PRELOADED BEARING CHARACTERISTICS UNDER AXIAL VIBRATION

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ABSTRACT

This paper describes the findings of a study into the dynamic response characteristics of a range of bearing configurations under vibration. This practical study aims to improve the understanding of different bearing design parameters on the structural behaviour and on the potential for excessive loading, which may affect the life or performance of bearings in space applications.

The experimental data from dynamic testing of a range of configurations forms the basis for this study. The data is used to confirm the validity of any predictions and to aid the correlate models, as well as providing additional qualitative data about the observed behaviour of the different configurations. This is then used to draw some general conclusions regarding the response characteristics of gapping bearings. A number of important observations are discussed in relation to the use of axial snubbers, damping characteristics and damage susceptibility. Furthermore, practical considerations for the design and testing of gapping bearings shall also be discussed.

1. INTRODUCTION

This document describes the findings of a study into the characteristics of a range of bearing configurations under vibration. The aim was to improve the understanding of a number of different bearing design parameters on the structural behaviour and on the potential for excessive loading, which may affect the life or performance of bearings in space applications.

A number of identical bearings were tested under ambient test conditions. While efforts were taken to ensure that the testing and hardware were generally representative of real life applications, the objective of the testing was to understand the relative characteristics of different designs. As such, the conclusions of this document should be used as a guide only and may not be directly comparable to real life applications where different parameters are varied

In performing an empirical, parametric study into the response of different bearing systems, it is hoped to cover the wide range of configurations being employed in spacecraft mechanisms today. The main objectives of the test programme are as follows:

- Assessment of the validity of bearing system load vs. deflection models and their application in predicting dynamic behaviour, including the accuracy of linear approximations.
- Develop an understanding of the damage process, if observed, and the influence of a range of parameters.
- Measure levels of damping with respect to these parameters.
- Investigation into the use of snubbers as a means to prevent damage in highly asymmetrical gapping bearing systems.
- Provide recommendations, if required, for the revision of gapping guidelines, extended to include recommendations on the design approach and how to model such bearing systems at a higher level e.g. instrument level.
- Provide data for the correlation of dynamic models with experimental results.
- Retention and archiving of test data to include time domain data for future use. For example, the assessment of shock behaviour of gapping bearings.
- Recommendations for further work – for example development of new structural analysis techniques to model bearing system non-linearities.

2. BACKGROUND

It is rare for any space mechanism to be developed without dynamic testing being a key activity. However, the understanding of gapping and the influence of different design parameters and features, such as the use of snubbers, is not well understood by industry or well documented in the available literature.

ESTL have previously carried out investigations into gapping bearings and, as a result, have developed an understanding of the behaviour of angular contact bearings under these conditions. Previous publications describe the results of static indentation tests performed on bearings and vibration tests performed to relate any damage to the equivalent static stress levels. It was found that peak Hertzian stresses up to 5000MPa were

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virtually impossible to detect at the raceway, but indentation damage, when observed was consistent with prediction [1].

Gapping was investigated further in order to explore levels of gapping and ball loads which can be generated in angular contact ball bearings under vibration. Verification of the quasi-static approach to gapping and stress estimation, and development of a method to monitor the relative motions of the bearing rings was also performed [2]. Further investigations included an approach whereby direct laser displacement measurement of the rings was performed and the effects of different levels of (predominantly sine) vibration and gapping on bearing in-vacuo torque were examined [3]. A model to predict the onset of raceway damage through vibration and hammering was developed, and this appeared to show good agreement with the test results and the subsequent degradation of torque performance.

Finally, some design guidelines were generated to ensure that where gapping occurs, this does not give rise to damaging stresses within the bearings [4]. In addition, the gapping analysis approach was revised based on the findings of this later work, in which a large number of tests were performed on the test bearings. The performance of lubricants following degradation due to vibration and tilt measurements during vibration was also investigated, showing that, in the case of lead, recovery to the original torque behaviour could be expected and life-time was not thought to be significantly affected. Importantly, it was also concluded that some gapping is tolerable without the expectation of damage to the bearings.

In light of these findings, and because gapping in a bearing is often perceived to be tolerable without risk to the lubricant or counter-faces, bearings are increasingly being flown in this configuration with gapping being limited by design or by snubbers. Correspondingly, ESTL have developed a much better understanding of the axial load deflection characteristics of angular preloaded bearings. It is known that the load deflection characteristics of a bearing system are sensitive to a number of different parameters. ECSS states that gapping is allowable provided that the adequacy of lubricant and potential consequential mechanisms damage or degradation due to bearing components or shaft motion are demonstrated to conform to the specified functional performance and lifetime [5]. However, it is clearly desirable to understand at an early stage the effect of the available design options such that the risks are understood from the start.

In addition to understanding the effect of bearing configuration on bearing system design and margins, it is also important from a wider system design and analysis perspective. Clearly, there is a need to understand when and why any approximations are valid, and to correlate these with available test data.

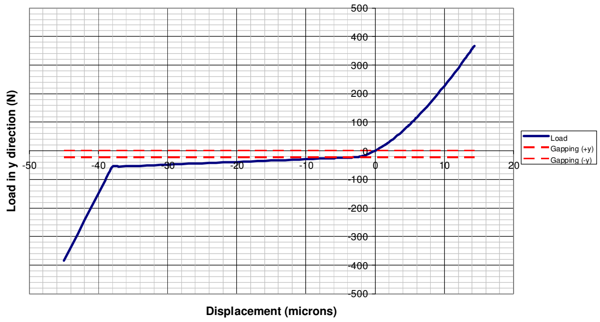

Figure 1 - Bearing load deflection curve (soft preload with snubber)

While damage to the bearings directly may not always be a concern, one would expect a difference in the dynamic response of a gapping bearing system as opposed to a higher preloaded bearing pair. However it is unclear whether these differences are significant and to what extent they contribute, either positively or negatively, to the behaviour of the system. Successful modelling of bearing behaviour requires an understanding of the input parameters and how they vary under different conditions. Approximations are often made with respect to the non-linear bearing characteristics (particularly non-linear and asymmetric in the case of gapping bearings as shown in Figure 1 above) and this study was proposed to show the extent to which these approximations are valid. Sarafin discusses the appropriate statistical extreme load levels [6] and the extent to which these are relevant to assessment of peak loads in the bearings will be discussed.

This work varies from previous studies in that it provides a set of structural predictions and measurements along with the dynamic response data for each configuration. This includes time domain data, which may provide a useful resource for further modelling and analysis, where correlation of the dynamic behaviour of these bearings against models may be performed. This body of data can be used in a number of ways in the future to assess the dynamic characteristics of different bearing configurations from a number of different perspectives.

3. TEST PROGRAMME

3.1. Test Overview

The test programme consisted of parallel testing of a number of bearing housings of different configurations. The following parameters were varied for the different configurations using bearings selected as appropriate to the needs of the test programme:

- Preload magnitude
- Preload stiffness – a range of 'hard' and 'soft' preloads
- Input load – sine and random
- Snubber gap – three gap sizes
- Snubber stiffness – three snubber stiffness options

The baselined tests enable a number of key relationships to be investigated including (axial unless stated). These use a number of dimensionless parameters to allow comparison between different configurations:

- Onset of gapping vs. preload - validation of calculated 'Preload ratio' (Gapping load/Preload).
- Damping ratio vs. gapping – comparison of damping ratio against normalised gapping (input load/gapping load).
- Resonant frequency vs. input load - validation of linearised stiffness from model, load dependent eigenfrequencies.
- Snubber gap vs. resonant frequency - linearised stiffness and snubbing as a means to elevate the natural frequency.
- Damping ratio vs. Snubber stiffness.
- Overall system response of snubbed and non-snubbed bearing systems with various degrees of gapping

The dynamic testing for this TAP was performed at the STFC's Rutherford Appleton Laboratory (Figure 2).

Figure 2 – RAL Vibration Facility

25 bearing housings underwent a series of tests as part of this test campaign. These were tested on a rigid base-plate holding up to 12 bearing housings at a time, with the number of housings reconfigured between test runs. The fully loaded vibration test adaptor, as tested on the first two test runs, is shown in Figure 3. The masses were loosely held with tape to prevent excessive rotation, without influencing the overall response.

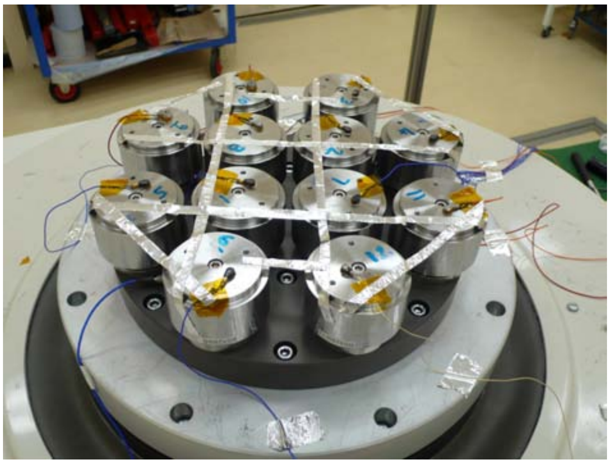

Figure 3 – Bearing Test Setup

To investigate the onset of damage and to correlate damage models, bearing torque measurements were performed before and after vibration testing to try and identify changes in the torque performance. Additionally, a visual inspection was performed on the bearings seeing the highest loads during vibration.

3.2. Test Configuration

The configuration of the test items is shown in Table 1. Testing was performed in the four groups indicated, using easily reconfigured bearings where possible. Replicated tests, shown in red, were included to confirm the consistency between identical units. Within each group the test housings were subjected to a number of sine and random runs at different levels. The final group also saw higher level vibration in order to investigate the onset of raceway damage.

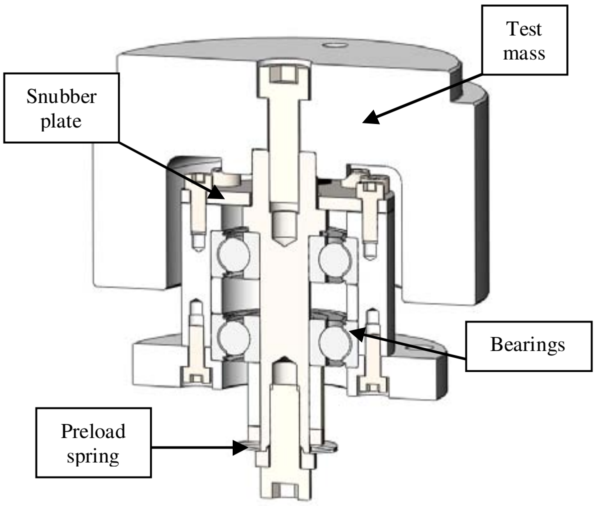

Figure 4 – Test Bearing Unit

Test groups 1 and 2 had a full complement of bearing housings (shown in Figure 4) with and without snubbers respectively (the 's' in the bearing unit ID indicates a snubber was present). Snubbers were simple plates to limit the axial motion of the shaft and these were manufactured with different thicknesses, to give a range of stiffness, and shimmed to set the appropriate gap. Group 3 had fewer bearing housings which were evenly distributed on the test adaptor. Group 4 tested a number of bearing housings to higher levels. Within each group, housings were removed between runs so that they did not experience excessively high levels based on the measured g_{rms} response.

Table 1 – Bearing configurations

Test Group	Test ID	Bearing Unit ID	Preload (N)	Preload Stiffness (N/um)	Snubber Gap (um)	Snubber Stiffness (N/um)	Test Mass (Kg)
1	1.1	#1s	20	0.25	25	25	1.25
	1.2	#2s	20	0.25	50	25	1.25
	1.3	#3s	20	0.25	75	25	1.25
	1.4	#4s	20	0.9	25	25	1.25
	1.5	#5s	20	0.9	50	25	1.25
	1.6	#6s	20	0.9	75	25	1.25
	1.7	#7s	20	0.25	50	5	1.25
	1.8	#8s	20	0.25	50	25	1.25
	1.9	#9s	20	0.25	50	100	1.25
	1.10	#10s	20	0.9	50	5	1.25
	1.11	#11s	20	0.9	50	25	1.25
	1.12	#12s	20	0.9	50	100	1.25
2	2.1	#1	20	0.25			1.25
	2.2	#2	20	0.25			1.25
	2.3	#3	20	0.25			1.25
	2.4	#4	20	0.9			1.25
	2.5	#5	20	0.9			1.25
	2.6	#6	20	0.9			1.25
	2.7	#13	20	5			1.25
	2.8	#14	20	Hard			1.25
	2.9	#15	80	0.25			1.25
	2.10	#16	80	0.9			1.25
	2.11	#17	80	5			1.25
	2.12	#18	80	Hard			1.25
3	3.1	#19	160	0.25			1.25
	3.2	#20	160	0.9			1.25
	3.3	#21	160	5			1.25
	3.4	#22	160	Hard			1.25
	3.5	#23	320	0.9			1.25
	3.6	#24	320	5			1.25
	3.7	#25	320	Hard			1.25
4	4.1	#1	20	0.25			1.25
	4.2	#2	20	0.25			1.25
	4.3	#3	20	0.25			1.25
	4.4	#4	20	0.9			1.25
	4.5	#5	20	0.9			1.25
	4.6	#6	20	0.9			1.25

3.3. Test Setup and Instrumentation

The test item was mounted directly onto the shaker head with all testing being performed in the axial direction. No testing was performed in either lateral axis. Where appropriate, a bearing housing would be removed to avoid excessively high loads once the peak allowable response had been achieved. Because of the variation between housings, this limit would be reached at different times for different housings.

A number of accelerometers were placed on the test item during testing. These consisted of a single axial channel on each test mass, a pair of single axis control accelerometers (for redundancy), and two out of axis channels on the test adaptor to rule out any large out of axis responses.

4. RESULTS SUMMARY

4.1. Sine surveys

Sine surveys were carried out throughout the testing to identify changes in frequency, indicating structural changes such as loss of preload or other settling. In general the frequencies stayed within 10% of the starting value. However, in one case, a significant drop in frequency occurred, indicating a loss of preload so data relating to this housing was disregarded.

In one or two cases there were indications of frequency shifts, however on closer inspection of the data it was found that the relative height of adjacent peaks had changed causing the data-processing macro to wrongly identify the peak value for a given mode.

4.2. High Sine Testing

High sine frequency response data was obtained for a range of input levels over the range 25Hz – 1kHz. This was done to explore the variation in response due to increasing input levels. This was subsequently post-processed to enable the presentation of results for a variety of parameter combinations.

The input levels varied between 0.41g and 13g with peak responses well in excess of 100g being measured. Q-factors varied widely, generally within the range 3 to 30.

As expected, frequency changes resulting from the varying input levels were observed. Both ‘softening’ and ‘stiffening’ were seen due to the onset of gapping and the start of snubber interaction respectively. Due to the nature of the non-linear response, it should be noted that these frequencies may not be resonances in the classical sense, but rather the frequency at which the peak was observed.

The large amount of data generated means that even the results tables are quite extensive and so these are not presented in this paper.

4.3. Random Testing

The overall response PSDs were obtained for all random runs and the g_{rms} response calculated for all of the bearing housings for a range of random input levels.

The peak transient acceleration values (peak g) were also obtained from the time domain data. These are absolute levels and the direction in which these were observed is not considered at this stage.

5. DISCUSSION OF RESULTS

5.1. Load vs. Deflection Models

To assess the extent to which the stiffness predictions of highly non-linear bearing configurations are valid, the stiffness of the bearing housings were calculated with and without snubbers. Table 2 shows the predicted natural frequencies for the mean positive maximum response at ~1.25σ (the mean of the Rayleigh random variable) and compares these with the measured values from the sine sweep. The possible range of frequencies based on linearisation of the load deflection curve is also shown for information.

Table 2 – Frequency Predictions and Results

Test Run	Test ID	Bearing Unit ID	Natural Frequency 1.25δ (Hz)	Frequency Range (Hz)	Sine Survey Resonance (Hz)	Prediction Error (%)
1	1.1	#1	749	324-930	418	79%
	1.2	#2	639	249-835	426	50%
	1.3	#3	567	225-765	343	65%
	1.4	#4	575	388-938	424	36%
	1.5	#5	646	344-843	466	39%
	1.6	#6	573	322-772	477	20%
	1.7	#7	513	248-591	441	16%
	1.8	#8	639	249-835	405	58%
	1.9	#9	677	249-932	320	112%
	1.10	#10	537	341-619	311	73%
	1.11	#11	646	344-843	342	89%
	1.12	#12	680	344-935	342	99%
2	2.1	#1	450	214-651	378	19%
	2.2	#2	450	214-651	380	18%
	2.3	#3	450	214-652	328	37%
	2.4	#4	453	302-655	393	15%
	2.5	#5	453	302-656	430	5%
	2.6	#6	453	302-657	428	6%
	2.7	#13	580	567-671	373	55%
	2.8	#14	931	810-1080	848	10%
	2.9	#15	449	293-794	516	13%
	2.10	#16	452	359-814	603	25%
	2.11	#17	652	637-900	529	23%
	2.12	#18	1106	1106-1181	1216	9%
3	3.1	#19	448	355-937	790	43%
	3.2	#20	451	421-954	580	22%
	3.3	#21	762	677-1033	1041	27%
	3.4	#22	1310	1288-1319	1532	14%
	3.5	#23	742	505-1108	1024	28%
	3.6	#24	1118	764-1182	773	45%
	3.7	#25	1539	1518-1539	1461	5%

The table shows that the stiffness predictions for the snubber cases are typically over-estimated using this simplification. Also, in general, it appears hard to predict the stiffness of bearings with any degree of accuracy. This is not surprising giving the high variability in resonant frequency for different input amplitudes, and the widest of these are the hardest to predict with any real accuracy. The hard preloaded bearings were all predicted to within 14% which represents a reasonable accuracy for predictions made on the basis of basic load deflection measurements of the bearings and hand calculations only.

The first modes obtained from the lower level random tests were generally consistent with the low level sine responses. However, this was not necessarily the case as the input levels were increased. Some configurations exhibited significant changes with increasing input amplitude as can be seen in Figure 5 and Figure 6. In the case of the bearings without the snubber, the frequency would drop significantly due to the reduction in stiffness with the onset of gapping. The housings with snubbers exhibited a similar reduction before the contact with the snubber would increase it again for yet higher input amplitudes.

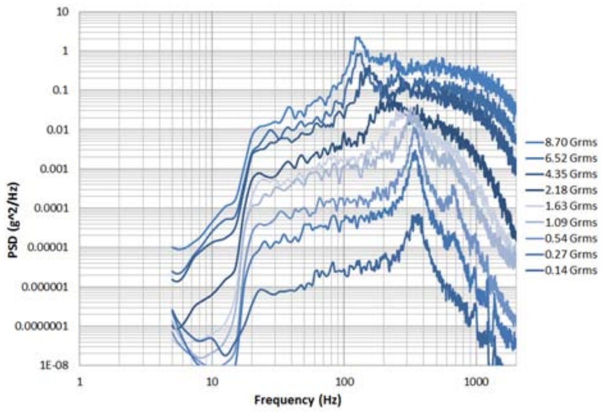

Figure 5 – Bearing #1

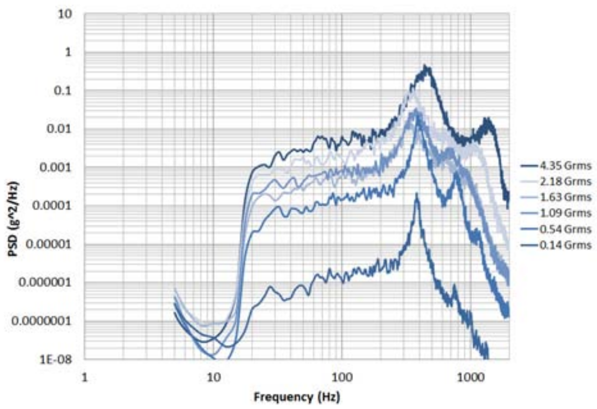

Figure 6 – Bearing #1s

5.2. Prediction of Damaging Response Levels

One of the main questions surrounding the use of gapping bearings is the validity of any assumptions relating to bearing loads. While the linear configurations may be treated in the same way as any other linear elastic system, the non-linearities introduce some additional uncertainty. As such, it is not clear whether damage is more likely for gapping bearings or if snubbers actually reduce the peak loads seen at the bearings. It is noticeable from the examples shown in Figures 4 and 5 that significant additional high frequency energy is observed at higher amplitudes. The shocks introduced during the impacts with the snubbers and raceway are therefore contributing more high frequency content which may be damaging to components sensitive to high frequency excitation such as nearby optics or electronics.

The response data from the random runs was processed and an overall random amplification factor was calculated for each run at each level. This gives the ratio of the total r.m.s. input to output for each. It is noticeable from the processed data that the highest amplifications are observed in the hard preloaded and un-snubbered bearing assemblies. Interestingly, the overall transfer functions are lowest for the most heavily gapping bearing assemblies. While, at first glance, this may look like a good way to minimise the loads going through the system, further investigation suggests that, while the overall responses are lower in the non-linear systems, the extreme peak values may not be correspondingly low.

The peak accelerations were measured directly from the time domain data and presented as a function of standard deviations of the random response. These sigma values can be compared with the predicted maxima using the approach from [6]. These are within the range 4 to 6 for a typical vibration test duration. However, it is clear from the test data that, while the more linear systems conform relatively closely to the peak loads expected from the measured r.m.s. responses, the heavily gapping bearings and snubbered bearings can exceed these levels considerably with peak accelerations in excess of 10σ.

An interesting observation is that the level of effective damping and the level of the peak response appear to effectively cancel across the different configurations. As such, those with apparently higher damping tended to have the highest peak instantaneous responses giving relatively comparable peak loads across all configurations. Table 3 to Table 5, calculated from the measured peak accelerations from the time domain data give the most direct indication of the loads experienced in the bearing housings.

Table 3 – Measured reacted forces group 1

		Peak Force (N)											
	Test ID	1.1	1.2	1.3	1.4	1.5	1.6	1.7	1.8	1.9	1.10	1.11	1.12
	Bearing Unit ID	#1s	#2s	#3s	#4s	#5s	#6s	#7	#8	#9	#10	#11	#12
Random Input (grms)	0.14	21.3	13.4	12.2	14.8	17.4	19.4	13.5	15	14.8	13.4	14.2	13.8
	0.27												
	0.54	111	109	86.5	144	149	112	126	118	80.3	87.4	90	110
	1.09	237	197	206	239	237	213	204	229	184	229	173	251
	1.63	359	469	334	336	352	305	440	456	574	383	404	492
	2.18	517	467	465	484	503	400	572	571	557	528	398	518
	4.35	1318	1421	1145	1612	1323	1006	1793	1722	1479	1474	1323	1421
	6.52												
	8.70												
	13.04												
	13.5												
	14.8												
	17.2												

Table 4 – Measured reacted forces group 2

		Peak Force (N)											
	Test ID	2.1	2.2	2.3	2.4	2.5	2.6	2.7	2.8	2.9	2.10	2.11	2.12
	Bearing Unit ID	#1	#2	#3	#4	#5	#6	#13	#14	#15	#16	#17	#18
Random Input (grms)	0.14	14	13.5	17.1	21.4	16.6	13.2	18.9	34.1	37.7	23.6	23.1	31.9
	0.27	31.5	36	47.3	63	38	33.6	50.6	62.3	78	49.7	40.4	74.2
	0.54	89.7	89.1	105	119	101	84.5	144	128	164	131	53.4	129
	1.09	183	147	248	213	221	193	306	223	260	284	241	258
	1.63	451	384	423	338	359	304	402	332	381	400	319	406
	2.18	641	704	546	589	649	577	572	415	797	681	544	675
	4.35	2183	2313	1803	1690	1911	1385	1580	837	1906	1318	1307	1369
	6.52	4297	3108	3421	2739	2732	2481	2593	1264	3127	2841	2504	1781
	8.70												
	13.04												
	13.5												
	14.8												
	17.2												

Table 5 – Measured reacted forces group 3

		Peak Force (N)						
	Test ID	3.1	3.2	3.3	3.4	3.5	3.6	3.7
	Bearing Unit ID	#19	#20	#21	#22	#23	#24	#25
Random Input (grms)	0.14							
	0.27							
	0.54							
	1.09	158	229	148	237	185	256	281
	1.63	310	363	314	333	331	392	398
	2.18	464	430	468	473	542	559	534
	4.35	1008	1324	874	904	1256	949	905
	6.52	2140	2384	1213	1097	1875	1250	1378
	8.70	3277	3474	1616	1510	2656	1604	1972
	13.04	6411	6612	3223	2032	4417	2948	2376
	13.5							
	14.8							
	17.2							

5.3. Use of Snubbers

The results presented in the previous section indicate that the loads experienced by the housings with snubbers are significantly less than on the same housings with the snubbers removed. The lower peak forces suggest that the increased non-linearity afforded by the snubbers goes some way to reducing the loads at the bearing raceways and would therefore reduce the likelihood of damage. It is also clear that the overall response levels are reduced as result of implementing snubbers and so this may influence the amount of vibration transmitted into the surrounding structure. There is certainly no evidence to suggest that the use of snubbers is damaging in any way.

While different snubbers of varying stiffness were tested, there appears to be no clear trend to indicate any preferable stiffness, however the larger gaps do appear to yield slightly reduced responses.

5.4. Damping

The findings of the previous section point towards higher damping in the gapping bearings. While the more linear elastic, high and hard preloaded, bearings are expected to exhibit lower levels of damping, there is also a clear variation in the effective damping observed in the system with increasing input amplitude. It does appear that damping is low for lesser amplitudes, reaching a peak just after the onset of gapping, and reducing again as amplitudes increase further.

The onset of gapping occurs at a load typically between 1 and 3 times the preload. For compliantly preloaded systems this is close to 1, while it approaches 3 for hard preload systems. In the case of the bearings tested here there is a clear relationship between the response, as a proportion of the load required to produce gapping, and the damping ratio (Figure 7).

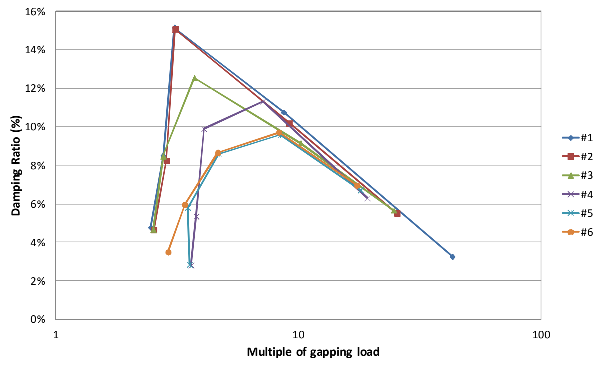

Figure 7 – Response vs. damping ratio bearings #1 - #6

For the other bearing housings the trend is less clear. This may be attributable to errors in the calculation of the exact load at which gapping occurs, which is subject to errors. However, a similar characteristic curve may also be observed in the majority of cases. The highest hard preloaded bearings have been omitted from the plot since their resonant frequencies were too high to be calculated from the sine sweep data (Figure 8).

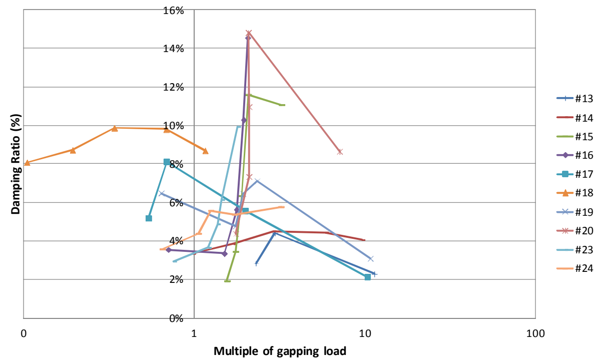

Figure 8 - Response vs. damping ratio bearings #13 - #24

It should be noted that these levels of damping are observed in the axial direction only and, as such, cannot be assumed for the mechanism as a whole. It is likely that cross axis damping will be lower, since these exhibit much more linear behaviour. The test bearings also used rigid test masses and these findings may not be applicable for all supported mass arrangements. In general, however, the lowest level of damping observed was 2% with effective modal damping of up to 15% observed in a number of bearing housings.

5.5. Qualitative Observations

It is worth indicating the characteristic features of the sine plots so that these features may be identified easily in the future. It may also help to indicate faults during vibration, since component loosening or unwanted contact may exhibit similar features.

The plot in Figure 9 shows these features clearly, while the phase plot in Figure 10 gives further information about the nature of the response at these frequencies of interest.

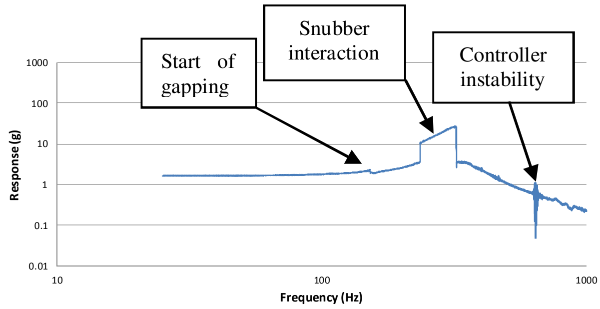

Figure 9 – Bearing housing #10 magnitude plot (high sine)

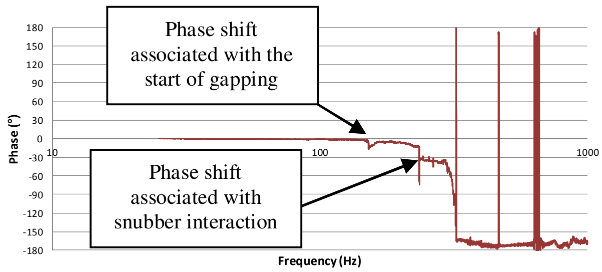

Figure 10 – Bearing housing #10 phase plot (high sine)

The plots above show essentially rigid body motion up until approximately 150Hz, whereupon there is some discontinuity in the response and a small phase shift between the input and the output accelerations. This is the onset of gapping with the relative motion of the shaft relative to the housing on the input side being close to the compression in the bearings at around 6 to 8 microns. The next feature is a jump in the response of the shaft and a significant phase shift to around 30°. Once again this is consistent with the expected snubber gap, with the calculated amplitude of this response being around 40 microns, while the intended gap for this snubber was 50 microns. The characteristic trapezoidal shape increases in width as the input levels are increased and so should be relatively easily identified during testing.

This continues until the point at which the response levels at the given frequency result in an insufficient amplitude to give contact and the response dies away almost instantly. Because the stiffness drops at this point, the input frequency is suddenly beyond the natural frequency of the bearing system and the phase therefore tends towards -180°.

5.6. Bearing Damage Assessment

It was confirmed following testing that the bearings, which saw peak values up to the ISO-76 limits, were not significantly damaged by testing at these levels. This is consistent with previous experience. Only minor witness marks were observed under high magnification, showing light polishing of the raceways from the fretting motion during vibration. No brinelling was observed. Torque tests also did not indicate any appreciable difference between pre and post-test measurements.

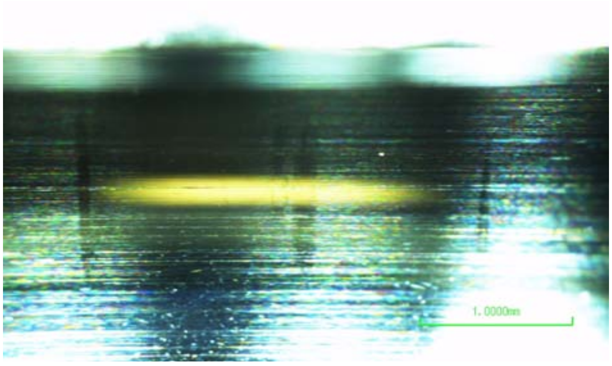

Figure 11 – Witness Marks

6. CONCLUSIONS

This investigation has yielded a number of useful results. The main conclusions are as follows:

1. The bearing response frequency is difficult to predict in the case of highly gapping bearings or those employing snubbers. Only for hard preloaded bearings which exhibit much more linear behaviour can the stiffness be easily estimated with any degree of accuracy.
2. The fundamental modes of vibration for highly non-linear and/or asymmetric systems likely to vary significantly with increasing input amplitude.
3. The overall random amplification tends to be much lower for the highly gapping bearings, suggesting that a much higher level of damping can be expected for gapping bearings.
4. However, the ratio of peak responses from time domain acceleration data to the overall r.m.s. response was much higher for the gapping bearings, though the absolute peak response for all bearings tended to be reasonably consistent for a given input.
5. For the more linear bearing arrangements the responses were consistent with our previous experience and conformed to the predicted peak response of approximately 4.5δ based on the overall g_{rms} response.
6. The shock events associated with collisions at the raceway and snubbers introduces significant high frequency content into the system and the surrounding structure. This may be detrimental to susceptible items such as optics or electronic components.
7. The use of snubbers appears to reduce the peak loads seen at the bearings when compared with the same bearings without snubbers.
8. The change in effective damping with the onset of gapping was confirmed. It was shown to increase significantly once gapping starts before reducing again once anti-phase hammering occurs.
9. Damping within the range 2% to 15% was observed.
10. The characteristic responses relating to common gapping and hammering phenomena have been identified to aid interpretation of future test data.

7. ACKNOWLEDGEMENT

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