Experimental Investigation of Torque Noise in Satellite Bearings

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Many spin-stabilized spacecraft use precision ball bearings to support and to align an antenna or instrument platform relative to the rotor. The pointing accuracy of the instrument platform is affected by the dynamic fluctuations in running torque or torque noise of the bearings. Although many previous investigations have been concerned with average (or dc) torque, there is a lack of information on the spectral distribution of the torque noise of lightly loaded, large-diameter bearings at slow speeds. This paper summarizes experiments performed on seven pairs of bearings of two bore sizes. Limited measurements of bearing profilometry were made. A special fixture was used to measure torque noise over a range of preload, rotational speed, and temperature. The bearing torque noise was processed by a computer to obtain torque spectral density and rms. Changes in the torque noise with changes in the following conditions were investigated: a) bearing manufacturing tolerances; b) bearing bore size; c) preload; d) rotational speed; e) temperature, and f) retainer configuration. On the basis of experimental data, a model was developed of torque spectral density for control system preliminary design.

I. Introduction

PRECISION ball bearings provide mechanical support and alignment for the payload of dual-spin satellites. A servo system orients the payload and is susceptible to bearing torque noise. The ability of the servo loop to maintain pointing accuracy of the payload is degraded by variations (or torque noise) about the average running torque of the platform support bearings. Consequently, a method of characterizing bearing torque noise is needed in the design of high-accuracy pointing servos.

Bearing torque noise may be caused by imperfections due to manufacturing processes[1-3] or by elastic deformation.[3,4] At the present time, a priori calculation of the amplitude and frequency of the torque noise of a specific pair of ball bearings is a problem in statistics and geometry that has not received a thorough theoretical analysis. Although the running torque of large, lightly loaded bearings for gimbals and antenna supports has been measured, it was stated that the dynamic effects (torque noise) were so complicated that there are "no systematic or analytical approaches to this problem."[5] Gyro bearing torque variations have also been measured,[6] and predictions of torque variations without frequency content have been made.[7] Another study[8] correlated rms torque with rms race profilometry and concluded that additional work should include power spectral measurements. Previous experiments and torque spectral density estimates were made with small-diameter bearings to investigate the effects of retainer configuration[9] and of installation tilt.[10]

Because of lack of information on the torque noise of larger-diameter (>60mm), lightly loaded bearings at slow speeds, the present experiments were performed‡ to characterize bearing torque noise in terms of axial preload, angular rate, temperature, and bearing geometry (size, configuration, and surface topography) and to indicate areas for future study. In this study, seven pairs of single-row, angular-contact ball bearings of 110- and 150-mm bore sizes were tested as a function of axial preload, rotational speed, and temperature. The time trace of torque noise was then processed by a computer to obtain the torque spectral density and the mean square torque noise. The test setup and data processing procedures are described in Sec.II, and the effects of various design and operating parameters are summarized in Sec. III.

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II. Experimental Program

A. Instrumentation

The fixture, Fig. 1a, permitted application of a constant axial preload to a pair of angular contact ball bearings mounted back to back. Preload was provided by a calibrated, spring-loaded, bearing-mounting fixture, installed on the position servo table. The position servo torque acting in a torque balance servo loop (Fig. 1b) maintained the outer races in a null position, while a velocity servo (Fig. 1c) held the bearing inner race rotation rate constant. A precision resistor was used in series with the torquer coil circuit to obtain a voltage signal proportional to the applied torque. This voltage was amplified and recorded on a magnetic tape. The average running torque during a test was nulled out by a d.c. offset voltage so that only torque noise was recorded. Spurious (background) noise was reduced to at least an order of magnitude lower than the bearing torque noise (Fig. 2) by support of the position servo table on low-spring-rate, torsional flexure mounts located on a seismic block.

Profilometry measurements on the outer and inner races and of a random sample of four balls on each of the four 110-mm bore-diameter bearings were made to evaluate affect of bearing precision. The measurements (Table 1) include: raceway waviness, the plus and minus fluctuations about a mean radius; raceway and ball surface finish; roundness, the minimum radial separation between circles inscribing and circumscribing the profile; eccentricity, the distance between two circles circumscribing the ball track and either the inner or outer race diameter; and ball sphericity, measured in a single plane.

B. Data Processing

The recordings of bearing torque noise, Fig. 2a, were band-pass filtered and then digitized at a sufficiently high rate to avoid aliasing of the data. The autocorrelation function of the discrete data runs was computed with the Goertzel algorithm and weighted by the Hanning lag window.[11] Torque spectral

Table 1 Profilometry measurement results (110-mm bore-diam bearings)

	Bearing serial numbers (SN)			
	1	2	3	4
Outer race				
waviness (μin.)	30	25	23	30
@(cycle/rev)	@(60-70)	(50-60)	(60-70)	(50-100)
roundness (μin.)	53	160	68	3900
eccentricity (μin.)	20	150	15	250
maximum surface finish (μin., arithmetic average, AA)	2.2	2.3	2.3	1.2
Inner race				
waviness (μin.)	5	45	5	40
@(cycle/rev)	@(60-70)	(50-60	(60-70)	(60-70)
roundness (μin.)	11	65	16	50
eccentricity (μin.)	30	15	10	23
surface finish (μin. AA)	1.1	3.4	0.8	15.5 (125μin. scratch)
Ball 1 surface finish (μin., AA)	1.2	1.6	1.3	2.5
sphericity (μin.)	13	17	15	23
Ball 2 surface finish (μin., AA)	2.2	1.2	0.7	3.7
sphericity (μin.)	27	21	14	37
Ball 3 surface finish (μin., AA)	1.45	0.9	0.8	4.0
sphericity (μin.)	18	13	12	40
Ball 4 surface finish (μin., AA)	1.9	0.65	1.5	4.4
sphericity (μin.)	19	12	18	40

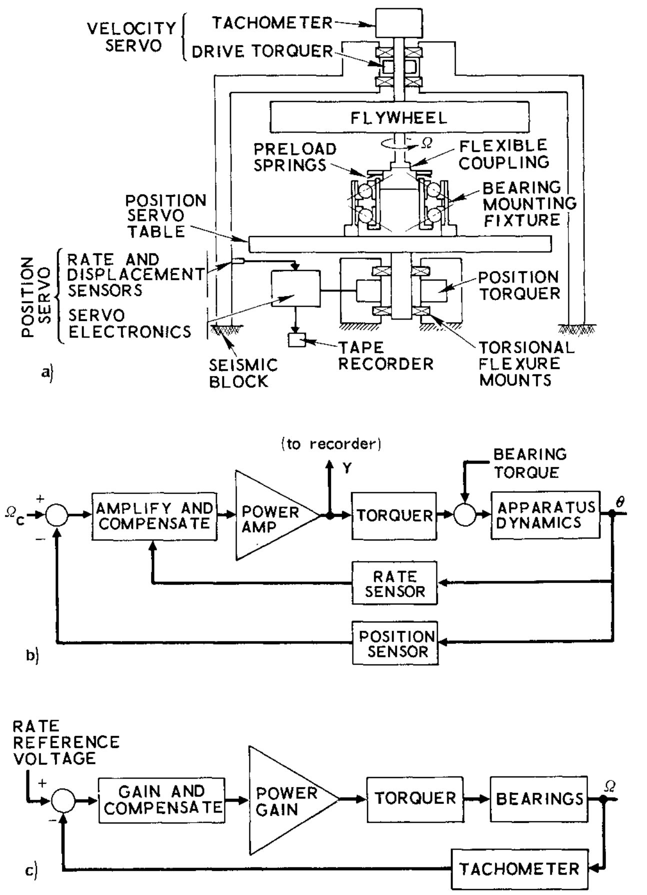

Fig. 1 Apparatus: a) Test apparatus schematic; b) Position servo (15-Hz bandwidth); and c) Velocity servo (10-Hz bandwidth).

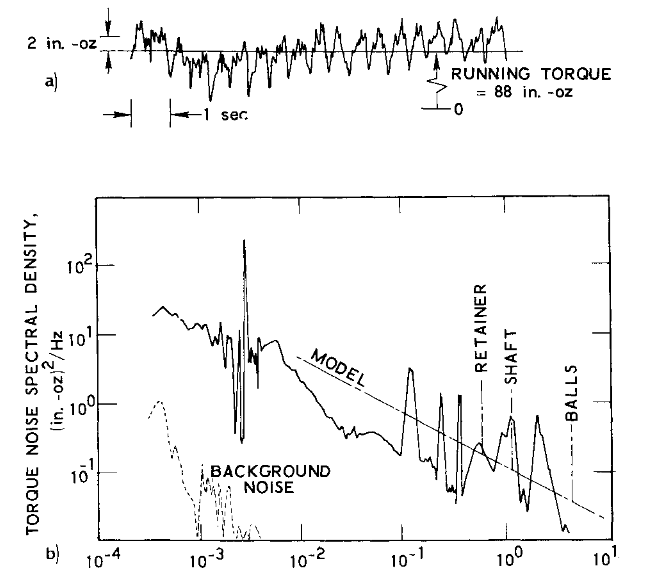

Fig. 2 Bearing torque noise and spectral density: a) Torque noise vs time; and b) Test data, 150-mm-diam ABEC 9.

density estimates were made by computation of the Fourier transform of the autocorrelation function.[12] The torque spectral density was integrated over the frequency range of interest for servo system design to obtain the mean square torque noise. Both the estimated torque spectral density, Fig. 2b, and the mean-square torque noise, Fig. 3, were then plotted as functions of frequency.

A smoothing effect occurs in spectral density at frequencies greater than 0.004 Hz, where the resolution of the data processing filter was coarsened for reasons of computational practicality. Although a coarse value of resolution smoothes the peaks in spectral density, its integral, the mean square torque noise is not affected (Fig. 3), since the noise energy is redistributed.

C. Procedure

Before the bearings were installed in the mounting fixture (Fig. 1), they were carefully cleaned and lubricated with 3 cc of Apiezon-C oil§ by a hypodermic syringe. After fixture

§The effects of amount of oil and possible contaminants were not investigated in this study.

Table 2 Empirical torque noise model[a]

Diam, mm/no. of balls	ABEC class contack angle, deg.	Retainer configuration	Serial no.	Speed, rev/sec	Preload, lb	Temp, °F	Empirical constants a(in.-oz.)^2/Hz	k
110/20	5/15	Race-guided	1 & 2	1.0	200	75	0.096	−0.933
110/20	5/15	Race-guided	3 & 4	1.0	200	75	0.0598	−0.690
150/22	7/12	Race-guided	...	1.1	200	75	0.122	−0.770
150/22	7/12	Race-guided	...	1.1	100	75	0.130	−0.666
150/22	7/12	Race-guided	...	1.1	200	75	0.023	−0.985
150/22	9/12	Ball-guided (reworked)	7 & 8	1.1	200	75	0.130	−0.773
150/22	9/12	Ball-guided	7 & 8	1.1	200	75	0.016	−0.973
150/22	9/12	Ball-guided	3 & 4	1.0	150	75 & 32	0.064	−0.526
150/22	9/12	Race-guided	16 & 17	1.0	150	75 & 32	0.133	−0.734
150/22	9/12	Ball-guided	3&4	1.0	150	75	0.190	−0.643

[a]Empirical torque noise model $W(f) = a f^k$ for frequencies from 0.01 to 10 Hz.

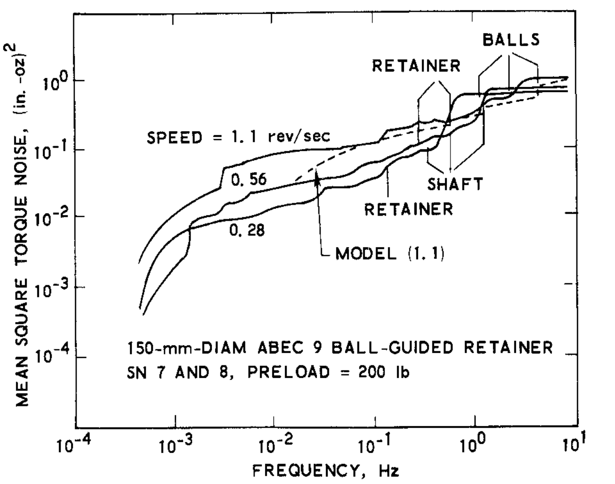

Fig. 3 Effect of speed on mean square torque noise.

assembly, the bearings were run in at the selected speed until stable operation was indicated by an rms instrument measurement of torque noise.

Dynamic torque was measured over the following range of controlled variables that are typical of spacecraft applications: axial preload, 20 to 365 lb; angular velocity, 0.28 to 1.1 rev/sec; and operating temperature, 32° and 75°F. Descriptions of the angular-contact ball bearings tested are presented in Table 2. The 110-mm bore-diam bearings were tolerance class 5, as designated by the Annular Bearing Engineers Committee (ABEC).[13] The 150-mm bore-diam bearings included ABEC class 7 and ABEC class 9. Four configurations of phenolic ball retainers were investigated: a) a ball-guided retainer (BGR); b) an inner-race-guided retainer (IRG); c) an outer-race-guided retainer (ORG); and d) a reworked ball-guided retainer (RBGR) with ball pockets of enlarged diameter.

III. Results

A. General Discussion

A typical time trace of torque noise is shown in Fig. 2a for a 150-mm-diam bearing at a speed of 1 rev/sec with a 200-lb preload. Processing the data results in the torque spectral density shown in Fig. 2b. Note that many peaks are observed in the spectral density. Some of the peaks occur at rotational frequencies of various moving components of the bearings. These frequencies can be calculated,[14] and they are identified for the shaft, the retainer,¶ and the balls. Further investigation of the peaks in torque spectral density, in terms of bearing kinematics and profilometry, are required to improve the prediction and control of bearing torque noise. Integration of the torque spectral density causes the peaks in spectral density to appear as step increases in mean square torque noise. (Fig. 3).

¶Retainer squeal frequency, which occurs at about 10 times the shaft speed, is not shown because it is outside the frequency response range for typical servo systems.

B. Design Parameters

1. Precision (surface irregularities and manufacturing tolerances)

The results of the profilometry measurements are summarized in Table 1. Comparison of the bearing topography of bearing pair SN 1 and 2 with SN 3 and 4 shows that the latter pair has a coarse inner race surface finish (15.5 μin.) and its outer race is out-of-round by 3,900 μin. and eccentric by 250 μin. These imperfections could produce fluctuations in torque due to elastic deformation strain energy[8] and to periodic sliding and skidding[15] between the balls, the retainer, and the races. Similarly, scratches and roughness might permit periodic metal-to-metal contact or variable friction forces.[4]

Unexpectedly, the mean square torque noise of bearing pair SN 3 and 4 was lower than that of bearing pair SN 1 and 2 (Fig. 4), and the former pair was smoother running than the latter. Thus, it was concluded that the limited profilometry

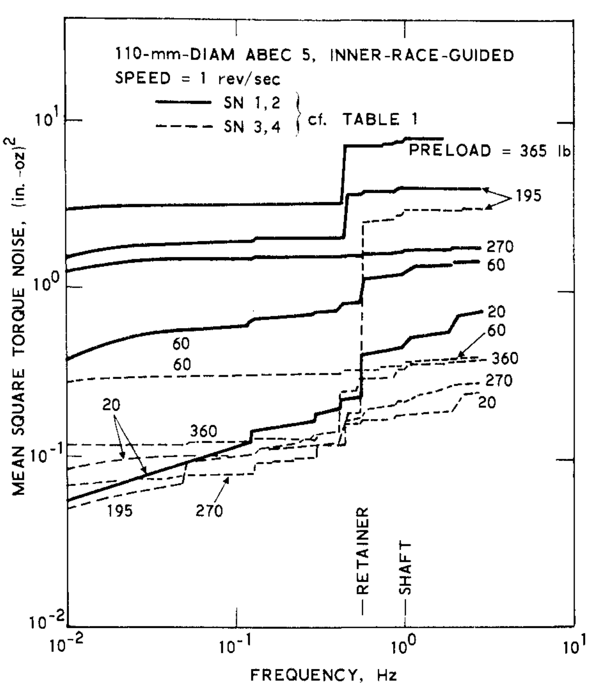

Fig. 4 Effect of surface irregularities on bearing torque noise at various preloads.

measurements were insufficient to correlate with torque noise. Further study is needed to quantitatively relate the effects of surface irregularities and of geometric imperfections to bearing torque noise.

A similar lack of correlation appears in Fig. 5 between pairs of bearings of different manufacturing tolerances (ABEC 7 and 9) and mean square torque noise. It should be noted that the qualitative similarity between ABEC 7 and 9 and the effect of different retainers, inner-race-guided and ball-guided, could cause the inconclusive nature of Fig. 5.

2. Size

A comparison between Fig. 4 ($D = 110$ mm) and Fig. 5 ($D = 150$ mm) shows that, for the conditions tested, size apparently has very little effect on the mean square torque noise. This comparison for three classes of bearings (5,7,9) further substantiates the lack of correlation between bearing precision class and torque noise.

3. Retainer

To investigate the retainer effects on torque noise, two pairs of ABEC 9 bearings were tested under the same conditions but with different retainers. The results, Fig. 6, indicate that over a frequency range of 0.01 to 4 Hz, torque noise is nearly an order of magnitude greater for the outer-race-guided retainers than for the ball-guided retainers. The three tests run on the outer-race-guided retainer configuration bearing, Fig. 6, indicate the variation from run to run of the peroidic components of torque noise that have been observed. Past experience shows that this variation was most prevalent on bearings with outer-race-guided retainers, whereas the ball-guided retainers exhibited the least variation.

Also shown in Fig. 6 is the effect of removing the peaks in spectral density (which appear as step increases in mean square torque noise) that are associated with the race and retainer rotation rate and other periodic disturbances attributed to bearing geometry. With the periodic components removed, the primarily random torque noise remaining can be regarded as a measure of torque noise inherent in each configuration, and the torque noise of the ball-guided retainer is still much lower than that of the outer-race-guided retainer configuration. Because of this, the effect of four different retainer configurations on bearing torque noise was investigated. These tests, with 25-mm-diam bearings, confirmed that retainer changes affected the spectral distribution of the torque noise.[9]

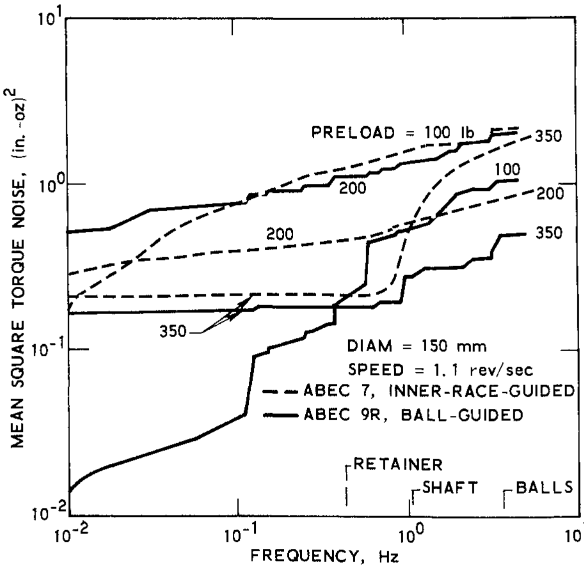

Fig. 5 Effect of manufacturing tolerances on bearing torque noise at various preloads.

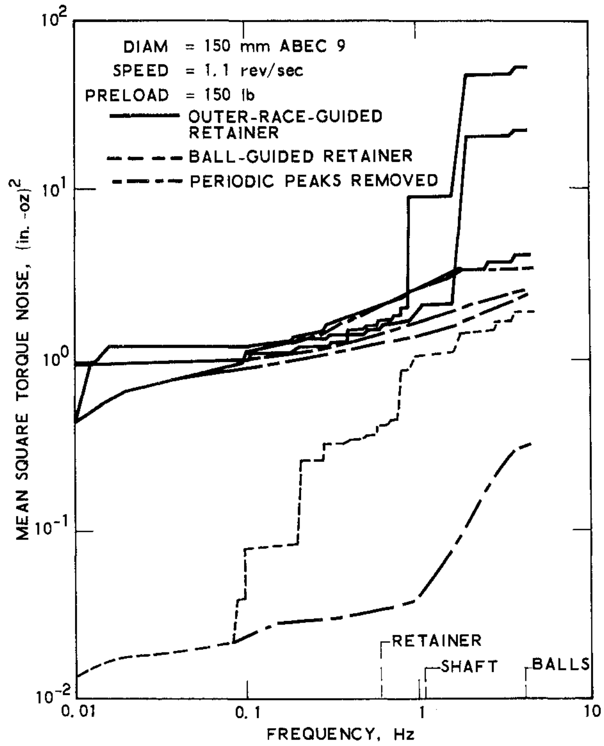

Fig. 6 Effect of retainer configuration.

C. Operating Parameters

1. Preload

Axial preloading of smaller-diam (<60 mm) instrument bearings was reported[16] to have a direct effect on 0.5-Hz sinusoidal torque variations at high rotational speeds (200 rev/sec) and on vibration.[2,17] A curvefit analysis of 60-mm-diam bearing torque noise at a rotational speed of 0.0028 rev/sec indicated that the ratio of rms to running torque was an inverse function of axial preload to the two-thirds power over a 0.02 to 15 Hz bandwidth.[10] To investigate the effects of preload on the torque noise of larger-diam bearings under conditions more typical of satellite application, experiments were performed at slow speed, 1 rev/sec, with preloads ranging between 20 and 365 lb.

Figure 7 shows the effect of preload on rms torque noise for both pairs of 110-mm bearings. Except for an unexplained dip at preloads of 270 and 365 lb, an apparent trend of increasing rms torque noise with increased preload is observed. This trend must be validated by a larger sample of tests.

2. Rotational speed

The effects of speed on mean square torque noise for the class 9 ball-guided retainer bearings with constant preload show a general increase in torque noise level with speed (Fig. 3). Note that the peaks in torque noise associated with the shaft, ball, and retainer of the bearings increase in frequency as the shaft speed increases.

3. Temperature

The effect on torque noise of lowering the ambient temperature to 32°F was evaluated for two sets of ABEC 9 bearings at a constant preload of 150 lb and at a constant speed of 1 rev/sec. The torque noise of bearings SN 3 and SN 4 with ball-guided retainers, Fig. 8, shows an increase in mean square torque noise with decreased temperature, which was attributed to differential expansion between the phenolic retainer and the metal balls and, possibly, to changes in lubricant viscosity. The relatively high rms torque noise (approximately 9 in.-oz.) of bearings with outer-race-guided retainers (not shown) did not change significantly at low tem-

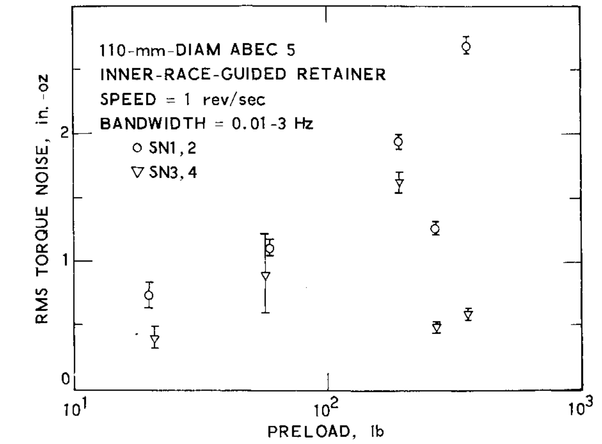

Fig. 7 Effect of preload.

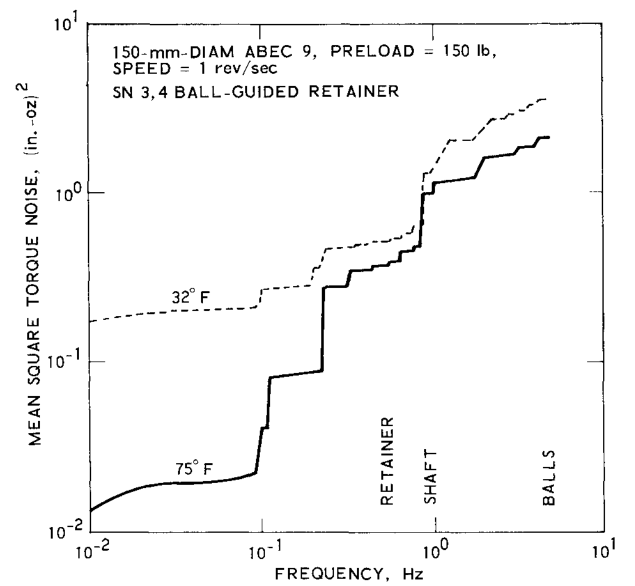

Fig. 8 Effect of temperature on mean square torque noise.

perature. Thus, it was concluded that the retainer configuration design does affect the sensitivity of bearing torque noise to temperature changes.

D. Empirical Model

A measure of the energy in the peaks in spectral density was obtained from the area under the peaks, i.e., the change in the mean square torque noise (the integral of the spectral density) from the start of the peak to the end of the peak (cf. Figs. 2 and 3). By this procedure, the periodic peaks in torque spectral density associated with the retainer, shaft, and ball kinematics were removed, and a curvefit analysis was used to obtain an empirical model for torque spectral density for use in servo control system preliminary design. The best fit from 0.01 to 10 Hz has the functional form: torque spectral density $= a(f)^k$ where f (Hz) is the frequency of interest. The resulting empirical constants (a and k) for various bearings and test conditions are given in Table 2.

To complete the model, the periodic torque noise was included by superimposing a spike at the retainer, shaft, and ball frequencies on the smoothed spectral density model shown in Figs. 2 and 3. The area under each spike was determined by the amplitude change of the experimentally determined mean square torque noise that occurs at the same frequency. It should be emphasized that this empirical model provides only a preliminary estimate of torque spectral density and is limited to the bearings and conditions of these tests. Additional study is needed to verify its application to other bearings.

IV. Summary and Conclusions

Because the torque noise of bearings used in satellite instrument platforms can affect the pointing accuracy of the servo control system, exploratory experiments were performed to characterize bearing torque noise in terms of design variables and operating parameters.

The test results were used to develop an empirical model for estimating torque spectral density for control system preliminary design. It was found that changes in retainer configuration could decrease bearing torque noise. Increases in rotational speed and in preload were generally accompanied by increased rms torque noise. A decrease in temperature also increased rms torque noise and was attributed to differential thermal expansion between components.

Peaks in torque spectral density were observed at the rotational frequencies of the retainer, the shaft, and the balls. Unidentified peaks occurred at lower frequencies. There appeared to be little difference in torque noise between the 110 and the 150-mm-diam bearings, and any correlation between manufacturing tolerance and torque noise was obscured by the effects of different retainer designs. It was found that profilometry measurements that completely describe surface irregularities and geometric imperfections are needed to relate these data to torque noise. The effects of retainer configuration on bearing torque noise and the improved modeling of torque spectral density appear to be promising areas for further investigations.

References

[1] Poritsky, H., "Stochastic Processes of Mechanical Origin," *Random Vibration,* edited by S.H. Crandall, Wiley, New York, 1958, pp. 57–62.

[2] Ruffini, A.J., "Bearing Noise," *Machinery Design,* pp. 232–235, May 1963.

[3] Scanlan, R.H., "Noise in Rolling-Element Bearings," ASME Paper 65-WA/MD-6, Machine Design Div., Winter Annual Meeting, Nov. 1965, Chicago, Ill.

[4] Tabor, D., "The Mechanism of Free Rolling Friction," *Journal of the American Society of Lubrication Engineers,* Nov.-Dec. 1956, pp. 379–386.

[5] Burgmeier, L., "Predicting Bearing Torque in Large-Diameter, Thin-Section Bearings," *Machinery Design,* Jan. 1964, pp. 118–123.

[6] Kingsbury, E.P., "Torque Variations in Instrument Ball Bearings," *American Society of Lubrication Engineers Transactions,* Vol. 8, 1965, pp. 435–441.

[7] Leveille, A.R., "Slow Speed Torque Variation," *Bearing Conference Proceedings,* American Ordnance Association, Hanover, N.H., Sept. 1968.

[8] Mullin, J.V., and Speece, A.L., "Prediction of Bearing Torque Variation Based on Profilometry Data," ASME Paper 69-WA/Lub-4, Lubrication Div., Winter Annual Meeting, Nov. 1969, Los Angeles, Calif.

[9] Greer, H., "The Effect of Retainer Configuration on Bearing Torque Noise," TOR-0073(3901-03)-8, Aug. 1972, The Aerospace Corp., El Segundo, Calif.

[10] Greer, H. and Shibata, H., "The Effect of Installation Tilt on Bearing Torque," TOR-0073(3901-03)-3, Sept. 1972, The Aerospace Corp., El Segundo, Calif.

[11] Dick, D.E. and Wertz, H.J., "Analog and Digital Computation of Fourier Series and Integrals," *IEEE Transactions on Electronic Computers,* Vol. EC-16, Jan. 1967, pp. 8–13.

[12] Bendat, J.S., and Piersol, A.G., *Measurement and Analysis of Random Data,* Wiley, New York, 1966, pp. 66–161, 224.

[13] *Bearing Tolerances,* Anti-Friction Bearing Manufacturers' Association, Standards Section, No. 3, Revision No. 8, March 1965.

[14] Harris, T.A., *Rolling Bearing Analysis,* Wiley, New York, 1966, pp. 203–220.

[15] Poplawski, J.V. and Mauriello, J.A., "Skidding in Lightly Loaded High-Speed Ball Thrust Bearings," ASME Paper 69/Lub-20, Lubrication Div., Lubrication Symposium, March 1969, San Francisco, Calif.

[16] Ballas, T.A., "Periodic Noise in Bearings," SAE Paper 690756, Cleveland, 1969.

[17] Baranov, I.A., *Effect of Axial Loading on the Magnitude of Vibration Generated by Ball Bearings,* Moscow Technological Institute, 1967; available as FTD-HT-23-1011-69, Supt. of Doc., Washington, D.C.