EVALUATION OF CRDOA BEARING REPORT FROM PUBLIC SERVICE COMPANY OF COLORADO

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July 1, 1985

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Summary

An independent review of the Public Service Company of Colorado letter report indicates the conclusions reached within the report to be generally valid and that the replacement bearings are statistically and operationally as good as the original CRDOA bearings. These conclusions are based upon an independent review of technical literature on ball bearing failure, dry molybdenum disulfide lubrication, ball bearing materials, bearing clearances and tolerances, life testing, life calculations, contact stresses, duty cycles and loadings. Where enough data was provided, calculations described within the Reference 1 report were independently spot checked and found to be correct. The Public Service Company of Colorado CRDOA Report provides positive support for the NRC Regulatory approval for all proposed changes to the CRDOA hardware described by this report.

Object

The object of this report was to review the Public Service Company of Colorado report (Reference 1) to determine if the Fort Saint Vrain control rod device and orifice assembly replacement ball bearings were equivalent to or at least as good as the original bearings used prior to January 1985.

Introduction

This letter report was prepared under the provisions of the NRC Form 189 for FIN A6701, Task Order R, to provide technical assistance to the NRC Region IV office.

The report (Reference 1) reviewed herein summarily describes actions taken by Public Service Company of Colorado and their subcontractors (GA Technologies, Industrial Tectonics, Inc., and SKF Industries) to justify the use of replacement ball bearings differing only slightly from the original ball bearings for the Fort Saint Vrain control rod drives and orifice assemblies.

The items discussed below follow the format and chronological order of the Reference 1 report. Page and paragraph numbers cited refer to this report.

Commentary Review

All the bearing raceways and some of the balls are 440C stainless steel (pg. 2, paragraph 3). This material is recognized as an excellent ball bearing material (Reference 2), though it is not as commonly used as E52100 type steels. The 440C materials have an upper limit on hardness of Rockwell C60 (200°F temper), while the E52100 steels have a Rockwell C64 hardness upper limit (200°F temper). Reference 2 (pg. 433-438) generally indicates that bearing load capacity and fatigue life are both very

sensitive to hardness. A few points of increase in hardness can easily double the load capacity of a given material and maximum fatigue life is also obtained with the materials of highest hardness. However, in the presence of moisture and oxidizing contaminants, 440C bearing materials may be superior (Reference 2, pg. 324) to the E52100 steels because of corrosion and oxidation resistance. In either case, 440C is an excellent material for this application. The tungsten carbide balls, 17-4 PH and nitralloy 135M cages are also generally recognized (Reference 2) as being good materials for these applications. Though Reference 1 was generally lacking in detail, the sintered bronze lubricant reservoirs are expected to be equivalent to the "Bearite" material used earlier in the original designs.

The choice of dry molybdenum-disufide powder, (pg. 2, paragraph 3) as a lubricant is also excellent for this application, given that the high radiation level and operating temperature make the use of common oil and grease lubricants impractical. The applicable temperature range for this lubricant exceeds 600°F (Ref. 2, pg. 212 and pg. 234, Ref. 9, pg. 26, Ref. 3, pg. 145-146, Ref. 4, pg. 163-164). Molybdenum disulfide appears to be the best choice in dry lubricants for temperatures around 200°F to 250°F; for higher temperatures, PbO and CaF_2 exhibit superior performance. It should be noted that molybdenum disulfide lubricants are not all the same; depending upon the binder material and other additives, the performance of this lubricant can be variable (Reference 2, pg. 212). The Reference 1 report gives no details regarding the specific molybdenum disulfide lubricant used and how it compares with the original lubricant. However, the testing program does at least partially verify that the lubricant used was of high grade and applicable for this service.

The dimensional differences between original and replacement bearings, including reductions in the number of balls, do not appear to contribute to altering the reliability of the control rod drive mechanisms (pg. 3, paragraph 1). Though the dimensional data given in Appendix A of Reference 1 are incomplete, the tolerances and clearances for corresponding parts in the original and replacement bearings are very similar. A comparison of ISO Tolerance Tables (starting on page 228 of Ref. 5) indicates that there are no significant differences in the original and replacement bearings. This conclusion is also supported by data from Reference 6 on bearing tolerances, though, as might be expected, the tolerances for instrument bearings listed in Reference 6 are in general tighter than those used for these dry lubricated ball bearings.

Though Appendix A (Reference 1) does contain limited data on comparative surface finishes for the retaining rings and cages, no such data is listed for the balls and raceways. Reference 2, pg. 436-438 indicates that bearing life (fatigue) is very much a function of surface finish. Better surface finishes in general exhibit significantly longer fatigue life. Ball bearing rolling resistance, and in particular the "breakaway torque" for dry lubricated ball bearings, are very much a function of ball and raceway surface finish. Not all surfaces on a ball bearing are critical; the roughest surfaces have about a 32 finish while ball and raceway finishes are generally much better than this. A

comparison of replacement and original ball bearings with regard to this important factor is not possible without this data. Also, a comparison of the actual critical surface finishes for original and replacement balls and cages is more important than a comparison of the specified values. In this regard, Reference 1 is deficient.

The finish specifications given for the retaining ring (Part Number SLR-D1210-222, Appendix A, pg. 2) are at best very very rough. The 63 and 32 outside diameter surface finishes for the cage (Appendix A, pg. 3) are also relatively rough for critical parts such as these. However, the cage outside diameter surface finish is probably relatively unimportant compared to that of the ball pockets. All parts which touch the ball bearings themselves and the inner and outer raceway critical surfaces should have RMS surface finishes of 4.0 or better. In particular, the cage ball pockets, the sintered bronze lubricant reservoirs, and other surfaces of the cage which rub or roll against the inner or outer races should have RMS surface finishes of 4.0 or better, but certainly no rougher than RMS 10 at the very worst. The retaining ring specified surface finish (63 RMS original, 125 RMS replacement) is probably not critical. The actual surface finish is probably much better than this. However, lacking the lack of surface finish data for other critical parts is an error of omission for a design parameter which strongly influences bearing life and "breakaway torque", the primary failure mechanism of this entire control rod device. A 125 RMS surface finish is typically what results from a rough lathe turning or roughing cut on a milling machine with no surface polishing or sanding.

In order to facilitate comparison of original and replacement bearings, all critical surfaces should have a listing of both specified and actual as-measured surface finishes. While it may be that the replacement bearings have critical surface finishes equal to or better than the original bearings, Reference 1 provides no evidence of this.

Reference 1 (pg. 3, paragraph 3) indicates the design parameters considered critical in assessing bearing performance to be, applied loads, internal geometry, materials, and lubrication. In this application in particular, contamination should also be considered as a critical parameter. The failure of the control rod drives has been suspected of being associated with or linked to the presence of water and water vapor in and around these bearings. Reference 2, pg. 215-216 indicates that contaminants are generally undesirable. Some contaminants can significantly increase the wear rate of the bearing though no specific reference was found in the literature search to water or water vapor causing premature failures. In this respect, both original and replacement bearings must be considered equally resistant to contamination.

The 80/20 figure cited (pg. 4, paragraph 2) for assumed duplex bearing loading distribution seems like an excellent choice. The actual load distribution is probably much better than this. However, lacking more specific data, this assumption is probably conservative.

The radial load data (pg. 4, paragraph 2) is impossible to evaluate without more data. However, the loads listed seem reasonable by superficially considering the geometry of the parts, the loads applied to the cable by the control rods, and the reduction ratio of each stage of the gear box. If the diagram (Appendix A, pg. 1) had the critical dimensions, these loads could be verified. The accuracy of the fatigue life calculations is directly related to the accuracy of the bearing loadings. In similar fashion, the validity of the testing program is dependent on accurately modeling the actual bearing loads.

The bearing operational life cycle data (pg. 5, paragraphs 1 and 2) is presented in sufficient detail that the numbers can be verified by calculation. Verification calculations indicate all the numbers calculated within these paragraphs to be both reasonable and correct.

The general worth and applicability of the fatigue life calculations described in Reference 1 (pg. 5, paragraph 3 to pg. 8, paragraph 4) are deserving of comment.

1. The fatigue life of a bearing is related to a subsurface material failure and as such is not the surface failure mechanism observed for these bearings. This statement is supported by the observation that the bearings with the lowest predicted life (pg. 8 tabular L-10 lives) have never failed in service. In fact, the only bearings that have failed (due to increases in surface roughness) have fatigue lives of between three and four orders of magnitude greater than the bearings that are predicted to be the shortest lived. Fatigue life calculations are worth while to facilitate comparison of original and replacement bearings, but fatigue failures apparently have little or nothing to do with the failures observed for the control rod drive bearings. The fatigue life of original and replacement bearings, while not exactly equal, are roughly equivalent (pg. 8, tabular data).

2. The results of the fatigue life calculations are based upon the commonly used 10% failure criterion and as such are statistically highly variable. To quote from Reference 2, pg. 167:

   "If a number of similar bearings are tested to fatigue at a specific load, there is a wide dispersion of life among the various bearings. For a group of 30 or more bearings, the ratio of the longest to the shortest life may be of the order of 20 or more. A curve of life as a function of the percent of bearings that failed can be drawn for any group of bearings. For a group of 30 or more bearings, the longest life would be of the order of four or five times the average life. The term life, as used in bearing catalogs, usually means the life that is exceeded by 90 percent of the bearings. This is the so-called B-10 or 10-percent life. The 10-percent life is one-fifth the average of 50-percent life for a normal life-dispersion curve".

Considering that the 37 control rod drives each employ 14 ball bearings, three of which are acknowledged to be critical, the probability of an early fatigue life failure significantly differing from the mean predicted life is very high. If the predicted fatigue lives were not so long (51 years minimum continuous operation), this would justify concern. Even if the fatigue life were reduced by a factor of 100 for the 111 (37 units x 3 bearings/unit) critical shim motor bearings, the resulting shortest fatigue life would exceed 1000 years.

3. The statements made (pg. 8, paragraph 2) regarding the applicability of fatigue life calculations for oil lubricated bearings not being directly applicable to dry film lubricated bearings are true. Reference 2, (pg. 383, paragraph 3) verifies this statement by saying:

   > "All experimental evidence obtained to date indicates that the inverse cubic relation between load and life, which was found to exist for point contact with conventional bearing materials with mineral-oil lubrication, is approximately true for other materials and lubricants and for bench-type fatigue testers used for studying the effects of different variables on rolling-element-bearing fatigue".

4. The Hertzian contact stress criterion of 368 KSI being one-third the Brinnel hardness number was not verified by a review of NASA-SP38 (Reference 2). This is a large reference; perhaps the author missed finding the words to verify this. Also, since no data was given for bearing material hardness, it was not possible to verify that 368 KSI was one-third the Brinnel hardness number. The 368 KSI contact stress cited seemed reasonable when compared to Hertzian stresses described in Reference 2 (pg. 383 to 384). It should be noted that calculated Hertzian contact stresses are not particularly sensitive to applied radial loads. Minor errors in the calculation of the applied radial loads (pg. 4, paragraph 2) will not result in significant errors for calculated Hertzian contact stresses because the contact stress varies as the cube root of the normal force (radial load). The predicted fatigue life, on the other hand, is very sensitive to errors in predicting radial loads. The predicted fatigue life of a ball bearing varies inversely as the cube of the load. Because the lowest predicted fatigue life (pg. 8, tabular data) is on the order of 51 years of continuous operation, only gross errors in the radial load prediction (pg. 4, tabular data) would give reason for alarm. However, as stated earlier, the radial loads were not verifiable from the data provided in Reference 1. If for any reason the actual loads are later found to be higher than the predicted loads, these calculations should be repeated.

The physical testing programs described (pgs. 9 to 11) seem reasonable. However, the conclusions drawn from a small sample size can be

very misleading and very inaccurate. The quote cited earlier from Reference 2, pg. 167 provides ample evidence that the life of seemingly identical ball bearings is statistically highly variable. To test two or three bearings of a given type and then to predict or even hint that 111 (3 per drive unit and 37 drive units) of these bearings will last as long is not prudent. This is probably the biggest single factor that makes the test results inconclusive.

The 30 oz-in. bearing torque criterion (pg. 9, paragraph 2) described seems to be a questionable choice for a failure criterion. The shim motors each employ three bearings; a cumulative (3 bearings) frictional bearing torque exceeding 15 in.-oz constitutes a failure of the control rod drive mechanism. For this reason, a more realistic failure criterion might be based upon the total frictional torque from any three bearings exceeding 15 in.-oz. Any conclusions drawn on the basis of tests with a two bearing 30 in.-oz failure criterion are inconclusive. The point in time when the cumulative bearing torque from two bearings exceeded 2/3 x 15 in.-oz = 10 in.-oz might also be meaningful.

The reasoning behind the choice of 65 lb and 15.3 lb radial loads was not clear (pg. 10, paragraphs 3 and 4). The table on page 4 lists loads of 50.8 lb and 12.2 lb as being closest to the test loads for the shim motor. Though the failures experienced in shim motor bearings are not directly related to fatigue life failures, the fact that fatigue life is known to vary inversely with the cube of the radial load should provide motivation to very carefully select the test loads. Perhaps the differences in these figures is an indication of additional static loading not included in the load table on page 10. The Reference 1 report provides no explanation for these differences.

Some tests did use a helium environment and select other operational duty cycle test parameters which were conservative (more demanding) than the actual duty cycles. However, no tests were done at elevated temperatures duplicating that of the real operational environment of 200° to 250°F. The lubrication of the bearings has been acknowledged as being critical to bearing life. Though molybdenum disulfide lubrication is capable of much higher temperatures (exceeding 600°F for some applications), certainly life tests run at room temperature are not necessarily applicable to operational temperatures of 200° to 250°F. Also, prolonged temperatures of 250°F will reduce the hardness of the 440C bearing races if a temperature of less than 250°F was used to temper these parts when originally heat treated. As stated earlier, ball bearing life has been shown to be directly related to material hardness. A few points reduction in hardness can significantly reduce ball bearing life. In this report, both original and replacement bearings are expected to be equal.

The lack of including both radiation and pressure effects in the test series seems justified because the author has found no evidence to indicate that bearing life is sensitive to these parameters.

Conclusions

This author is in general agreement with all the conclusions from physical testing (pg. 11, paragraphs 2 to 4). In particular, this author believes the replacement bearings are roughly equivalent to the original bearings. Though the test results are inconclusive, and some of the test parameters and criteria are questionable, the author believes the replacement bearings to be equivalent to the original and suitable for use in the control rod drives.

REFERENCES

1. D. W. Warembourg, Public Service Company, Colorado, letter report to E. H. Johnson, USNRC, Region IV, Evaluation of CRDOA Bearings, June 13, 1985.

2. E. E. Bisson and W. J. Anderson, Advanced Bearing Technology, Cleveland, Ohio, NSA SP-38, Lewis Research Center, 1964.

3. P. Freeman, Lubrication and Friction, London, Whitefriares Press Ltd., 1962.

4. F. P. Bowden and D. Tabor, The Friction and Lubrication of Solids, Clarendon, Oxford University Press, 1958.

5. A. Palmgren, Ball and Roller Bearing Engineering, Philadelphia, third edition, S. H. Burbank and Co. Inc., 1959.

6. Anti-Friction Bearing Manufacturers Association, Inc., "AFBMA Standards for Instrument Ball Bearings," Section 12, Revision 2, AFBMA, New York, October 1966.

7. Plant Engineering Training Systems, Unit 4, Bearings Lubrication, Edited by Sayre, Clifford R., Technical Publisning Company, Barrington, Ill., 1970.

8. Plant Engineering Training Systems, Unit 2, Lubrication, Edited by Sayre, Clifford R., Technical Publishing Company, Barrington, Ill., 1970.

9. FMC Corporation, Link-Belt Bearing Division, Bearing Technical Journal, Indianapolis, Indiana, First Edition, 1970.