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SPACE SHUTTLE MAIN ENGINE TURBOPUMP BEARING ASSESSMENT PROGRAM

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Science and Engineering Directorate

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This report documents the work done on the bearing assessment program over the past 2 1/2 years. The objective of the program is to develop a nondestructive evaluation system for the SSME HPOTP's which would be used to detect anomalies in installed bearings without engine disassembly. Data bases of various signatures are obtained by slowly turning the pump shafts before and after an engine firing. These signatures are then analyzed and compared to the original signatures to more accurately predict bearing wear.

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TECHNICAL MEMORANDUM

SPACE SHUTTLE MAIN ENGINE TURBOPUMP BEARING ASSESSMENT PROGRAM

I. INTRODUCTION

The space shuttle main engine (SSME) was the first reusable large liquid rocket engine in the world. With a planned design life of 55 starts, it promised to be the most cost effective and optimum engine for space travel. However, since its inception, it has had bearing problems which limit its reusability. Bearings within the engine's four high speed turbopumps must withstand a very harsh environment and operate at peak performance to avoid catastrophic engine failure. As a safety precaution, bearings within one of the SSME turbopumps are replaced every three flights, which is time-consuming, costly, and frequently appears to have been unnecessary.

Of the four turbopumps on an engine, the high pressure oxidizer turbopump (HPOTP) by far has the most unpredictable and troublesome bearing wear problems. It contains two matched pairs of angular contact bearings. Two of these are located near the pump end and the other two near the turbine end. For easy reference, the bearings are numbered from one to four from the pump to the turbine end. Most wear is found on the pump end bearings, particularly bearing number two.

HPOTP bearings deteriorate quickly for many reasons. The balls wear the fastest of all the bearing components and, in the process, lose their preload. Liquid oxygen is their only lubricant and a poor one at that. It oxidizes all exterior surfaces, thus promoting metal to metal contact. Pump speeds in excess of 27,000 r/min and high temperatures in the contact stress areas are also significant contributing factors.

During flight, no internal instrumentation is allowed inside the turbopumps. Only externally mounted strain gauges and accelerometers monitor bearing health. When an engine returns for postflight checkout, various methods are used to determine the condition of the bearings. The pump is disassembled if any bearing wear indicators are found in the flight data. However, by the time external instrumentation indicates a problem, significant wear has already occurred. If no indicators are found, inspection methods are used to better determine the actual degree of bearing wear.

Several diagnostic methods are used on all of the turbopumps. It is standard checkout procedure to perform manual torque tests on them to verify that their shafts are free to rotate. Another inspection method is the microwiggle test whereby the shaft is rotated and radial clearance is measured instead of travel. This test provides some HPOTP pump end bearing information as well as turbine end; however, it is still in the development stage.

Two inspection procedures are available for the HPOTP turbine end bearings. The first is called microshaft travel and involves a push and pull test of the turbopump whereby actual shaft displacement is measured. This diagnostic method reveals reliable data about the actual condition of the two bearings as a pair. Second, the number three bearing can be borescoped for evidence of visible damage to the bearing components (i.e., spalling, pitting, etc.).

Unlike the turbine end bearings, the HPOTP pump end bearings currently have no reliable inspection method or diagnostic tool. Consequently, a conservative 2,000-s limit is placed on these bearings to ensure mission safety. This amounts to a maximum of three flights before pump disassembly. If this limit could be replaced by a reliable inspection method, then the engine could possibly fly for a total of nine flights before it would have to be torn down. At an expense of $1.5 million per tear down, that would amount to a cost savings of $3 million per engine.

Recognizing the critical need for such a diagnostic tool, Marshall Space Flight Center (MSFC) engineers have developed the bearing assessment program. This research and development program has been in existence for the past $2^{1}/_{2}$ years, and, in that time, we have studied several different methods used to determine bearing wear. We have also developed a prototype unit called the automated torque sensor (ATS) and are in the process of qualifying it for service.

In this report we describe the program and methodology used to develop such a nondestructive evaluation (NDE) system. In sections III, IV, and V, the testing programs involving acoustic emission, vibration, and torque are discussed and their results are presented. The prototype diagnostic tool and its use are explained in sections VI and VII. In section VIII, an overall summary is given and our recommendations are presented. The report concludes with a discussion of our future plans for the program.

II. PROGRAM PLAN

The Support Equipment Branch, EP44, of MSFC's Propulsion Laboratory has been working on an innovative program to develop a bearing wear detection system for the SSME HPOTP's. This NDE system would be used to detect anomalies in bearings on assembled SSME's without component disassembly. The plan is to develop a data base of various types of signatures obtained from slowly turning the HPOTP shafts before and after an engine firing. These signatures would in turn be analyzed and compared to the original signatures to more accurately predict bearing wear.

We began by conducting literature searches and researching past work done on similar projects. We investigated the current methods used to determine bearing wear and identified the areas we were most interested in pursuing. Then we laid the ground work by establishing a test facility, acquiring a test article, and forming a working group.

To expedite testing, we obtained a spare HPOTP for our test facility. This would enable us to determine concept feasibility more quickly and concisely. From the beginning, we were always cautious not to develop a technique that would work well in a laboratory setup but fail miserably in real life application. Every concept we consider must be capable of operating in a test stand environment.

The first pump we received was HPOTP unit 2315, which we had briefly before it was replaced with HPOTP unit 0810. Most of our testing has been done with this pump. It is a phase II development pump with several configuration modifications. The pump end bearings have accumulated approximately 6,000 s of run time while the turbine end bearings have 10,000 s. Even though the bearings have much time on them, we still consider the pump to have good bearings. A microwiggle test was performed and no appreciable wear was found on any of the bearings. This only verified the predictions made from the internal isolator strain gauge hot fire data which showed low amplitude cage frequency responses which can be correlated to uneven ball wear in the HPOTP pump end bearings.

The next logical step after preparing a test facility was to assemble a bearing assessment team. We composed this team of individuals from three different areas of expertise. The following disciplines were represented: mechanical, electrical, and data analysis.

Once a foundation was in place, our first task was to develop a system to dry-spin the turbo-pump. For this, we first used as much off-the-shelf hardware as possible and then eventually used the ATS itself to drive the pump. We designed and even fabricated the support and interface hardware.

With the ability to turn the pump came the question as to how fast we could rotate it dry without damaging the bearings or any other component. Rocketdyne, primary contractor for the SSME, has never given us a maximum speed limitation but advised us to turn as slowly as possible. To be conservative, we decided to limit our speed to a maximum of 8 r/min. With a functional test article in place, we were now ready to begin a comprehensive testing program.

III. ACOUSTIC EMISSION TESTING

One of the testing areas we have had little success with is acoustic emission (AE). AE's are stress waves that are produced when materials are stressed. A structure must be sufficiently loaded in order to produce stress waves. This method detects movement, not existing geometric discontinuities. Since MSFC government employees had limited experience with acoustic emission testing, the industry was surveyed.

Two independent contractors performed preliminary AE testing on HPOTP unit 0810. The results were inconclusive in one case and in the other the NDE method was deemed not viable at such low speeds. Turning at 0 to 8 r/min, the system did not generate enough energy to trigger the sensors. Even though the results were not promising, we still plan to continue testing in order to disprove or prove feasibility. With the required authorization, we will increase the speed of the turbopump up to 50 r/min and test again. Perhaps this will generate enough energy to excite the sensors.

IV. VIBRATION TESTING

The second testing area we explored involved conventional vibrational analysis. As bearings begin to wear, they demonstrate marked increases in vibration levels. Piezoelectric accelerometers are traditionally used to convert this vibratory motion into an electrical signal. These transducers are widely used in industry to determine bearing health and predict failure.

In our own vibration testing, we attached surface-mounted transducers to the pump housing, slowly rotated the shaft, and analyzed the output signal for bearing signature characteristics. To begin, we had Physical Acoustics Corporation and Brüel and Kjær Instruments perform preliminary testing on HPOTP unit 0810. Both companies' tests were inconclusive since the background noise was indistinguishable from the pump noise. They did, however, recommend continued testing with seismic accelerometers.

Before we could begin our own test series, we needed to know the SSME bearing fault frequencies we would be looking for. These frequencies are a function of speed and bearing geometry. The following equations were used to calculate them:

Outer Race Defect

$$f(Hz) = \frac{n}{2} f_r \left(1 - \frac{BD}{PD} \cos\beta\right)$$

Inner Race Defect

$$f(Hz) = \frac{n}{2} f_r \left(1 + \frac{BD}{PD} \cos\beta\right)$$

Ball Defect (Ball Spin)

$$f(Hz) = \frac{PD}{2BD} f_r \left[1 - \left(\frac{BD}{PD} \cos\beta\right)^2\right]$$

Cage Imbalance

$$f(Hz) = \frac{f_r}{2} \left(1 - \frac{BD}{PD} \cos\beta\right)$$

where

n = number of balls

BD = ball diameter

PD = pitch diameter

f_r = rotation speed in Hz

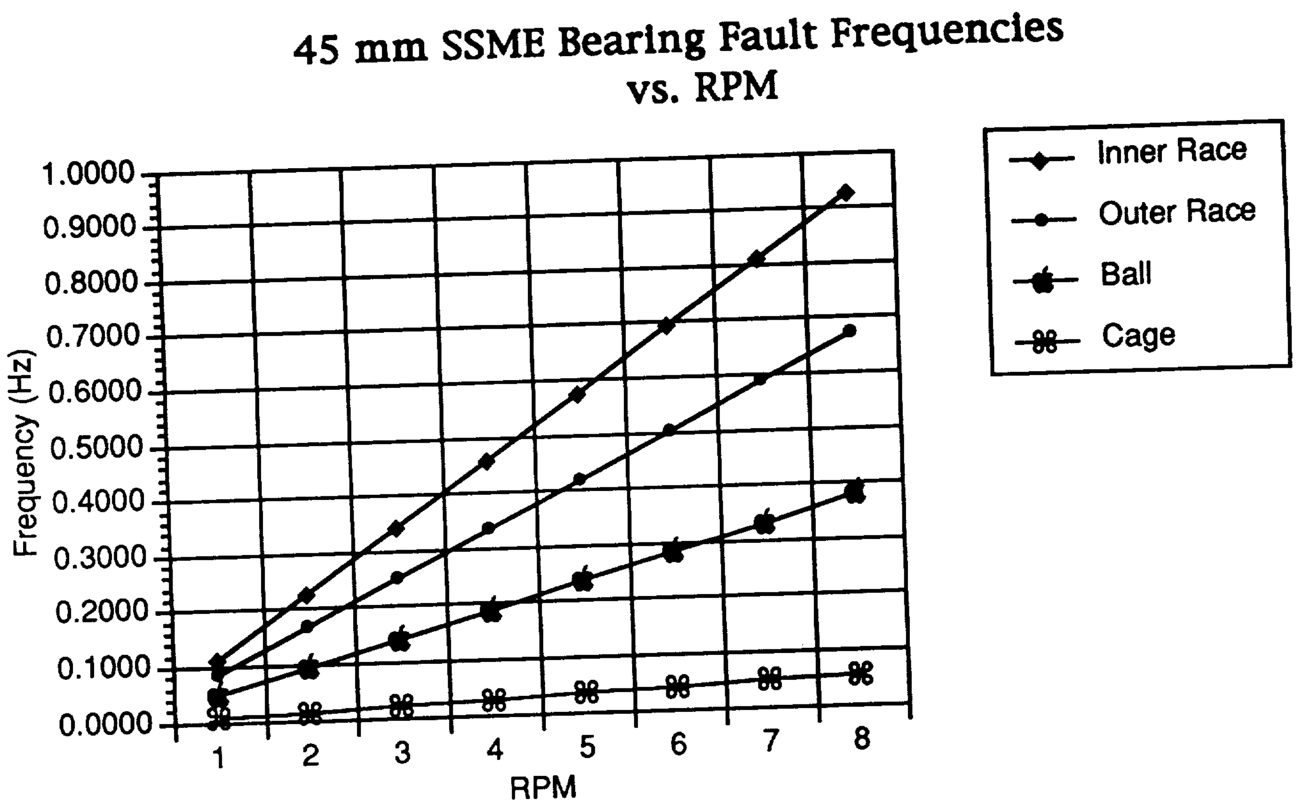
β = contact angle in radians.

The results can be found in table 1 and are displayed graphically in figure 1. We could see that the data analysis would be challenging because at testing speeds of 0 to 8 r/min, fundamental SSME bearing fault frequencies are all less than 1 Hz.

Since those preliminary tests, we have conducted three significant test series. In our first series, we utilized a spare keel latch motor, which was developed for the Hubble space telescope maintenance mission, to rotate the pump as shown in figure 2. We designed the spool and internal spline to mate with the pump. The motor only ran at a constant speed of 8 r/min, which limited our testing, since throttle capability would have allowed us to better distinguish pump rotational driven spectral components from stationary background and electronic noise. The test series consisted of four 15-min runs at this constant speed. We used three different types of accelerometers: four miniature compression driven piezoelectric sensors with internal electronics, three seismic spring-mass type units sensitive in frequency to dc, and four shear-type piezoelectric sensors driven by external charge amplifiers. These off-the-shelf transducers were mounted in three different tiers: the preburner pump flange, the HPOTP housing weld 3 region, and the G3 flange.

Table 1. Values for SSME bearing fault frequencies.

VARIABLES		45mm Bore	57mm Bore		
		Pump End	Turbine End		
n	number of balls	12	13		
BD	ball diameter (in)	0.4375	0.5000		
PD	pitch diameter (in)	2.5600	3.1700		
beta	contact angle (radians)	0.3840	0.3840		
Parametric Study					
for 45 mm					
RPM	*Rotational Speed in Hz*	*Inner Race*	*Outer Race*	*Ball*	*Cage*
1	0.0167	0.1158	0.0842	0.0475	0.0070
2	0.0333	0.2317	0.1683	0.0951	0.0140
3	0.0500	0.3475	0.2525	0.1426	0.0210
4	0.0667	0.4634	0.3366	0.1902	0.0281
5	0.0833	0.5792	0.4208	0.2377	0.0351
6	0.1000	0.6951	0.5049	0.2852	0.0421
7	0.1167	0.8109	0.5891	0.3328	0.0491
8	0.1333	0.9268	0.6732	0.3803	0.0561
Parametric Study					
for 57 mm					
RPM	*Rotational Speed in Hz*	*Inner Race*	*Outer Race*	*Ball*	*Cage*
1	0.0167	0.1242	0.0925	0.0517	0.0071
2	0.0333	0.2484	0.1850	0.1034	0.0142
3	0.0500	0.3725	0.2775	0.1551	0.0213
4	0.0667	0.4967	0.3700	0.2068	0.0285
5	0.0833	0.6209	0.4625	0.2585	0.0356
6	0.1000	0.7451	0.5549	0.3102	0.0427
7	0.1167	0.8692	0.6474	0.3619	0.0498
8	0.1333	0.9934	0.7399	0.4136	0.0569



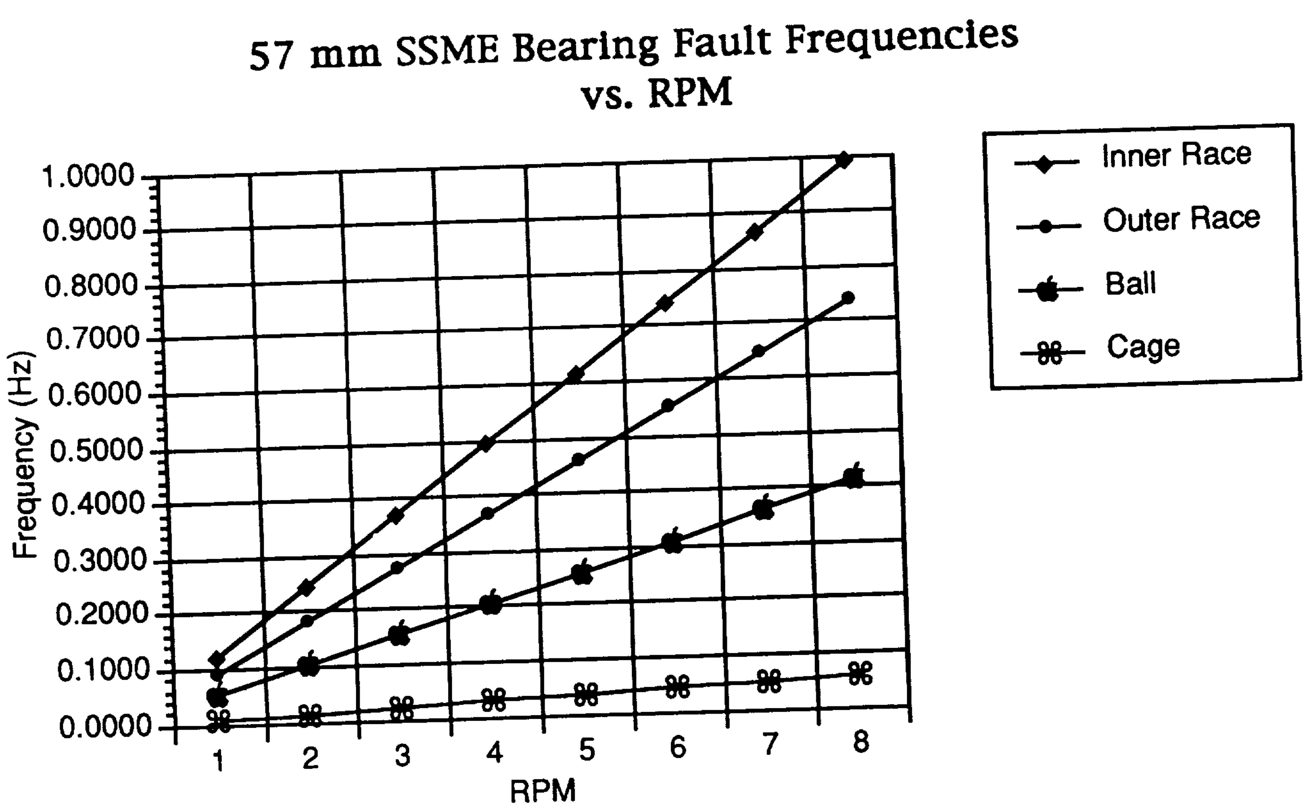


Figure 1. Plots of SSME bearing fault frequencies.

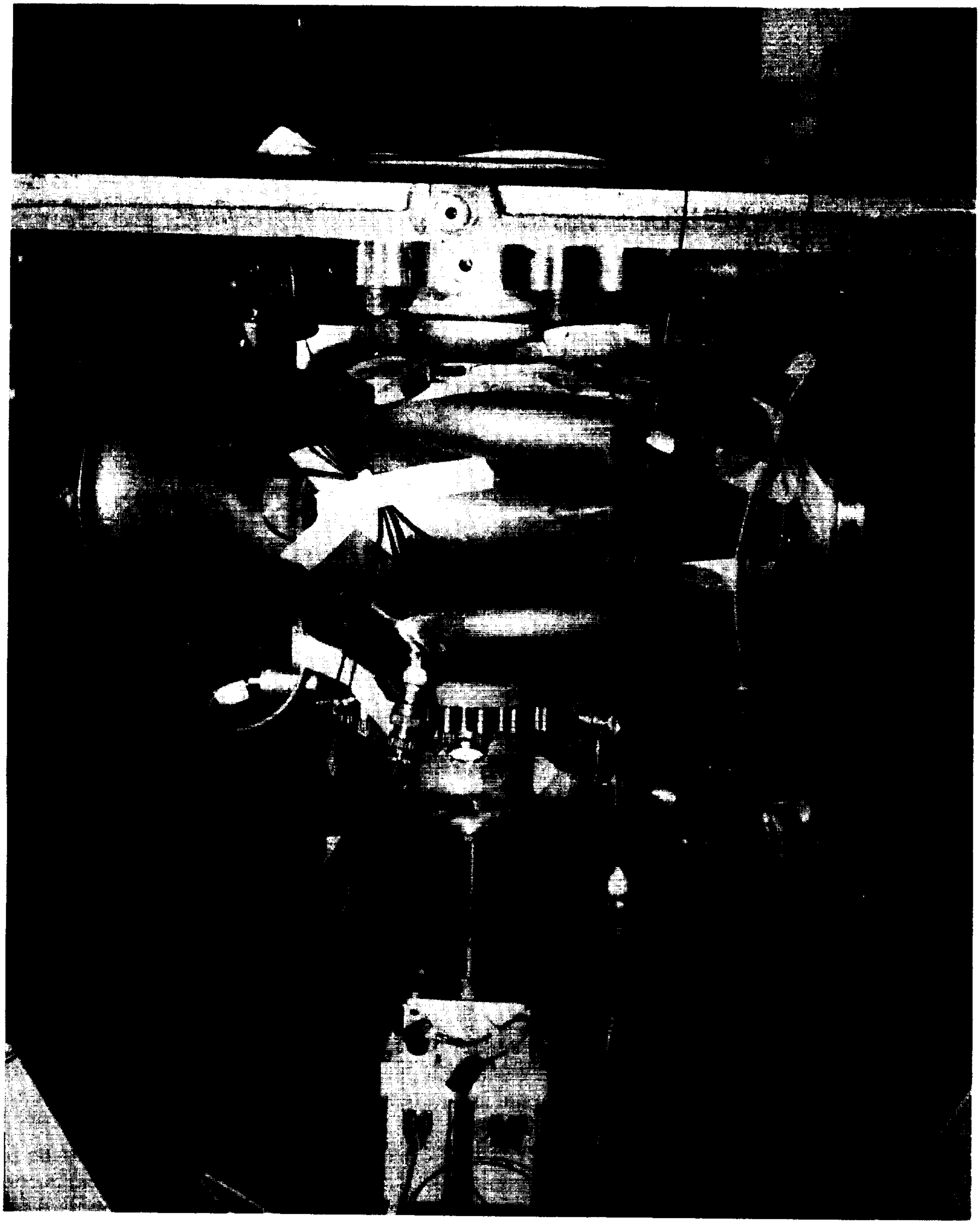


Figure 2. Keel latch motor setup.

Our test objectives were to optimize the selection of acquisition equipment and the mounting locations of the transducers. To accomplish this, we classified the dynamic noise floor data prior to the pump rotation and then determined the external drive's influence on the data. The dynamic signals from the dry-spin testing were evaluated and the external environment influences on the dynamic data were identified.

It was obvious from our test results that there was not enough energy in the system. The miniature accelerometers had not functioned properly. We found that we needed to work on understanding the frequencies from external sources. As for identifying bearing defect frequencies, the overall results were inconclusive.

We did learn that we needed the ability to vary the motor speed. This would make it much easier to distinguish bearing related spectral components from environmental and instrumentation noise. A slow throttling capability would also prove beneficial. Prior to the actual test, we should have performed decoupled testing in order to isolate the feed through from the drive mechanism to the HPOTP data. Also, we needed to record a shaft rotational speed data channel, i.e., key phaser information, to better enable us to locate bearing related frequencies.

V. TORQUE TESTING

In addition to the standard NDE methods of AE and vibration, we also investigated a new approach—dynamic torque analysis. Little if any work has ever been done investigating SSME torque signatures for bearing signature content. Since torque checks must be performed on all pumps, it seemed logical to tap into this available resource.

Currently torque tests are routinely performed on turbopumps during checkout to determine breakaway and running torque. The test consists of a technician with a Snap-on torque wrench hand turning the pump while a quality inspector reads the dial indicator upside down. Needless to say the test is rather crude, and a more sophisticated test could possibly produce useful data. Since the test must be performed any way, it seemed reasonable to have it automated so that the data could be stored and analyzed for later use. Because the unit is hard mounted to the pump, the tests are more accurate and consistent. No human error is involved, and no side loads are induced into the pump. The automated test is definitely more precise in measuring static torque, since the accuracy of a hand held torque wrench is only plus or minus 2 percent within the upper 80 percent of the scale. Moreover, the dynamic content of the available torque signature could prove useful in determining bearing health.

VI. AUTOMATED TORQUE SENSOR

The ATS was envisioned from the very beginning of the bearing assessment program. The prototype was planned as an engineering unit for development purposes. Its requirements gradually evolved over time. The finished product is fully functional and reliable.

The ATS is battery powered, computer controlled, and easy to use. It was designed to be portable and durable for operation on test stands. The complete ATS system is shown in figure 3 and is composed mainly of two parts, a torque test head and a data acquisition system. The entire unit fits in two cases (fig. 4) which have a combined weight of less than 40 lb.

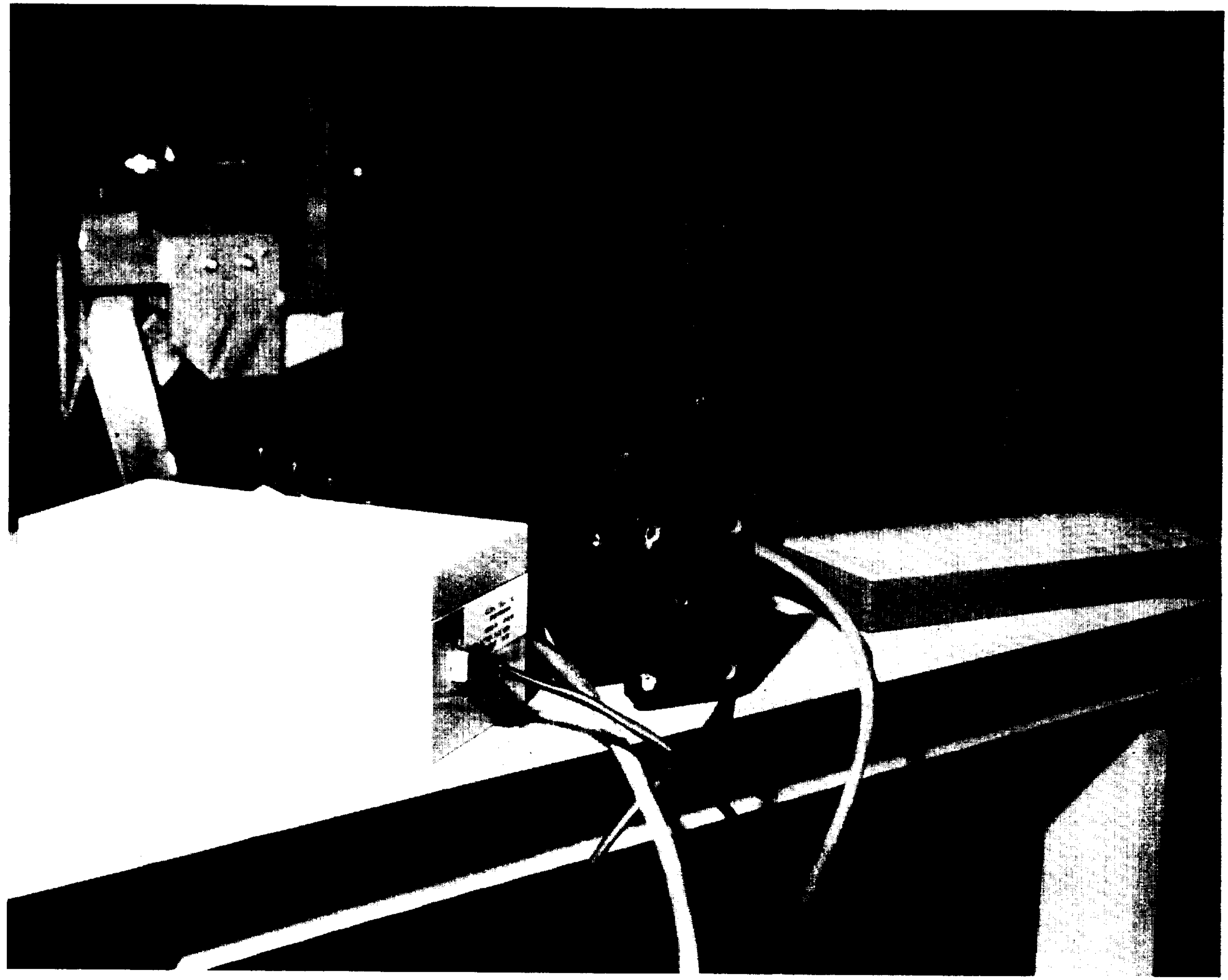


Figure 3. ATS.

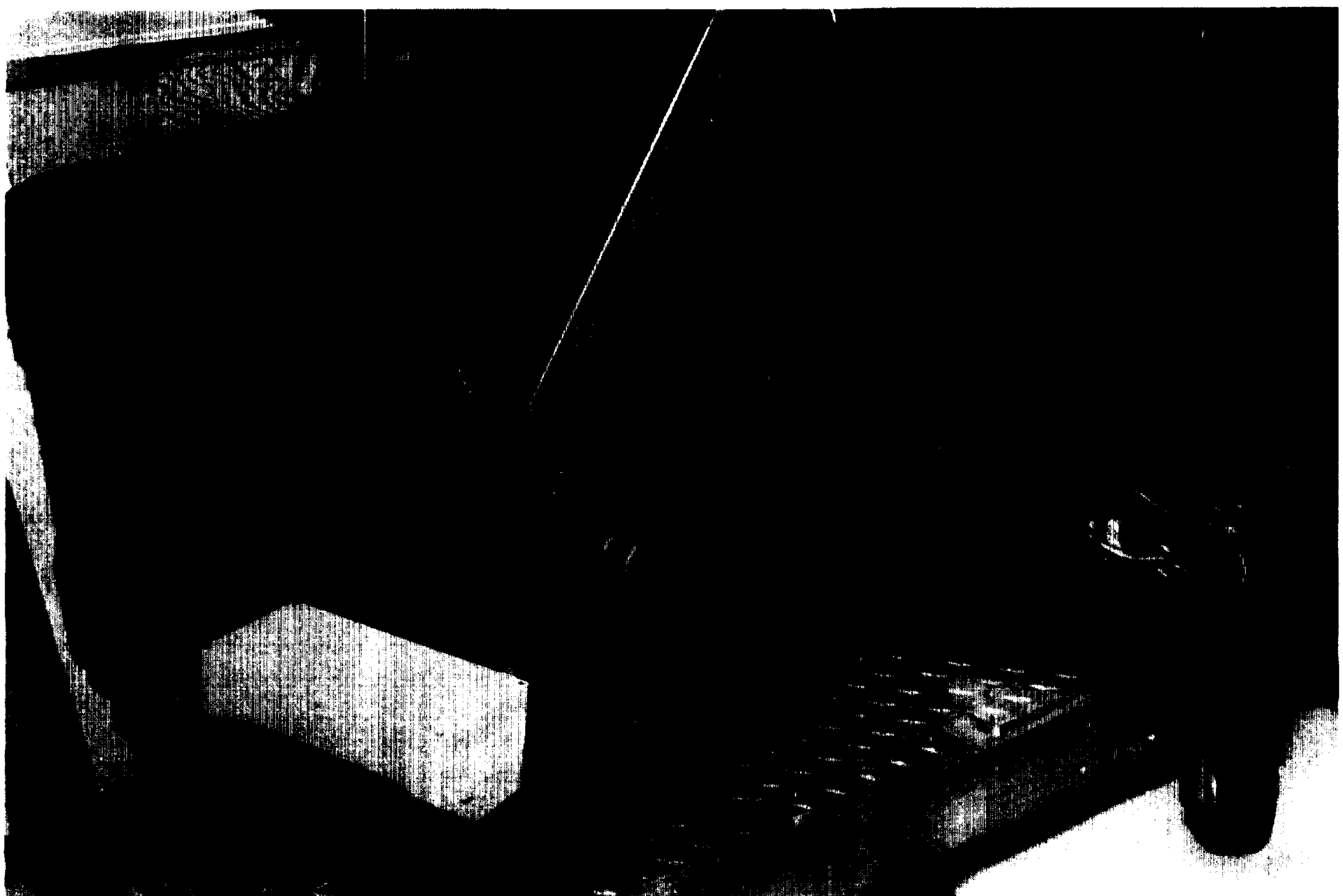


Figure 4. Portable ATS.

The torque test head (fig. 5) contains a 100 in-lb torque transducer, a motor with a speed reducer, an optical encoder, various couplings, a ball bearing, and a torque sensor motor control printed wiring assembly. All of these components are encased in a container with a removable lid to protect them from the elements. The test head mates with an interface rod via a $^{3}/_{8}$-in square drive at the container's open end and hard mounts to the pump by way of a flange.

The data acquisition system (fig. 6) consists of a 486SLC-25 notebook PC and a docking station. The laptop is configured with an 80-megabyte hard drive, 4 megabytes of RAM, a 3.5-in floppy disk drive, an ac-dc adapter, and an Ni-Cad rechargeable battery pack.

Other important hardware included with the system are the ATS battery pack and its accompanying battery charger, an interface cable, and a Bayonet Neil-Concelman (BNC) breakout box. This box serves as an interface between the laptop and a data recorder. The ATS has a BNC output of the unfiltered analog torque signal for recording to high-frequency analog tape. From the beginning, the bearing assessment team felt that the ATS should have a BNC connector in order to record the analog signal before the A/D conversion process. This valuable information can be used to verify the ATS's proper operation and to expand our high frequency data processing options.

The accompanying software program was written especially for the ATS and controls its entire operation. The program is menu driven and very user-friendly. It offers two levels of security with both an operator and a supervisor password. All activity is recorded in a log that can only be accessed by the supervisor. In order not to damage a pump by accidentally exceeding its maximum torque limit, the value can be preset. The entire unit will shut down if this maximum torque is exceeded.

To run a test, the operator can either select a previously loaded test set or select new parameters via pulldown menus. Many options are available and can be set in any sequence. The speed at which the pump turns can be set anywhere from 0 to 4 r/min. The length of the test can be established by either selecting the number of revolutions or number of seconds. One to four data points can be recorded per degree of rotation. It is also possible to always start a test in a designated zero position. If this option is selected, the unit will rotate to the start position before the test begins.

Once all the parameters have been chosen, the operator must verify his selections before the test can begin. During an actual test, the torque, angular position, and r/min levels are dynamically displayed (fig. 7). Upon completion, the results can be viewed along with the average values for the running and breakaway torque. Data can be stored in ASCII on either the hard disk or a diskette.

VII. COMBINATION TESTING

After the first test series, we were eager to begin recording torque and vibration data simultaneously. The team configured an off-the-shelf torque sensor, motor, and tachometer into a workable unit (fig. 8). We also designed and fabricated the support and interface structure ourselves to expedite testing.

Before the test began, a manual torque check was performed on the pump. We then simultaneously recorded the data from the torque sensor, the tachometer, and the accelerometers. The same transducers were mounted in similar locations to the previous test series. During our test, we gradually increased the pump's speed to a maximum of 4 r/min during a 30-min time interval. We slowly ramped up to 4 r/min, held this speed for 15 min, and then slowly ramped back down. The results of this successful test will be mentioned later.

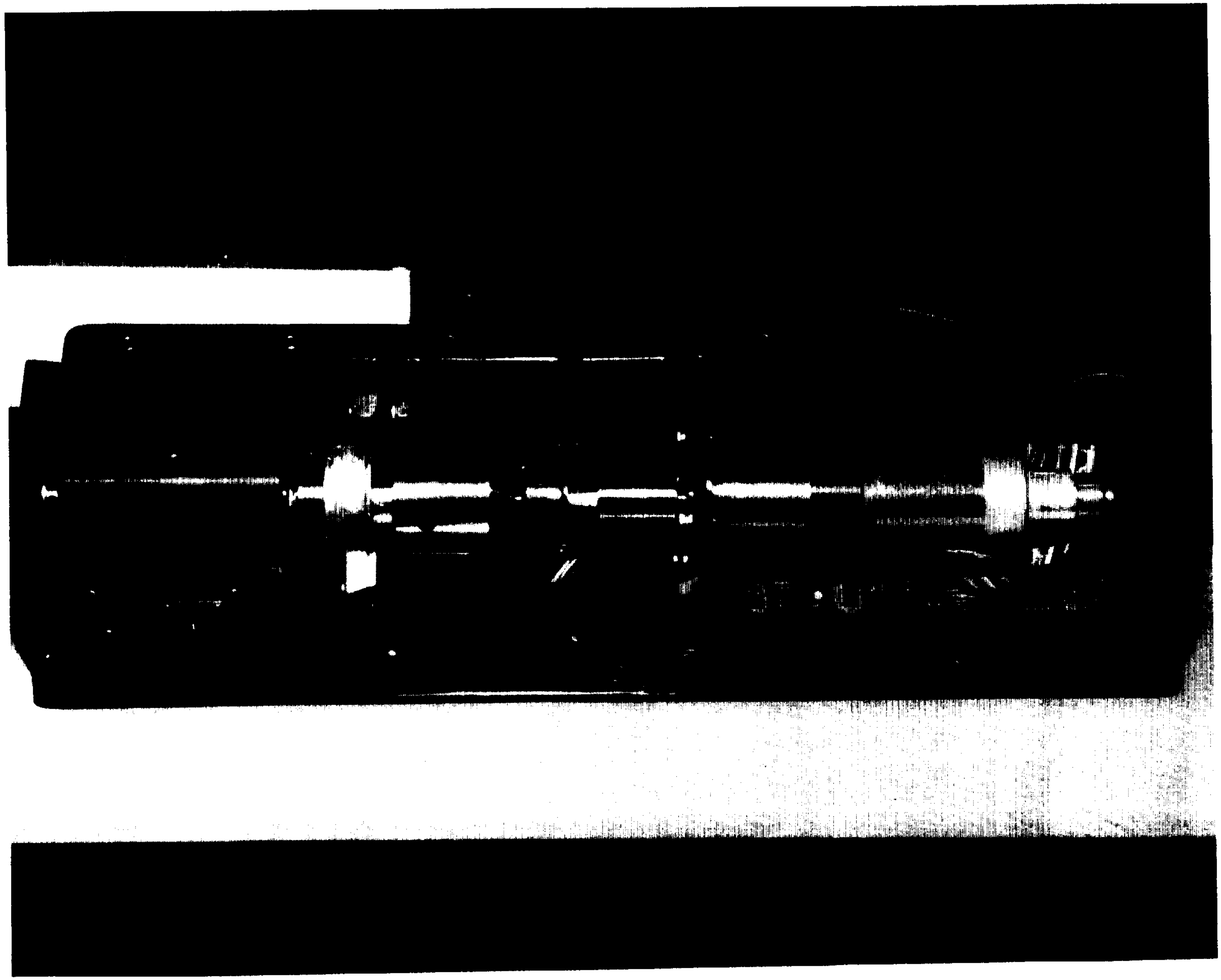


Figure 5. Torque test head.

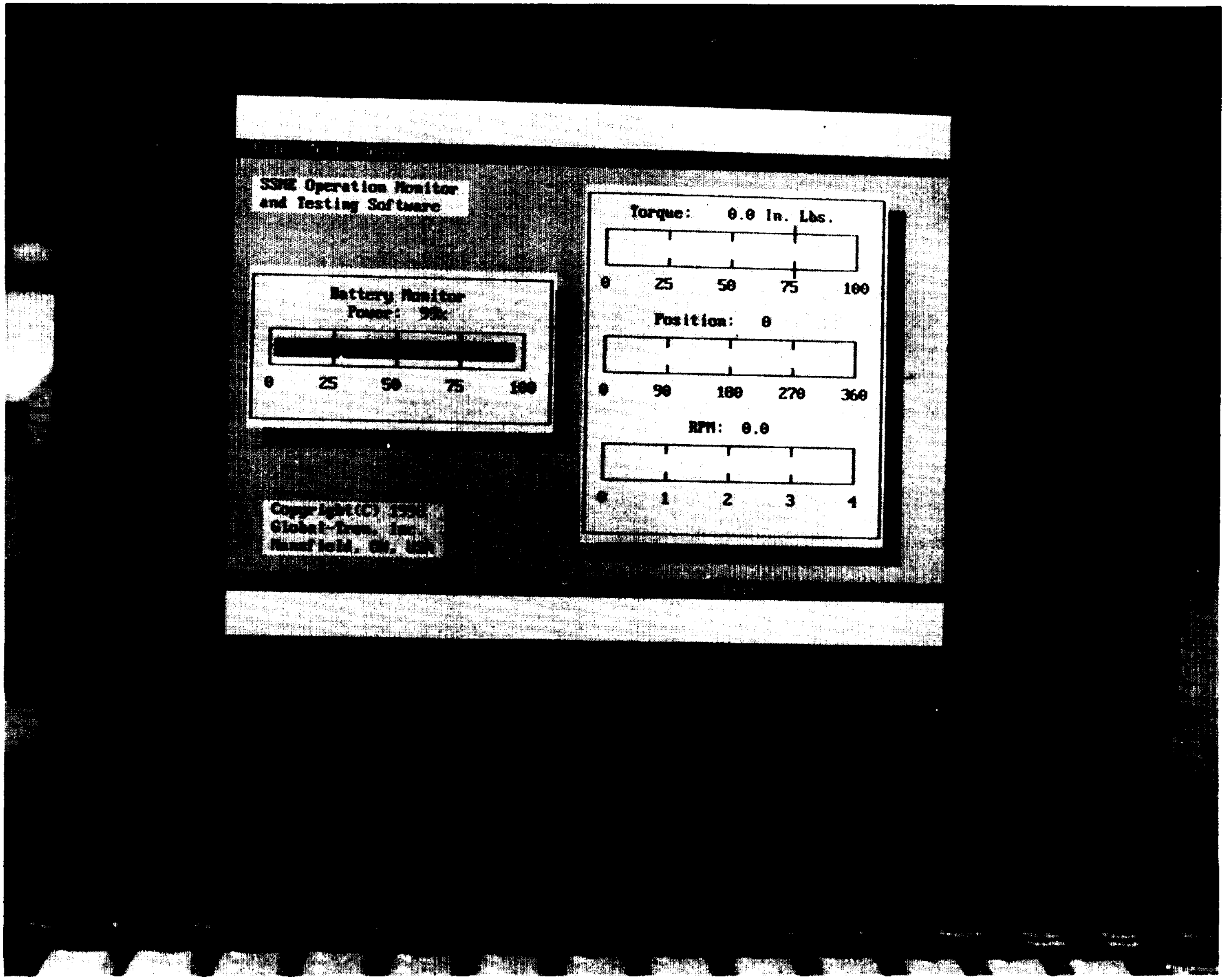


Figure 6. Data acquisition unit.

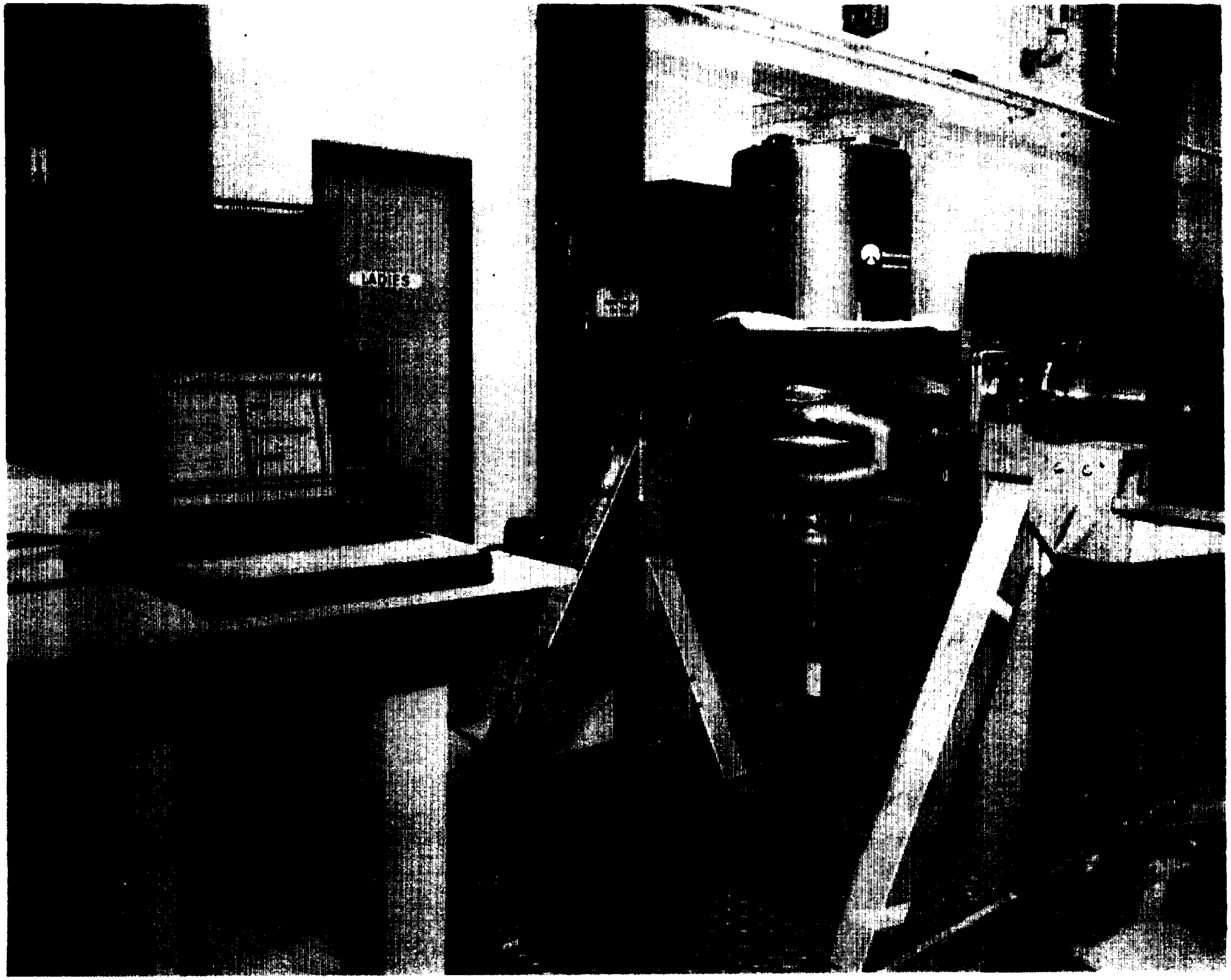


Figure 7. HPOTP undergoing actual torque test.

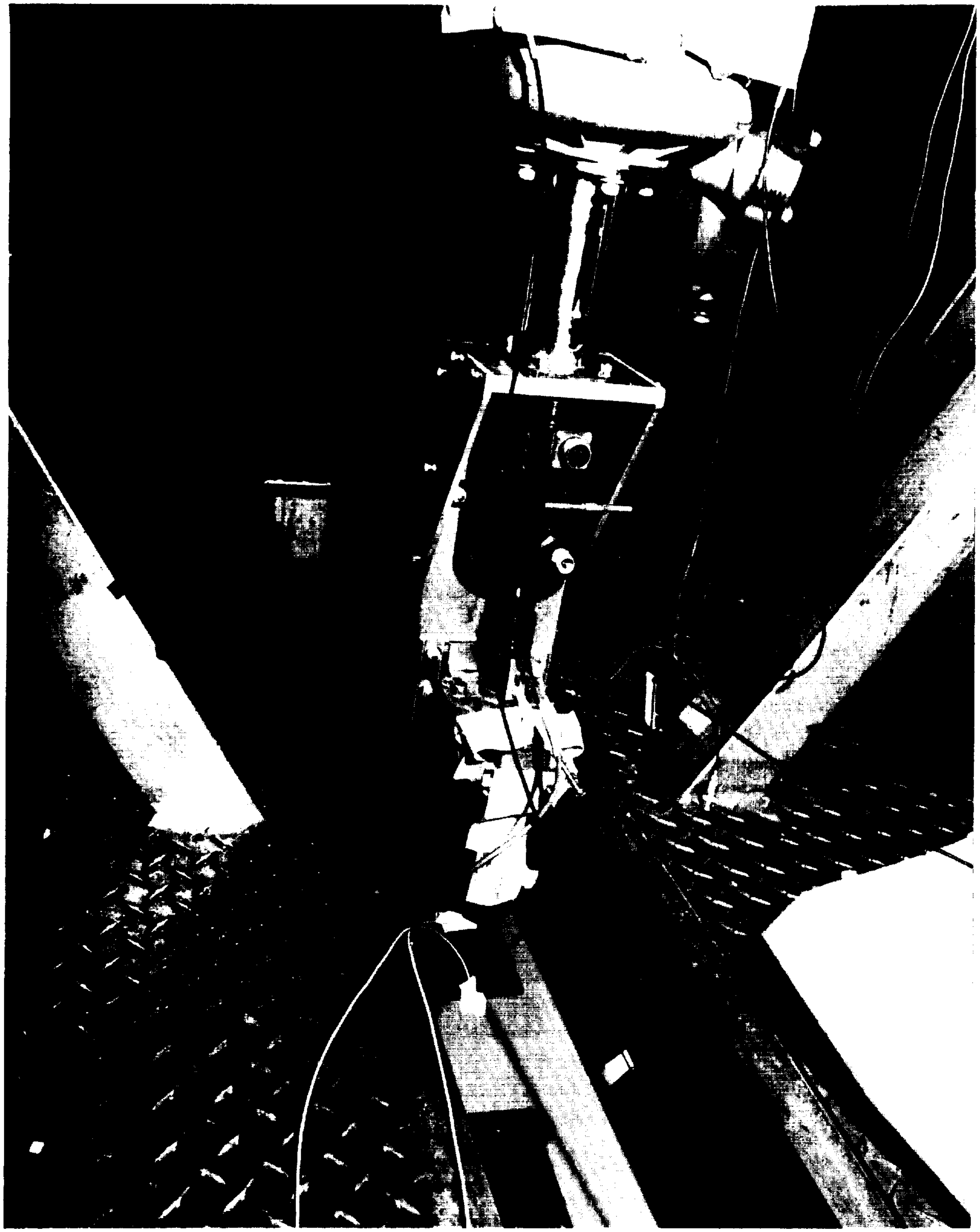


Figure 8. Off-the-shelf component test.

When the ATS was delivered, we were again enthusiastic to begin a third test series. In this test, we could achieve a twofold objective. We wanted to baseline our pump with good bearings and verify that the ATS was recording torque data accurately. In order to do this, the team used state-of-the-art accelerometers and seismic transducers. The ATS drove the pump at a constant speed of 4 r/min. Three different types of data were recorded which included high frequency torque data, key-phaser data, and acceleration data. The actual test ran for 10 min, with 9 min at maximum speed and 30 s at the beginning and end at zero speed.

In the data reduction, two different frequency regions were explored, those below 100 Hz and those below 20 kHz. In the low frequency analysis, the acquired torque signatures were examined for fundamental bearing defects. In the high frequency analysis, bearing diagnostic information was searched for using envelope detection techniques.

In the time and waveform analysis, the results looked promising. As expected the accelerometers registered very low energy levels. However, the general shape of the torque signature was consistent with those previously taken in test series one and two. Comparing the last plot in figure 4A with the second plot in figure 5A, which was taken during a previous test (01810R2-0006), we noticed the two were strikingly similar. This showed that the torque signature from the pump was repeatable using totally different sensors and data acquisition systems.

The frequency analysis also demonstrated favorable results. In the 0- to 100-Hz spectrum, no distinguishable components were found in the accelerometer data. However, some discrete low frequency activity was noted in the torque data. The 0- to 20-Hz spectrum taken from the ATS (fig. 7A) is remarkably similar to that taken in a previous test series (fig. 8A).

The high frequency torque data looked very promising since the signature content was repeatable from test to test with different transducers. No obvious bearing defect frequencies were found as we had expected with good bearings. However, some unidentified spectral components were noticed which provide merit for further study. In conclusion, the team decided that a HPOTP with severely worn bearings was needed as a test article to unequivocally prove concept feasibility. For an in-depth discussion of the this test and its results, please refer to the appendix.

VIII. CONCLUSION

Over the last 2$^{1/2}$ years, the bearing assessment team has made important strides. Of the three different NDE methods that were considered, we have enjoyed the most success with torque. AE and vibration still need much more work for their verification. Our accomplishments are many. We have baselined an HPOTP which to the best of our knowledge has good bearings, and thus have excellent reference data to be used for later comparisons. The very development of a prototype torque device was significant in itself. We have verified that the ATS can record Rocketdyne HPOTP static torque data accurately and consistently.

Thus, it is our recommendation that the ATS be incorporated into the standard turbopump checkout procedures to replace the manual torque checks for both the Rocketdyne and Pratt and Whitney HPOTP's. There is no doubt that this automated method is better than the currently used manual one. In addition, the data taken can be stored and later retrieved for study and analysis. More importantly, this information has the potential of one day being used to determine the amount of HPOTP pump end bearing wear. Additional systems could be developed for use at Stennis Space Center, Kennedy Space Center, and Rocketdyne.

Mission safety has always been a primary goal of the space shuttle program. This ATS would only enhance the SSME's performance and reliability. It would greatly improve bearing diagnostics and thus provide another degree of confidence to ensure shuttle safety. The savings in time and money are secondary but are still worth serious consideration during these times of severe budget cuts.

IX. FUTURE PLANS

Although much has been accomplished, much more work is left. The program is by no means finished. Our research with acoustic emission and vibration sensors will still continue, but with a new angle. The team is in the process of designing an advanced ATS that will be capable of turning the turbopump a maximum of 50 r/min. Faster rotational speeds will increase energy levels in the system and hopefully better excite the monitoring sensors. This advanced ATS will also have an increased motor and torque capacity to handle the alternate Pratt and Whitney HPOTP which breaks at a much higher torque level of 370 in-lb.

Our ultimate goal to determine the relationship between torque signatures and bearing wear is still foremost in our minds. Since the likelihood of obtaining a turbopump to test with severely worn bearings is not favorable, the bearing and seal materials tester will be used. This unit is conveniently located at MSFC and runs a set of bearings until they are worn. Our objective will be to establish a torque signature for the unit with new bearings and then acquire the torque signature at the end of the bearing test when there is likely substantial bearing distress. Upon disassembly the bearings will be analyzed and the wear correlated to our data observations. This approach will give us fast results and enable us to familiarize ourselves with data and to develop data analysis criteria.

As a parallel effort, we will test the ATS on an assembled SSME. However, in order to accomplish this, precision interface hardware must be designed to mount the unit to an installed HPOTP. A support bracket will hold the test head in place and guide the interface rod to mate with the pump through the preburner pump inlet duct. Since attachment involves a blind installation through a duct, this guide is needed so the ATS will not induce any additional loads into the HPOTP. Once it is demonstrated that we can take successful measurements on both Rocketdyne and Pratt and Whitney HPOTP's, a fairly extensive data base of flight and developmental HPOTP's can be established. Then our formidable task of data analysis will begin.

The ATS itself will be optimized by making the unit lighter and shorter for ease of handling and installation. The software package will be enhanced to accommodate more user-defined parameters and to enable the operator to manipulate the data immediately following the test. Finally, this system may one day be adapted for industrial use to perform quick and reliable bearing checks and thus provide another level of quality assurance.

APPENDIX

National Aeronautics and
Space Administration

George C. Marshall Space Flight Center
Marshall Space Flight Center, Alabama 35812

NASA

JUN 17 1993

Reply to Attn of: ED23-93-045

TO: EP65/Barbara Spiegel

FROM: ED23/Tom Zoladz

SUBJECT: Dynamic Assessment of SSME High Pressure Oxidizer Turbopump (HPOTP) Unit 0810R2 Dry-Spin Data in Support of SSME Bearing Degradation Program

Intent of Analysis

The actual testing and supporting analysis of HPOTP unit 0810R2 was performed in order to baseline the pump dynamic signal content as monitored through external housing accels and a high frequency torque sensor contained in the Global-Tron Automated Torque Sensor (ATS) system. Attention in the analysis was focused on recovery of possible bearing defect signature content within the acquired data.

Test Setup

The tested unit, HPOTP unit 0810R2, is a Phase II development pump with several configuration modifications including a 15 vane main pump inlet along with a thin blade inducer, ion implanted bearings, and FEP coated cages. During developmental hot fire testing, the pump-end bearings accumulated approximately 6,000 seconds of run time while the turbine-end bearings have experienced over 10,000 seconds. Following this series of testing, unit 0810R2 did not receive a full tear down which would have allowed for complete inspection and measurement of the bearings. Instead, a partial tear down along with "microwiggle" and "squeeze" testing was performed, and no appreciable bearing wear was found (<1 mil) in either the pump or turbine-end bearings. These results confirmed wear predictions made using internal isolator strain gage hot fire data which showed low amplitude bearing cage frequency responses. Prior to the current test, the unit also accumulated over 2000 seconds of dry-spin run time at speeds up to 8 RPM.

In the current test configuration, the Global-Tron ATS system was used to drive the HPOTP shaft assembly at a constant speed of 4 RPM. Along with high frequency torque and key-phaser data supplied by the ATS, acceleration data was taken at various locations along the pump and turbine-end flanges of the HPOTP external casing. Figure 1 shows both the locations and nomenclatures for the acquired channels. The subject test, 0810R2-010, ran for a total duration of 10 minutes with 9 minutes at 4 RPM and 30 seconds at no rotation at both the start and end of the run for background level assessment purposes. All channels were recorded with a wide-band analog recorder (0-20 kHz), and subsequent data reduction was performed using the ED23 Operator Interactive Signal Processing System.

Data Reduction/Analysis Procedure

Two major analysis directions were taken during reduction of the dry-spin data of test 0810R2-010. The first concentrated on the low frequency dynamic content of the acceleration and torque data (<100 Hz) while looking for fundamental bearing defect signatures such as cage, ball pass, and ball spin frequencies. The second effort involved analysis of the high frequency dynamic content of the acquired signatures (<20 kHz). During this high frequency analysis, resonant structural or instrumentation carrier frequencies, possibly excited by bearing defect pulsations, were sought after for attempts at bearing diagnostic information recovery via envelope detection techniques. In both passes at the data, both time and frequency domain (including linear and nonlinear) analysis techniques were relied on.

Results

Time/Waveform analysis

As expected, with such low rotational speeds being applied to the HPOTP shaft assembly via the ATS drive system, acceleration levels registered on the turbopump housing were minimal. Figures 2 and 3 show wide-band (0-20 kHz) time histories of all channels including high frequency torque (key-phaser/speed data is not shown since it was suspect). Peak-to-peak acceleration levels (Gpp) were greatest at the ATS casing location, as would be expected, with levels of approximately 1.1 Gpp over background readings. Highest acceleration levels recorded on the turbopump casing after start (>28 seconds) were registered on the radially mounted seismic accels at the 180° and 225° circumferential positions on the pump-end flange with increases of $3.5(10^{-3})$ Gpp and $7.0(10^{-2})$ Gpp over background levels, respectively. The radial high frequency accels (pbp45sh and hpot135sh) showed little if any response at the initiation of dry-spinning. The torque sensor did show a substantial response during dry-spinning with an increase of ~1.7 Vpp. Using vendor supplied calibration information, this translated to a peak-to-peak variation of ~30 in-lb which is suspect since typical running torque for the pump averaged ~30 in-lb also. Since a dynamic variation in torque of the same order of magnitude as the static is doubtful, the scaling of the torque output is questionable. Possible sources of error could be found in internal ATS gains or in the actual data recording. Even though the amplitude of the torque signature was suspect, the general shape of the wave form was consistent with those acquired in previous dry-spin testing using a different torque sensor/drive motor configuration. The final plot of figure 4 shows the torque wave form over approximately one revolution of the LOX pump shaft for the current test at ~4 RPM. Clearly, there is discrete content within the signal. Figure 5, plot 2, shows a torque signature taken during previous testing (0810R2-006) of unit 0810R2 using a different torque sensing device while running at ~4.1 RPM. Notice that the torque wave forms in the two figures are very similar. Using the old data as a standard, the new torque variation as taken from the ATS sensor, is approximately an order of magnitude high. This amplitude scaling discrepancy should be addressed prior to continued testing.

Frequency Analysis

Frequency analysis performed by ED23 concentrated first on identifying fundamental bearing defect components in the 0-100 Hz spectrum. With the turbopump dry-spinning at ~4 RPM, all fundamental bearing defect frequencies would be less than 1 Hz. No distinguishable spectral components were found in this band using accelerometer data. However, some discrete low frequency activity was noted in the torque data. Figure 6 is a 0-10 Hz Power Spectral Density (PSD) plot showing the torque signal energy content versus frequency. The peaks at 0.2 Hz (labeled a), 0.5 Hz (2a), 0.7 Hz (3a), 0.9 Hz (4a), and 1.1 Hz (5a) have shown to be harmonically related through nonlinear analysis. The fundamental at 0.2 Hz is approximately 3 times rotor synchronous (3N) and may or may not be bearing related. Also, observe in Figure 6 the high distribution of energy in the 4.5-5.5 Hz band. This band of energy corresponds to the fundamental periodicity (T= ~0.2 seconds) seen in the torque wave form of figure 4. The 0-20 Hz spectrum taken from the ATS torque sensor in the current test (figure 7) is amazingly similar to that taken in previous testing using a different torque sensing device (figure 8) in previous testing. Apart from amplitude scaling differences, the frequency content of the signatures taken from two different sensors agree very well. As to identifying the source of the 4.5-5.5 Hz band of energy seen in the current torque data, continued analysis is necessary .

Some low amplitude (<2.6(10^{-4}) Grms) discrete components were common to several of the accels and the torque sensor. Figures 9 and 10 show discrete components at 31.9 Hz (labeled A), 39.4 Hz (B), and 43.8 Hz (C). Analysis showed the peaks to be coherent across all channels pointing to a common source. These frequency components are not line noise related as seen in the topographical isoplot in figure 11. The components wander slightly in frequency and appear only during actual pump rotation (unlike the readily apparent line-noise also shown in the figure). As with the previously mentioned unidentified frequencies, further testing and analysis must be performed in order to identify the source of these frequency components.

With regard to the 0-20 kHz data, no appreciable frequency content was noted in any of the channels past 100 Hz. No structural of transducer resonances were found.

Conclusions/Recommendations

In terms of assessing the bearing health of HPOTP unit 0810R2, the dry-spin test was inconclusive. Analysis has shown that energy levels within the LOX turbopump shaft assembly as registered through external housing accels are very low making the project very challenging. The high frequency torque data is promising since the signature content was repeatable from test to test with different transducers. No obvious bearing defect frequencies are apparent in the monitored data. However, some unidentified spectral components provide motivation for continued testing within the SSME Bearing Degradation Program. Results from unit 0810R2 dry-spin testing should be correlated to future tear-down inspection findings for the unit. Moreover, a HPOTP unit with documented bearing defects should be used as a test article in future testing.

Thomas F. Zoladz

Enclosures

Approval:

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Luke A. Schutzenhofer
Chief
Structural Analysis Division

6-17-93
Date

cc:
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EP65/Mr. Cross
ED33/Mr. Jones
ED14/Mr. Earhart
EP62/Mr. Genge

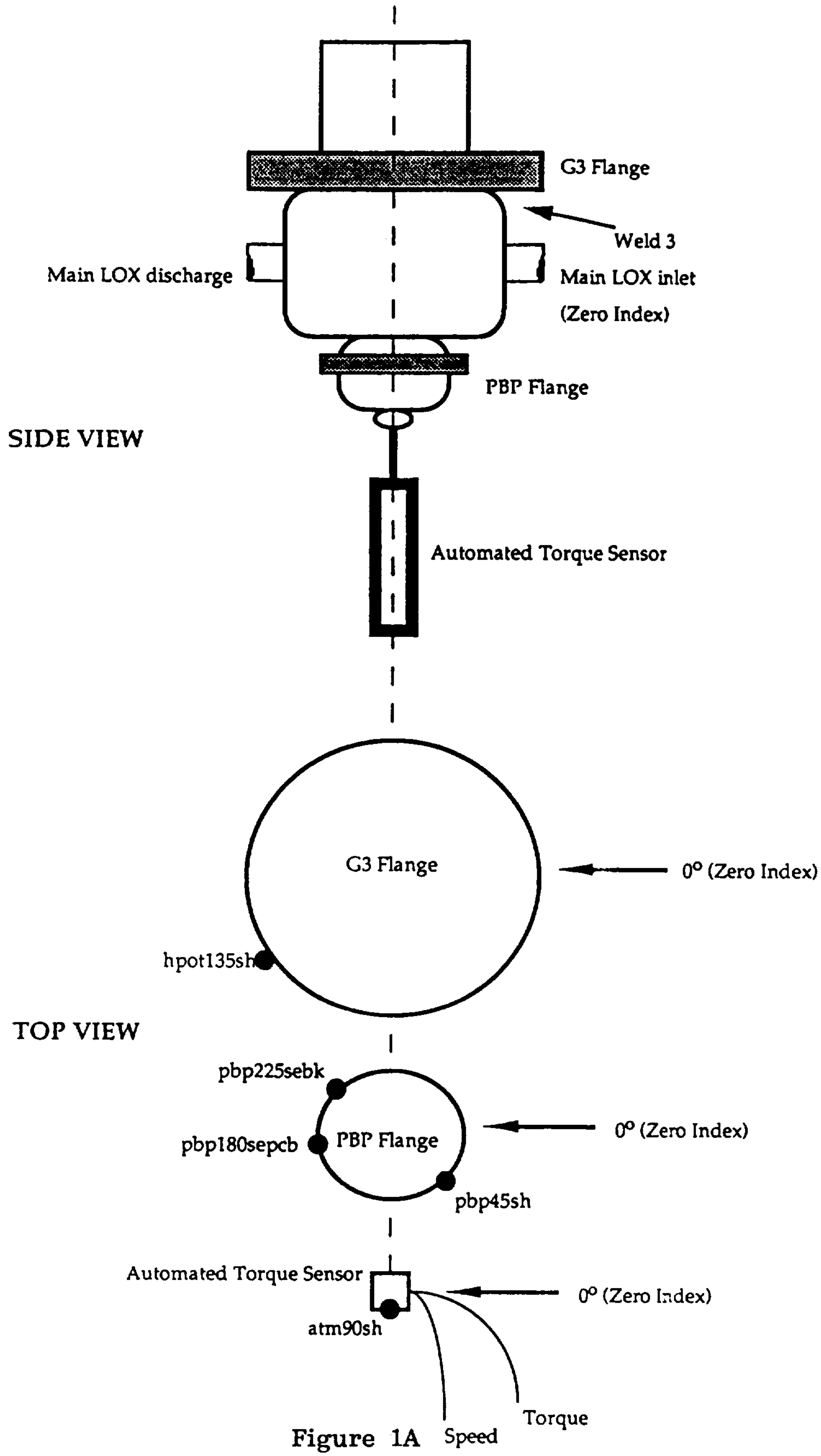

G3 Flange
Weld 3
Main LOX discharge
Main LOX inlet
(Zero Index)
PBP Flange
SIDE VIEW
Automated Torque Sensor
G3 Flange
0° (Zero Index)
hpot135sh
TOP VIEW
pbp225sebk
pbp180sepcb
PBP Flange
0° (Zero Index)
pbp45sh
Automated Torque Sensor
0° (Zero Index)
atm90sh
Speed
Torque

Figure 1A

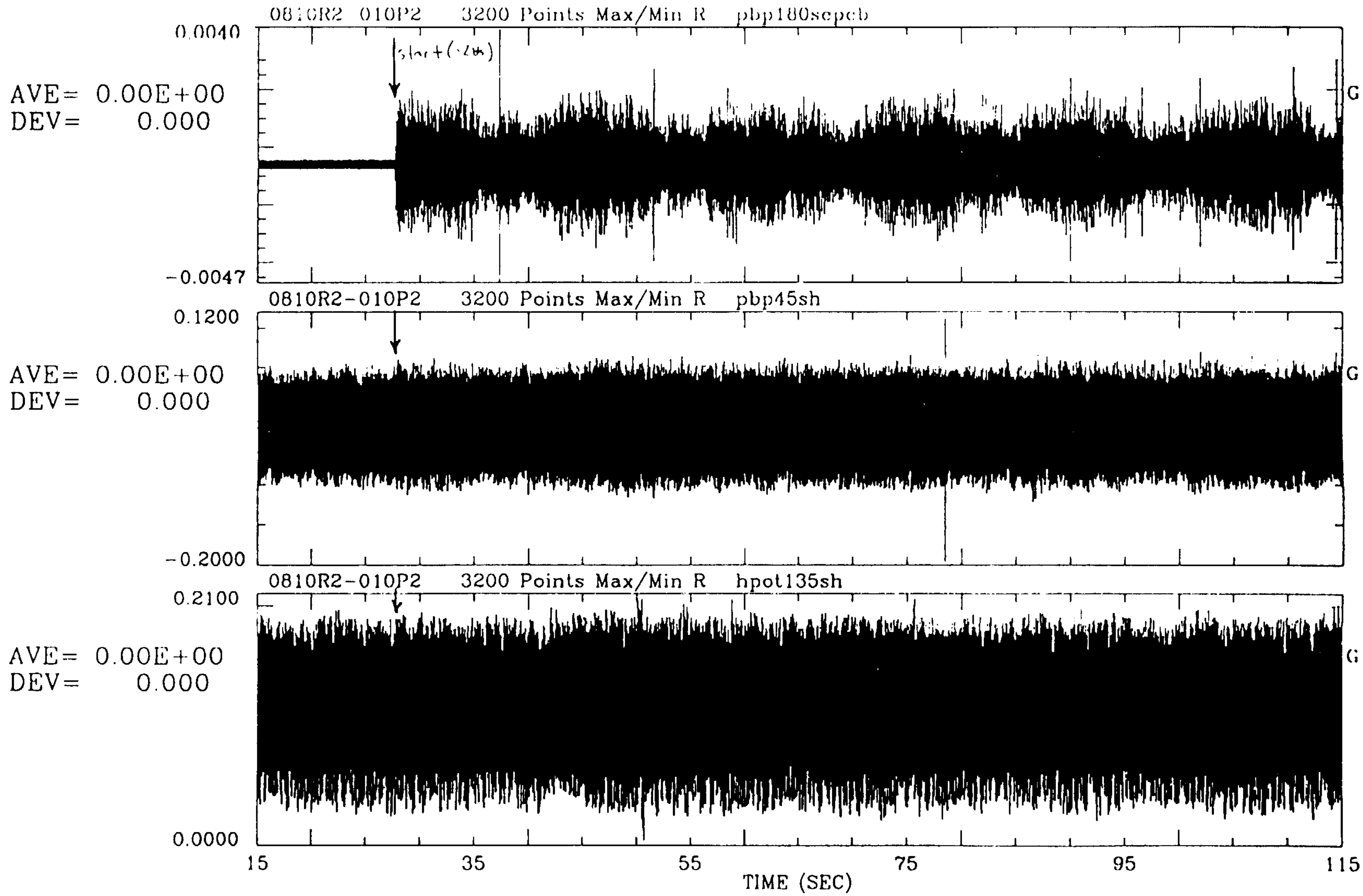
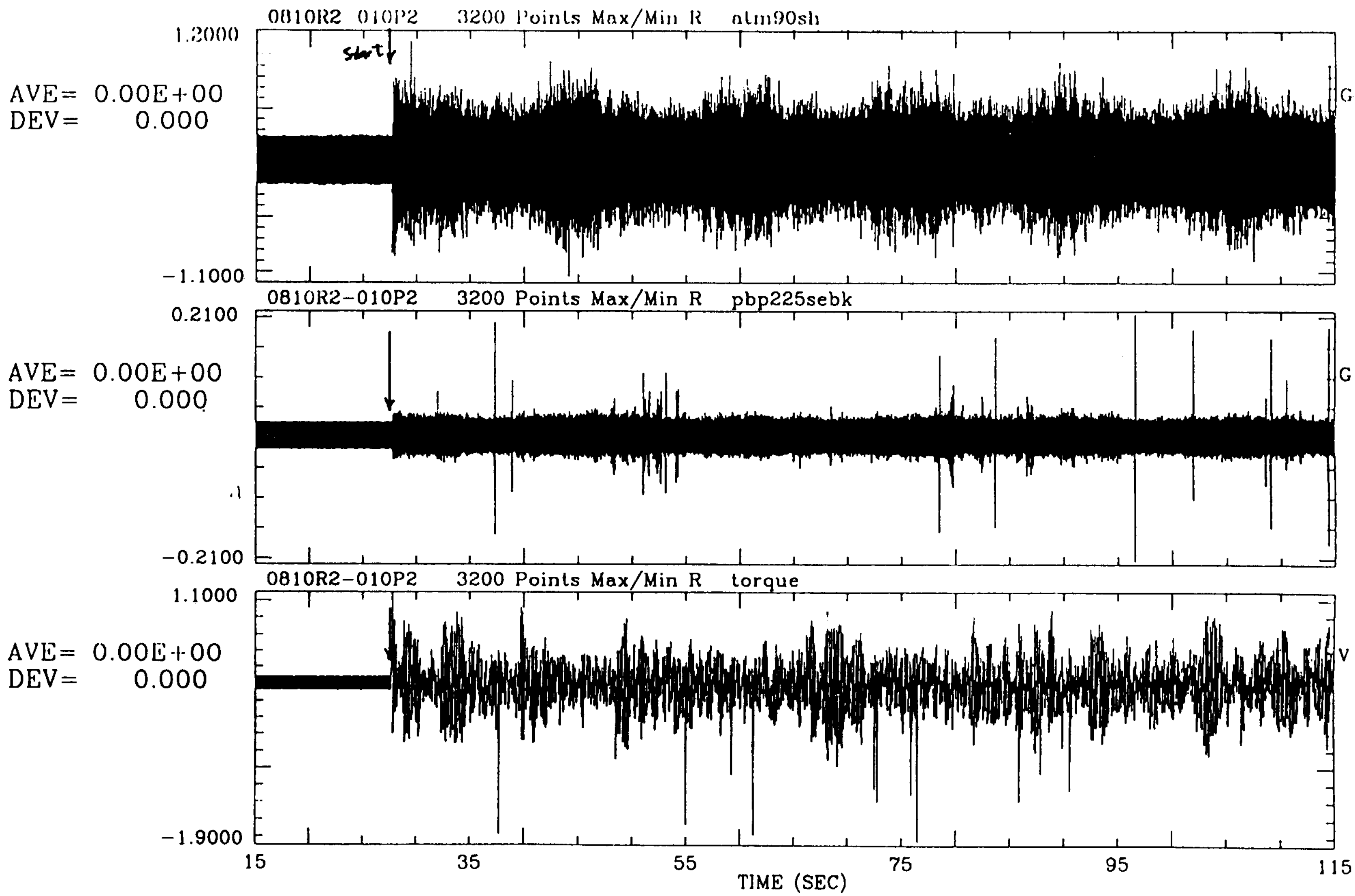

0810R2 010P2 3200 Points Max/Min R pbp180sepcb
0810R2-010P2 3200 Points Max/Min R pbp45sh
0810R2-010P2 3200 Points Max/Min R hpot135sh
AVE= 0.00E+00
DEV= 0.000
0.0040
-0.0047
0.1200
-0.2000
0.2100
0.0000
G
15
35
55
75
95
115
TIME (SEC)

Figure 2A



0810R2 010P2 3200 Points Max/Min R atm90sh
1.2000
AVE= 0.00E+00
DEV= 0.000
start
-1.1000
G
0810R2-010P2 3200 Points Max/Min R pbp225sebk
0.2100
AVE= 0.00E+00
DEV= 0.000
-0.2100
G
0810R2-010P2 3200 Points Max/Min R torque
1.1000
AVE= 0.00E+00
DEV= 0.000
-1.9000
V
15
35
55
75
95
115
TIME (SEC)

Figure 3A

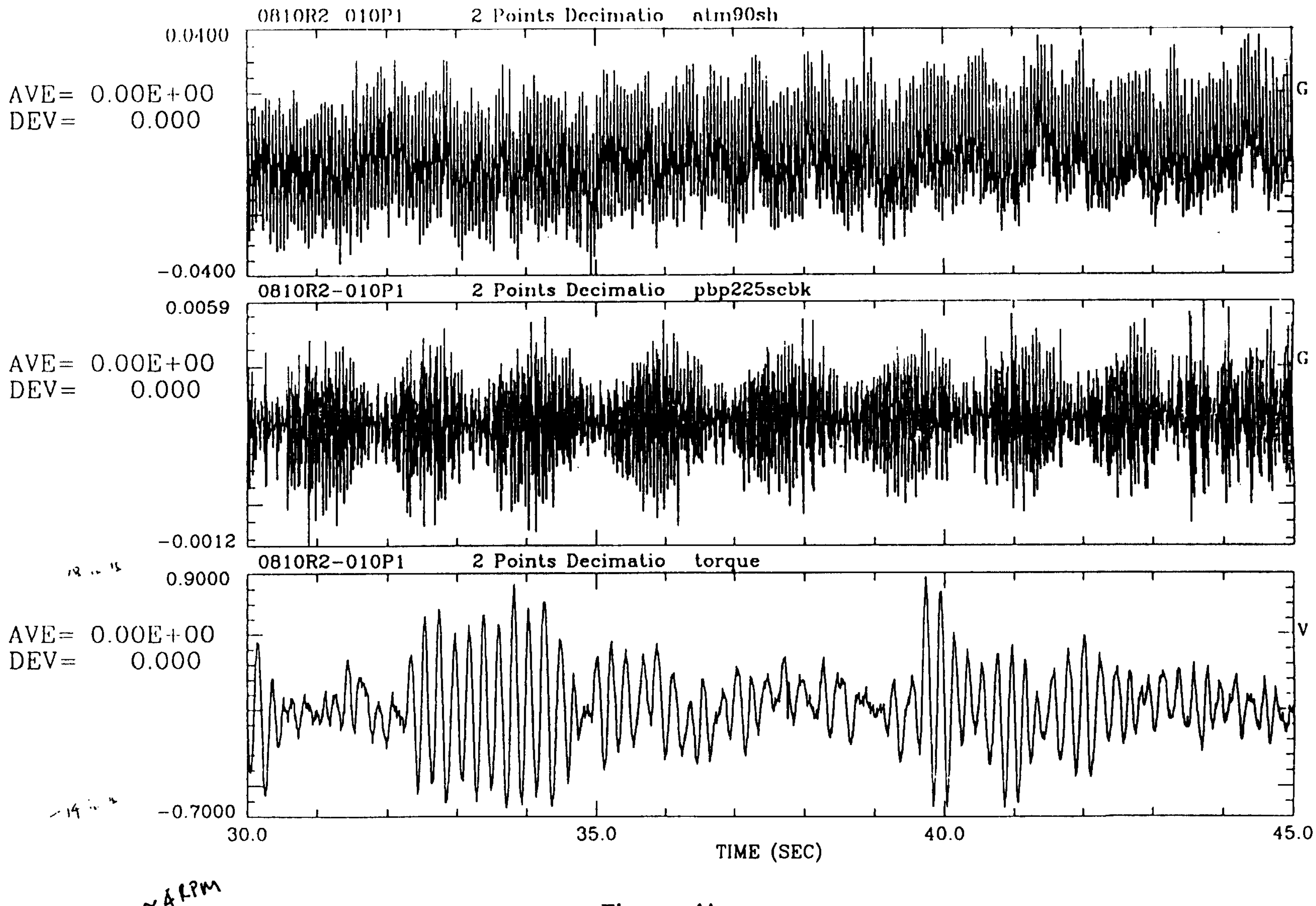
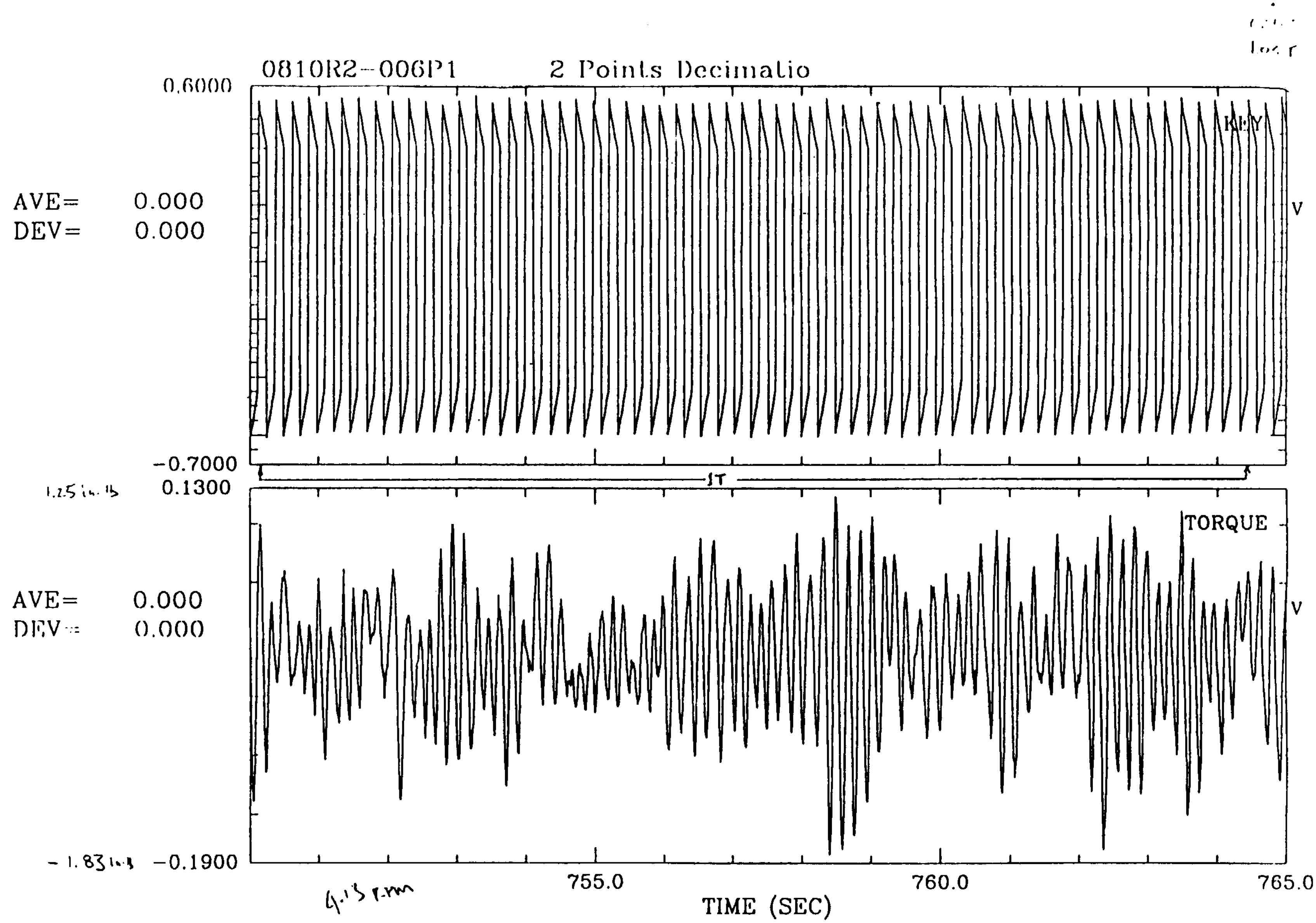

0810R2 010P1 2 Points Decimatio atm90sh
0.0400
AVE= 0.00E+00
DEV= 0.000
-0.0400
G
0810R2-010P1 2 Points Decimatio pbp225scbk
0.0059
AVE= 0.00E+00
DEV= 0.000
-0.0012
G
0810R2-010P1 2 Points Decimatio torque
0.9000
AVE= 0.00E+00
DEV= 0.000
-0.7000
V
30.0
35.0
40.0
45.0
TIME (SEC)
~4 RPM

Figure 4A



0810R2-006P1
2 Points Decimatio
0.6000
AVE= 0.000
DEV= 0.000
-0.7000
KEY
V
0.1300
AVE= 0.000
DEV= 0.000
-0.1900
TORQUE
V
755.0
760.0
765.0
TIME (SEC)

Figure 5A

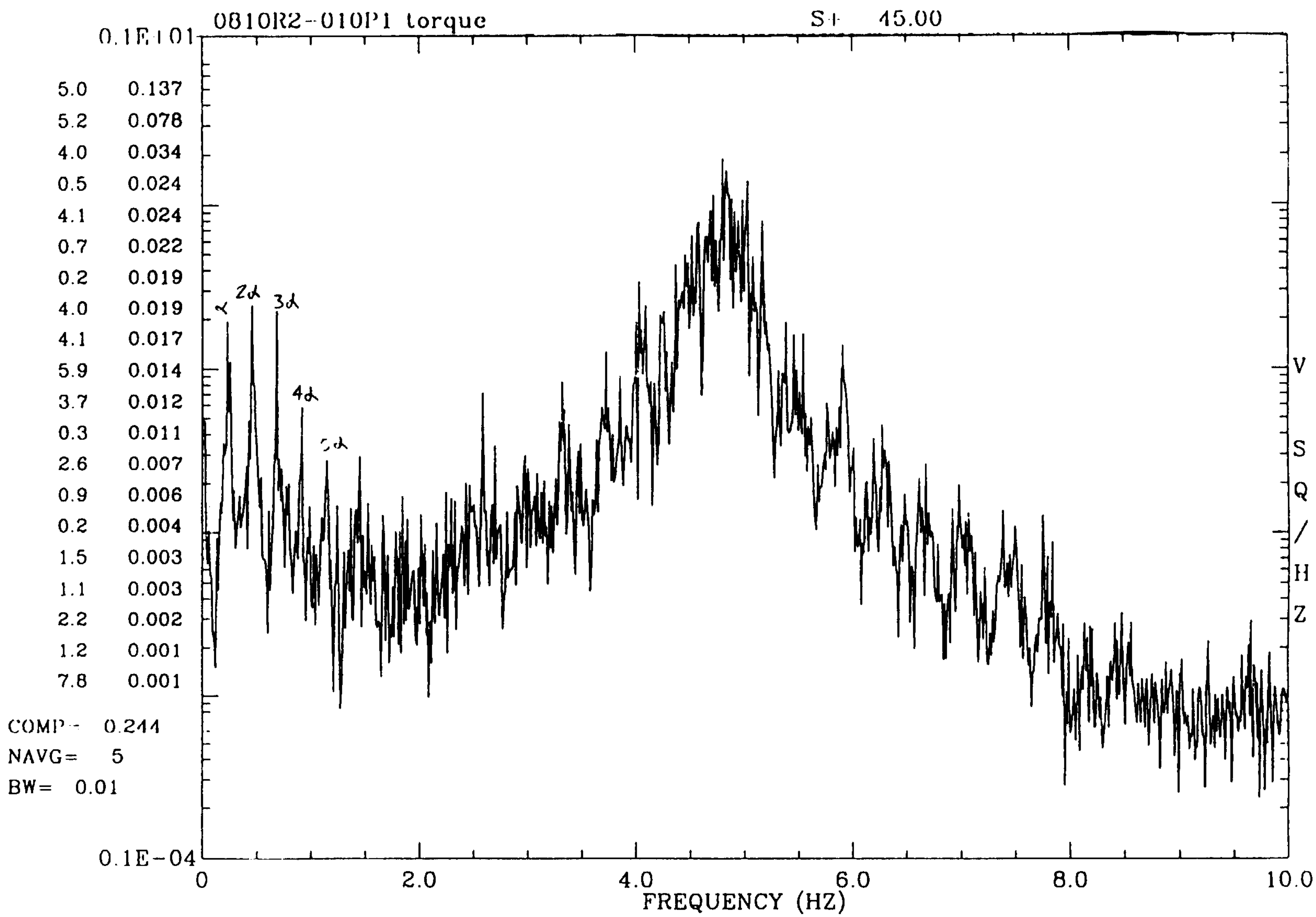

0810R2-010P1 torque
S+ 45.00
0.1E+01
0.1E-04
5.0 0.137
5.2 0.078
4.0 0.034
0.5 0.024
4.1 0.024
0.7 0.022
0.2 0.019
4.0 0.019
4.1 0.017
5.9 0.014
3.7 0.012
0.3 0.011
2.6 0.007
0.9 0.006
0.2 0.004
1.5 0.003
1.1 0.003
2.2 0.002
1.2 0.001
7.8 0.001
COMP= 0.244
NAVG= 5
BW= 0.01
α
2α
3α
4α
5α
V
S
Q
/
H
Z
0
2.0
4.0
6.0
8.0
10.0
FREQUENCY (HZ)

Figure 6A

CURRENT TEST/ATS TORQUE TRANSDUCER

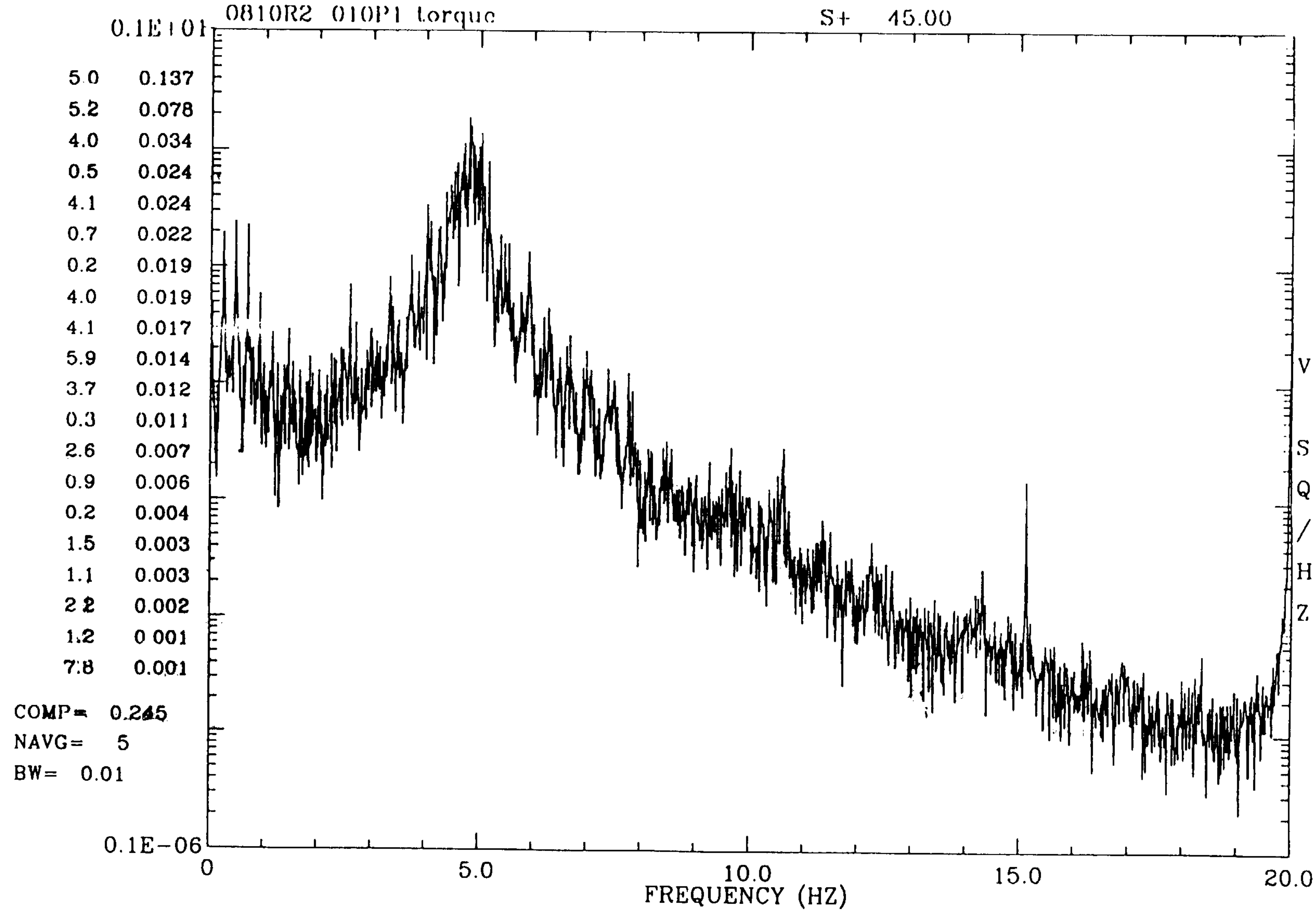


Figure 7A

PREVIOUS TEST

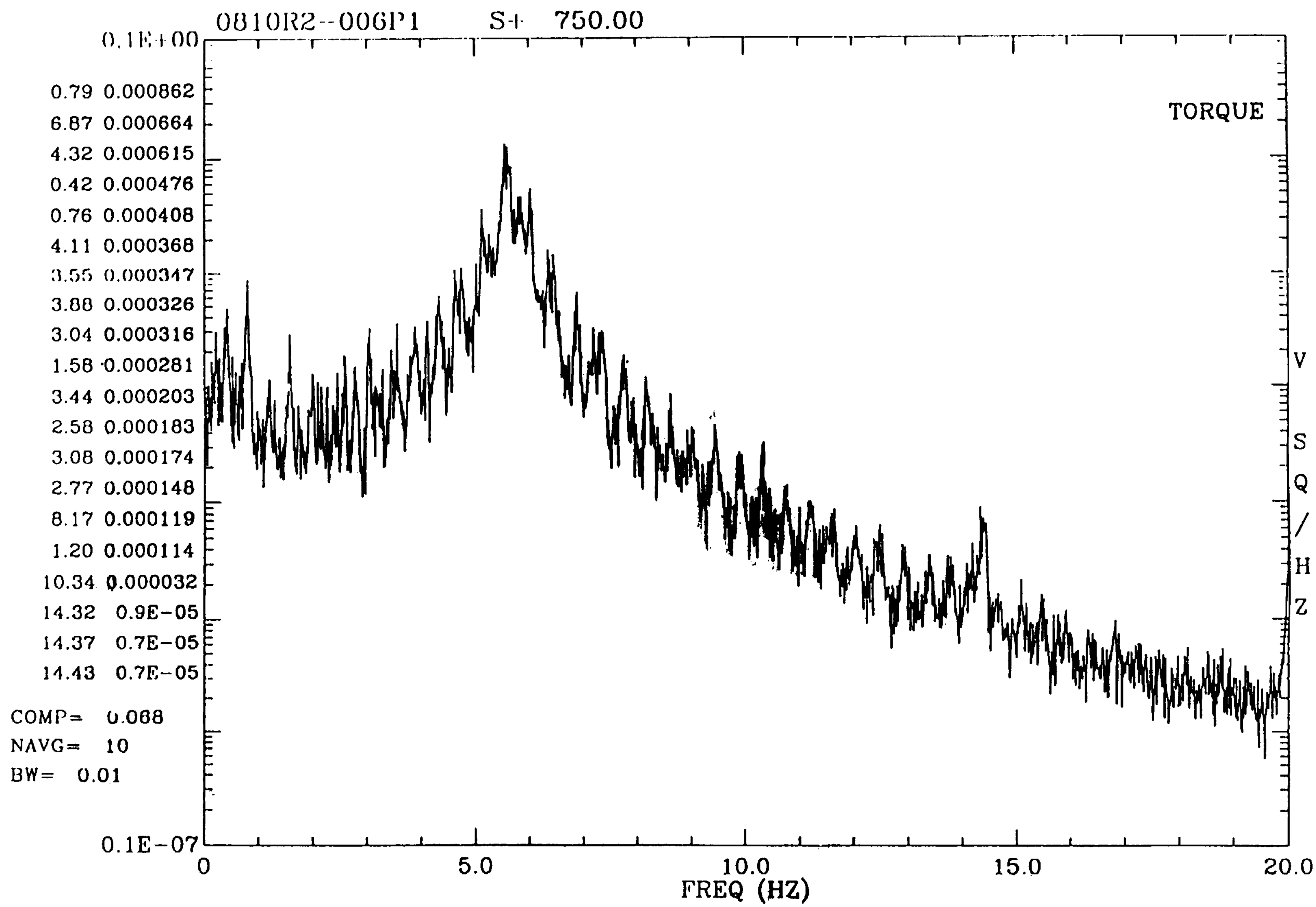


Figure 8A

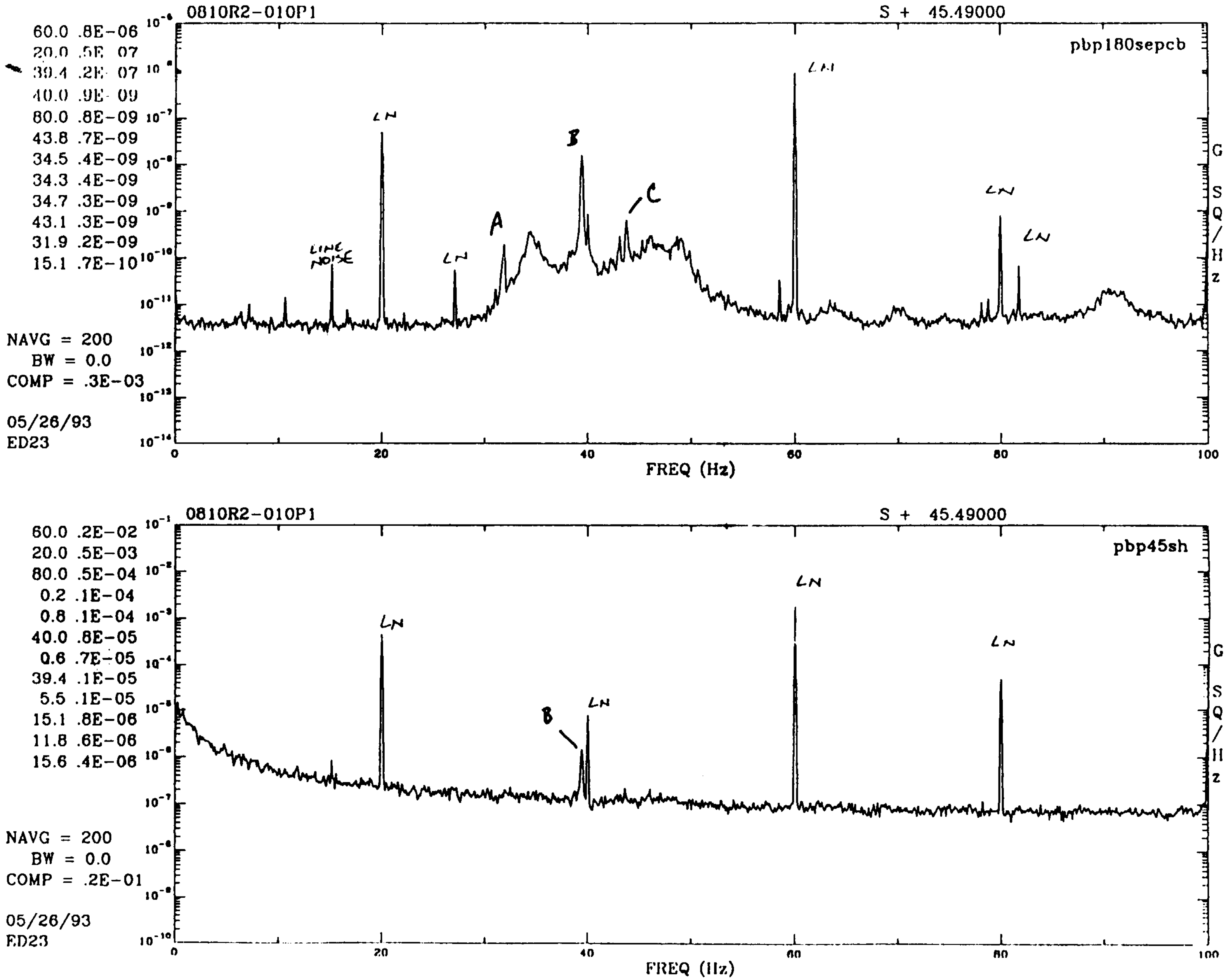

0810R2-010P1
S + 45.49000
pbp180sepcb
60.0 .8E-06
20.0 .5E-07
39.4 .2E-07
40.0 .9E-09
80.0 .8E-09
43.8 .7E-09
34.5 .4E-09
34.3 .4E-09
34.7 .3E-09
43.1 .3E-09
31.9 .2E-09
15.1 .7E-10
NAVG = 200
BW = 0.0
COMP = .3E-03
05/26/93
ED23
LINE NOISE
LN
A
B
C
GSQ/Hz
FREQ (Hz)
0810R2-010P1
S + 45.49000
pbp45sh
60.0 .2E-02
20.0 .5E-03
80.0 .5E-04
0.2 .1E-04
0.8 .1E-04
40.0 .8E-05
0.6 .7E-05
39.4 .1E-05
5.5 .1E-05
15.1 .8E-06
11.8 .6E-06
15.6 .4E-06
NAVG = 200
BW = 0.0
COMP = .2E-01
05/26/93
ED23
LN
B
GSQ/Hz
FREQ (Hz)

Figure 9A

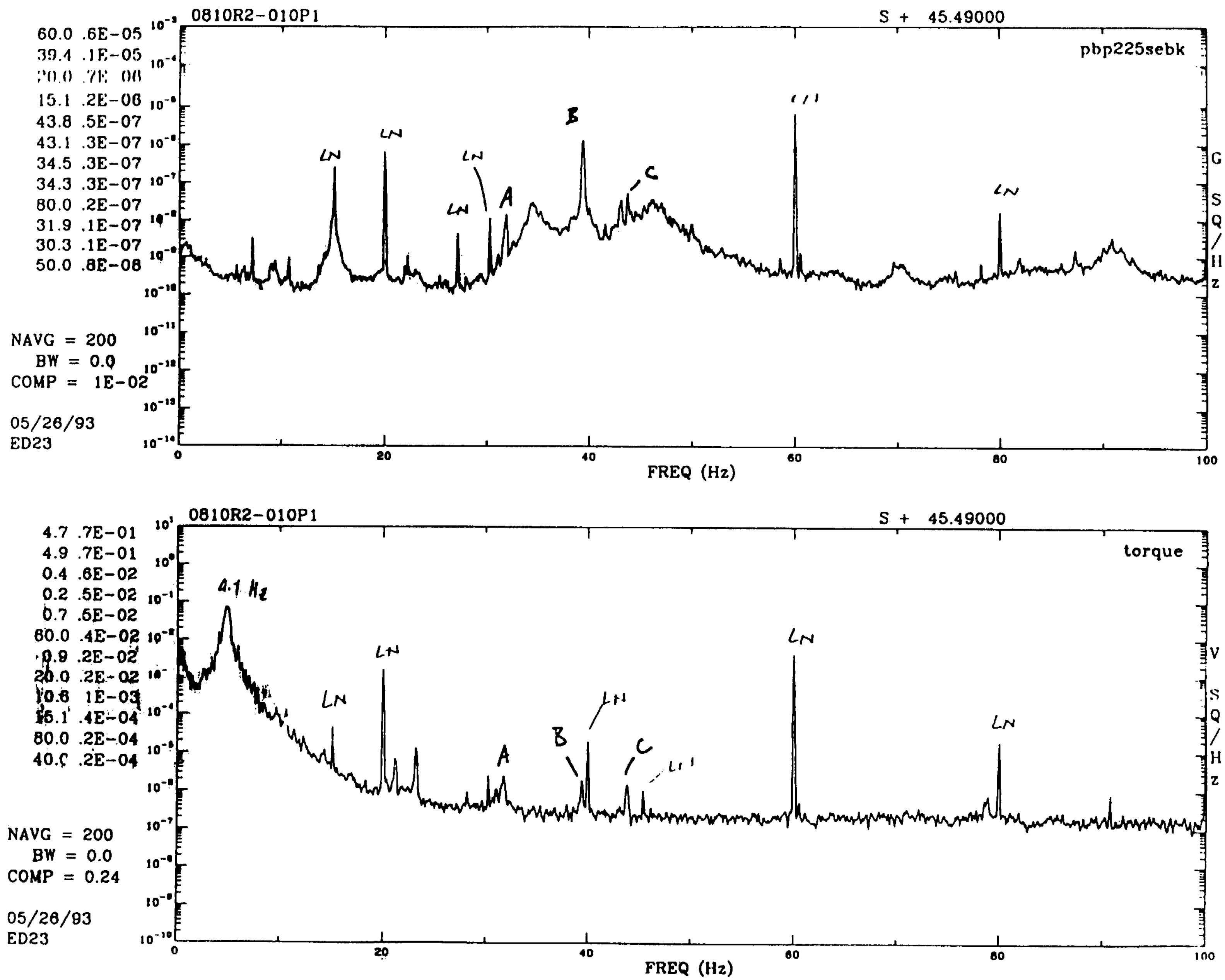

0810R2-010P1
S + 45.49000
pbp225sebk
60.0 .6E-05
39.4 .1E-05
15.1 .2E-06
43.8 .5E-07
43.1 .3E-07
34.5 .3E-07
34.3 .3E-07
80.0 .2E-07
31.9 .1E-07
30.3 .1E-07
50.0 .8E-08
NAVG = 200
BW = 0.0
COMP = 1E-02
05/26/93
ED23
LN
A
B
C
FREQ (Hz)
GS Q/Hz
0810R2-010P1
S + 45.49000
torque
4.7 .7E-01
4.9 .7E-01
0.4 .6E-02
0.2 .5E-02
0.7 .5E-02
60.0 .4E-02
80.0 .2E-04
NAVG = 200
BW = 0.0
COMP = 0.24
05/26/93
ED23
4.1 Hz
LN
A
B
C
FREQ (Hz)
VS Q/Hz

Figure 10A

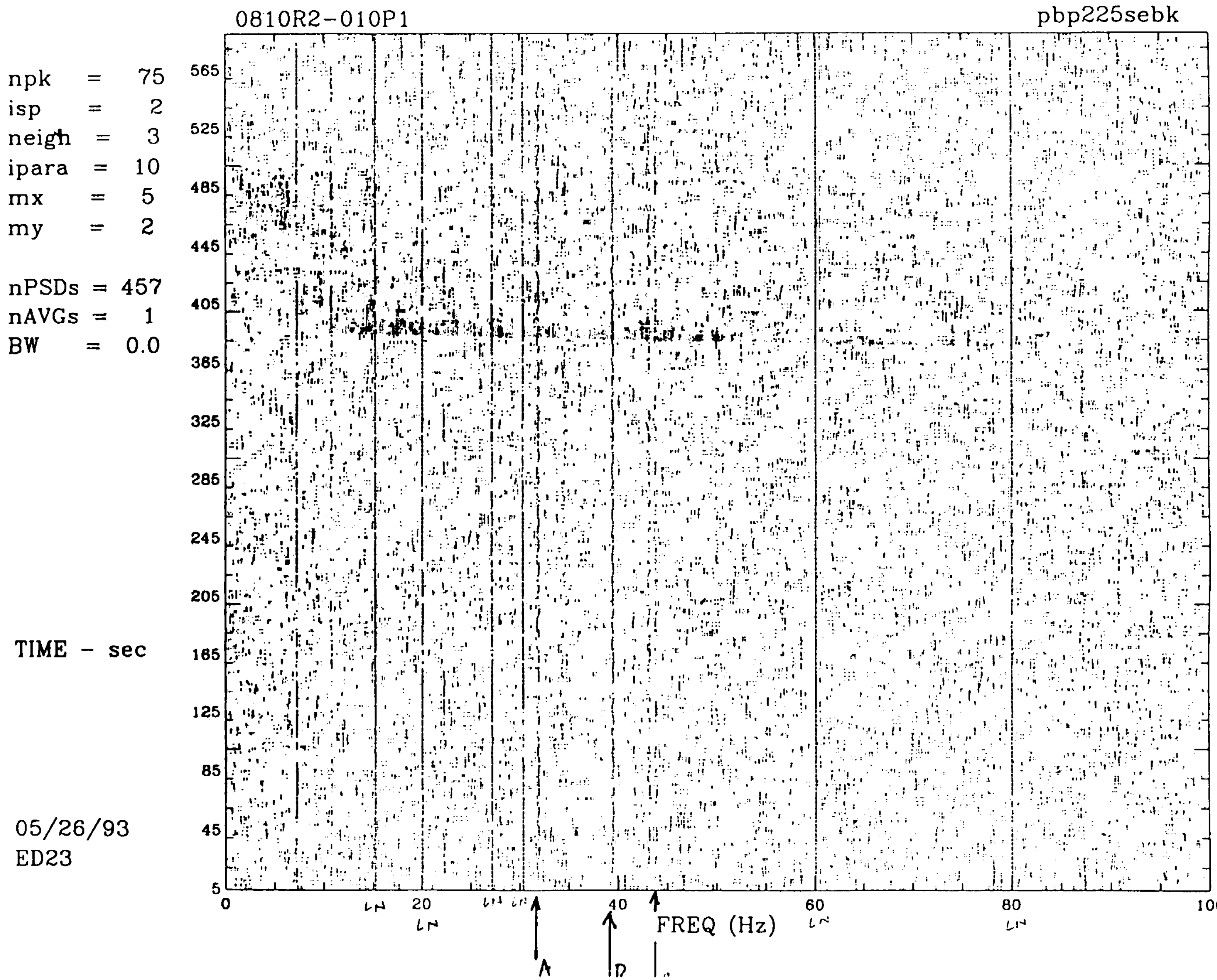

0810R2-010P1
pbp225sebk
npk = 75
isp = 2
neigh = 3
ipara = 10
mx = 5
my = 2
nPSDs = 457
nAVGs = 1
BW = 0.0
TIME - sec
05/26/93
ED23
565
525
485
445
405
365
325
285
245
205
165
125
85
45
5
0
20
40
60
80
100
FREQ (Hz)

Figure 11A

APPROVAL

SPACE SHUTTLE MAIN ENGINE TURBOPUMP BEARING ASSESSMENT PROGRAM

By B. Spiegel Breithaupt

The information in this report has been reviewed for technical content. Review of any information concerning Department of Defense or nuclear energy activities or programs has been made by the MSFC Security Classification Officer. This report, in its entirety, has been determined to be unclassified.

for C. Verschoore

J.P. McCARTY
Director, Propulsion Laboratory

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