DESIGN, ASSEMBLY AND PRELOADING OF BALL BEARINGS FOR SPACE APPLICATIONS – LESSONS LEARNED AND GUIDELINES FOR FUTURE SUCCESS

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ABSTRACT

The use of ball bearings in the space industry is commonplace, with a broad range of applications from the most precise pointing mechanisms to extreme long -life and performance-demanding reaction wheels and simpler single-shot devices where precision, performance and life may be less difficult to achieve. Though most application developments are ultimately successful, the lessons learned from the incorrect implementation of ball bearings are, usually for understandable commercial reasons, not widely distributed – but often extremely valuable for industry.

The organizations contributing to this paper have an unprecedentedly clear view of numerous spacecraft applications, both successful and otherwise, together with many years of experience in design, handling, lubricating, preloading and testing ball bearings for space applications.

In order to help the space community to avoid design or handling errors in future, ESA has mandated bearing manufacturer ADR, and the European Space Tribology Laboratory (ESTL) to create a useful guideline which summarises both best practice and a distillation of the lessons learned from many programs.

This paper presents a selection of the more valuable and generally applicable lessons learned by these organizations in the last 30-40 years together with an overview of the recently published guideline document itself [1] which embodies this experience and contains recommendations concerning ball bearing selection, design and conceptual rules and recommendations for assembly, preloading and verification testing.

The novelty of this paper comes from the broad range of experiences and applications, both good and bad that the organisations concerned have observed and the publication (for the first time) of guidelines for use by the space mechanisms community in Europe.

1. BALL BEARING CONSIDERATIONS FOR SPACE MECHANISM DESIGNERS

The selection and application of ball bearings for spacecraft mechanisms must take into account a number of considerations and even rules which are in general different from other applications, e.g. ground-based industrial or defence applications. Some of the main considerations related to ball bearing selection, specification and use for spacecraft mechanisms are listed below:

- In addition to the mechanism specification, a wide range of space industry (e.g. ECSS-E-ST-33-01, AIAA-S-114-2005), aerospace (e.g. AS9100) and industrial standards (e.g. ISO76) may be implicitly or explicitly applicable.
- Bearings, materials and processes must be space-approved or specifically Qualified for new space applications by a dedicated validation campaign. Furthermore there is usually a requirement for traceability at all stages of the manufacturing process (for example through manufacturer or processor serialisation, batch traceability, materials certification, Certificates of Conformity etc.).
- Bearing systems must be designed with the intent of achieving optimally low mass and (usually) high stiffness, combined with high reliability, minimal inter-unit manufacturing/performance variability, and the capability to sustain launch vibration-induced loads without damage or degradation. Yet typically development programmes might involve manufacture of only very few units prior to in effect "freezing" the design and all processes for Qualification and overall the quantity of units manufactured is rarely sufficient for any meaningful statistical analysis.
- In contrast to industrial use, fits selected for spacecraft bearings must usually be compatible with ease of dis-assembly both to facilitate inspection after test and also because of the often

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high manufacturing cost of interfacing parts (which could not be scrapped due to any potential bearing related problem).

- In general both solid and fluid lubricants may be options, the selection often being dictated by contamination or ground-test requirements, yet the lifetime of the lubricant in the application must be demonstrated by an appropriate life test as part of the full Qualification test campaign for any newly developed hardware (including conditioning of tribological components by application-enveloping vibration/shock tests prior to subsequent thermal vacuum life test).
- After launch, very low torque and torque noise performance must be sustained despite exposure to a challenging thermal and vacuum environment over (often) long lifetimes.

Space mechanism designers are therefore faced with a wide range of performance factors and design considerations to be addressed in order to make an appropriate choice of bearing and lubricant for any new application. Once chosen a high level of "attention to detail" underpinned by sound selection and application, analysis and test is required in order to ensure that the few items of hardware which will typically be built have the best chance of achieving the required performance and lifetime in order to minimize overall development risk and cost.

Over many years, ESA has supported industry by funding generation and dissemination of new data and tools to assist in this process, for example the publication of the Space Tribology Handbook [2] and the development of the bearing analysis code CABARET [3]. These are supplemented by several other ESA-supported handbooks which relate to other areas of mechanism performance. However it is only recently that emphasis has been placed on providing specific guidance for the most detailed design, selection and application of ball bearings for space applications, motivated in part by some notable but perhaps un-attributable and certainly non-public-domain issues which ESA/ESTEC has observed and wishes to avoid in future programmes.

2. LESSONS LEARNED

Such issues then must be considered as "Lessons Learned" and the list presented below is a selection of some which need to be highlighted because of their potential to result in costly investigative work, re-design or even premature failure on orbit.

2.1. Confirmation of Bearing Seating

Unlike terrestrial applications, spacecraft mechanisms typically need to be dis-assembled in a non-destructive manner in order to preserve post-test features and demonstrate compliance with ECSS mechanism (ECSS-E-ST-33-01 Qualification Test Success Criteria) requirements. For this reason despite the need for minimal misalignment of precision parts, it is usually the case that interference fits are avoided. In consequence bearings may be fitted with minimal assembly force but due to this it is still more critical to ensure they are fully seated on installation prior to clamping load being applied. The lesson here is that precision and depth of bearing seats, width of bearing rings and of any spacers must be controlled and confirmed in order to ensure correct bearing installation and preloading. This can also be achieved by detailed measurements of piece parts and sub-assemblies and facilitated by provision of appropriate probe-holes in bearing clamps together with accessible datum surfaces on the outside of housings in order to confirm the bearing location. Provision at the design stage of access to shafts for measurement of axial and radial runout and introduced misalignments is also a valuable aid to assembly verification.

2.2. Clamp Load Setting

Where excessively high clamp loads are used, there is a risk of bearing ring and raceway deformation which will result in unexpected stresses within the bearing and potentially anomalous torque or stiffness behavior. Experience suggests it may be relatively easy to exceed a recommended bearing clamp load that generally should be between 3 and 5 times higher than the preload. For example, a large number of parallel fasteners all with modest torque will result in a high cumulative clamping load if this is controlled by fastener torque alone. The lesson here is to monitor carefully and ideally control by design the clamping loads introduced in order to secure the preload without deforming the bearing rings.

2.3. Bearing Preload Setting and Monitoring

Loss of preload after vibration test, can be indicative of excessive settling at component interfaces. Whilst some small degree of settling is relatively normal, ring axial motion which is sufficient to lose preload is not acceptable and may indicate issues with bearing housing geometry or clamping. Even in the absence of vibration, preload may be lost due to ring wear during ground test if the lubricant becomes ineffective. For this reason one lesson learned is that not only is it essential to verify set preload after assembly and run-in (as required by the ECSS Mechanisms standard) , but also to make provision for its measurement and re-confirmation at higher levels of sub-assembly or development (e.g. instrument sub-assembly level). Whilst not a formal requirement, a design which permits this would certainly facilitate diagnosis of test anomalies/confirmation of suitability for flight.

2.4. Problems with preload setting and sliding interfaces

Neither hard nor soft preload applied to bearings is entirely without potential for incorrect application or inherent failure-promoting possibilities.

In both cases the order of bearing mounting/fitting and preloading operations can be significant to the preloaded outcome and the correct sequence needs to be maintained for each assembly manufactured.

Setting and measurement of hard preload, especially for duplex (matched) bearing pairs seems straightforward, the "free-state" (without housing or shaft) preload being set and measurable by the bearing manufacturer. However it is often the case that the "as-assembled" preload (with housing and shaft) may be increased, and especially for solid lubricated bearings may be found to have been increased by the presence of the solid lubricant (presumably due to its impact on bearing internal clearance or local conformity) and this is, so-far at least, not easily predictable. Furthermore it is very difficult to measure directly and accurately the "as-assembled" preload of a hard preloaded bearing system. Whilst the use of a pre-instrumented shaft/housing can be effective, it is more likely that this confirmation will need to be obtained by pre-characterisation of the torque versus preload behaviour of the bearings (for example in a spring-preloaded test setup as a calibration of preload) enabling subsequent fitted preload to be estimated from low-speed torque measurement.

In contrast, soft preloads (whether set in the preferred method using a compliant diaphragm, or the less favoured method of using sliding interfaces at shaft or housing) is relatively easily and accurately measurable – for example, directly by external press load or from the load versus deflection characteristics of the compliant element. However issues have been recently reported with bearing ring rotation in operation (which might lead to debris and potentially fretting damage), failure of sliding interfaces (causing the nominally soft preloaded system to behave as if hard preloaded) and even unexpected bearing ring wear. From the above it is clear that a general lesson learned is the need to take special care in the selection of bearing ring fits, the details of bearing clamping as well as the specification of geometric tolerancing, surface treatments and /or lubricants in order to avoid as far as possible unexpected performance problems.

2.5. Management of Assembly Hyper-statism

By nature, a hard-preloaded configuration is considered to be "hyper-static" which means that assembly tolerances between the bearing and its interfaces can have a large impact on the bearing system by introducing additional unexpected loads or load paths.

This "hyper-statism" is usually managed in one of two ways:

1) By use of select-on-assembly parts to accommodate tolerances.

2) By tolerance budget control – that is, by ensuring that regardless of the tolerances of individual parts within their respective tolerance budgets the configuration is always preloaded in accordance with the design intent.

Specifically the aim of these control measures is that when the preload gap set-up by the manufacturers is known, the gaps on outer and inner clamps, Fig. 1, can be managed taking into account the stiffness of the clamp rings and fasteners, but without excessive gap when fully clamped.

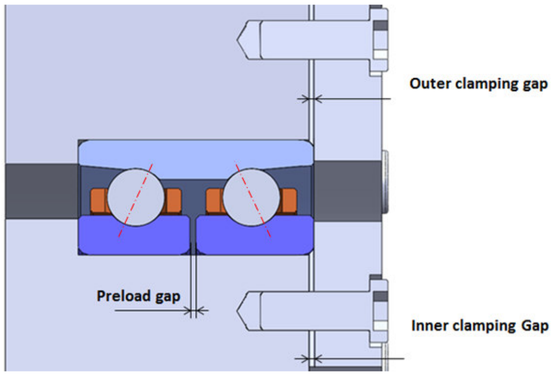


Figure 1. Gaps in hyper-static hard-preloaded bearing system during assembly (but prior to final clamp load application).

2.6. Lessons Learned Summary

The recently published Guideline document [1] provides additional guidance on the above issues and a wide variety of other practical points relevant to the design, assembly and testing of ball bearings for space applications, for example, appropriate handling of bearings, verification of preload and running-in with different lubricants.

In the remainder of this paper we will highlight some of the more important information provided in the guideline document.

3. BEARING CONCEPTS

The guideline provides an overview of the main characteristics of the different ball bearing types used including both the common deep groove (radial) and angular contact bearings but also more specialised super-duplex bearings which can provide some advantages in terms of ring misalignment and the custom "integrated ball bearing" designs available from some manufacturers. The custom designs offer integration of flanges holes and other features to facilitate assembly and minimise interface compliances and tolerance effects providing the following benefits in space applications:

- Reduced part count, assembly geometric dispersion and interface count which may be advantageous in applications with high positioning accuracy (see Fig. 2 below).
- Allow unusual shaped features to be fully integrated and, where integrated clamping is provided can be less likely to suffer adverse effects of variable clamping loads as bearing grooves can be manufactured with rings pre-clamped.

  The guideline also draws attention to the need to consider the high sensitivity to stress corrosion cracking of some materials used in custom designs. One recommendation which must be noted is that that threads should not be used in custom designs, and that the geometry should be such that the application of the preload should not create any tensile stress in the loaded parts of the bearing ring adjacent to the balls (load path from the ball contact points to the surfaces clamped by fasteners).

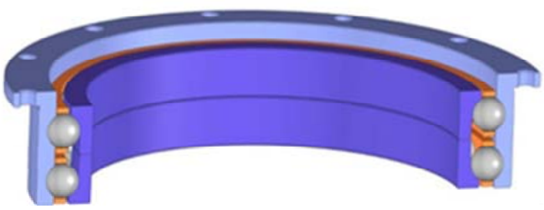


Figure 2 Ball bearing with custom design (integrated ball bearing flange)

In addition to providing an overview of the bearing types, the Guideline also provides an overview of the applicability and features of common cage (also called separator or retainer) designs (Fig. 3) and their role in lubrication – though cage design itself is a specialist function not included in the Guideline and ultimately verified for each application only by test in an appropriate environment.

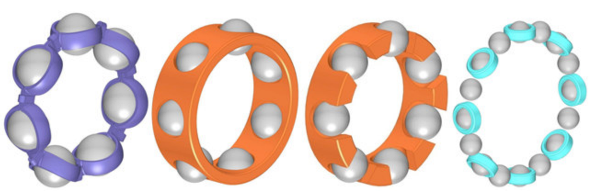


Figure 3. Most common cages (Two-piece pressed, one piece cage, snap-in crown, toroid)

4. CONTACT STRESS

The Guideline provides details of the necessary steps in selecting and sizing bearings for space applications, in particular the following points are highlighted and discussed:

- Identification that ball bearing material fatigue is rarely a consideration in space applications – life being more often limited by lubricant performance degradation.

- Sizing of bearing for launch vibration based on the limitations on bearing static (quasi-static in the case of launch vibration) load related either to sub-surface yield, or truncation. Truncation is the point at which as the axial load is increased, the elliptical ball/ raceway contact formation is disrupted by the close proximity of the contact to the edge of the groove (land) leading to unpredictable, high and damaging local stresses, see Fig. 4.

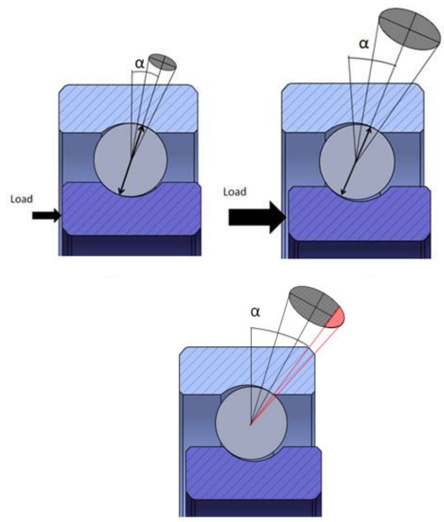


Figure 4. As load increases, ball-raceway contact ellipse size grows, contact angle α increases until ultimately truncation occurs (lower image).

- Explicit definition of ECSS-allowable (de-rated) peak Hertzian contact stress limits for ball bearings manufactured from various materials.

- A straightforward approach for making an **initial** sizing estimate for the bearing (which is to select a bearing with catalogue static rating at least twice the required load rating for worst case launch vibration) prior to detailed bearing design optimisation and analysis.

5. PRELOAD CONCEPTS

The Guideline provides an introduction to the different conceptual means of achieving either hard, or soft preloads and some of the main points highlighted are summarized below for each.

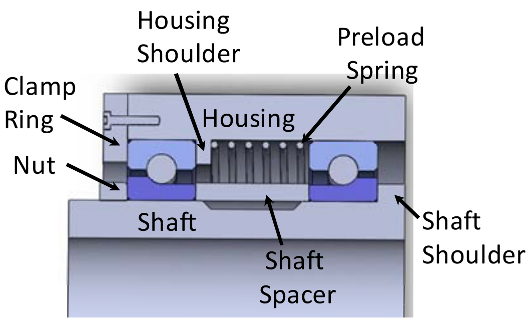
5.1. Soft preload

Classically soft preload, also called "elastic" or "compliant" preload, is applied by use of an axial spring element or a flexible diaphragm.

Clearly, as compared to hard preload, the main advantages of soft preload for space applications are that a soft preload renders the bearings more tolerant of thermal gradients, wide ranges in operational temperature, debris or uncontrolled lubricant transfer (for example in self-lubricating bearings) because the low stiffness of the spring or compliant element in series with one bearing of the pair limits the change in effective preload and so torque that is produced in each case. Furthermore this insensitivity also means that under adverse thermal conditions thermally induced offloading and gapping (as preload reduced to zero) is unlikely.

However compliantly preloaded systems are by nature asymmetric with respect to response to external axial loads (including dynamic response to vibration loads) and in general offloaded by lower loads relative to preload than would the equivalent hard-preloaded bearing system.

The Guide shows schematically the different configurations (e.g. Fig 4. and Fig. 5) and presents recommendations for their successful implementation.



Schematic Representation
(Black rectangles indicate blocked/rigidly located)

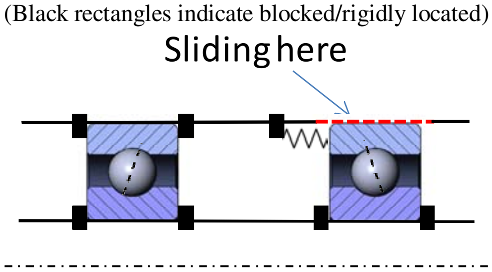
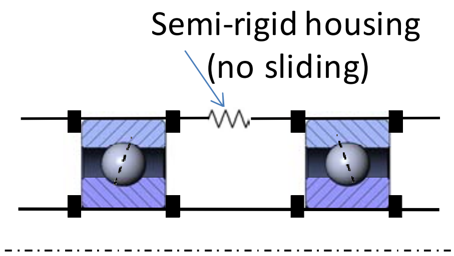


Figure 4. A typical soft bearing preload configuration (with sliding interfaces) in reality and schematic form.

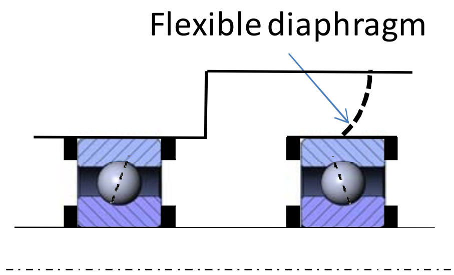
For example, with respect to the compliant preload with sliding interface as shown Fig. 4 above, the Guide recommends:

- Selection of appropriate fits, geometry, thermal matching, location, hardness and lubrication (in accordance with ECSS-E-ST-33-01) of the sliding bearing seat.
- Provision of overload and over-travel protection (for the shaft), stiffness characterisation of both the compliant element and any spacer and means for confirming assembled precision and preload within appropriately toleranced bearing seats.
- Use of helical or other coil springs or wave-springs is recommended, whereas Belleville type springs are excluded in flight applications due to low stability.

Alternatively, where a semi-rigid housing or flexible diaphragm is used to provide the preload (as shown in Fig. 5) then the Guide recommends most of the above points plus:



Back-to-Back Soft (compliant) Preload With Semi-Rigid Housing (No sliding)



Back-to-Back Soft (compliant) Preload With Flexible Diaphragm (No sliding)

Figure 5. Soft preload bearing configurations (without sliding interfaces) in schematic form.

- Matching as far as possible flexible diaphragm/ semi-rigid housing seat coefficient of thermal expansion to those of the bearing to minimise changes in fit with thermal strain.
- Appropriate stress relief of the flexible element.
- A relatively tight fit to diaphragm housing for floating bearing.

5.2. Hard preload

Configurations for hard preload, especially of matched duplex pairs which are relatively widespread in space applications are also presented.

Whilst bearing manufacturers can typically manufacture hard preloaded bearings with a “free-state”(without shaft or housing) preload tolerance of around 10-20% of nominal preload the role of clamping loads and fits need to be assessed as they can have a large impact on bearing performance.

When correctly implemented (e.g. Fig 7), hard preload can provide a higher bearing stiffness, better positioning accuracy and lower torque variability between identical units than would an equivalent soft preloaded (compliant) bearing system.

However correct implementation requires a good knowledge of the immediate thermal environment of the bearings (especially thermal gradients) in order to analyse the thermal effects on the preload and hence friction torque. Since hard preloaded systems are more thermally sensitive and prone to torque noise due to lubricant debris they may not be adequate for all application cases.

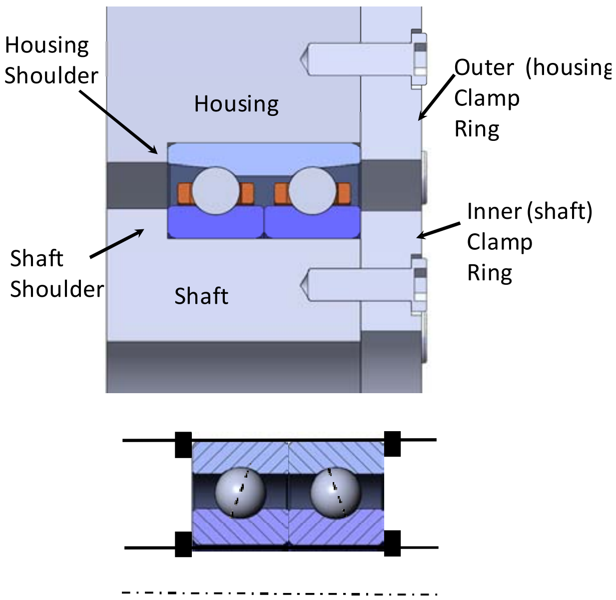


Figure 7. A typical hard preloaded (back-to-back) bearing configuration (without sliding interfaces) in reality and schematic form.

6. BEARING GAPPING

The Guideline also discusses the impact of launch vibration on ball bearings in which the preload is insufficiently high to avoid bearing “gapping”.

This is the phenomenon in which, under high external loads and moments compared to initial preload (typically during launch vibration), some balls within the preloaded bearings will be on-loaded (i.e. experience increased load) whilst others will experience a reduction in load until becoming totally off-loaded (free). If the external load or moment is further increased beyond this point there will be a nominal clearance (or gap) between some balls and the raceways.

This is most easily envisaged as well as most potentially damaging for axially loaded bearings, in which case, as shown below (Fig. 8), a large external axial load will on-load the left-hand bearing whilst all balls in the right hand might in the limit become off-loaded, permitting a gap between balls and raceway. Gapping under exposure to launch vibration loading is a concern because of the dynamic non-linearity this presents to the bearing system and the possibility of damage to balls, raceways or lubricant due to “hammering” when the balls come back into contact as the load is reduced or its direction reversed.

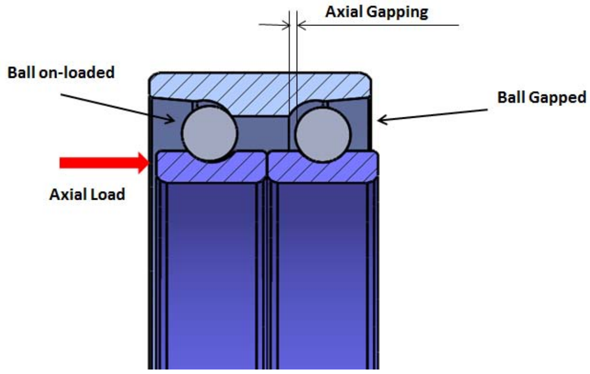


Figure 8. Bearing ball gapping under axial load

Note that whilst for a hard-preloaded system as shown the gapping is symmetric with direction of loading, for a compliantly preloaded (more widely known as “soft” preloaded) system it is highly asymmetric with a fixed axial load producing potentially some 10s or 100s of microns of axial gapping in one direction (with spring in the load line) and essentially none when applied in the opposite direction (with the spring element out of the load line).

Whilst gapping is not prohibited by ECSS-E-ST-33-01, it is required that ”if bearing gapping occurs during vibration, adequacy of lubricant and potential consequential mechanisms damage or degradation due to bearing components or shaft motion shall be demonstrated to conform to the specified functional performance and lifetime.”

Furthermore apart from the potential to affect system lifetime, gapping MAY be undesirable for system reasons.

Whilst industrial “rules of thumb” exist which state allowable (axial) gapping amounts (typically a few 10s of microns), a global recommendation concerning a maximum allowable gapping value cannot be substantiated experimentally hence no industry standards exist in this area, though ESTL has provided guidelines and recent experimental studies which discuss this issue.

7. EXCESSIVE CLAMPING LOAD

When the clamping load is excessive, the internal geometry of the bearings is modified due to the deformation of the rings in all directions (see Fig. 9 below).

An excessive clamping load may generate:

- Overstress on balls / raceways (axial + radial)
- Overstress on shaft / housing (radially)
- Bearing system contact angle / stiffness modified
- Modified effective/fitted preload, stiffness and torque (unpredictable in magnitude)

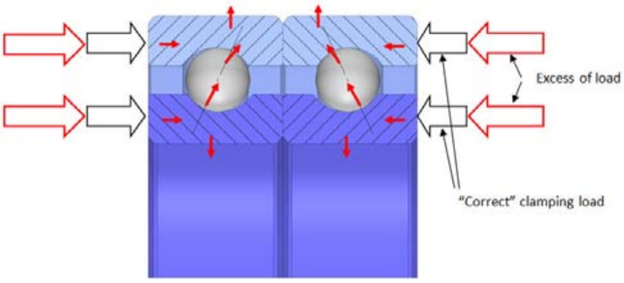


Figure 9. Influence of excessive clamping load on inner and outer rings (back-to-back configuration.)

An excess of load on inner rings and outer rings also generates deformation of the ball grooves which can lead to a high variability of the stiffness or torque characteristics of the bearings (this is unpredictable and in some cases excessive clamp load at one interface may actually REDUCE stiffness and torque of the bearing system). The deformations of the rings may also introduce large hysteretic frictional effects which can impact the bearing system during launch or vibration tests.

For these reasons excessive clamping load must be avoided and good spacecraft mechanism designs/assembly processes will feature means of ensuring the clamping load is within a certain acceptable tolerance limit to ensure known, predictable bearing fitted preload and hence good performance.

For a hard preloaded system as shown in Fig. 10 below, the clamping loads to be considered do not only concern the rings which must be clamped in order to close the "preload gap" set by the manufacturer. In fact the load lines in three different loops must be understood.

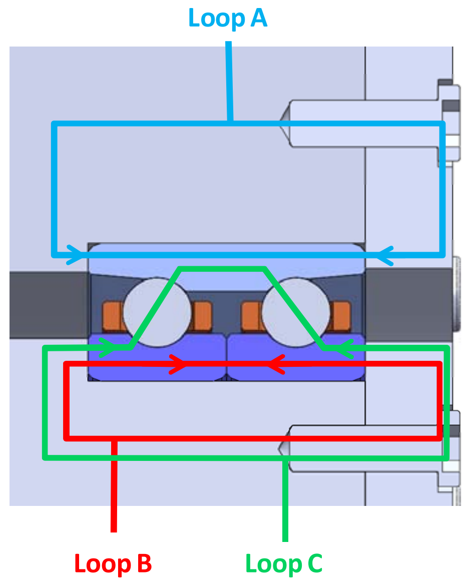


Figure 10. Preload and bearing clamping loops in back-to-back hard-preload configuration

- A = load to maintain the position in space of the bearing system
- B = Inner clamping loop: load on the rings to maintain the bearing at the shaft interfaces including preload loop C.
- C = Preload loop: the clamping load to maintain the preload alone (load needed to close the preload gap for the as-manufactured bearing).

The most important load loop for the bearing is Loop C. Unfortunately whilst load C is set by the manufacturer during the finish machining of the bearing pair offset, the total clamping load (B plus C) is defined only during assembly and the use of excessive clamping load will lead to ring deformation which could modify the distribution of load within the bearing (i.e. reduce or increase load C). Hence the total of loads B and C must be well controlled during assembly of the bearing pair into the spacecraft mechanism to avoid this undesirable outcome.

The A and B loops maintain the complete bearing system interfaces. Usually the load in these loops must be relatively high (usually significantly higher than the bearing preload for example) in order to prevent separation at the preloaded interfaces between clamps and bearing rings (and hence to prevent the possibility of unwanted ring motions) during launch or vibration tests.

If Loop B requires a high clamping load to maintain the bearing in place during launch, engineers must ensure that only a part of that load passes through the bearing (Loop C). The mechanism design must be adapted in order to ensure that the load in all preloaded loops remains at reasonable values.

As for a hard preloaded system the clamping loads in soft preloaded bearing systems also feature three main load loops as shown in the typical system below in Fig. 11.

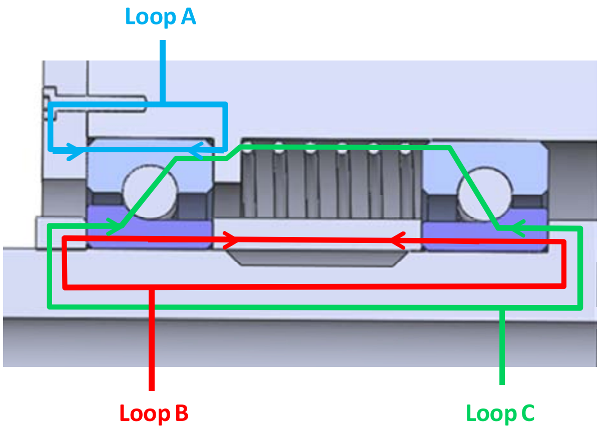


Figure 11 – Preload and bearing clamping loops in back-to-back soft preload configuration.

- A = Outer clamping loop: load to maintain the position in space of the bearing system
- B = Inner clamping loop: load on the rings to maintain the bearing at the shaft interfaces including preload loop C. (Clamp force = B+C)
- C = Preload loop: the clamping load to maintain the preload alone (load needed to compress the preload spring or flexible diaphragm if used by the defined amount to apply the required soft preload).

As mentioned earlier, the use of an appropriate clamping load ensures that the bearing remains seated during launch vibration AND avoids deformation of the bearing rings. A notional relationship between preload and clamping load is provided in Fig. 12 below.

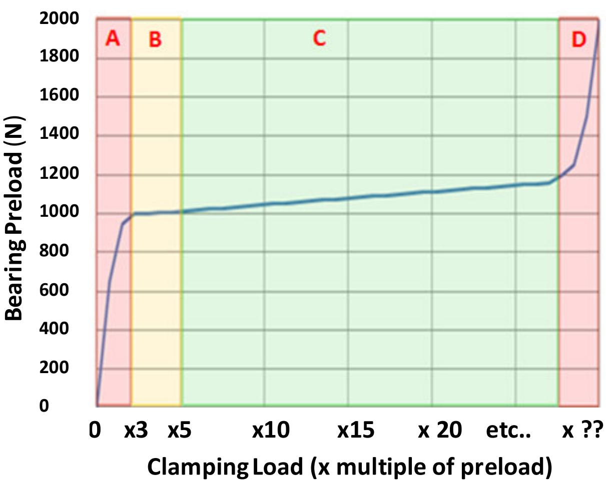


Figure 12 – Assembled preload versus bearing clamping load

A = Low clamp load: zone of deflection of the bearing up to the preload

B = Ideal clamp load : ~ x3 to x5 the original preload of the bearing

C = Acceptable clamp load : depends on dimensions and geometry of the bearings (but could be up to the catalogue load capacity of the bearing or higher)

D = Excessive clamp load : deformation of rings (preload / torque increase).

As a "rule of thumb", it can be considered as reasonable to apply a clamping load in a range from approximately 3 to 5 times the nominal preload of the bearing pair up to the catalogue axial load capacity of the bearing pair (always safe with respect to vibration loads (which are at most 50% of catalogue load capacity in any direction)).

The extra clamping load (above the preload set by the manufacturer) must pass through the interfaces and not the bearing balls.

The bearing manufacturer may be consulted for confirmation of the appropriateness of any intended preload and clamping load.

8. CONCLUSION

This paper provides an overview of some ball bearing lessons learned by the authors, together with a summary of the content of the recently released Guideline document which is available from ESTL and ADR for use by organisations and entities in the ESA member states and subject to the terms of the ESA contract under which it was developed.

It is hoped that this paper and the Guideline document itself [1] can become a useful reference for newcomers in space mechanism design, such that basic mistakes may be avoided and/ or not discovered too late during a test campaign.

For more established readers, the recommendations and considerations highlighted in the Guideline will also be useful in the form of a checklist to promote adequately detailed consideration and justification of design choices and pre-empt review questions.

In cases of doubt, or the need for further clarification, both ADR and ESTL can provide assistance to industry.

The authors welcome feedback from industry both in the content of this paper and on the Guideline document itself which is already proposed for update to accommodate latest observations and industrial practices where these can be made public domain.

9. REFERENCES

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