SOFTWARE EVALUATION OF BALL BEARING PERFORMANCES (STRUCTURAL, TRIBOLOGY): COMPARISON OF NUMERICAL RESULTS OBTAINED BY FIVE DIFFERENT ENGINEERS FOR THE SAME SET OF INPUT

Alain BLANC [(1)]

[(1)] European Space Agency, ESTEC, Keplerlaan 1, PO Box 299
NL-2200 AG Noordwijk, The Netherlands, Email:alain.blanc@esa.int

ABSTRACT

Ball bearings are commonly used in space mechanisms. Those critical mechanical components are at the heart of space mechanism sizing. They shall both comply with a specific set of space related technical requirements and increasingly also with demanding programmatic challenges (in the mechanism design phase there is a strong need to make the right choices quickly & safely , but also to minimise non-recurring and recurring costs , and finally to comply with tight schedule).
Ball-bearing simulation software is a recognised tool useful not only to help engineers to better support early design trade-offs while giving the possibility to evaluate a large number of solutions through fast iterations but also of use when it is necessary to verify the selected ball-bearing design suitability during the design development phases (eg. PDR, CDR). As a summary, using software for ball bearing sizing appears a good answer to the technical and programmatic challenges raised.
The current landscape of simulation software for space ball bearings is composed of in-house software (mainly developed by bearing manufacturers like ADR, Cerobear, GRW and SKF), and a couple of specific commercial software tools. In Europe the most frequently used software tools are from CNES (RBSDYN) and from ESA/ESTL (CABARET) are both freely available with no-cost licence agreements available.
With the intent to build a software-improvement road map and associated R&D activities answering industry needs, a Working Group has been established involving main space ball bearing players (Satellite Primes, mechanism suppliers, ball bearing suppliers and agencies).
The working-group has recently evaluated the different software capabilities through:
- A structural case study
- A tribology case study

This benchmarking of the most commonly used software has highlighted concerning differences in the results obtained when analysing specific typical ball bearing loading cases – in some cases due to misunderstanding of the way to setup the problem, but in others due to analysis differences between codes which need to be better understood.
The goal of the paper is to share the outcome of this case study (results comparison, lessons learnt and recommendation)

1. INTRODUCTION

The current landscape of simulation software for space ball bearings is composed of in-house software (mainly developed by bearing manufacturer like ADR, Cerobear, GRW and SKF), and a couple of specific commercial software tools. In Europe the most frequently used software tools are from CNES (RBSDYN) and from ESA/ESTL (CABARET). Both are freely available with no-cost licence agreements.
With the intent to build a software-improvement road map and associated R&D activities answering industry needs, a Working Group has been established involving main space ball bearing players (Satellite Primes, mechanism suppliers, ball bearing suppliers and agencies).
The working-group has recently evaluated the different software capabilities through "back -to-back" comparisons of predictions of various software tools in two types of case study:
- A tribology case study
- A structural case study

The results of this benchmarking of the most commonly used software is presented here.

2. COLOUR CODE

The following colour code is used in this paper:

S1 – participant 1	(blue)
S2 – participant 2	(orange)
S3 – participant 3	(yellow)
S4 – participant 4	(purple)
S5 – participant 5	(green)
S6 – participant 6	(black)
Test datas	(grey)

Table 1. Colour code

3. TRIBOLOGY CASE STUDY

3.1. General presentation

The reference test proposed has been defined with ESTL, with the following logic:
- Evaluation of a **solid lubrication**, varying preload
- Evaluation of a **fluid lubrication**, varying speed

The case study selected represents around 70% of space mechanism cases and has the advantage of having experimental data without restriction of confidentiality.

All cases analysed are bearing pairs used in back-to-back configuration and having compliant preload. The actual value of the compliance has not been defined, as this should not affect anything in the analysis as we are not changing the temperature.

3.1.1. Solid lubrication

3.1.1.1. Case 1

<u>Input</u>

Parameters for the evaluation of the solid lubrication case (MoS2) are the following:

Ball diameter	6.35 mm
Pitch diameter	31mm
Free contact angle	15 degrees
Ball complement	11
Inner conformity	1.07
Outer conformity	1.08
Inner diameter	20 mm
Outer diameter	42 mm
Bearing Width	12 mm
Material ball	AISI 52100
Material inner ring	AISI 52100
Material outer ring	AISI 52100
Cage mass	3.29g
Mounting configuration	Back-to-back
Ball roughness RMS	0.018µm
Race roughness RMS	0.075µm

Table 2. Solid lubrication Case 1, bearing parameters

Preload 1	45 N
Preload 2	160 N
Preload 3	400 N

Table 3. Solid lubrication Case 1, Preload variation

Rotational speed	2 rpm
Rotating ring	inner
MoS2 friction coefficient in vacuum	0.015
Cage friction coefficient (Sintimid 15M-HT on steel in vacuum)	0.12
Temperature	22°C

Table 4. Solid lubrication Case 1, Operational parameters

<u>Output</u>
The following output parameters are analysed:

- Contact stress [MPa] function of axial preload [N]
- Mean torque [Nmm] as a function of axial preload [N]

3.1.1.2. Case 2

<u>Input</u>
Case 2 differs from Case 1 by the following modified parameters:

Ball diameter	7.14 mm
Ball complement	10
Inner conformity	1.14
Outer conformity	1.14
Cage mass	5.46g

Table 5. Solid lubrication Case 2, bearing parameters

Preload 1	45 N
Preload 2	300 N
Preload 3	500 N

Table 6. Solid lubrication Case 2, Preload variation

Cage friction coefficient (Duroid on steel in vacuum)	0.22

Table 7. Solid lubrication Case 2, Operational parameters

<u>Output</u>
The following output parameters are analysed:

- Contact stress [MPa] function of axial preload [N]
- Mean torque [Nmm] as a function of axial preload [N]

3.1.2. Fluid lubrication

<u>Input</u>

The following parameters for the evaluation of a liquid lubrication, Nye 2001 have to be considered.

Ball diameter	6.35 mm
Pitch diameter	31mm
Free contact angle	15 degrees
Ball complement	11
Inner conformity	1.07
Outer conformity	1.08
Inner diameter	20 mm
Outer diameter	42 mm
Bearing Width	12 mm
Material ball	AISI 52100
Material inner ring	AISI 52100
Material outer ring	AISI 52100
Cage mass	1.46g
Mounting configuration	Back-to-back
Ball roughness RMS	0.018µm

Race roughness RMS	0.075µm

Table 8. Liquid lubrication case, bearing parameters

Preload 1	48N

Table 9. Liquid lubrication case , Preload

Lubricant name	NYE 2001A		
Oil type: multiply-alkylated cyclopentane, no additives			
Density @ 20°C	0.841 g/ml		
Viscosity index	135		
Kinematic viscosity ν_0 (cSt) (ASTM D445)	20°C	40°C	100°C
	302	106.7	14.3
Pressure viscosity coefficient (inverse asymptotic) a* (m^2/N) @20°C	2.00e-8		
Thermal Conductivity K_f (W/m/K) @30°C	0.084		
Specific heat C_p (cal/g) @30°C	0.52		
Effective boundary friction coefficient @20°C	0.14		

Table 10. Liquid lubrication case, lubricant parameters

Rotational speed 1	2 rpm
Rotational speed 2	100 rpm
Rotational speed 3	500 rpm
Cage friction coefficient (effective boundary cof, for wet phenolic cage vs steel in vacuum)	0.14
Temperature	22°C

Table 11. Liquid lubrication case, Operational parameters

<u>Output</u>
The following output parameters are expected:

- Bearing resistive torque [Nmm] as a function of speed [rpm]
- Specific film thickness as a function of speed [rpm]

3.2. Main results

3.2.1. Solid lubrication Cases 1 & 2

Not all participants have been able to provide answers to this case study.

We observe that the data varies significantly between participants. A good illustration of this is offered by considering the mean torque predictions corresponding to Case 1 , 45 N preload::

- The experimentally observed mean torque measured by ESTL for this bearing system was **0.75** [Nmm].
- Mean torque predictions from the codes for this Case showed a wide range, with a mean of all codes being 0.74Nmm, and a spread as below::

Mean of predictions	0.74 [Nmm]
Min prediction	0.17 [Nmm]
Max prediction	1.60 [Nmm]
Standard deviation	0.6 [Nmm]

Table 12. Solid lubrication Cases 1 & 2, Case 1, 45 N preload, predicted mean torque summary

Same trend is observed for the 6 cases studied:

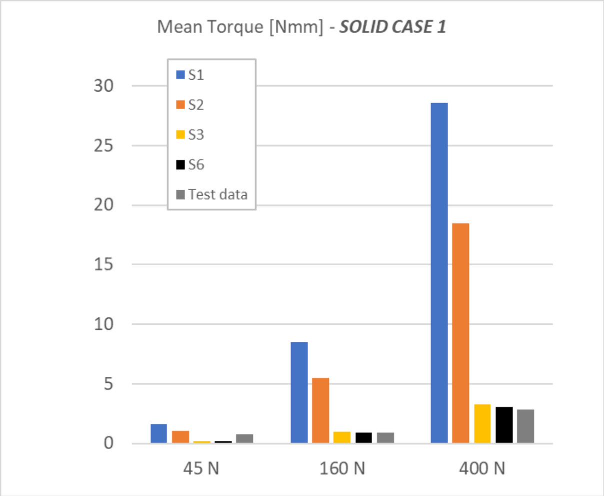


Figure 1. Solid lubrication Case 1, mean torque

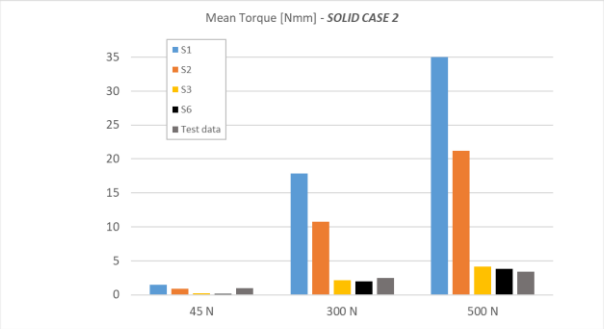


Figure 2. Solid lubrication Case 2, mean torque

3.2.2. Liquid lubrication case

Not all participants have been able to provide answers to this case study (note that the 4th participants is not the same for solid and liquid case)

<u>Bearing torque values</u>
We observe that the data varies significantly between participants
Illustration per case 500 RPM, bearing torque estimate (same trends is observed for the 3 cases studied):

- The experimentally observed mean torque measured by ESTL in this case was 14.7 [Nmm]
- Results obtained by analysis are :

Mean of predictions	11.56 [Nmm]
Min prediction	8.80E-01 [Nmm]

Max prediction	23.30 [Nmm]
Standard deviation	8.10 [Nmm]

Table 13. Liquid lubrication case, 500 RPM, mean torque

Same trend is observed for all cases studied:

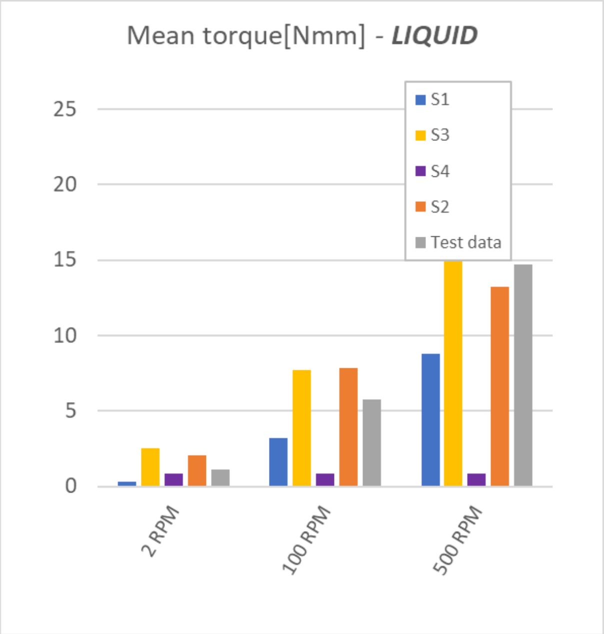


Figure 3. Liquid lubrication case , mean torque

Evolution of specific film thickness

The evolution of film thickness shows fairly close values with low dispersion between participants.

Illustration per case 500 RPM , film thickness, in µm :

Mean of predictions	0.31 [µm]
Min prediction	0.253 [µm]
Max prediction	0.360 [µm]
Standard deviation	0.040 [µm]

Here is the overall summary:

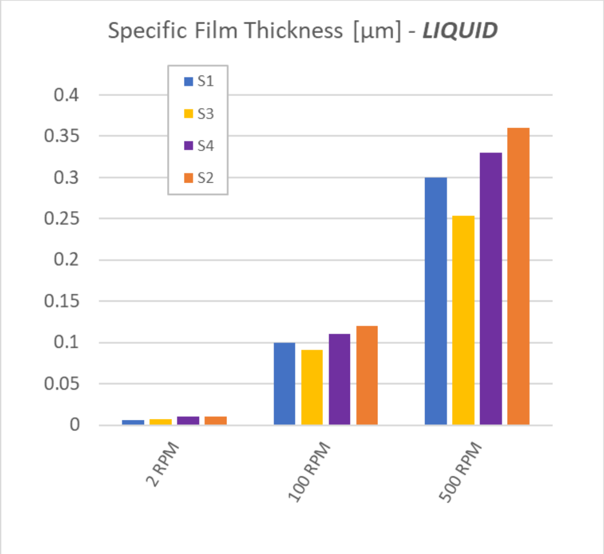


Figure 4. Liquid lubrication case, Specific film thickness

3.3. Findings, conclusion and recommendations

The main findings are the following:

- **Not all** participants have been able to provide answers to the tribological case study (solid or liquid lubricant)
- We observe that the for both solid and liquid lubricated cases the predictions of the codes for the bearing mean resistive torque parameter varied **significantly between participants.**
- In the case of liquid lubricant, **the evolution predicted film thickness shows fairly close values with low dispersion between participants**.

The predictions of bearing torque (tribological predictions) amongst the benchmarked codes differ widely. All codes provide a correct approximate “order of magnitude” torque prediction, which agrees approximately with experimental data and so can be considered to provide valuable engineering guidance, based on analytical results obtained.

However, it remains a strong recommendation **to fully characterise by representative test the ball bearing resistive torque at an early stage in new mechanism development.** Such a test removes modelling uncertainty and permits lower motorisation margins to be applied at the design stage.

4. STRUCTURAL CASE STUDY

4.1. General presentation

4.1.1. Configuration

The structural analysis reference case is based on bearing test units that were created by ESTL for a vibration test campaign. Those test units are based on a pair of preloaded bearings with a test mass fixed on the shaft.

Ball diameter	5.556 mm
Pitch diameter	20 mm
Free contact angle	15 degrees
Ball complement	9
Inner conformity	1.04
Outer conformity	1.06
Inner diameter	10 mm
Outer diameter	30 mm
Bearing Width	9 mm
Inner land diameter	17.15 mm
Outer land diameter	22.85 mm
Inner spacer length	0 mm
Outer spacer length	8 mm
Distance between bearing centre planes	17 mm
Distance between left bearing center and load point	8.5 mm
Material ring	440C
Material inner ring	440C
Material outer ring	440C
Material shaft	440C
Material housing	440C
Test mass	1.25 Kg
440C Young's modulus	209 GPa
440C Poisson's ratio	0.3
440C Density	7830 Kg/m^3
440C Thermal expansion coefficient	11.5 $x10^{-6}$ K^{-1}

Table 14. Structural case study, parameters

Bearing A is the bearing close to the mass and Bearing B is the other one. For soft preload cases, Bearing B is the bearing subjected to the preloading spring.

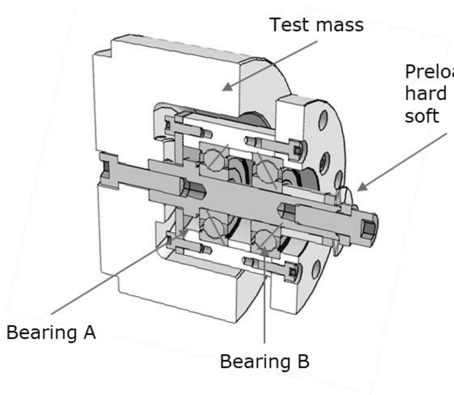


Figure 5. Structural case study

4.1.2. Load cases

14 cases were identified for inclusion in the code comparisons:
- 6 in hard preload configuration
- 8 in soft preload configuration.

CASE	PRELOAD			LOAD
	Value	Type	Stiffness	
	N		N/mm	Load Fa, Fr : N Torque M : N/m
1	160	Hard	-	-
2	80	Hard	-	-
3	20	Hard	-	-
4	20	Soft	250	-
5	20	Soft	900	-
6	160	Hard	-	Fa = 20, Fr = 50, M = 2
7	160	Hard	-	Fa = 0, Fr = 800, M = 5
8	160	Hard	-	Fa = 800
9	20	Soft	250	Fa = 10
10	20	Soft	250	Fa = 30
11	20	Soft	250	Fa = -30
12	20	Soft	900	Fa = 30
13	20	Soft	900	Fa = -30
14	20	Soft	250	Fa = 0, Fr = 800, M = 5

Table 15. Structural case study, Load cases analysed

4.1.3. Output

The following output parameter predictions have been compared:
- Inner / outer contact stresses for hard / soft preload cases
- Stiffness matrix of individual bearings for hard / soft preload cases
- Axial and angular stiffness of the pair of bearings
- Behaviour under gapping load (number of balls gapped and normal distance of gapping)
- Ellipse truncation
- Thermo-elastic behaviour
- Effect of mounting configuration

4.2. Typical results

4.2.1. Contact stress

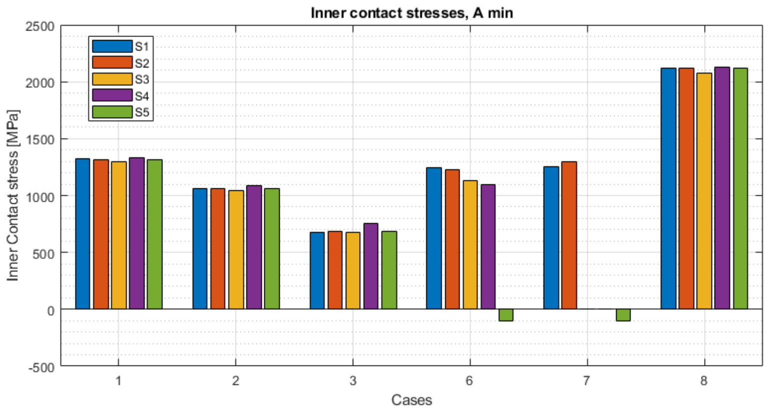


Figure6. Typical results, contact stress, hard preload

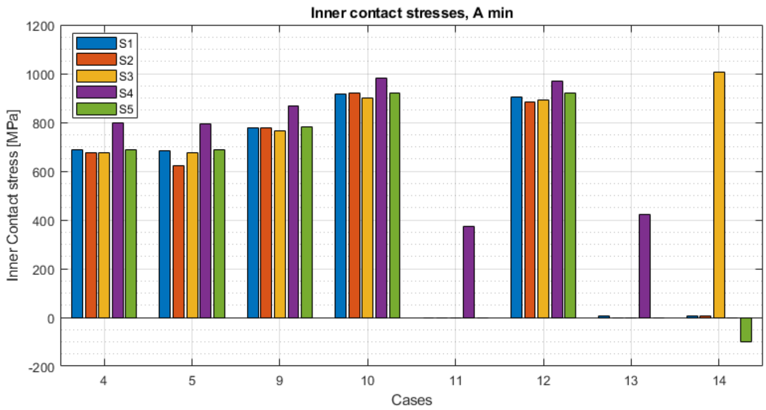


Figure 7. Typical results, contact stress, soft preload

4.2.1. Stiffness bearing system

Hard preload

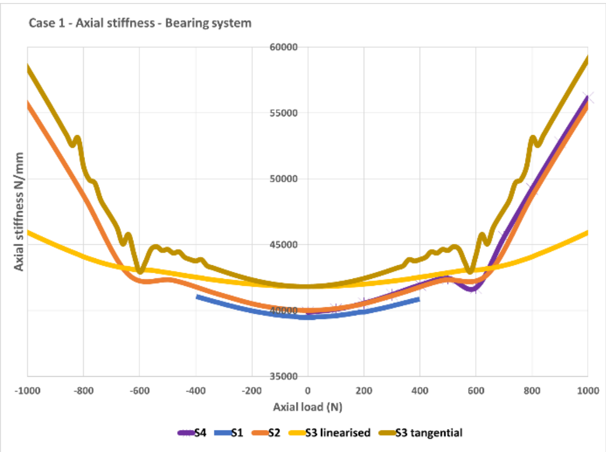


Figure 8. Typical results, axial stiffness

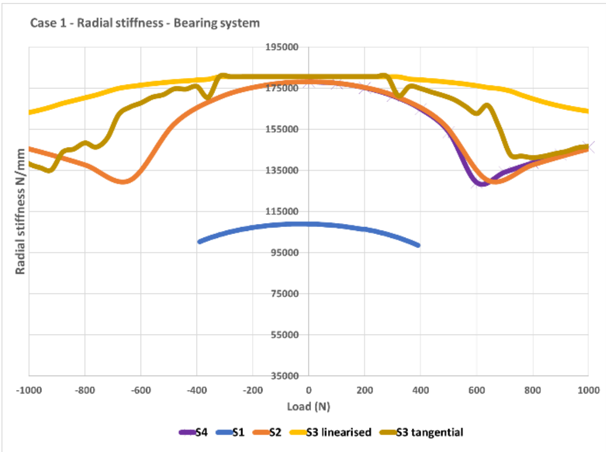


Figure 9. Typical results, radial stiffness

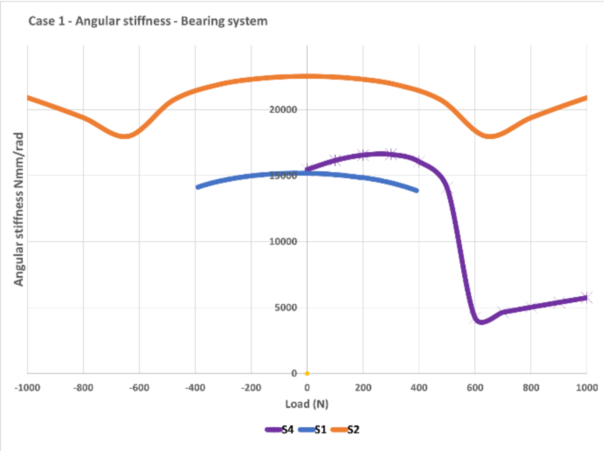


Figure 10. Typical results, angular stiffness

Soft preload

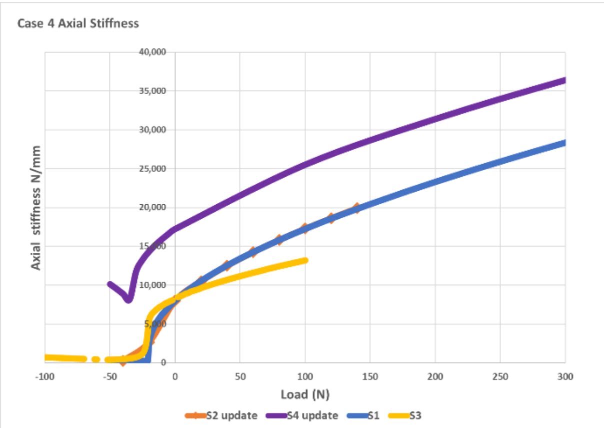


Figure 11. Typical results, axial stiffness

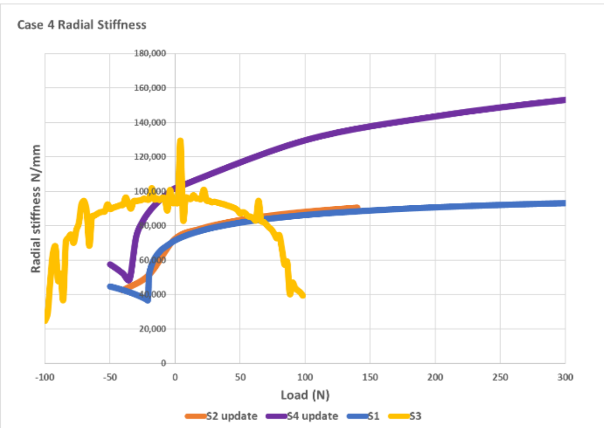


Figure 12. Typical results, radial stiffness

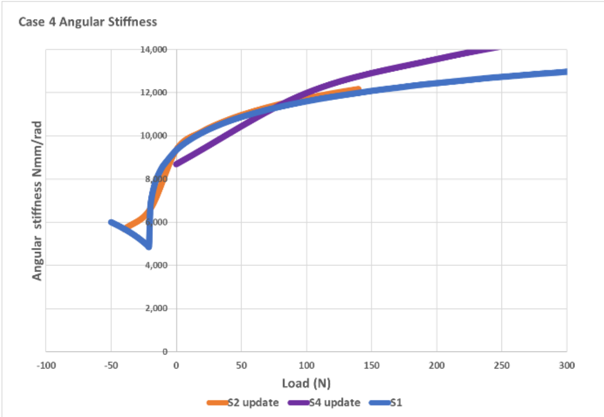


Figure 13. Typical results, angular stiffness

4.3. Observations

Contact stress

- Hard preload*:*
  - The higher the *preload*, the better (lower dispersion) are the code prediction results between participants.
  - The easiest *load case* (axial load only) produced the lowest dispersion of predictions between participants
- Soft preload*:*
  - Compared with hard preload, the differences observed between participants is higher for the soft preload
  - The higher the preload stiffness device is, the greater the dispersion of results between codes.
  - The easiest load case (axial load only) is, produced the lowest dispersion of predictions between participants.

Stiffness

- Single bearing*:*
  - Very high dispersion is observed, between code predictions for all case studies, including also the simple case.
  - The higher the complexity of the load case, the worse are the results (higher dispersion)
- Bearing Pair (System):
  - Not all participants were able to provide predictions, meaning obtaining the predicted stiffness curve of a bearing pair is not straightforward.
  - Axial stiffness is well predicted by all participants, with low differences (both for the shape of the stiffness curve, and the specific axial stiffness values at the set preload point)
  - We observe high dispersion for radial and angular stiffness (not only for the shape of the curves but also the comparison of

- values at the preload point) between participants (30 to 40%)
    - The theoretical onset of gapping for hard preload is not respected, by any participants. Instead, gapping occurs at ~600 N for all of them

Gapping

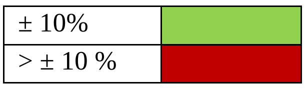
- <u>Number of balls gapping:</u> dispersion is observed between predictions of all participants. Nevertheless, the number of balls subjected to gapping is not a criterion.
- <u>Maximum normal gapping distance</u>: high dispersion is observed on the values of the maximum normal gapping distance (up to 30 microns delta)

Mounting configuration, thermal influence, ellipse truncation

- <u>Effect of X configuration versus O configuration</u>: only 2 participants have provided results here, with high dispersion between results available for radial and angular stiffness. Only 1 participant presents results consistent with RD01
- <u>Effect of temperature & its gradient on the results of Case 1</u>: the implementation of thermal gradient within the bearing leads to significant increase of differences between the predictions amongst participants.
- <u>Axial force needed to have the first ellipse truncation</u> for case 1 is well managed by all participants (<1% delta in the evaluation)

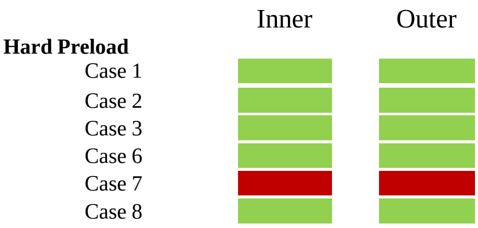
4.4. Summary of results

After iterations with participants to understand as best as possible the differences between results obtained, it appears that realizing a ball bearing structural analysis is not easy: this document shows that, with the same input, significant differences are observed.
Color code defined in this summary is the following:



The cell is green only if all participants having provided results are presenting results with the differences between min and max value in the range 0-20%, so ±10% wrt average value.

Contact stress:



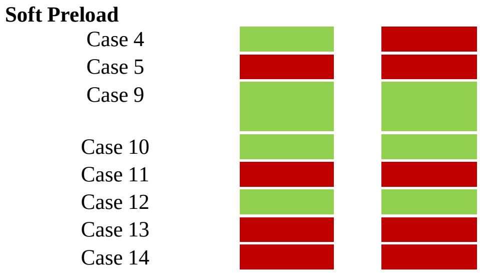


Figure 14. Structural case study, contact stress, results summary

Stiffness, individual bearing:

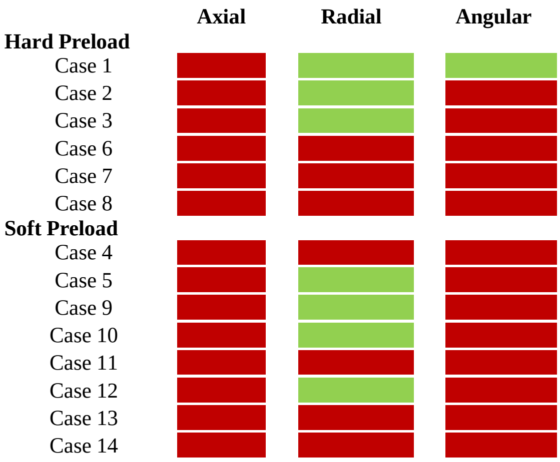


Figure 15. Structural case study, Stiffness, individual bearing, results summary

Stiffness, bearing system:

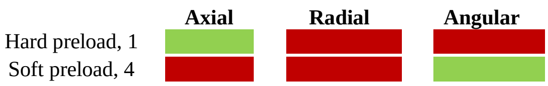


Figure 16. Structural case study, Stiffness, Bearing system, results summary

4.5. Discussion: limitation of the study and evaluation of the origin of dispersions observed

4.5.1. Limitation of the study.

Although we tried to make the case definition as unambiguous as possible, having a single point of origin of case definition is not conducive towards a scientifically robust methodology of accounting for analysis error/uncertainty, as it adds up the bias of case definition to that of the analysis method/tool and that of the operator. This is a limitation of this study.

4.5.2. Evaluation of the origin of dispersions observed

Three potential origins of the observed dispersions are identified.

Model/coding error. The present Case study is a good help for software developer to realise or improve their codes.

Model/coding differences (these could be errors or fundamental differences in the modelling approach between codes).

User misunderstanding in the problem setup.
Based on previous ESA evaluations with similar case studies, but 1 single engineer using different software to run the cases, small dispersions were observed in overall results.
In the present paper, the same case study was analysed by several engineers using different software. As a result, a lot of "human factor" or "user-generated" dispersion has been observed:

- Even if the case study definition includes many details, it was found in default for:
    - Gravity orientation
    - Reference frame definition and loads application
    - Tolerances selected by mechanism designer for the shaft and the housing shall be provided as an input (or defined with the ball bearing supplier) for implementation in the analysis
- Unexpected interpretation coming from the understanding of the engineer doing the analysis (bearing denomination)
- A common mistake also observed is the good application of the load signs, and their implementation in the software (in particular due to software sign convention) Recommendations

4.5.3. Recommendation

User misunderstanding in the problem setup (input and output)
User misunderstanding in the problem setup is a "human factor"/"user-definition" error. Solution for this is a training / dialogue both in setup of the problem in the code and support for results interpretation.
This includes in particular a definition as accurate as possible of the analysed configuration, and a common understanding / definition of the expected output between all parties involved.

Modelling: cross check and parallel analysis
Each of the results are attached to one model (note that models from participants have not been discussed), and any differences, interpretation or simplification in the content of the model can generate significant differences. Some results have been significantly modified after first iteration to understand results discrepancies and check the overall consistency between participants.

It is fair to say that the approach followed in this case study is not typical of a standard project: in general results are not challenged (except if found incongruous) and considered as valid.

A strong recommendation is to challenge the modelling and results obtained, by having at least a double check of model and ideally a parallel analysis (with an alternative code for instance)

Others
Other recommendations are planned to be discussed within the Working Group, such as the definition of a check-list (input, output) to set-up the problem,

Other evaluations are on-going to enrich the structural Case study with existing test results

5. ABBREVIATIONS AND ACRONYMS

ECSS: European Cooperation for Space Standardization

FEM Finite Element Model

6. REFERENCE DOCUMENT [RD]

RD01 Guidelines for space mechanism ball-bearing design assembly and preloading operations / ESA-ESTL-TM-0136

RD02 ECSS: Space Engineering – Mechanisms, ECSS-E-ST-33-01C, Rev. 2 (01/03/2019)

RD03 Space Tribology Handbook by ESTL, 5th Edition (Version 2).

RD04 Rolling Bearing Analysis by T.A. Harris, 4th Edition

RD05 Lewis, S., Gaillard, L., Seiler, R., Parzianello, G., Le Letty, R.: Recent steps towards a common understanding of ball bearing load capacity. Proceedings ESMATS, 2015.

RD06 Hinque, V., Seiler, R. Dynamic behavior of ball bearing under axial vibration.

RD07 Munro, G., Checkley, M., Forshaw, T. , Seiler, R.: Preloaded bearing characteristics under axial vibration.Guay, P.: BM5371 V1 - Roulements – Calculs – Techniques de l'ingénieur.

RD08 Ball bearing stiffness. A new approach offering analytical expressions. Pascal GUAY, Ahmed FRIKHA ESA SP-737, September 2015. Proc. '16th ESMATS 2015', Bilbao, Spain, 23–25 September 2015

CONCLUSION

The benchmarking of the most commonly used ball bearing software realised by the Working Group has highlighted differences in both the tribological and structural predictions obtained from bearing analysis codes when analysing specific typical ball bearing configurations.

As a conclusion, we can highlight the following:

- **Model validation is essential**: all models should be able to predict some real-world results with reasonable accuracy.
- **Try to cross-check ANY model results** to the extent this is possible (with other codes/ or with same code/different users/ or with test)
- **Try to correlate model with experiment/test** (in the case of bearing resistive torque, an early test is useful not only to check the model but also to allow reduction in the motorisation factor applied under ECSS but also mass/power optimisation).
- **Don't assume the same code will give identical results in all cases when used by different users**: user experience in code use and results interpretation ("modelling skill") may be important - as in the case of FEM codes too.

ACKNOWLEDGEMENT

In the frame of this study, ESA gratefully acknowledges the cooperation and support of all the participants to the Software Ball Bearing Working Group :

From agencies

- ESTL : S. Lewis, A. Vortselas
- CNES, J.B. Mondier
- ESA : P. Eggers, L. Gaillard, R. Le Letty, R. Seiler

From Primes :

- AIRBUS Defense and Space : A. Gossant, P. Guay, I. Koeker
- Thales Alenia Space : S. Vezain
- OHB : S. Senese

From space mechanism suppliers

- ROCKWELL COLLINS : M. Ehinger
- SENER : P. Riera, E. Urgoiti, J. Vazquez
- RUAG : A. Lang

From space ball bearing suppliers

- ADR : C. Lebreton, F. Seguin-Henry, E. Videira
- GRW : T. Kreis, W. May, O. Ramin
- CEROBEAR : J. Kreuser, M. Martens