtained when a similar fitting is also used for the outer ring. The use of an aluminium seat for the outer ring can only be realised with a clearance for the differential thermal shrinkage. This gap also causes a delay in the cooling time (curve 7).
Finally the following angular contact bearing has been chosen:
71920 E TPA PA7 UL from IBC in "O" arrangement, with a 250 N pre-load and fitted with tungsten carbide balls. Both inner and outer races are mounted with press fit in seats machined martinsitic stainless steel. The thermal conductance of this bearing has been measured to be 0.7W/K at 300K and 0.3 W/K at 80K.

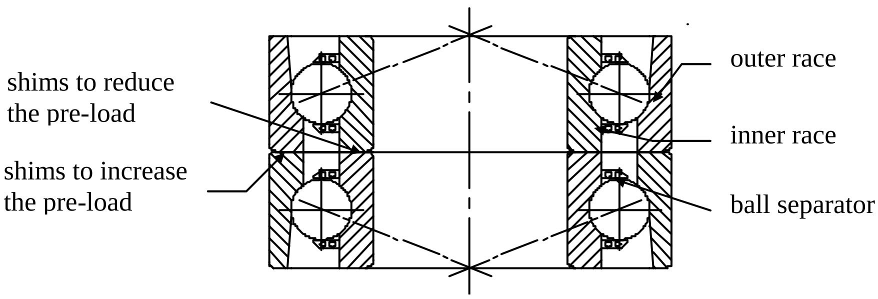

Figure 2 Pre-loaded pair of dismontable oblic contact ball bearings

Choice and application of the lubricant

Pure molybdenum disulphide (MoS2) is well known to be one of the most efficient lubricants under vacuum. This coating generally applied by sputtering suffers the disadvantage of being expensive and of having a rather long delivery time. Moreover MoS2 is a very good lubricant only under vacuum where when very pure it has its lamelar structure. In order to simplify the test procedure of the instrument functions it was very important to have a solution, which can be used, both in air and in vacuum at cryogenic temperature without significant differences. A series of test allowed us to qualify a commercial dry lubricant for this application.

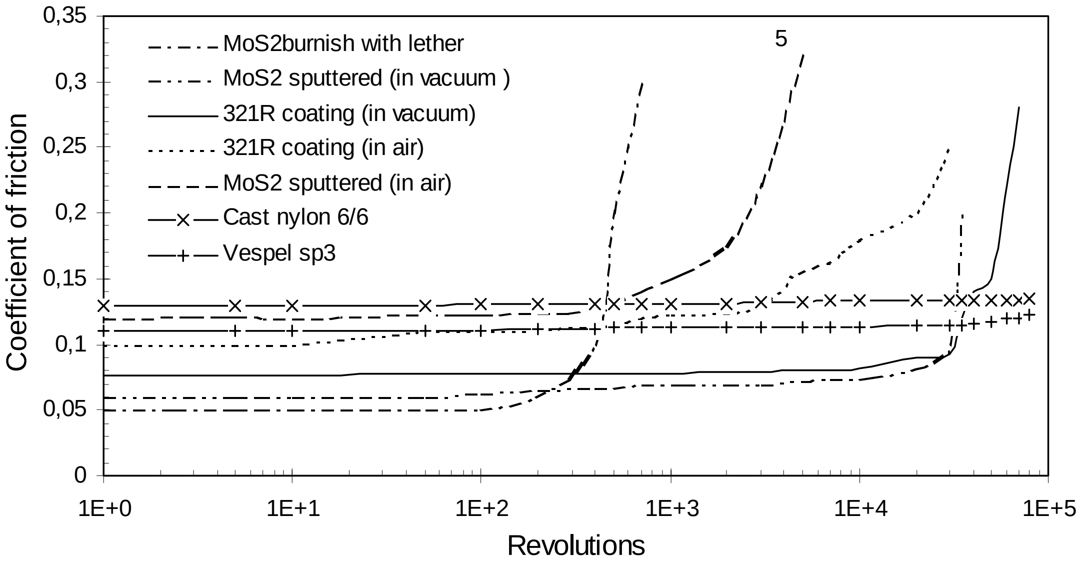

Figure 2. Coefficient of friction and life time comparison of various lubricants

Looking at the dynamic of a bearing we can see that the ball separator is the only part which is subject to sliding, whilst the balls are rolling on the races. Therefore special care will be taken when realising this cage for which a non-metallic self-lubricating material will be used. One of the best candidates for this application would be the Vespel SP3 from Dupont, which is a MoS2 filled polyamide. As this material is extremely expensive and almost unavailable in large sizes we started an investigation in order to find an alternative.

Therefore the following combination has been selected for the lubrication of the bearing:

- Lubricant: 321R from Dow Corning (Pel Kovenst. 152, D (0000) Munich)
- Ball separator: Cast nylon 6/6 from Nylacast Oilon ltd., Leicester, LE5 OHD, U.K)

The choice of a lubricant commercially available as a spray was a first step. This needed to be completed by the development of a method of application, which can guarantee the mechanical quality, and accuracy of the races. A variation of one micron in the thickness of the lubricant film is enough to cause a 2 arc second run out error. After many months of experimentation and various unsuccessful tries an application procedure has been developed which allows fully keeping the mechanical quality of the bearing and ensuring a smooth reliable operation at cryogenic temperatures.

The bearing is disassembled and carefully cleaned in an ultrasonic bath filled with acetone. The bearing is reassembled with a ball separator which has been generously coated .The pair is assembled with only 50% of the original pre-load for a first run-in operation (≈5000 rev). The run-in will continue with 3 more periods using respectively 75, 100 and 110% of the pre-load. During this process the lubricant is transported from the ball separator onto the races. Using different pre-load ensures to have a somewhat larger area of the races coated.

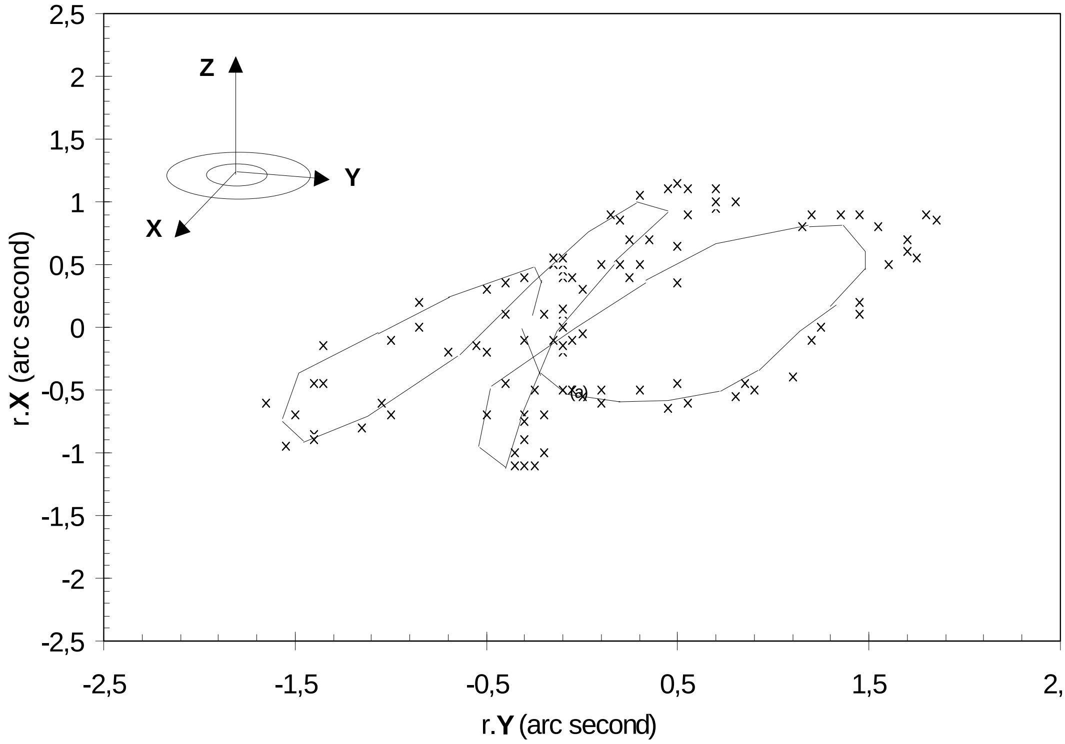

Figure 3. Run out error of the bearing after coating

The bearing is then disassembled and the races are wished with cleaning paper. The final ball separator is coated with a very thin film of lubricant. We take care to brush it strongly with a nylon brush in order to remove all the excess coating which later could be a source of pollution. The bearing can be finally assembled with the final tungsten carbide balls.

Results, status

In the pre-design phase two prototypes of this unit have been build in order to fully qualify the bearing technology and the drive principle. The design of the drive is based on a worm wheel mechanism powered with a 5-phase stepper motor. As the positioning accuracy rely only on the motor steps it is extremely important to keep friction and torque as low as possible along the complete cinematic chain. The running torque at 80K has been measured on ten sets of the selected bearing after complete adaptation. The dynamic torque is in all cases lower than 0.55Nm while the static torque was never measured to be above 0.6Nm.
The various measurements that have been carried out have clearly demonstrate that these two units met in all points the technical specification. At the end of the test the prototypes have been used for a lifetime test. In order to simulate an utilisation over a period of 10 years the unit has been rotated over 1 million of revolutions. During the test the positioning accuracy has been regularly measured. Figure 5 shows measurement recorded at two different stages of the test.

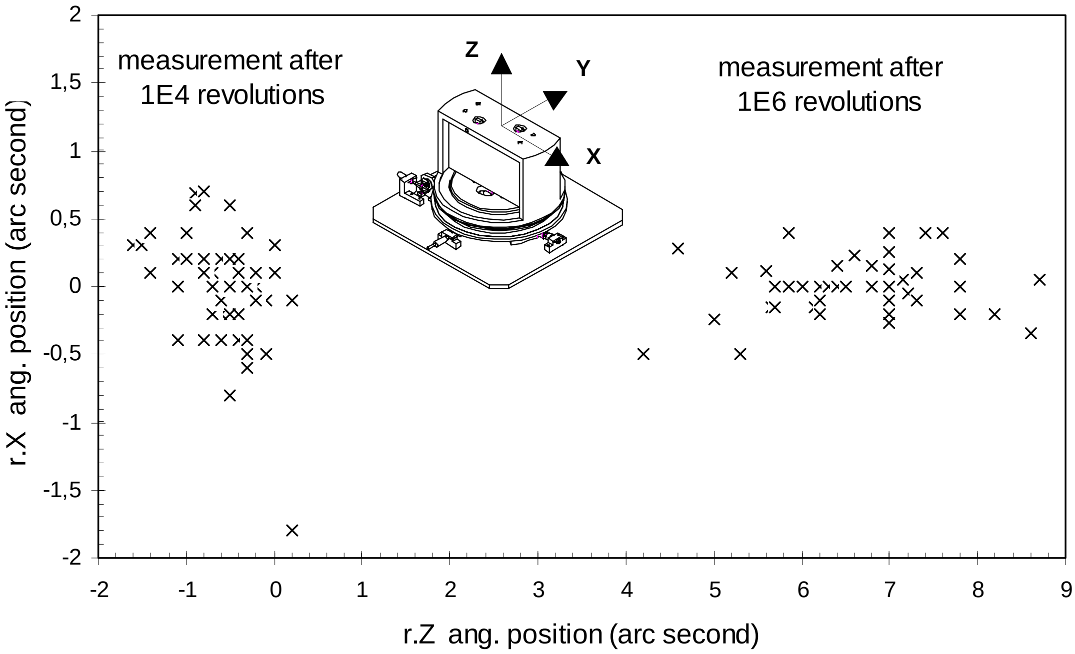

Figure 4. Angular positioning accuracy

The two units survived without any failure the 10 years lifetime simulation. Analysing the results of the positioning accuracy measurement we have to distinguish the two following cases. In the plane of the bearing (r.X) the accuracy improved, this parameter which depends directly from the bearing seems to indicate that a longer run-in would be benefit. The degradation of the angular positioning around the rotation axis (r.Z) is caused by two sources: the torque noise of the bearing which increased significantly during the test and the degradation of the drive which performed a total of 400 millions revolutions.
At this time the final grating unit as well as all other units of the instrument has been build and tested. The instrument is entering in the system test phase.

Conclusion

The lubricant and its application procedure have proven to be extremely reliable. The same coating has also been applied successfully to the roller screw, which provides the translation of the instrument collimator.
This technology is confirmed to be a very simple and cheep alternative for many mechanisms operating in a cryogenic environment.

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