RECENT STEPS TOWARDS A COMMON UNDERSTANDING OF BALL BEARING LOAD CAPACITY

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ABSTRACT

This paper discusses the derivation and substantiation of the load capacity standards currently used to define the de-rating of ball bearings to provide appropriate margins for space applications. It is noted that European and U.S. standards affecting this issue differ, leading to discrepancies in claimed margins and perhaps to mass and performance disadvantages for mechanisms designed strictly in accordance with the present European standard. On the basis of empirical and experimental data, this paper proposes a potential route toward a harmonisation of the load capacity approach for ball bearings both between different international standards and between the separate ECSS standards on mechanisms and structures. The effect of such an approach on bearings and other tribological components intended for space applications is briefly reviewed.

INTRODUCTION

When ball bearings are selected for a space application, they must be de-rated in terms of their load capacity to provide appropriate load or stress margins in-line with the relevant standards. In Europe, the present ECSS standard document on space mechanisms [1], in place since March 2009, requires de-rating of bearings by 25% in stress terms against yield.

This simple requirement presents some quite challenging questions:

1) What load should be used to determine the stress, and how can this be calculated?

2) What is the yield strength of the bearing material?

In the case of ball bearings subjected to launch vibration, the appropriate calculation is the peak Hertzian contact pressure [2], based on a quasi-static assessment of the launch loads to which the device will be exposed. It should be noted that though launch loads are dynamic in nature, the appropriate load capacity figure from bearing catalogue data is not the "Dynamic Load Capacity" (which relates to the probability of fatigue failure under normal bearing operational conditions and is covered by [3]), but instead the Static Load Capacity (defined in [4]).

It has been noted [5] & [6] that there are 3 regimes of steel behaviour when subjected to an applied Hertzian compressive load which are usually defined in terms of the peak Hertzian contact pressure (p_o) or mean Hertzian contact pressure (p_m) in relation to the **local** yield strength of the material in uniaxial tension (σ_y).

The relationships below are based on the so-called von Mises shear strain energy criterion of yield, considered appropriate to ductile materials [6], which states yield will occur when the maximum principal shear stress (τ_{max}) in the contact satisfies the condition

$$\tau_{max} > 3^{-1/2}\,\sigma_y \sim 0.577\,\sigma_y \qquad (1)$$

(see also von Mises equivalent tensile stress approach). For a material with Poisson ratio of 0.3, yield occurs sub-surface and approximately according to the three regimes identified below:

$p_o<1.65\sigma_y$ - there is purely elastic deformation

$1.65<p_o<4.5\sigma_y$ - there is elasto-plastic deformation – a zone with an increasing degree of local plastic deformation which occurs sub-surface and is constrained by the elasticity of the surrounding material

$p_o>4.5\sigma_y$ - there is full plastic deformation.

The most common international standard for bearing static load capacity is [4] according to which a total permanent surface deformation of 0.0001 of the rolling element diameter, d (i.e. for identical ball and raceway materials an indentation of 0.00005d) would not impair the bearing operation. Moreover the standard indicates that this occurs when a bearing ball/raceway contact reaches a peak Hertzian pressure of 4200MPa. The standard does not explain on what theoretical basis this assessment is made, but [7] has cited a number of references which confirm this approximation experimentally and in one case theoretically.

Since in Hertzian contacts [2] the maximum stress

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occurs sub-surface, this surface feature with total composite deformation depth 0.0001d is in fact not a bulk surface indentation, but the consequence of some yield of the material in shear at a small depth below the surface – the elasto-plastic deformation mode described above.

Whilst clear on bearings "manufactured from contemporary, commonly used, high quality, hardened bearing steel in accordance with good manufacturing practice", [4] gives no guidance on treatment of other materials, nor of different, newer steel grades. Indeed, different bearing manufacturers claim sometimes quite different material properties for the steel grades used, presumably due to their unique in-house processing. As bearing manufacturers also have freedom, whilst still adhering to [4], to apply arbitrary in-house de-ratings, or to design bearings which are limited in the first instance by ball over-ride of the land, rather than sub-surface yield, it can be difficult to clearly identify yield stress margins for space applications based on catalogue load capacity data.

One approach to determining the yield stress for the material used is to base the estimation on yield stress (or flow stress, or ultimate tensile stress) vs. Vickers hardness data for the material in question. This is based on an empirical observation that:

$$H_v \approx 3\sigma_y \qquad (2)$$

Where:

H_v is Vickers hardness in MPa
(i.e. converted from HV as measured in $kgfmm^{-2}$)
σ_y is yield strength in MPa

As discussed in [8], the relationship is approximate, the correlation to yield stress being closest for materials that are ideally plastic. Other authors (e.g. Tabor [9]) suggest a better correlation to a uniaxial flow-stress at some specified strain value (for a Vickers hardness indentation, the representative strain is ~8-10% typically [6, 9]) with the implication that Vickers hardness would correlate best the flow stress at this strain, with hardness being 3 times the corresponding flow-stress. Still other authors prefer a correlation between hardness and ultimate tensile strength, considered appropriate for some carbon and alloy steels.

Despite the limitation that it is not clear where the best correlation lies, the very significant advantage of using hardness to relate to yield or ultimate strength is that hardness tests can be easily and non-destructively carried out on a wide range of sample types (including bearing rings and other tribo-components) in order to determine explicitly the yield (or ultimate) stress in as-processed condition and so to estimate or confirm the margin against yield. This approach has been criticised (e.g. [10, 11]) however it remains a very useful approximation and for the commonest bearing steels today fully appropriate. According to an assessment of data from 9 (anonymous) sources including 7 bearing manufacturers the method suggested allowable peak Hertzian pressure of 4270MPa and 3962MPa for SAE 52100 steel and AISI 440C steels respectively based on the mean minimum hardness data as shown in Fig. 1 below.

Given the above, the established practice of allowing 4200MPa for conventional steels and 4000MPa for the (typically softer) corrosion resistant steels such as AISI 440C as indicators of the onset of yield appears to be fully justified and can be experimentally verified.

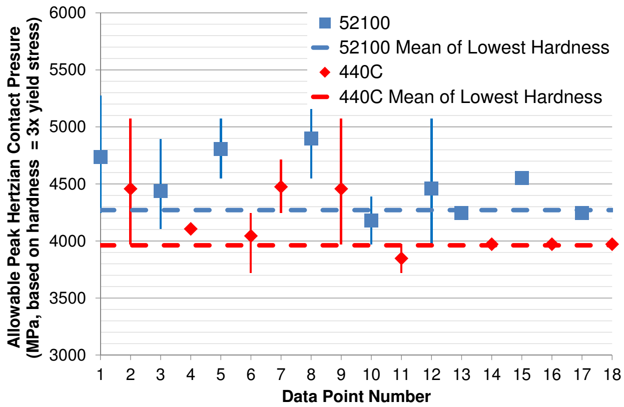


Figure 1, Predicted Allowable Peak Hertzian Pressures for Sub-Surface Yield (based on manufacturer-supplied hardness data and $H_v \approx 3\sigma_y$ vs. data points for SAE 52100 and AISI 440C steels

It might also be noted from the above, that even for AISI 440C, steels from some sources claim high hardness. In one case for example, a mean of ~61.5HRC (~735HV) is available, commensurate with allowable peak Hertzian pressure of almost 4500MPa. Given the relationship between load and peak Hertzian pressure, this allows loads ~40% higher than for some other bearing suppliers for whom the mean hardness may be closer to 58-59HRC (~650-680HV).

It may be noted that for a simple hardness test to predict a yield strength of 4000MPa, or above would require a minimum hardness of approximately 58.5HRC (approximately 664HV) for AISI 440C steel, whereas for a predicted yield stress of ~4200MPa the required hardness would be 59.5HRC (or 686HV) for SAE 52100 steel.

The approach of defining a minimum hardness as suggested above also removes some conservatism in the design process, because it relates directly and experimentally hardness and yield strength.

One misunderstanding experienced is the assumption that when a bearing is subjected to a load which will cause sub-surface yield, then the surface indentations will be easily visible. This is unlikely to be the case with standard laboratory/cleanroom inspection equipment, since the indentation is small, for typical space mechanism bearings with say 5mm diameter balls this amounts to around 0.5μm and its location not well defined. Using a simple static press and a ball on flat geometry, ESTL [12] applied loads to bearing steel surfaces and found that, even with oblique lighting and quite high magnification, indentations (also known as Brinelling marks) were not "easily" visible until their depth was significant compared to the roughness of the substrate. On a rough substrate (R_a~0.5micron), indentations could not be detected below $p_o \sim 1.9\sigma_y$ (~5500MPa), but on a smooth substrate (R_a~0.05-0.1micron) with oblique lighting and when the location of the indentation was known beforehand, indentations were detectable at $p_o \sim 1.5\sigma_y$ (4250MPa for the quite hard steel tested), approximately in-line with the predicted onset of elasto-plastic behaviour for the material under test. Clearly indentation damage was much more easily detectable in solid lubricated bearings than ones previously fluid lubricated due to the change in contrast at the indentation site.

Given the non-linear relationship between peak Hertzian pressure (p_o) and contact load (P), $p_o \propto P^{1/3}$ in ball bearings, the above observation suggests that a fluid lubricated bearing which has clear "Brinelling" marks visible on its raceway may have been very grossly overloaded by a factor of around 2 (i.e. peak Hertzian pressure 25% higher than the onset of yield to achieve a peak Hertzian pressure of ~5000MPa). Furthermore, as an aid to failure diagnosis, measurement of tribo-component hardness might enable the loads actually experienced by an indented raceway to be estimated.

BEARING FAILURE MODES

Assuming no damage during launch, in-flight failure modes mainly relate to lubrication & wear problems (possibly combined with adverse bearing dynamics including cage related issues), fatigue is therefore not usually a consideration. However exposure to launch vibration has the potential to cause other failures:

1) Permanent indentation to raceways
2) Land over-ride leading to damaged balls
3) Failure of hard coatings (e.g. TiC on balls)

In all three cases, increased torque noise (at minimum) will be one result.

When bearings are axially loaded during vibration, the contact angle increases to a value greater than for the preload alone (in-flight condition). However the quasi-static axial load also increases the size of the contact ellipse between ball and raceway. Typically the rate of change of this ellipse size with load greatly exceeds the rate of change of contact angle with load, such that there remains a significant overlap of the loaded zones under launch loads and in-flight conditions, which means that any deformation of the raceway under launch could be detectable in the bearing torque signature as balls roll over the affected zone of the raceway. The likelihood of this depends on the size of the actual zone of physical damage (i.e. plastic indentation or "Brinelling") in relation to the total contact zone itself.

Increasing the conformance (i.e. reducing the conformity number, c – the ratio of across-track raceway curvature radius to ball radius, typically in the range ~1.06-1.14) of a bearing will provide a higher potential load capacity, because the rate of contact pressure increase with load is reduced for close conformity bearings. However the effect makes the bearings more prone to mis-behaviour [2] as a result of misalignment and for a fixed geometry more prone to land over-ride (truncation) as shown in Fig. 2 below in which the tighter conformance bearing (c=1.06 has higher load capacity but is truncation limited, whereas the more 'open' conformance bearing c=1.14 has lower load capacity but is not truncation limited.

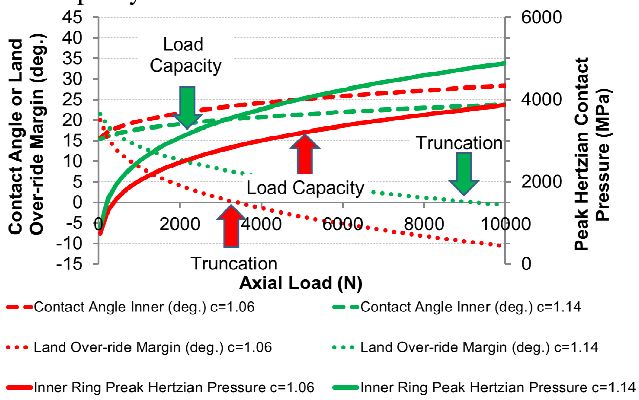


Figure 2, Peak Hertzian Contact Pressure, Contact Angle and Land-Over-ride Margin for a Typical 20mm Assuming Tight and Open Conformances of 1.06 and 1.14.

For bearings having ceramic balls, or having balls with hard coatings (e.g. TiC), the failure modes of ceramics need to be considered. Whilst early fears about the robustness of ceramic balls have been largely overcome, they have very high hardness, and durability, it remains the case that in marginal or un-lubricated conditions, tensile stresses at the edge of the contact can lead to the coating damage and ejection of hard particles from the ball surface due to un-sustainable tensile stresses at the edge of the contact zone.

Such particles will at minimum increase torque noise, but have in addition a capability to cause a failure. For TiC coated AISI 440C balls, also the likelihood of coating crazing damage to the balls due to hammering under gapping or due to ball over-ride of lands needs to be avoided. Hard coatings are typically thin (3-4 µm) with very fine grained microstructure. It is typically observed that at low contact pressure the ceramic coating deforms elastically and remains adherent to the substrate, while at higher pressures, if deformation of the substrate increases the hard coating may crack ("craze") due to un-sustainable tensile stresses ("ice-on-mud" failure) releasing hard ceramic particles, which detach from the ceramic coating.

When a ductile structure begins to yield there is some potential for it to further sustain a load until ultimate failure, a kind of graceful degradation. However since there is no bearing performance "graceful degradation", all of the three failure modes above can be considered to be a type of "ultimate failure" of the bearing – after which there is no potential for further in-specification function.

SIZING OF BALL BEARINGS AND LOAD DERIVATION

In practice, the above discussion means that for bearings rated according to ISO 76 [4] that the maximum allowable peak Hertzian pressure is 4200 MPa for bearings in commonly used, hardened bearing steel e.g. SAE 52100 (or 4000 MPa for bearings manufactured in stainless steel, e.g. AISI 440C considering some empirical de-rating, which seems justified by the hardness data – albeit based on a small sample). Because of the non-linear relationship between load and contact pressure, ball bearings not limited by ball over-ride or rated with internally applied manufacturers factors, are actually de-rated for space applications according to the actual ECSS-E-ST-33-01C [1] by approximately 51% in load capacity terms relative to their ISO 76 rated load capacity. This means that in a first selection iteration (and prior to any more detailed analysis), it is often sufficient to select a bearing which has a catalogue load capacity 2 or more times the required load. Following this very initial selection, a full bearing analysis is required using an appropriate bearing analysis code in order to confirm actual contact pressures for the design load cases.

In the U.S. there are various standards (e.g. [13, 14,15]) and in some cases programme specific margins, which in effect apply similar, though not identical de-rating strategies to [1] also aimed at assuring adequate margin for the avoidance of damage to bearings. The factors applicable to various US standards (with higher permitted contact pressures for non-critical than for "quiet running" applications) and to ECSS [1] are shown in Fig. 1 by comparison to the ISO76 [4] allowable pressures and load capacity. This shows that for a given applied design load case, the present ECSS factor, which provides a 25% margin against yield stress could be considered to be more conservative than the U.S. standards. This conservatism may mean:

1) For identical envelope and ball sizes, selection of a tighter conformity bearing (providing greater potential risk of bearing misbehaviour).
2) In some cases larger bearing physical size and mass leading to a larger mechanism mass.

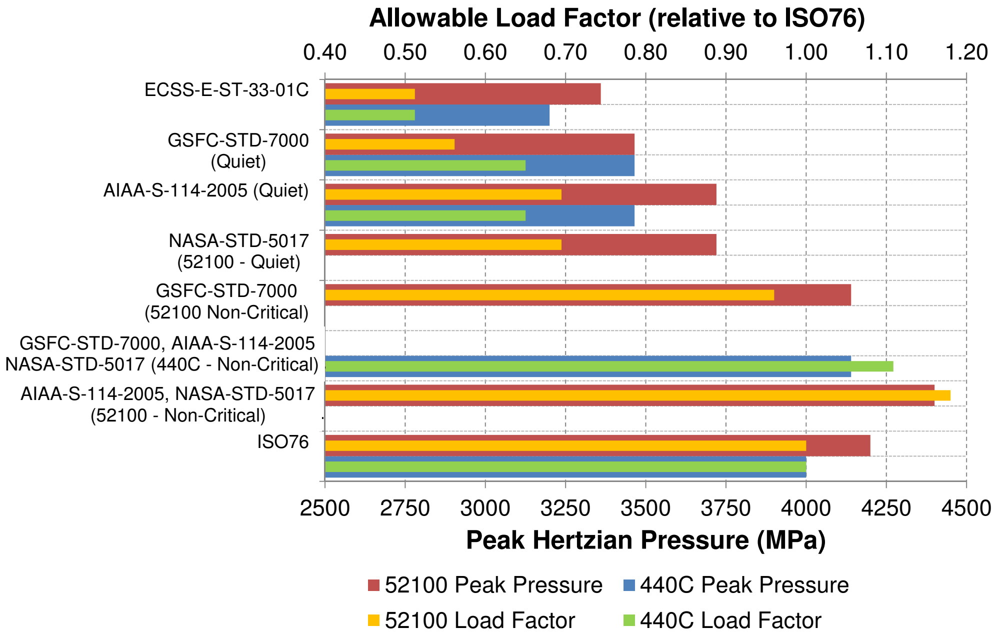


Figure 3. Comparison of Allowable Peak Hertzian Contact Pressure and Load Factor (compared to ISO76) for SAE 52100 and AISI 440C Steels in various Space Engineering Standards

A potential reduction of the ECSS margin below that proposed herein would require a still deeper study and understanding of the dynamic loads acting on ball bearings subjected to launch vibration.

DYNAMIC EFFECTS

Clearly, in predicting the maximum loads experienced by the bearings, it is relevant to have some knowledge of likely dynamic effects. The conventional approach is to apply a quasi-static approach in which the designer is required to take and substantiate in review a view on the expected modal behaviour, the input levels and the damping of the system.

Whilst in recent years, the effects of launch vibration on bearings has been studied by ESTL [16, 17], it is clear that there remains much work to be completed to develop a full understanding.

Considering only the axial vibration (perhaps the most non-linear case), Munro et al [18] have proposed that for hard preloaded systems, due to their relatively linear behaviour, the stiffness of the system is well known and response frequencies easily predictable. Furthermore, the amplitude of responses were typically within 4.5-sigma based on the overall g_{rms} response.

In contrast, for highly gapping bearings or those employing snubbers (typically polymeric elements which limit shaft axial travel under launch vibration), Munro et al found that response frequencies could not be predicted and furthermore that modes could vary with increasing input amplitude. It was however also observed that, given the overall lower random amplification of highly gapping bearings, their level of apparent damping seemed to be relatively high.

So whilst the response of the bearing systems are difficult to predict, it is also clear that, unlike for tests on shakers, in a real-world application there is no clipping of the input spectrum to ensure that it remains within 2 or 3-sigma of the random level. For example, Sarafin et al [19] have proposed that in fact in real world applications there will be multiple occurrences of >3sigma input peaks experienced during mechanism launch. Such occurrences could be potentially damaging and a conservative design approach should include the loads thus generated in some way.

The authors believe that the present practice for designers attempting to derive limiting bearing loads is to assume 3-sigma acceleration peaks above the nominal RMS level at the nominal modal frequency and mass participation and to make an assumption also on the level of damping (which might initially be ~2%, but gradually be refined to a slightly higher value late in development once structural data is available). This quasistatic approach, whilst frequently used lacks the capability to predict the ball-raceway contact pressures when bearings are gapping (balls offloaded due to instantaneous acceleration/response levels) under launch vibration, a capability feasibly realisable (but perhaps not yet fully validated) with dynamic bearing analysis codes.

Given the complexity of the behaviour as discussed above, the adequacy of such an approach, and its ability to generate sizing data for the bearings at the time in the development programme that it is needed (i.e. early in the programme) is questionable. Several authors including Munro [18], Sarafin [19] and Mondier [20] have observed that in the case of gapping, 3-sigma clipping is not guaranteed. It is clear however that this kind of approach is common industrial practice today, and that there is a need for the collection of further data to substantiate any change to this approach to avoid under- or over-conservative design practices being adopted. One improvement to conventional test methods could be the time-domain acquisition of the forces during sample vibration test. This would in due course permit the generation of a database which could enable a review of the derivation of design loads for bearings.

POTENTIAL UPDATE TO ECSS

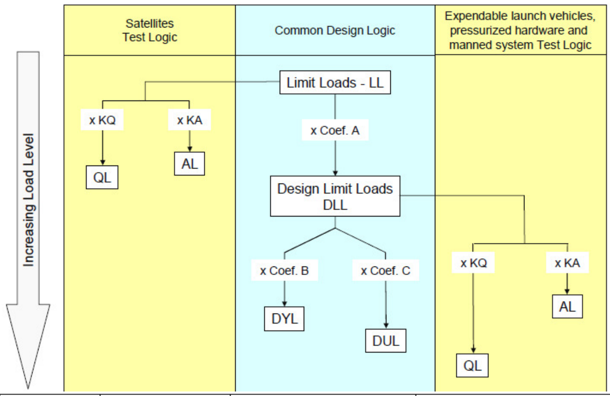
ESA intends to harmonise the load capacity calculation approach used for structures and for all types of mechanical components. This also permits the approach to load derivation to be clarified together with the selection of factors which, when taken together, would permit the approximate harmonisation of the allowable bearing contact pressures between ECSS [1] and various US standards (e.g. [13, 14, 15]).

The basis of this harmonisation, which may exclude bearings in which a hard coating is present that might affect the validity of the Hertzian contact analysis and perhaps those where extremely long or quiet running may be required is briefly summarised below.

ECSS FACTORS

ECSS-E-ST-32-10C [21] provides the structural factors of safety for spaceflight hardware and clarifies also the logic of factor application and the flow of load derivation. The standard also foresees the possible application of additional factors whose values depend on the specific design (see Fig. 4). For bearings, due to the possibility of permanent and in-elastic deformation, it is proposed that the load causing permanent deformation is treated as an “ultimate load” due to the presence of permanent deformation, or damage to hard coatings, which constitute irreversible damage to the bearing surfaces.

The factors of safety suggested for metallic parts are 1.1 for yield loads and 1.25 for ultimate load.



Coefficient	Satellite	Launch vehicles and pressurised hardware	Man-rated systems
Coef A or Design factor	$KQ \times K_P \times K_M$	$K_P \times K_M$	$K_P \times K_M$
Coef B	$FOSY \times K_{LD}$	$FOSY \times K_{MP} \times K_{LD}$	$FOSY \times K_{LD}$
Coef C	$FOSU \times K_{LD}$	$FOSU \times K_{MP} \times K_{LD}$	$FOSU \times K_{LD}$

Figure 4. Logic for Factor Of Safety Application [21]

The usage of one specific factor included in the mentioned ECSS (for example the "local design" factor K_{LD}) to be used for the evaluation of the loads on contacting surfaces in space mechanisms, would allow application of the same load derivation logic for all mechanical components, while maintaining margins against the failure modes for bearings discussed previously. The application of a factor KLD=1.15 on top of the ultimate load factor of safety (1.25) would bring the allowable load to approximately 69% of the load capacity, corresponding to a peak Hertzian pressure of 3720MPa, in line with the NASA standard for SAE 52100 steels. This proposal is summarised in Tab. 1 below in the context of the allowable contact pressures according to the present ECSS [1] standard, the relevant US standards [11, 12, 13] which differentiate quiet running and other implicitly non-critical applications (for which higher pressures are permitted) and ISO76 [4].

Remark: The use of a lower safety factor against yield might be substantiated, among others, on the basis of real micro-hardness measurements on representative bearing races, confirming a minimum required **local** yield strength, i.e. a verified material capacity with respect to peak Hertzian contact pressure.

In other words, there might be less remaining uncertainty with respect to the minimum guaranteed strength of the material than possibly assumed in the past (or elsewhere, compared to so-called material 'A values' for yield strength in structures), which might justify a lower yield safety factor, i.e. a less conservative approach.

Table 1. Comparison of Allowable Peak Hertzian Pressure, Load Reduction Ratios and Effective KLD for Factor of Safety 1.25 versus Ultimate Load According to Various Standards and a Proposed Revision to [1]

Peak Hertzian Pressure (MPa) for Material Indicated		Standard/Comment	Load Ratio (vs. ISO76)	Peak Hertzian Pressure Ratio	Inverse Load Ratio	KLD
AISI 440C	**SAE 52100**					
4000	4200	ISO76	1.00	-	-	-
-	4440	AIAA-S-114-2005 & NASA-STD-5017 (52100 - Non-Critical)	1.18	0.95	0.85	0.68
4140	-	GSFC-STD-7000, AIAA-S-114-2005 & NASA-STD-5017 (440C - Non-Critical)	1.11	0.97	0.90	0.72
-	4140	GSFC-STD-7000 (52100 - Non-Critical)	0.96	1.01	1.04	0.84
3543	**3720**	**Proposed Revised Criteria**	**0.69**	**1.13**	**1.44**	**1.15**
	3720	AIAA-S-114-2005 & NASA STD-5017 (Quiet)	0.69	1.13	1.44	1.15
3465		GSFC-STD-7000, AIAA-S-114-2005 & NASA STD-5017 (Quiet)	0.65	1.15	1.54	1.23
	3465	GSFC-STD-7000 (Quiet)	0.56	1.21	1.78	1.42
3200	**3360**	**ECSS-E-ST-33-01C**	**0.51**	**1.25**	**1.95**	**1.56**

The possibility to harmonise the values used by European and USA standards for space, therefore to increase the allowable Hertzian peak pressure for bearings in an update of the ECSS-E-ST-33-01C [1] could be beneficial, and should be accompanied by additional measures to ensure appropriate care is taken to avoid failures.

These measures comprise on the one hand further experimental investigation on the effects of the vibration loads on bearings, including also real force measurements and distribution, on the other hand the usage of appropriate analysis methods, including also dynamic analysis, to be implemented during the design phase, to carefully assess effects such as gapping and potential out -of -3-sigma events.

CONSEQUENCES FOR BEARINGS

The potential consequences of the above proposed changes for ball bearings and other components with circular or elliptical contacts could be:

1. For ball bearings not limited by land over-ride (e.g. green curves in Fig. 5), the proposal would result in an increase of 35% in permissible load compared to the presently allowed value (and in fact a similar proportionate change would be apparent for any elliptical or point contact within a device).

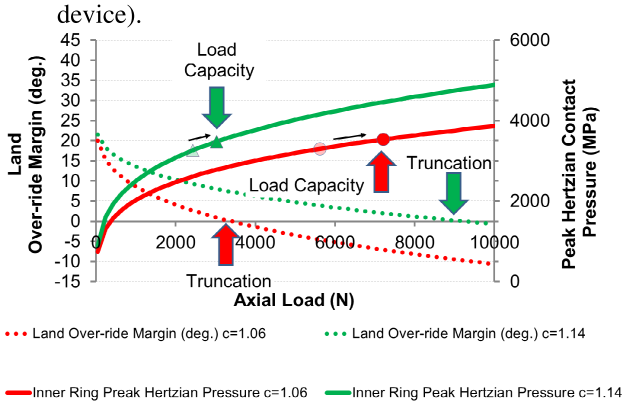


Figure 5, Peak Hertzian Contact Pressure and Land-Over-ride Margin for a Typical 20mm Bearing Assuming Tight and Open Conformances of 1.06 and 1.14 – Effect of Proposed Load Capacity Change.

2. Since allowable contact pressures are slightly higher, this could result in the potential use of more open conformity bearings which are known [2] to be less prone to mis-alignment and cage misbehaviour effects than very tight conformance bearings.

3. Bearings currently limited by Hertzian contact pressure could become limited by land over-ride (truncation) if the allowable contact pressure is increased, hence some care would be needed in the case of recurring products or applications.

4. A firm criterion for the load or geometric limitations with respect to land over-ride (truncation) presently prohibited by U.S. standards, but not explicitly prohibited by ECSS should be determined.

5. For hybrid ceramic bearings, it is considered likely that tight conformity bearings would be retained, though since the load capacity of such bearings is limited by yield of the steel rings, there may be the possibility of some small reduction in the bearing conformance even for such bearings.

6. In some applications the higher contact pressure allowable could be traded for envelope or preload (gapping resistance) – such that a physically smaller bearing potentially with lower preload might be selected in order to reduce the mass of the device (potentially smaller shaft, smaller housing and housing parts and in some cases a lower preload and even a smaller motor may be selectable).

7. The appropriate KLD for to be used for bearings with hard coated steel balls (e.g. TiC-coated) might need to be reviewed to avoid the possibility of initiating hard coating failure under launch vibration. Such a review would need to be supported by bearing level test data which seems not to be systematically available at the present time, necessitating some dedicated testing.

8. Since there is only a loose relationship between launch load capacity and on-orbit operational Hertzian contact pressures, there will be likely be no significant adverse impact on lubricant lifetime from this proposal.

CONSEQUENCES FOR OTHER CONTACT TYPES

For components having (when any edge contact stresses can be ignored) a nominal line-contact, for example spur gears or journal bearings, the relationship between peak Hertzian pressure (p_o) and contact load (P), $p_o \propto P^{1/2}$ and so the philosophy of the above proposal relating to ultimate load would result in an approximate 23% increase in allowable load for a given allowable peak Hertzian contact pressure. Once again the local load capacity of the gear or journal material in its present state may also be demonstrated using a simple hardness test to minimise any uncertainty relating to allowable loading.

It might be speculated that designers might opt to reduce gear mass (and for a plain (un-spoked) gear the mass would be approximately proportional to gear face width hence a 23% increase in allowable load might realise a similar proportionate reduction in gear mass.

CONCLUSIONS

A discussion on the bearing sizing methods and criticalities has been presented, including possible bearing failure modes and differences between the approaches of European and American space standards and that of ISO 76 [4].

With reference to a selection of bearing steel property data it has been shown that Hardness can be a good estimator of yield stress, and this could in future be employed as a verification of stress margins for hardware as built (or a diagnostic in development).

Moreover the approach to the derivation of dynamic limit loads for mechanisms and the effects of test execution and monitoring, have also been discussed and may also be considered as an approved methodology within ECSS in future.

A proposal for a harmonised approach relating to ball bearing load capacity assessment to be potentially implemented in ECSS has been outlined. This modification harmonises both the methodologies already used by ECSS for structures and (approximately) the factors presently used by U.S. standards [13, 14, 15] removing some inconsistencies.

It is shown that the application of this proposal both to ball bearings (as well as to other components having nominal spherical or elliptical contacts) and to other components having nominal line contacts provides potentially some significant advantages for load capacity and torque or options to reduce overall device mass which might be welcome within the European mechanism community.

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