NASA-CR-159586
19790017221

NASA CR-159586

T700 Power Turbine Rotor Multiplane/ Multispeed Balancing Demonstration

prepared by
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prepared for
National Aeronautics and Space Administration

NASA Lewis Research Center
Contract NAS 3-18520

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
NASA CR-159586		
4. Title and Subtitle T700 Power Turbine Rotor Multiplane/Multispeed Balancing Demonstration		5. Report Date February, 1979 6. Performing Organization Code
7. Author(s) G. Burgess and R. Rio		8. Performing Organization Report No. MTI 79TR29 10. Work Unit No.
9. Performing Organization Name and Address Mechanical Technology Incorporated 968 Albany-Shaker Road Latham, New York 12110		11. Contract or Grant No. NAS3-18520 13. Type of Report and Period Covered Contractor Report
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D.C. 20546		14. Sponsoring Agency Code

15. Supplementary Notes

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16. Abstract

Experimental tests were conducted to demonstrate the ability of influence coefficient based multiplane/multispeed balancing to control rotor vibration through bending criticals. Rotor dynamic analyses were conducted of the General Electric T700 power turbine rotor. The information was used to generate expected rotor behavior for optimal considerations in designing a balance rig and a balance technique.

The rotor was successfully balanced through 9500 rpm. Uncontrollable coupling behavior prevented observations through the 16,000 rpm service speed.

The balance technique has been shown to be practical and with additional refinement will meet production standards.

17. Key Words (Suggested by Author(s))	18. Distribution Statement
Balancing Flexible rotor balancing	Unclassified - Unlimited STAR Category 37

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	45	

* For sale by the National Technical Information Service, Springfield, Virginia 22161

NASA-C-168 (Rev. 10-75)

N79-25392 #

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1.0 SUMMARY

Rotordynamic analyses were used to guide in a multiplane/multispeed balancing experiment using actual engine hardware where practical. A rotor was selected which operates on squeeze-film dampers, and two bending criticals were traversed prior to reaching the rotor's service speed.

Single- and double-level modeling of the T700 power turbine assembly was conducted. A test rig was fabricated to support the forward power turbine bearing assembly and power turbine assembly. Displacement sensors were used to determine shaft orbits in three planes. Temporary weight addition was achieved using precision balance collars which clamped onto the shaft. Drive power was supplied by an in-line variable speed driver.

Analysis predicted lateral bending criticals at 8500 rpm and 14,000 rpm. An unexpected spline shift at 4200 rpm was responsible for shaft bending. This was controlled by a balance correction applied near the spline end of the rotor. The first critical speed occurred experimentally at 8200 rpm. Balance corrections in two planes (mid-span and near the first stage disk) significantly reduced shaft bending. Verification of behavior and balancing at the second critical could not be carried out because of limited rotational speed.

It was concluded that mathematical modeling accurately predicted flexible rotor behavior at the first bending critical speed; three-plane balancing is adequate to control rotor vibration; and the successful use of an engine module shows promise as a production balancing technique.

2.0 INTRODUCTION

2.1 Background

Developments in multiplane flexible rotor balancing technology(1,2,3)* have indicated that the influence coefficient balancing method has potential as a practical, cost-effective procedure during gas turbine manufacture and overhaul. During the past several years, this balancing method has been effectively used with several types of practical rotating machinery. The method has proven capable of overcoming axially-distributed unbalance caused by machining or assembly. The result is rotor operation which through bending critical speeds has lower vibration levels than would be possible using other balancing techniques.

Because of cost and performance benefits gained in other applications, a preliminary analytical investigation was conducted to study the requirements for applying the influence coefficient balancing method to the T700 engine power turbine rotor. This investigation, conducted under NASA contract NAS3-18520, was also performed because it would act as a complement to a current U.S. Army project - development of the General Electric T700 Gas Turbine Engine for new helicopters. The investigation showed the feasibility of performing assembly balancing of a power turbine rotor made up of a shaft which has not been previously balanced and a turbine assembly which has been only coarsely balanced at low speed. In addition, acceptable results were analytically demonstrated for rotors with increased concentricity tolerances; such parts were assumed to be burdened with five times the unbalance originally estimated.

After repeated rotor balancing simulations, it was concluded that assembly balancing using the influence coefficient technique was cost-effective and practical for the T700 Power Turbine Rotor. The successful nature of the analytical balancing simulation results allowed MTI to demonstrate the actual balancing process with a standard General Electric T700 engine power turbine assembly.

2.2 Program Demonstration

In the test program, demonstration balancing of a T700 engine power turbine assembly was conducted for the purpose of evaluating the multiplane balancing capability using production engine hardware.

*Numbers in parentheses indicate references at end of report.

The balancing operation was performed on a General Electric supplied rotor assembly. A prototype balancing fixture was designed and fabricated during the program. All sensors and instrumentation, and a computerized balancing system were provided by MTI. The drive train and general facilities are government property, designed and built by MTI under NASA Contract NAS3-16824 [3]

During the balancing demonstration, amplitudes and phase angles were recorded as functions of speed from all sensors in order to document before-and-after conditions. Finally, recommendations were made for production optimization of the fixture and process, including procedures and hardware/fixture configurations.

2.3 Acknowledgements

The investigation presented in this report was performed by the Mechanical Dynamics Department of Mechanical Technology Incorporated under Contract NAS3-18520 for the National Aeronautics and Space Administration, Lewis Research Center, Cleveland, Ohio. Dr. David Fleming was the NASA Project Manager and Mr. Richard Rio was the MTI Project Manager. The analytical studies used as reference in this experiment were performed by Mr. Juergen Tessarzik and Mrs. F. Gillham. The balancing demonstration was performed by Mr. George Burgess and Mr. Joseph Davis with assistance from Mr. Michael Talmadge.

3.0 ROTORDYNAMICS ANALYSIS

3.1 Description of the T700 Turboshaft Engine

The T700 helicopter engine, which was the focal point of the MTI analytical and experimental studies, is an excellent example of flexible rotor shafting. Developed by General Electric for the Army Utility Tactical Transport Aircraft System (UTTAS) and the Advanced Attack Helicopter (AAH), the T700 uses several high-technology elements to produce 1.1 MW from a 180 kg engine. Compared to engines used in similar applications, the T700 engine excels in the following technological advancements:

- 25 percent lower specific fuel consumption (SFC)
- 100 percent more power through dual engines at only 30 percent weight increase
- Threefold increase in the engine mean-time before overhaul (MTBO)

The five axial and one centrifugal stage compressor is driven by two air-cooled gas generator turbine stages to a speed of 44,740 rpm. The low-speed power turbine is two uncooled stages operating at a constant 20,000 rpm. (See Figure 1 for engine cross section.)

Design features which assist in achieving these technological advancements are the extremely slender power turbine shaft and the squeeze-film dampers. This power turbine assembly was the focal point of our analytical and experimental studies.

3.2 T700 Rotordynamics Analysis

Highlights of MTI's rotordynamics analysis of the T700 power turbine are presented in this section. The analytical results were used to select balance speeds, location of balance planes and sensor locations for an optimum balance operation. The information was originally presented in MTI Technical Report 75TR36.

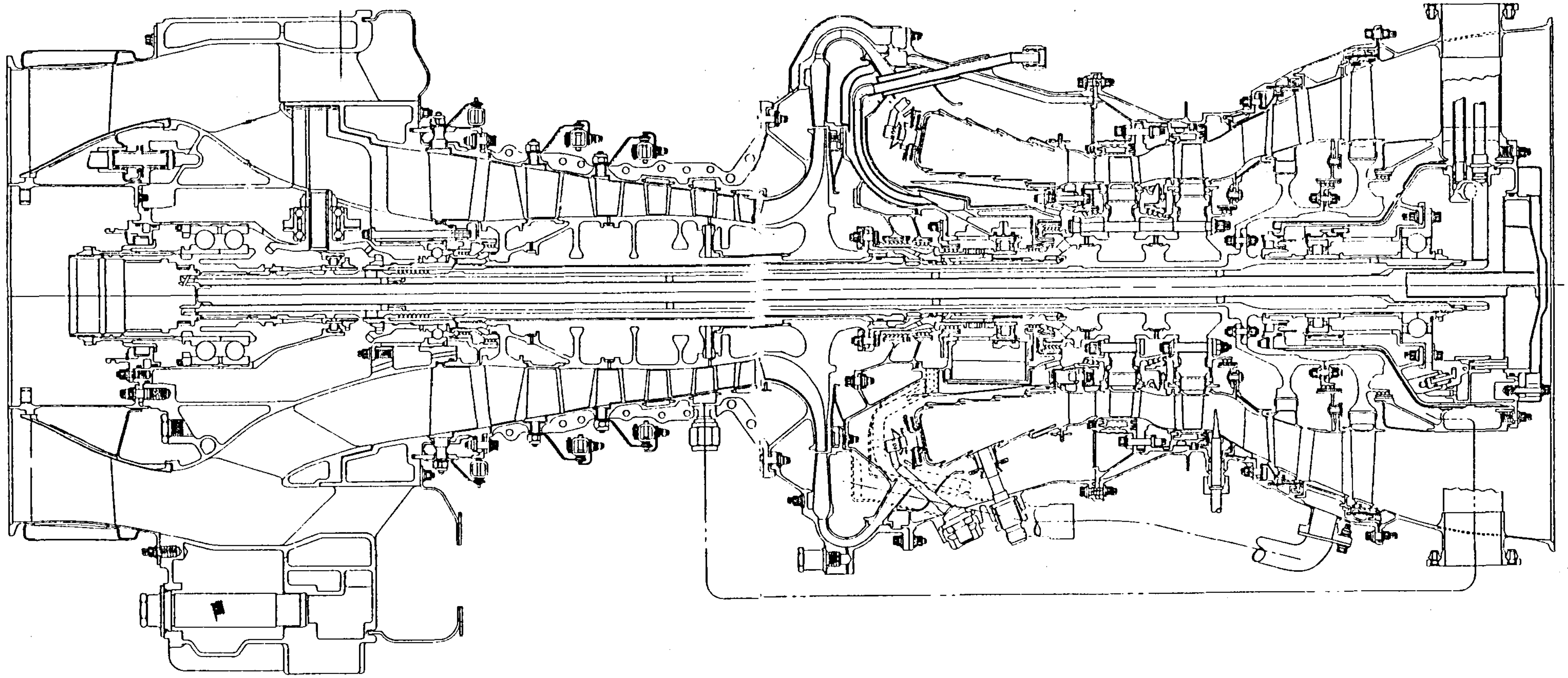

Figure 1 The General Electric T700 Turboshaft Engine Cross Section

3.2.1 Rotordynamic Model

For the T700 rotordynamics analysis, the actual rotor-bearing system must be represented by an analytical model. The complexity of the model often grows as the number of components that make up a rotor increases. However, the most important step in devising a realistic model usually lies in obtaining correct analyses of assembly fits and connections (such as shrink fits, bolted joints, spline couplings and sliding fits). Rotor models are most accurate when realistic sets of elastic stiffness and damping values are used for all interconnected rotor elements. Obtaining these values often requires experimental evaluation of component properties. Thus modeling a rotor system like the T700 power turbine relies upon experience using parameters derived experimentally. Simplifying assumptions can then be made during the preparation of the mathematical model. Such experience gained by the General Electric Company was offered in support of this effort, and proved to be of considerable value to MTI.

For the T700 power turbine rotor-bearing system, several calculations of undamped critical speeds were made for the evaluation of possible choices in the development of the analytical rotor model. The final rotor system selected for the balancing optimization dynamic analysis consisted of single shaft with branch, as shown in Figure 2. The turbine stages in the power shaft rotor are represented as a flexible branch, connected to the main rotor at one end and free at the other end. The torque tube is represented as a number of lumped masses connected to the main rotor at those points where either contacts or connections exist between the two. Part of the output shaft is lumped as a mass on the power shaft at the opposite end from the turbine. This simplified, yet representative model of the power turbine rotor-bearing system avoids the complexity and cost associated with the use of a multi-shaft model in subsequent dynamic response calculations.

3.2.2 Critical Speed Analysis

For complex rotating systems, such as the T700 engine's power turbine rotor, the calculation of undamped critical speeds serves an important purpose. The critical speeds may be determined quickly and rather inexpensively for various

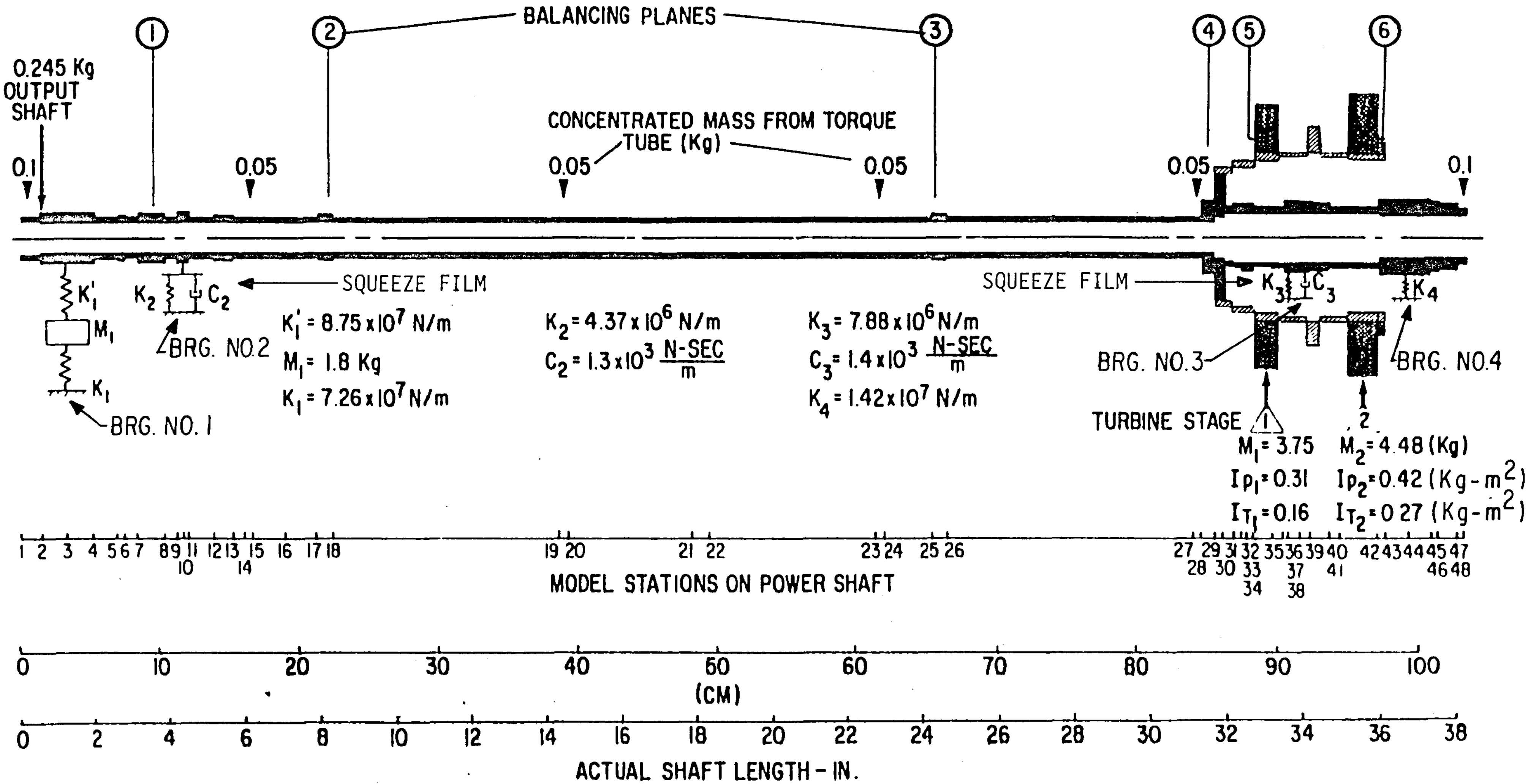

Figure 2 Analytical Model of the T700 Power Turbine Shaft for Critical Speed and Amplitude Response Calculations

bearing stiffnesses and for different rotor elastic and mass distribution characteristics. The simplicity of the process invites its use as a model-tuning tool for the evaluation of alternate analytical representations of the complex rotor system.

The second area of investigation for which undamped critical speed calculations were used was for the evaluation of the effects of variations in individual bearing stiffnesses upon the critical frequencies of the system. The two damper bearings, which have inherently low stiffness, may have considerable variations in stiffness, depending upon damper design and bearing support structure.

Once the rotor configuration and support details have been selected, they can be used to determine the nature of the rotor vibration modes. Comparison of these mode shapes for an early model (Figure 3) indicates that maximum rotor amplitudes occur slightly closer to the turbine end of the rotor (at about 0.5 m) at the first critical speed, and shift forward on the rotor (at 0.36 m) as the rotor approaches the second critical speed. From the unbalance response analysis, the shaft amplitudes at the first mode are much below those at the second mode because the first mode balance calculation was made at a speed further from the first critical frequency than was the case for the second critical. (See Figures 6 and 7). The general shape of both rotor modes was found to be remarkably independent of several features of the rotor model, such as inclusion of the torque tube, branching of the rotor for the power turbine, representation of the output shaft as a spring-supported pedestal.

The same features do have a definite effect upon the speed at which the mode occurs. The improved final model with number-two bearing stiffness of 4.4×10^6 N/m (Figure 4) resulted in critical speeds of 8,500, 13,740, and 35,700 rpm for the first three critical speeds. One early model's first three critical speeds were at 6,538, 15,377 and 35,371 rpm. Therefore, it is possible to provide for practical rotor balancing even though the balance rig stiffness is not an exact duplicate of the engine.

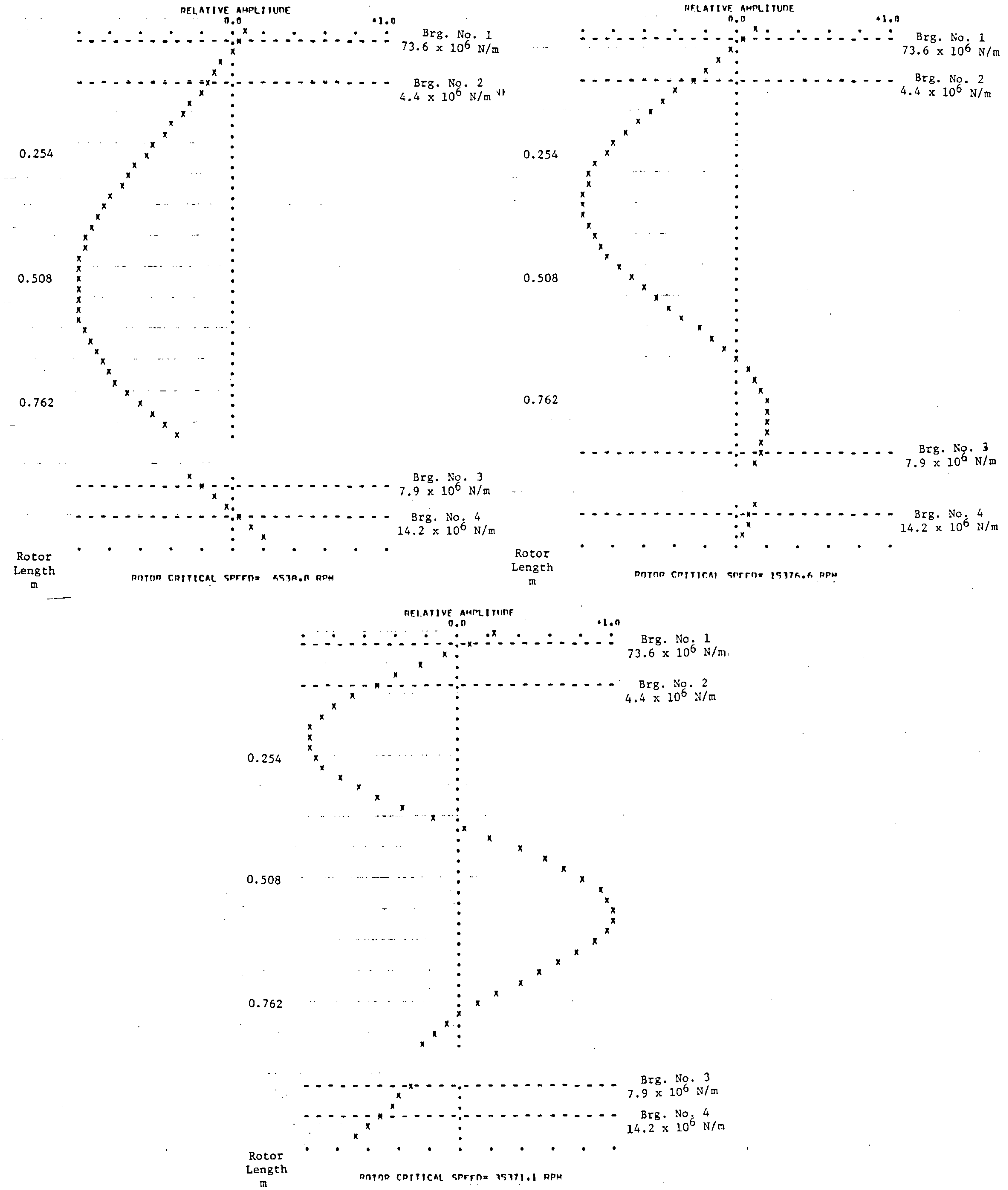

Figure 3 Undamped Critical Speed Mode Shapes of T700 Power Turbine Shaft Without Torque Tube (Turbine Wheels Concentrated on Main Shaft)

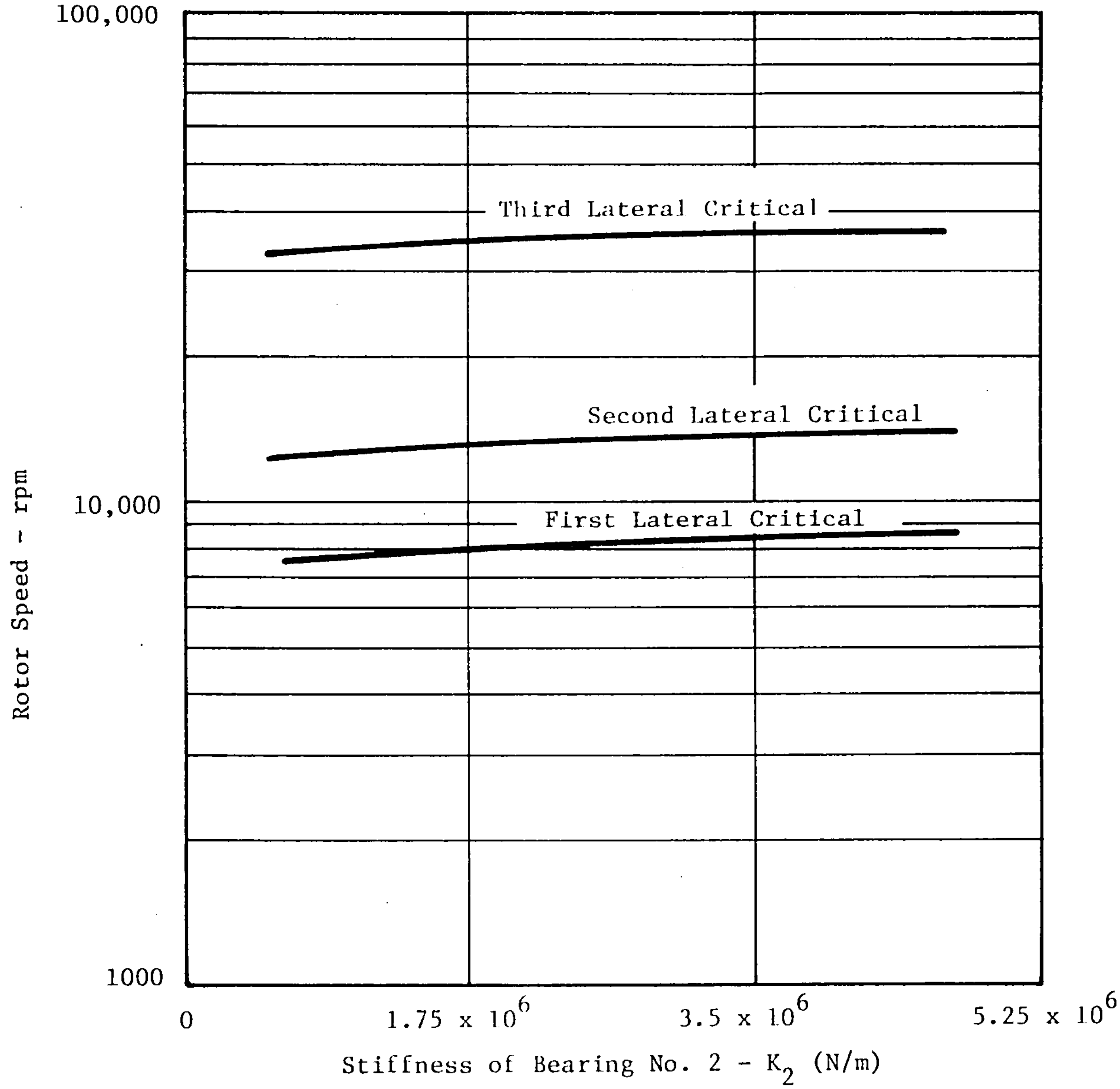

Figure 4 Undamped Critical Speeds of T700 Power Turbine Rotor as a Function of No. 2 Bearing Stiffness

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3.2.3 Rotor Unbalance Response Calculations

Rotor response calculations require two additional inputs beyond those supplied for the undamped critical speed analysis. These are bearing damping and unbalance distribution in the rotor.

The introduction of bearing damping into the rotordynamics calculations often results in amplitude maxima that are considerably lower than those predicted without damping, especially at resonant frequencies. Bearing damping may also completely suppress some system resonances, making it difficult to identify their presence unless undamped critical speed calculations are made. The amount of damping used in response calculations thus has a very significant effect on predicted amplitudes, and is a key element for an accurate assessment of the relative severity of the critical speeds.

Unbalance distributions used in rotordynamics calculations must, by necessity, be based upon estimates in the absence of precise distributed unbalance measurements. Using manufacturing tolerances, a worst possible case may be determined for any particular rotor. Additional unbalances which can be quite significant may be present in the actual rotor (e.g., unexpected material inhomogenities). Therefore, dimensional tolerances may only be used for first approximations to true rotor behavior. The modeling assumed that no balancing was done on individual rotor components.

Maximum response amplitudes calculated for the power turbine rotor are shown as the upper curve in Figure 5 . Calculations used the standard worst case unbalance distribution, with random distribution of unbalance along the shaft. Amplitude peaks are found at 8,700 and 13,875 rpm. These rotational speeds agree very well with the previously calculated first and second undamped critical speeds (see Figure 4) . The middle and lower curves in Figure 5 depict the predicted results of balancing. The general damped mode shapes of the rotor's elastic axis in the vicinity of the first two critical speeds are shown in Figures 6 and 7 for speeds of 7,000 and 13,875 rpm, respectively. The analysis assumed both measurement and balance planes coincide with illustrated balance planes.

The analytical model was used with the unbalance response computer program system to evaluate several arrangements of planes, speeds, and sensor locations

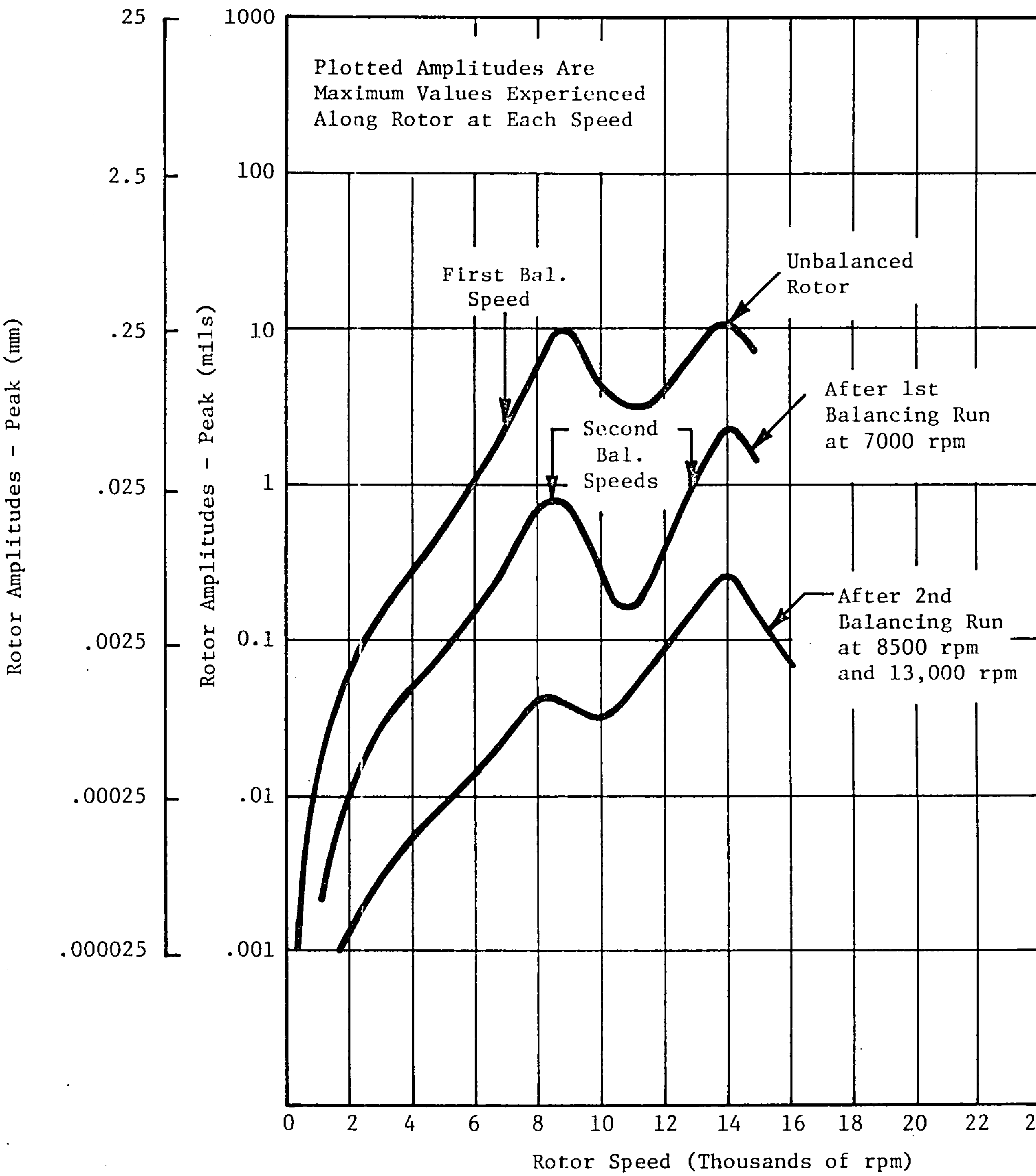

Figure 5 Calculated Maximum T700 Power Turbine Rotor Amplitudes with Initially Unbalanced Rotor and After Two Successive Balancing Runs (Standard Unbalance, Randomly Distributed)

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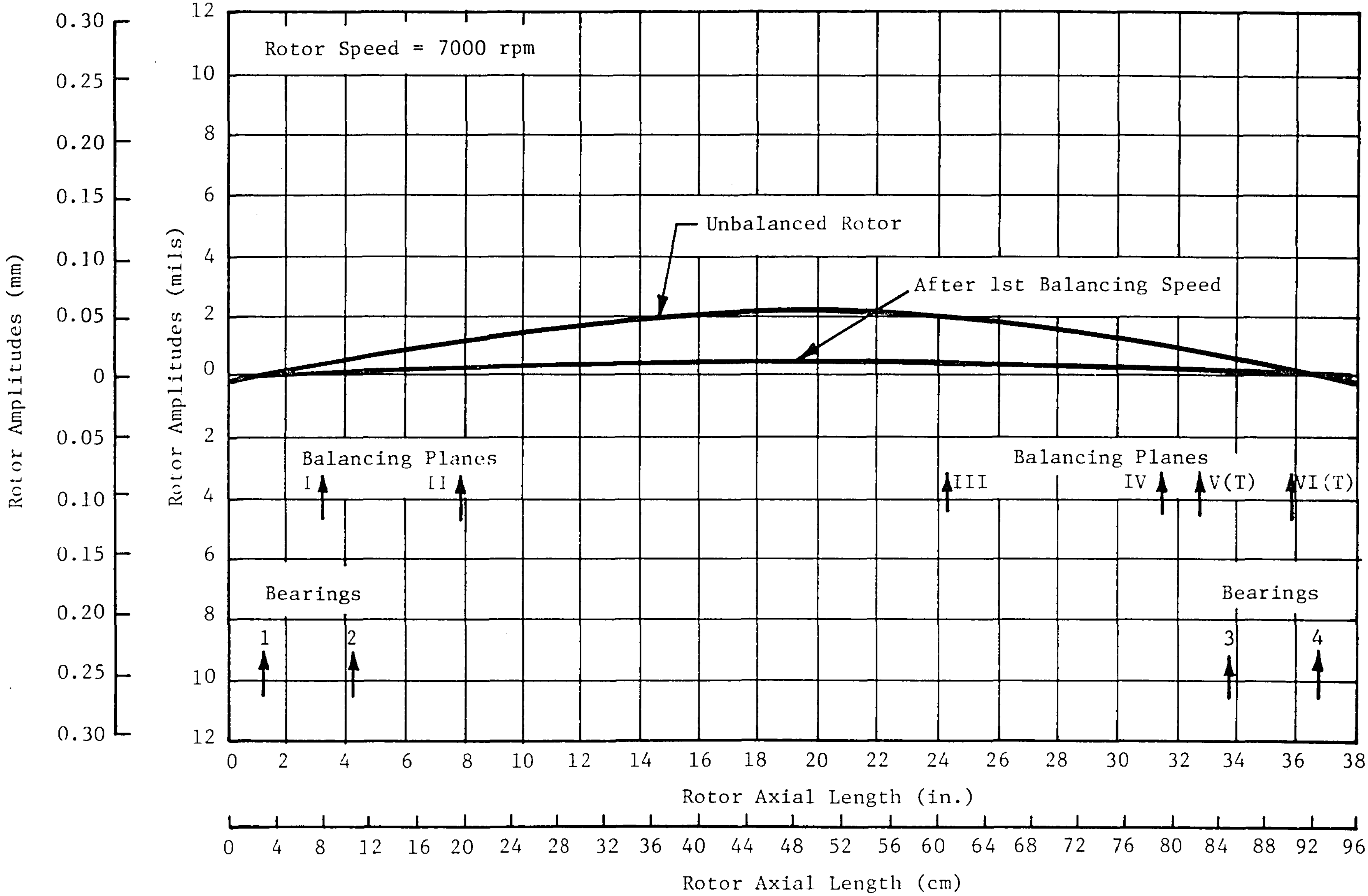

Figure 6 Calculated Damped Rotor Mode Shapes at 7000 rpm for Unbalanced T700 Power Turbine Rotor (Standard Unbalance) and After One Balancing Run

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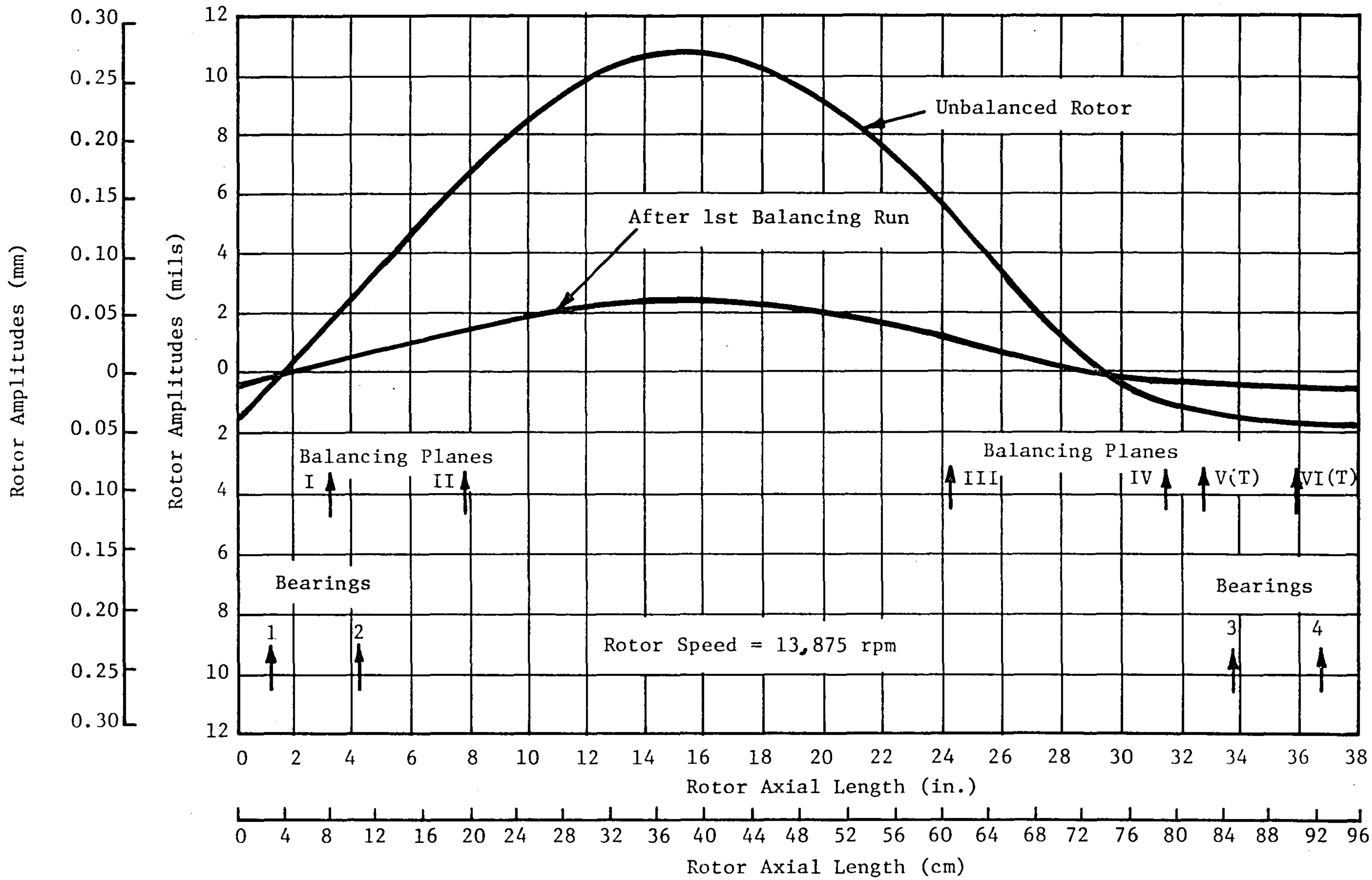

Figure 7 Calculated Damped Rotor Mode Shapes at 13,875 rpm for Unbalanced T700 Power Turbine Rotor (Standard Unbalance) and After One Balancing Run

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for multiplane-multispeed balancing. It was concluded that the rotor assembly, as presently manufactured, can be effectively balanced in two steps to achieve low vibration throughout the entire speed range, without any standard low-speed balancing of the shaft and shaft components. Moreover, it appears that the same result can be achieved with a shaft which has been manufactured to considerably less severe concentricity tolerances, such that it has initially five times the amount of unbalance assumed for a production-line shaft built to existing tolerances.

The rotor dynamics analysis described in this section provided adequate design requirements for a high-speed multiplane balancing rig and was effectively used to predict the required correction planes and balancing speeds.

The power turbine rotor was manufactured with six balance correction planes - four on the shaft and two on the turbine assembly (see Figure 2). An early prototype included an additional balance land at shaft midspan. Since the rotor mode shape indicates a large balance sensitivity at midspan, the existing balance planes 2, 4, and 6 are adequate for high-speed balance. Adding a mid-span plane might lower possible vibration levels or serve as a replacement for planes 2 and 3.

Vibration sensors may be located independently of the balancing planes. Sensors at other than nodal points can be used to improve vibration at that location. Given the mode shape and access constraints, noncontacting displacement sensors adjacent to balance planes 1, 2, 3, 4, and 6 are used to provide bearing load and rotor bending information. An accelerometer placed on the aft bearing module may also be useful. Balance speed selection was based on close proximity to the critical speed as well as safe vibration amplitude. The maximum vibration is designed to not exceed clearances of 0.014 mm (0.55 mils) for the forward, and 0.018 mm (0.70 mils) for the aft damper bearings (numbers 2 and 5, respectively). Assuming a randomly distributed unbalance, a balance speed of 7000 rpm would be adequate for the first mode and 13,000 rpm for the second mode.

4.0 BALANCING METHODOLOGY AND HARDWARE

This section describes the steps taken to make the transition from balancing theory to practice. Details of power turbine balance requirements, their execution in a test rig, monitoring instrumentation, and balance weight addition are discussed.

4.1 Current Balancing Scheme - Low Speed

The current power turbine balancing procedures comprise a series of subassembly balances and a final assembled rotor low-speed balance at 1500-2000 rpm. As illustrated in Figure 8, this requires a 6-step operation. Each blade is moment weighed and installed in the disk. The assembled disk is then low-speed balanced. Two such disks are married and balanced as a unit at low speed. A previously balanced power turbine shaft is mated to the torque tube and the resulting shaft assembly is low-speed balanced. Correction weight orientation and magnitude are determined in two planes using state-of-the-art practices. Since the shaft is recognized as being flexible, the balance weight requirement in each end plane is divided up and a specified portion moved toward midspan balance planes. The power turbine disk and shaft are then mated. The assembly is not balanced as a unit requiring a very precise locating procedure or potential assembly imbalances.

4.2 Proposed Balancing Scheme - Multiplane/Multispeed

The use of multiplane assembly balancing offers a streamlined balance operation as well as a potentially lower vibration at all engine speeds. Figure 9 illustrates that four of the present procedures are eliminated and replaced by a single high-speed balance of the assembled module. Since the module does not require any bolting or unbolting after balance, it leaves the balance machine as a ready-to-run item.

4.3 Modularity of Engine Components

The proposed balancing scheme emphasizes the modular nature of the T700 engine design. Significant engine maintenance savings are anticipated as a result of having four bolt-together modules:

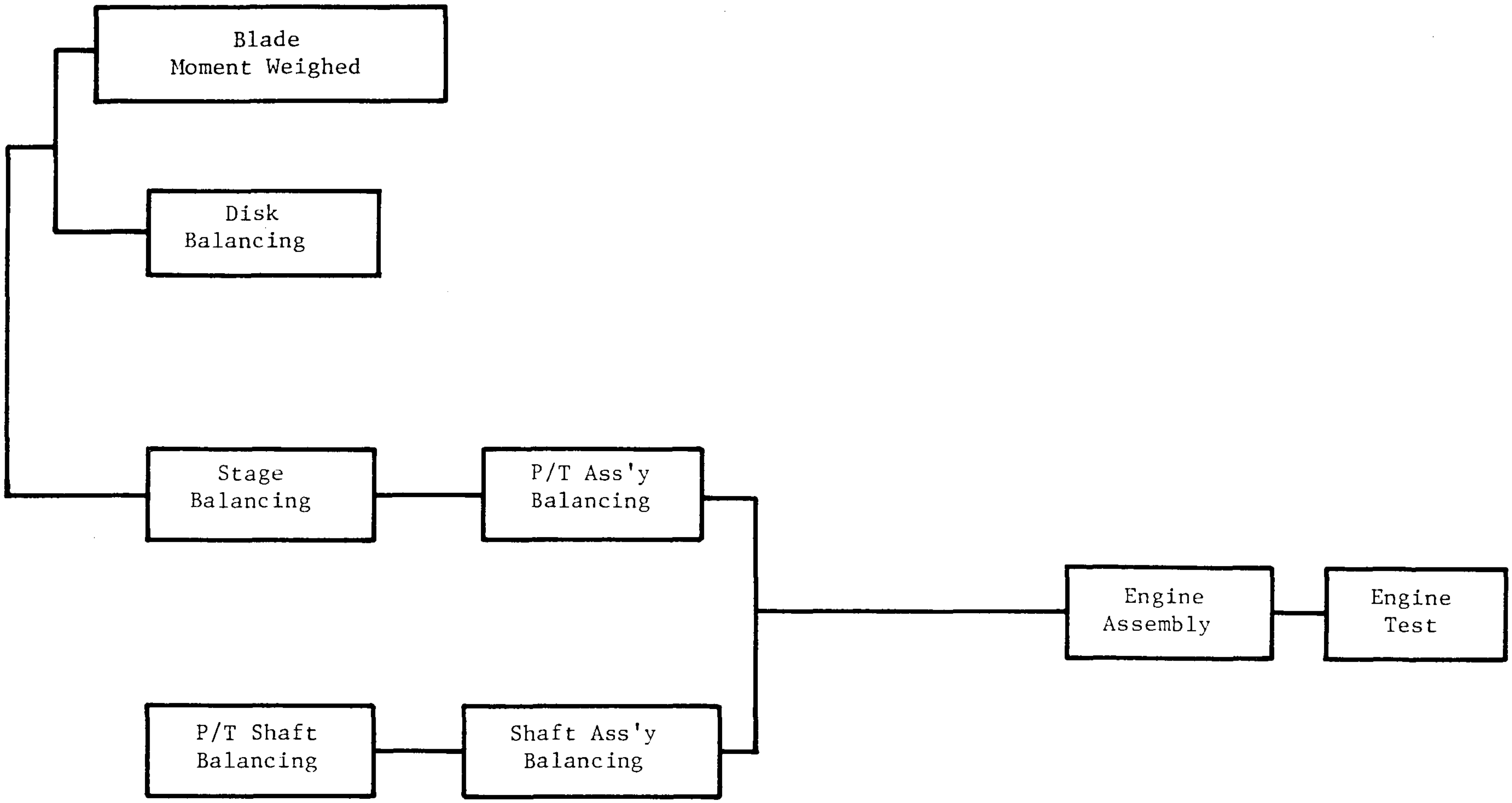

Figure 8 Conventional Low-Speed Balancing Procedure

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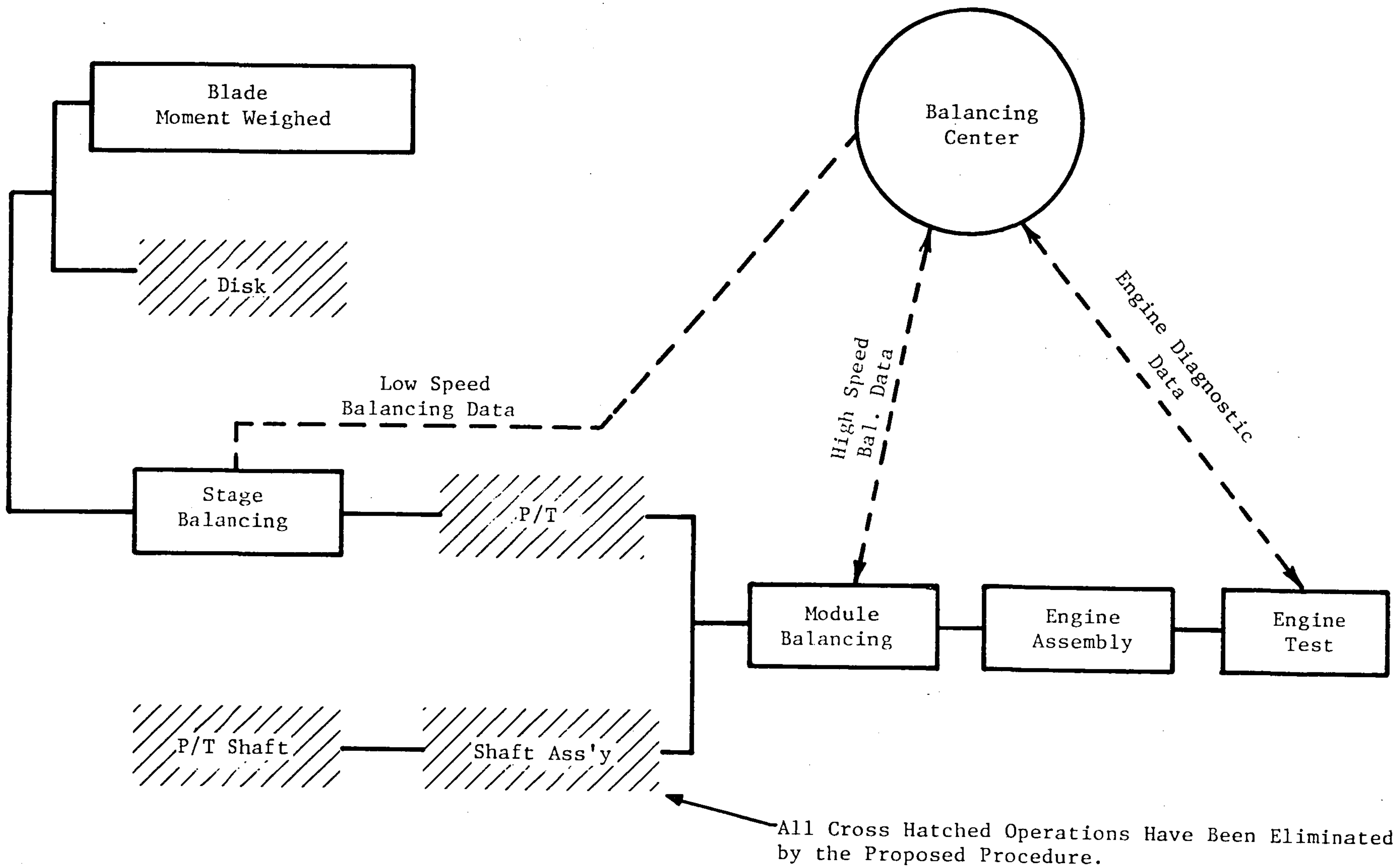

Figure 9 Proposed Computerized Balancing Procedure

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- accessory module
- cold section/compressor module
- hot section/turbine module
- power turbine module

4.4 Balancing Stand

The engine cross section (see Figure 1) shows the power turbine module assembled in the engine. For the balancing operation the power turbine and forward bearing assembly are installed in special design supports which duplicate engine condition (see Figure 10). Therefore, the bearing and spline characteristics are included as an integral part of the balance process - a function which cannot be accomplished on low-speed balance machines.

Drive power was supplied in an in-line configuration which uses the spline teeth to transmit rotation; the arrangement is similar to engine rotation characteristics. Hardware consisted of a variable speed motor, magnetic clutch, speed increaser, and flexible coupling. System rotation is counterclockwise as viewed from aft facing forward; this results in a backward turbine spin relative to the engine operation, but does apply spline tooth contact as would be experienced in the engine. (The drive system is capable of applying torque, but this feature was not used for the T700.)

4.5 Instrumentation

The goal of the balancing operation was to control rotor bending as well as bearing loading. An accelerometer placed on the power turbine casing and another placed on the forward support housing were to identify bearing vibration. However, the primary measurements were from Bently Nevada displacement probes placed adjacent to the existing balancing lands on the power turbine shaft. In addition, Bently Nevada probes were placed to monitor movement of the forward bearing housing.

Signals from the instrumentation were fed to oscilloscopes for visual display, to an X-Y plotter for record purposes, and to an MTI Command™ Balancing System for calculation of balance weights.

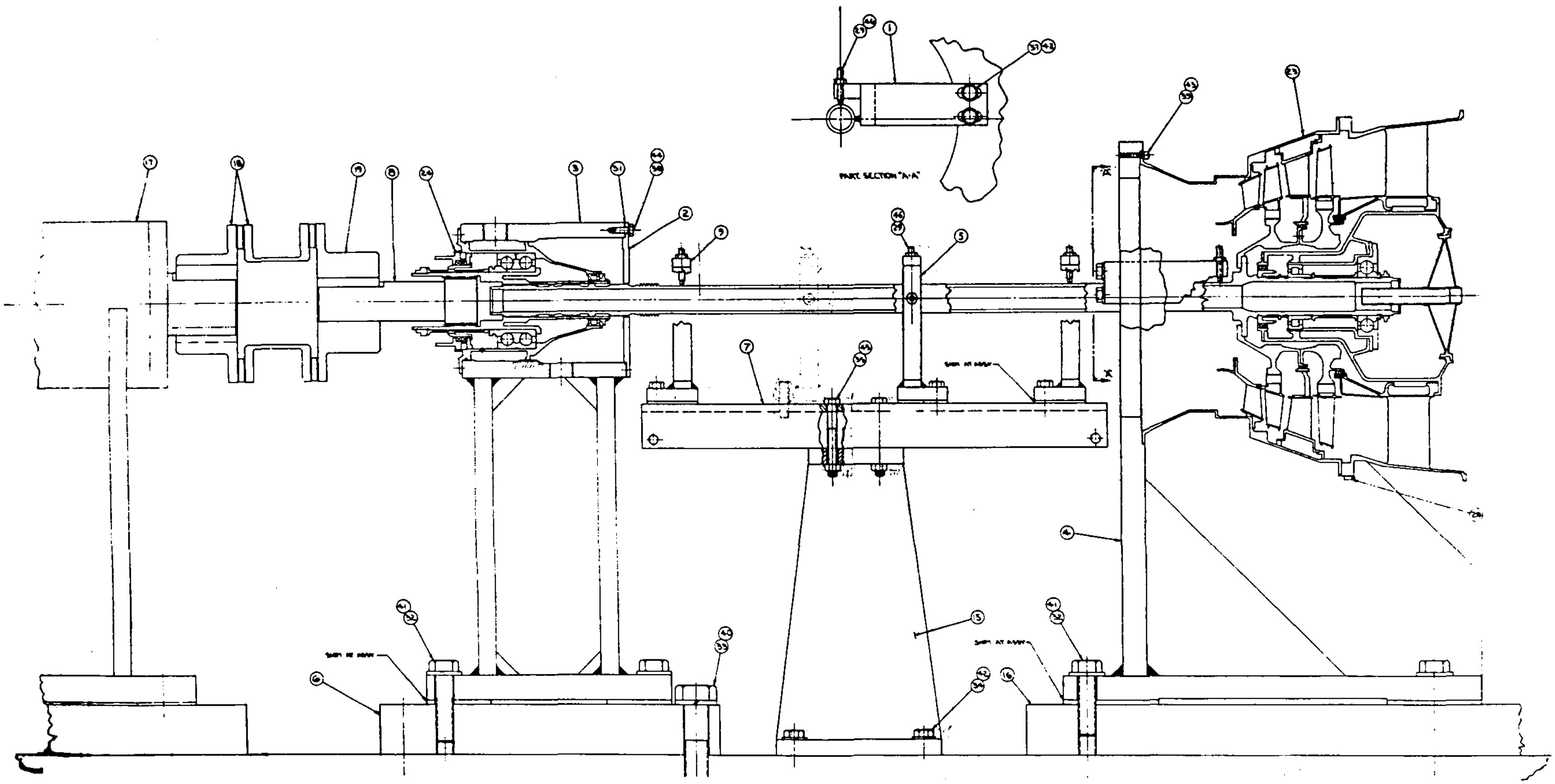


Figure 10 Prototype High-Speed Balancing Test Rig

Additional condition monitoring instrumentation included:

- RPM/ZERO PHASE indication from a Spectral Dynamics fiber optic system.
- AIR TEMPERATURE thermocouple sensing the vibration probe environment near the third turbine stage.
- PRESSURE & TEMPERATURE of synthetic lubricant.
- AIR PRESSURE to seals.
- SAFETY & OPERATIONAL MONITORS of the MTI drive system and its auxiliaries.

4.6 Balance Collars

The version of the T700 power turbine available for test had had its balance lands removed prior to arrival at MTI. Wrapping wire around turbine blades and applying heavy tape on shafting for balancing can be used only with limited success. Therefore an alternate means of balancing weight change was required.

The method developed by MTI in NASA contract NAS3-19408 was a precision balance collar, (see Figure 11). Predrilled and tapped holes in the collar permit weights to be added at a precise angular position. The design is capable of up to a 0.05 mm (2 mil) interference fit when installed on the shaft to assure positive positioning especially at high speed and high vibration.

In the previous NASA work, the balancing collars were designed to have negligible dynamic effect on a very sensitive, overhung test rotor. Results of the tests showed that the presence of a collar did not materially affect the vibration response of either a balanced or unbalanced rotor. The balance achieved using temporary weights in collars closely duplicated the balance obtained after permanent corrections were made and the collars removed. The success of the collar is traceable to a minimum size and weight, precision manufacture, and meticulous collar prebalance.

The T700 rotor diameter was larger than the evaluation rotor. During the balance collar upscaling operation, design changes were made to reduce manufacturing cost,

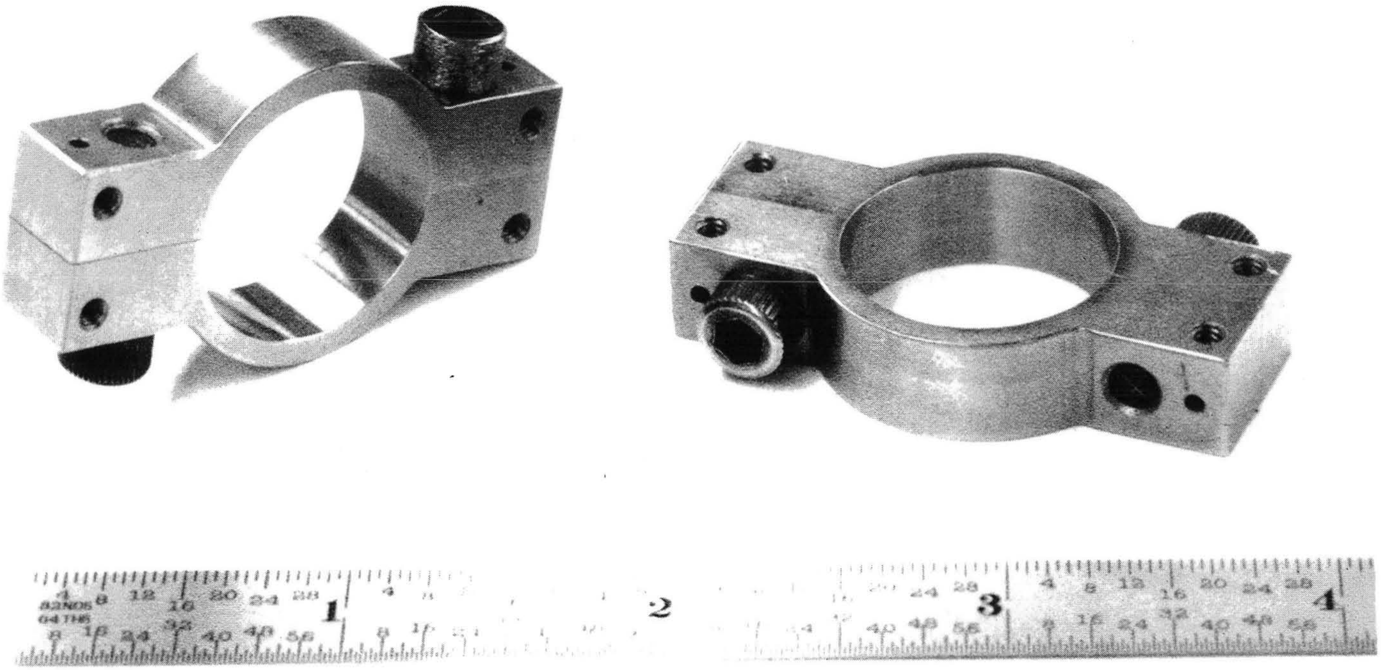


Figure 11 Prototype Precision Balance Collars Developed Under NAS3-19408

counteract stress raisers, and improve handling characteristics. The result (see Figure 12) met the objective without compromising dynamic properties.

The open design of the T700 spin-up hardware presented no restrictions to the collar installation (see Figure 13).

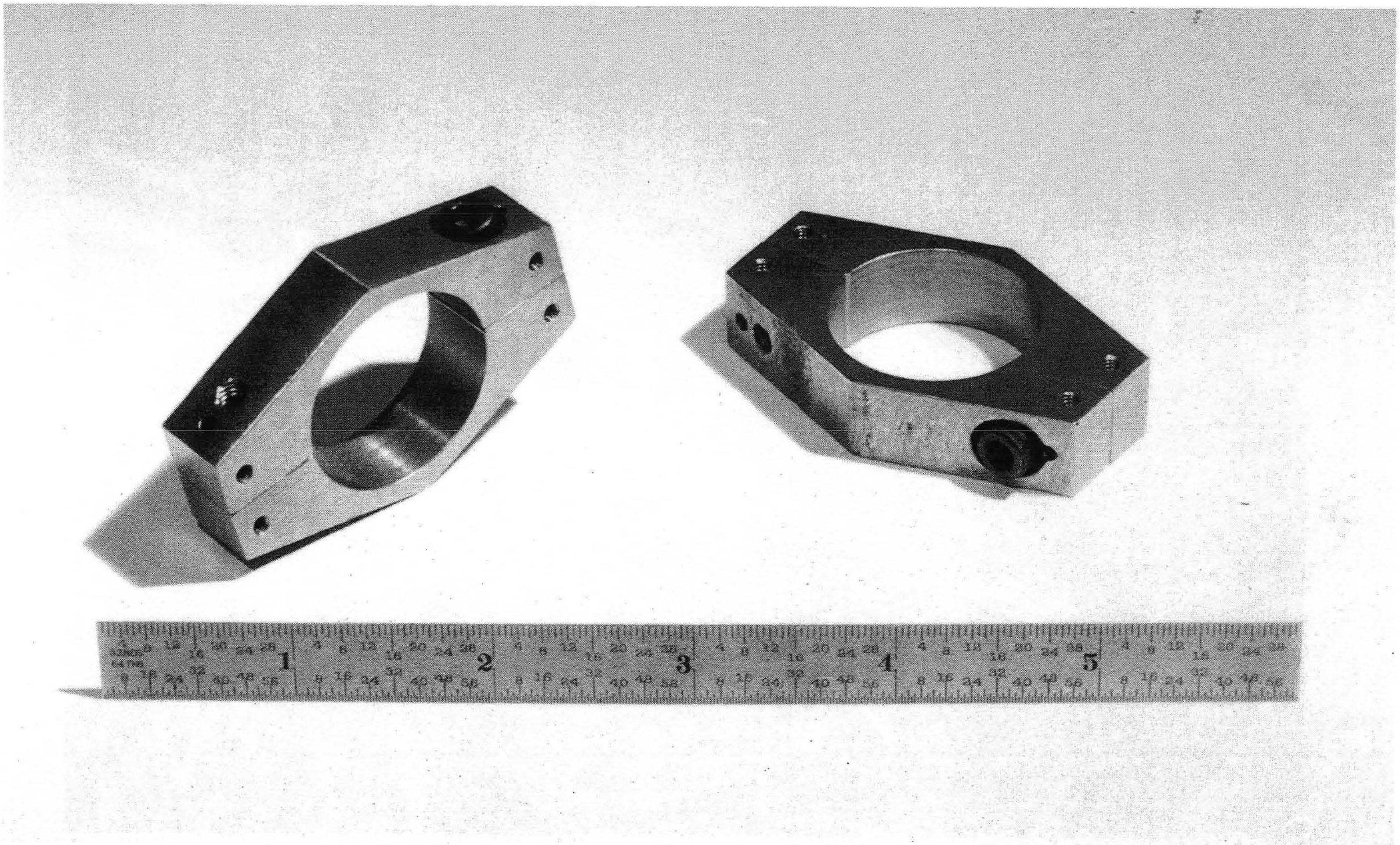


Figure 12 Precision Balance Collars for T700

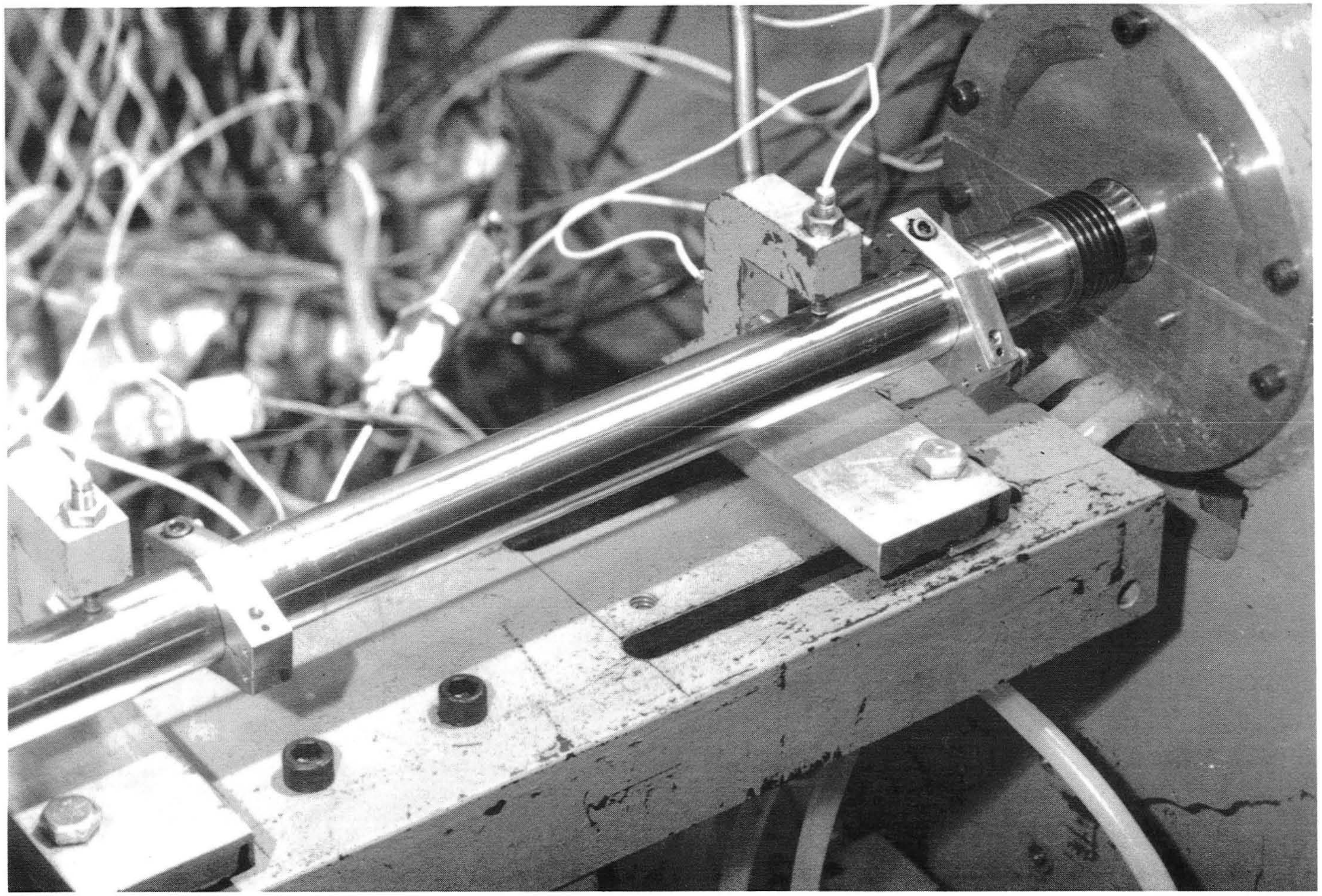


Figure 13 Location of Precision Balance Collars on T700 Power Turbine Shaft

5.0 HIGH-SPEED BALANCING DEMONSTRATION

Prior to the balancing operation, the hardware and electronic subsystems required a calibration-run and proof-test. The rotor was then run up in speed to obtain sensitivity data or influence coefficients. Section 5.0 describes the entire T700 power turbine rotordynamics experiments.

5.1 Test Rig Assembly

Because a General Electric T700 power turbine was going to be operated at high speed, MTI requested GE's advice concerning the proper operation of the module. In implementing their suggestions, several changes were made that had not been included in the original plan. One early modification was the use of a lubrication system capable of handling MIL-L-23699, a synthetic jet engine lubricant. The original lubrication system in the flexible shaft facility had provisions for only a petroleum-based oil supply. (The General Electric Aircraft Engine Group required that the T700 Power Turbine use the synthetic oil as a lubricant.) The existing in-house pumps and lube system rubber parts were either changed to VITON or BUNA N, or replaced with steel to avoid deterioration during operation.

One area of concern was the physical condition of the engine and related hardware. As received, there was noticeable spline wear (see Figure 14), fretting on the power turbine shaft (see Figure 15), fretting within the forward bearing assembly (see Figure 16) and damage to an electrical connector. Such evidence of hard wear generally suggests that bearings internal to the power turbine might also be heavily worn. All indications were that because of its "as received" condition this power turbine would be a demanding test of the high-speed balance procedure.

The rig side of the driveline also required special attention. There was very little design information about the GE female spline. As a result, the mating MTI stub shaft was manufactured to an inaccurate spline class specification. The resulting loose spline fit was compensated for by pressing a sleeve onto the stub shaft which fit line-to-line with the bearing assembly. The sleeve provided some centering capability not available from the loose spline under the low torque load.

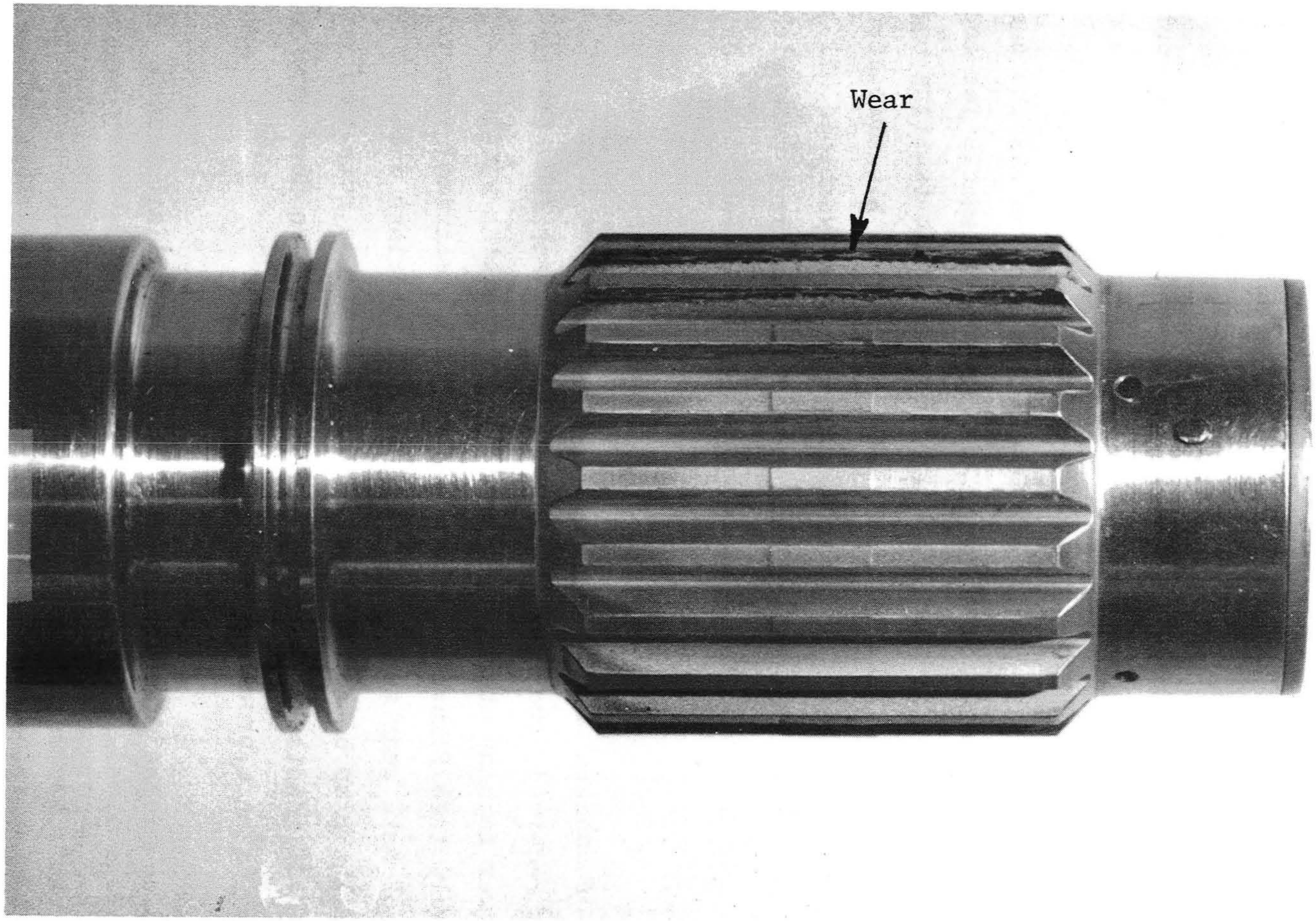


Figure 14 As-Received Condition of Forward Spline of the T700 Power Turbine Shaft

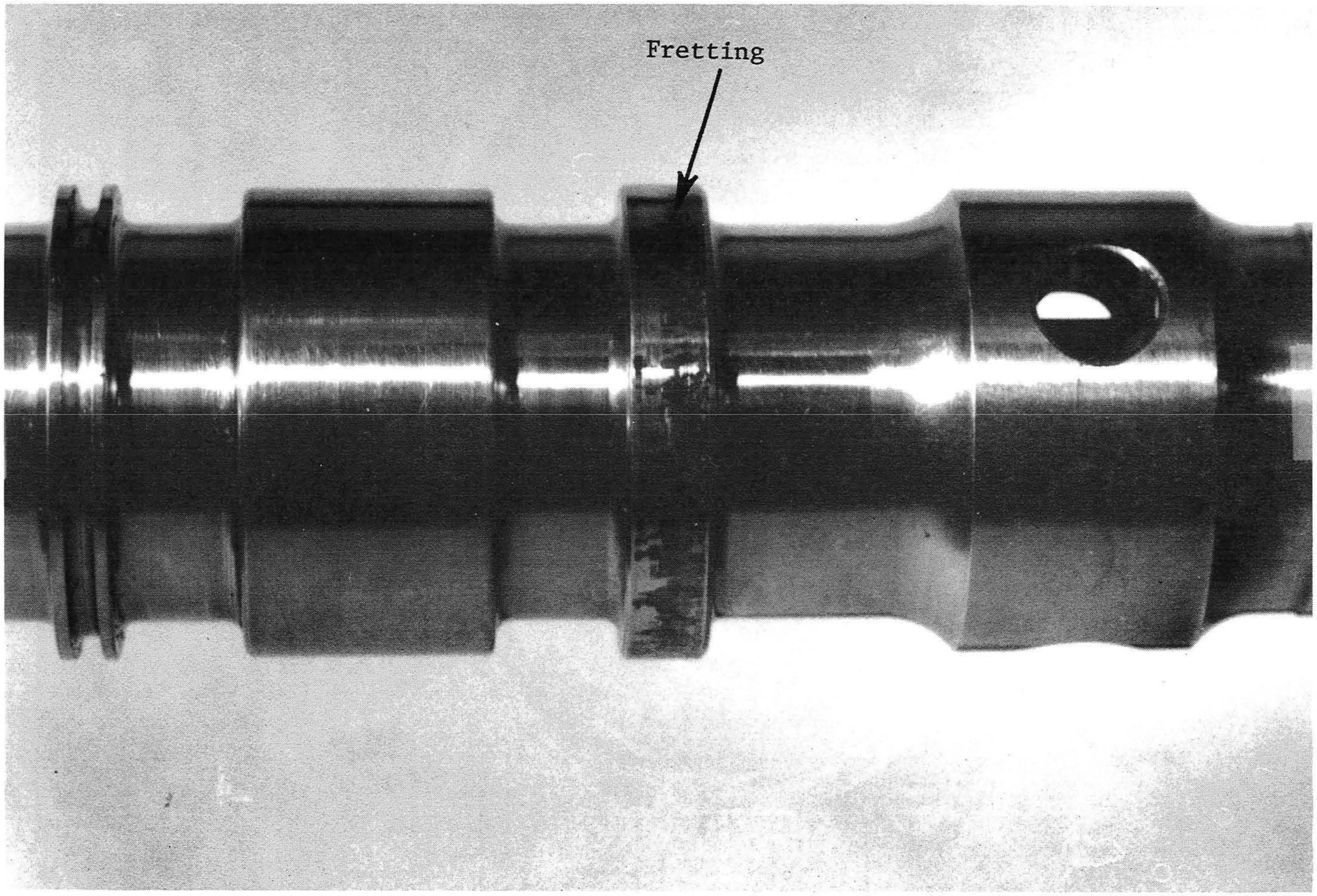


Figure 15 As-Received Condition of Forward Seal Surface of the T700 Power Turbine Shaft

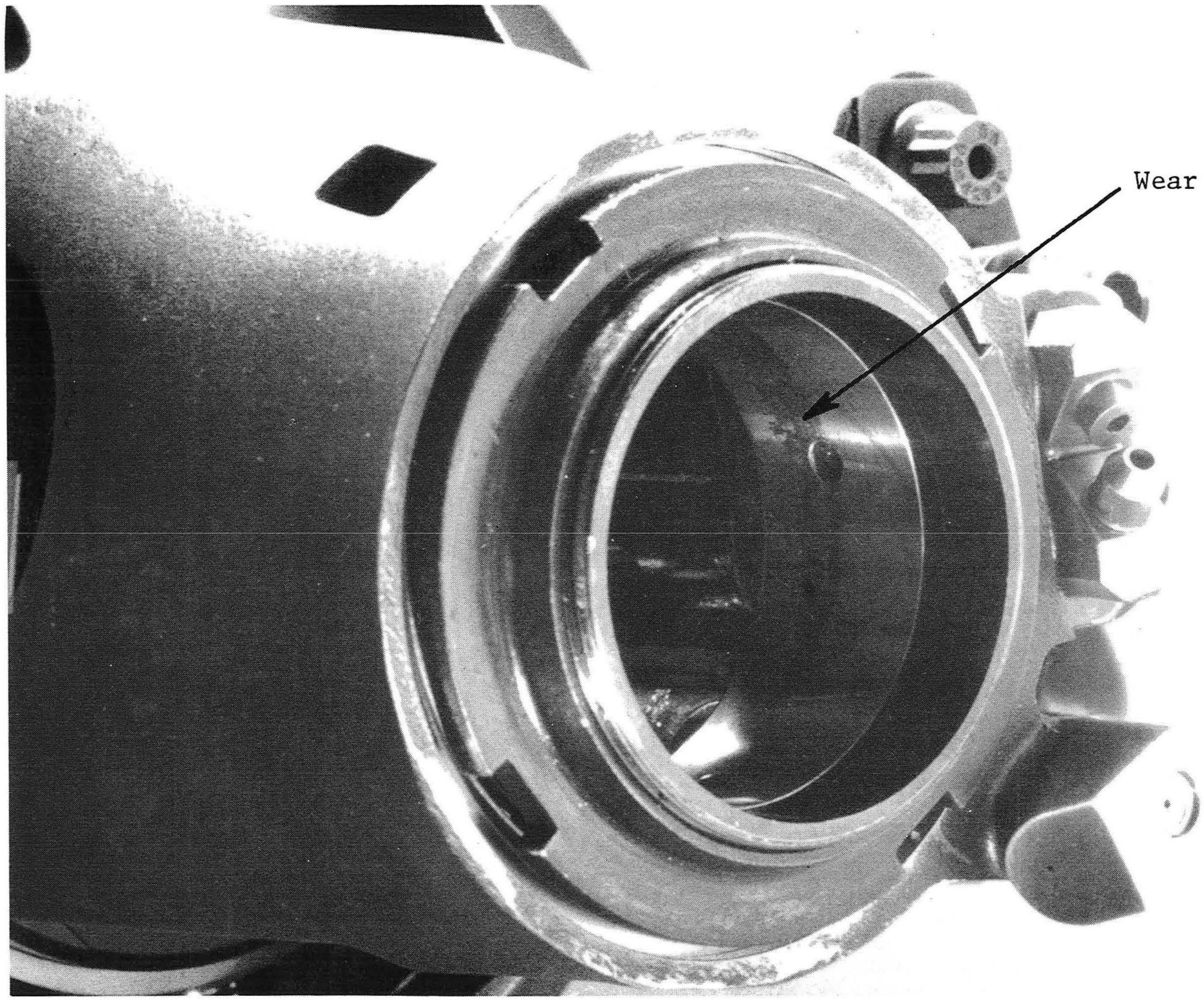


Figure 16 As-Received Condition of the Forward Bearing Assembly of the T700 Power Turbine Shaft

A time-consuming preparation of the standard Rexnord series 52 coupling was necessary for compatibility with the light T700 assembly. Because the coupling was originally sized to transmit 1.9 MW (2500 hp), it required considerable machining to tighten tolerances and reduce weight. Individual elements were dynamically balanced to an absolute minimum. After meticulous alignment to less than 1 mil, the two disk packs, which exhibited low angular stiffnesses, still had a tendency to cock the coupling spool to one side. The result was an unbalance that could not be controlled. The flange was reversed on the hub and assembled "inside" the spool. Unbalance was still a problem until one disk pack flexure was eliminated. With this final modification, the coupling was balanced and the drive system could be run to 10,000 rpm with low vibration levels.

5.2 Multiplane/Multispeed Balancing Procedure

The details of the balance routine were outlined in MTI Technical Report 75TR36. A brief description of the proposed routine is repeated here.

The balancing procedure can be divided into five sequential operations:

1. Rotor Preparation
2. Rotor Installation in Balancing Rig
3. Acquisition of Trial Weight Response Data and Calculation of Correction Weights
4. Preparation and Installation of Correction Weights
5. Verification of Rotor Balance Condition

5.2.1 Rotor Preparation

Trial weight and balance weight additions to the rotor may be made using controlled depth drilling or by providing tapped holes for weight attachment. A more temporary change (as by using a balance collar) was preferred for experimental investigations on the thin-walled T700 rotor.

Phase angle measurement was provided by creating an arbitrary zero degree location on the rotating system. The electronics system measured the difference in phase angle between the peak vibration signal and the arbitrary zero.

5.2.2 Rotor Installation in Balancing Rig

The bearing, seals, and probes were in place. The power turbine was bolted to its support stand (item 4 in Figure 10). The forward spline of the power turbine was inserted into the bearing support subassembly (item 3 in Figure 10) . The coupler to the existing driveline was assembled. Alignment of rotating parts, connection of air and oil supplies, and probe gap settings completed the installation.

5.2.3 Acquisition of Trial Weight Response Data and Calculation of Correction Weights

Step 1: Rotor speed is brought to about 600 rpm for acquisition of out-of-roundness data in all planes. Speed is increased up to the first balance speed (about 7000 rpm) unless unsafe vibrational amplitudes occur. (If the amplitudes are too high, the speed is reduced until amplitudes are 80 percent of the maximum permitted.)

Step 2: Rotor amplitude and phases are acquired and stored automatically by the Balancing System at the two safe speeds from Step 1.

Step 3: Rotor speed is reduced and the rotor brought to a stop.

Step 4: A trial weight is installed at rotor zero degrees in balance plane 1.

Step 5: Rotor speed is returned to the first (safe) balance speed and held there. Speed variation and repeatability is to be less than 0.5 percent (peak-to-peak) of nominal balance speed.

Step 6: Amplitude and phase information are automatically recorded.

Step 7: Rotor speed is reduced and the rotor is brought to a stop. Notes: Steps 4-7 can be repeated using the trial weight at the 180° position. Use of both 0 degree and 180 degree data minimizes the number of balance iterations.

Steps 8-11: Repeat Steps 4-7 with the same trial weight, but in balance plane 2.

Steps 12-15: Repeat Steps 4-7 with the same trial weight, but in balance plane 3.

Steps 16-19: Repeat Steps 4-7 with a smaller trial weight in balance plane 4. (The flange diameter in plane 4 is larger than the diameters used along the slender shaft.)

Steps 20-23: Repeat Steps 4-17 with a smaller trial weight for balance plane 6.

Step 24: Examine balance data and compute the required balance corrections.

5.2.4 Preparation and Installation of Correction Weights

Fabricate correction weights from standard set screws. Grind and weigh as required and add them at the specified angular position in each correction plane.

5.2.5 Verification of Rotor Balance Condition

Run rotor up to operating speed. Record rotor vibration at balance speed and operating spedd. Stop rotor. Compare observed vibration with desired vibration and remove rotor if satisfactory. If vibration is not satisfactory, the computer will calculate trim correction weights as required.

NOTE: The similarity in mode shapes at the 8500 rpm and 14,000 rpm criticals may result in acceptable vibration if only the 8500 rpm critical is balanced. If desired, Steps 2-24 may be repeated at 8500 rpm and 13,000 rpm for further reductions in vibration.

5.3 Multiplane/Multispeed Balancing Experiment

The procedure for balancing oulined in Section 5.2 was used as a blueprint for the actual balancing experiments. Modifications to this procedure were made only if rotor behavior was not as expected.

The power turbine exhibited expected vibration behavior up to 4,000 rpm, given the coupling behavior and predicted critical speed of 7,000 rpm. An unexpected vibration increase started at about 4,300 rpm. Several repeatability runs duplicated the phenomena during acceleration as well as deceleration. Upon close examination of the raw vibration data, a peak was also noted at 3,200 rpm (see Figure 17). Since analysis predicted 6,500 rpm as a "low" estimate for the first critical, this response was not the first bending critical speed. Multiple probes confirmed that although the entire rotor responded at this speed, the largest jump was noted in plane 3 near the forward bearing and P/T spline. The location of the maximum vibration and the sudden onset confirmed suspicions of a spline shift. Although the resonant amplification is small due to the distance away from the resonant speed, the pounding of the spline is sufficient to cause response in the first mode shape. The loose spline fit, worn spline teeth, excessive shaft unbalance and low centering torque are contributing factors to the spline shifting phenomenon.

Access to the balance land closest to the spline was not possible because of its location within the forward support assembly. Balancing was conducted at 4000 rpm and 4700 rpm with the restriction that balance weight computation be limited to plane #3 provided nearest the spline. Installing a 6.65 gm-in. correction reduced the response sufficiently to increase speed. The influence coefficient data indicated that the center of the shaft was the region most sensitive to unbalance. However, any balancing done to the shaft center would spoil efforts to balance the true first lateral critical. The vibration data shown in Figure 18 demonstrates the success of the balance. Note that these plots were generated directly from the probes and, therefore, remain uncorrected for runout.

The balancing experiments continued by increasing the rotational speed. The vibration response started a gradual climb and the speed was limited to 8000 rpm to prevent damage to the test rotor. Two balance speeds were selected, 6800 rpm and 7300 rpm along with two balance planes, one positioned at mid-span and the other on the turbine bolting flange. Influence coefficients were generated by installing a nominal weight at 0° and then 180° in the balance planes. The computer calculated that a 0.99 gm-in. correction was required at mid span and 1.67 gm-in. near the power turbine. The corrected vibration response, see Figure 19 , indicates the success of the balance. The net vibration, corrected for runout, dropped from 0.06 mm (2.5 mils) to less than 0.03 mm (1 mil) at 7,300 rpm. The balance effort limited vibration to a high of 0.1 mm (4 mils) P-P at 9,500 rpm.

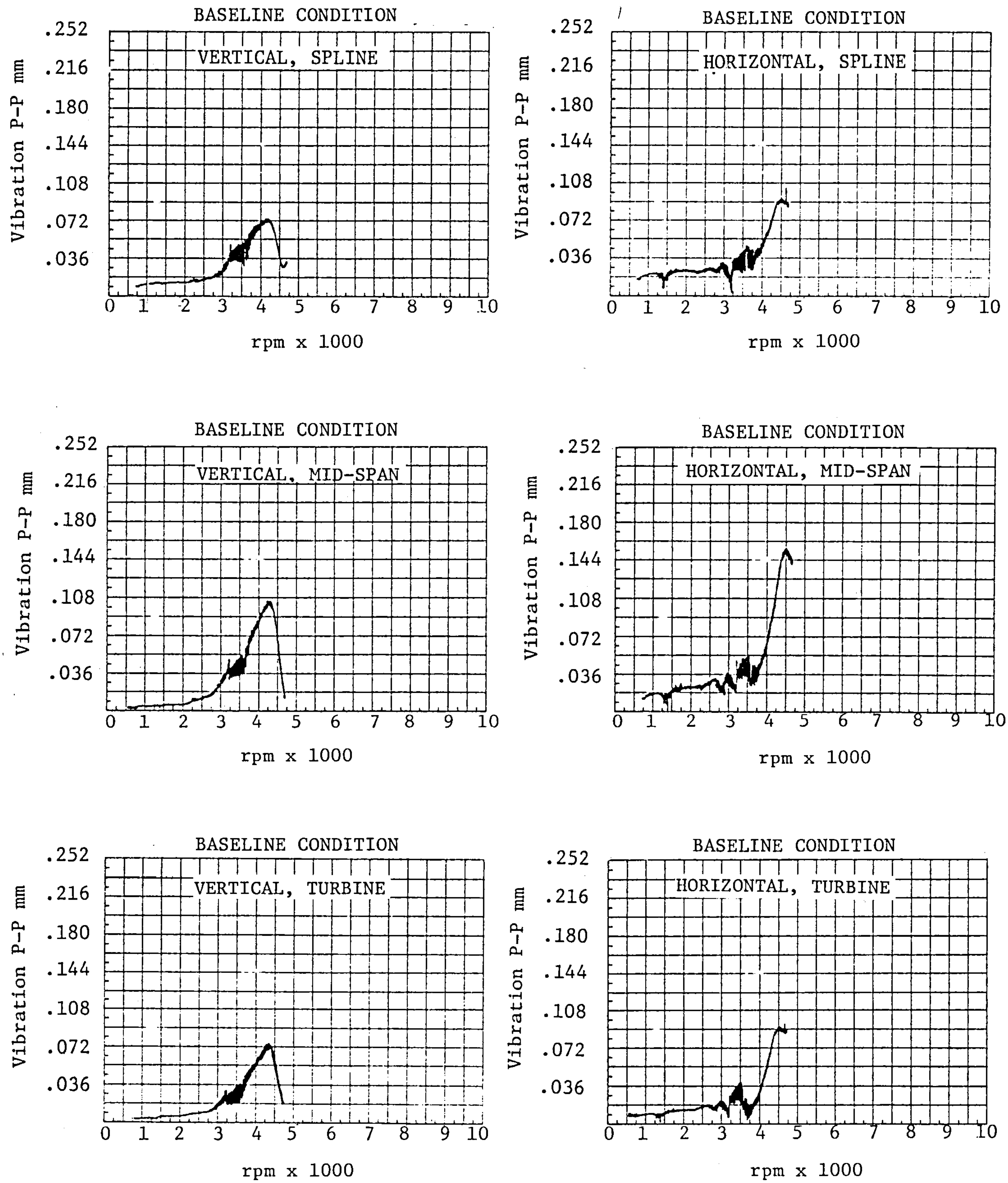


Figure 17 As-Received Condition of Balance, T700 Power Turbine

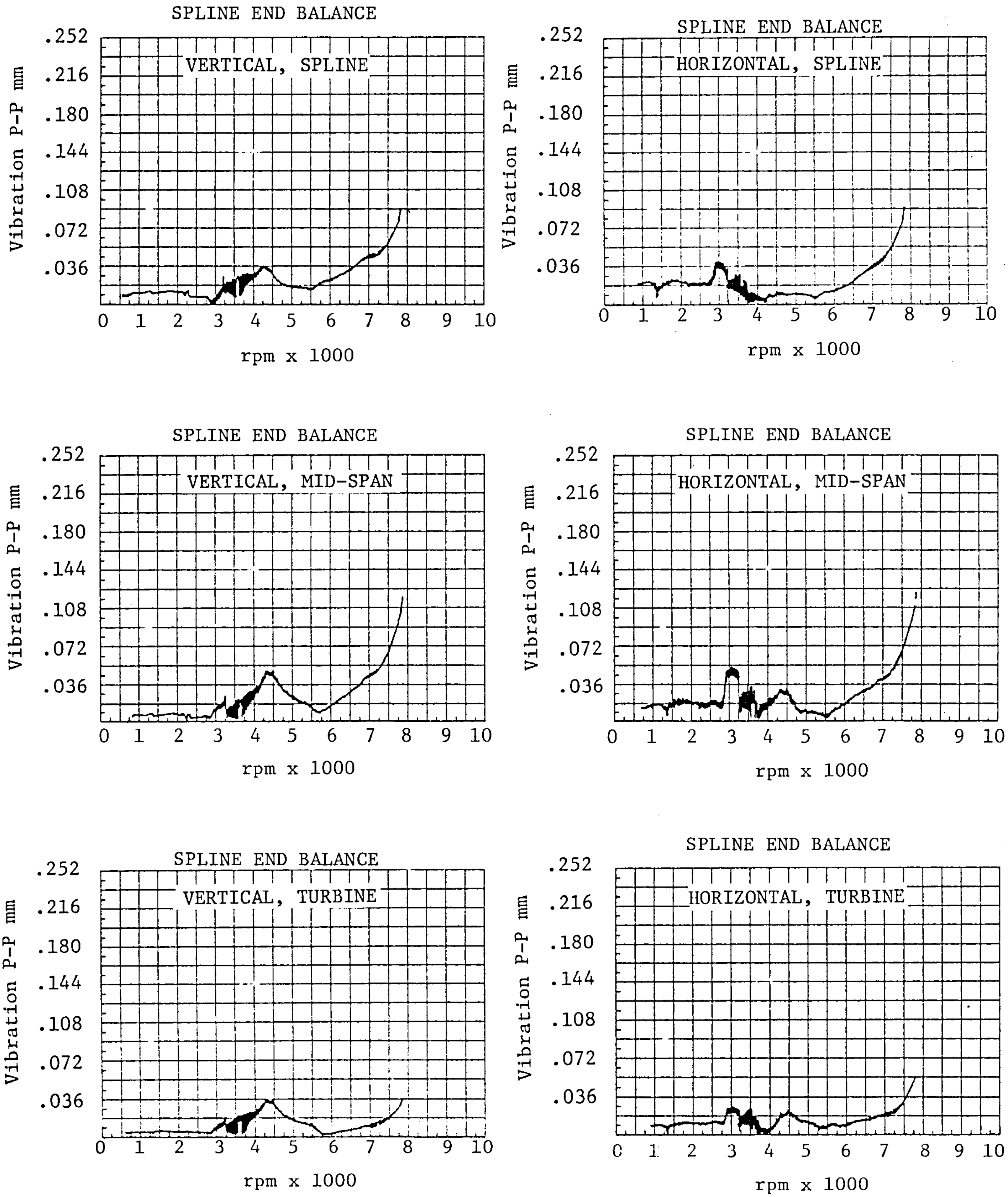


Figure 18 Condition of Balance After Correcting for Spline, T700 Power Turbine

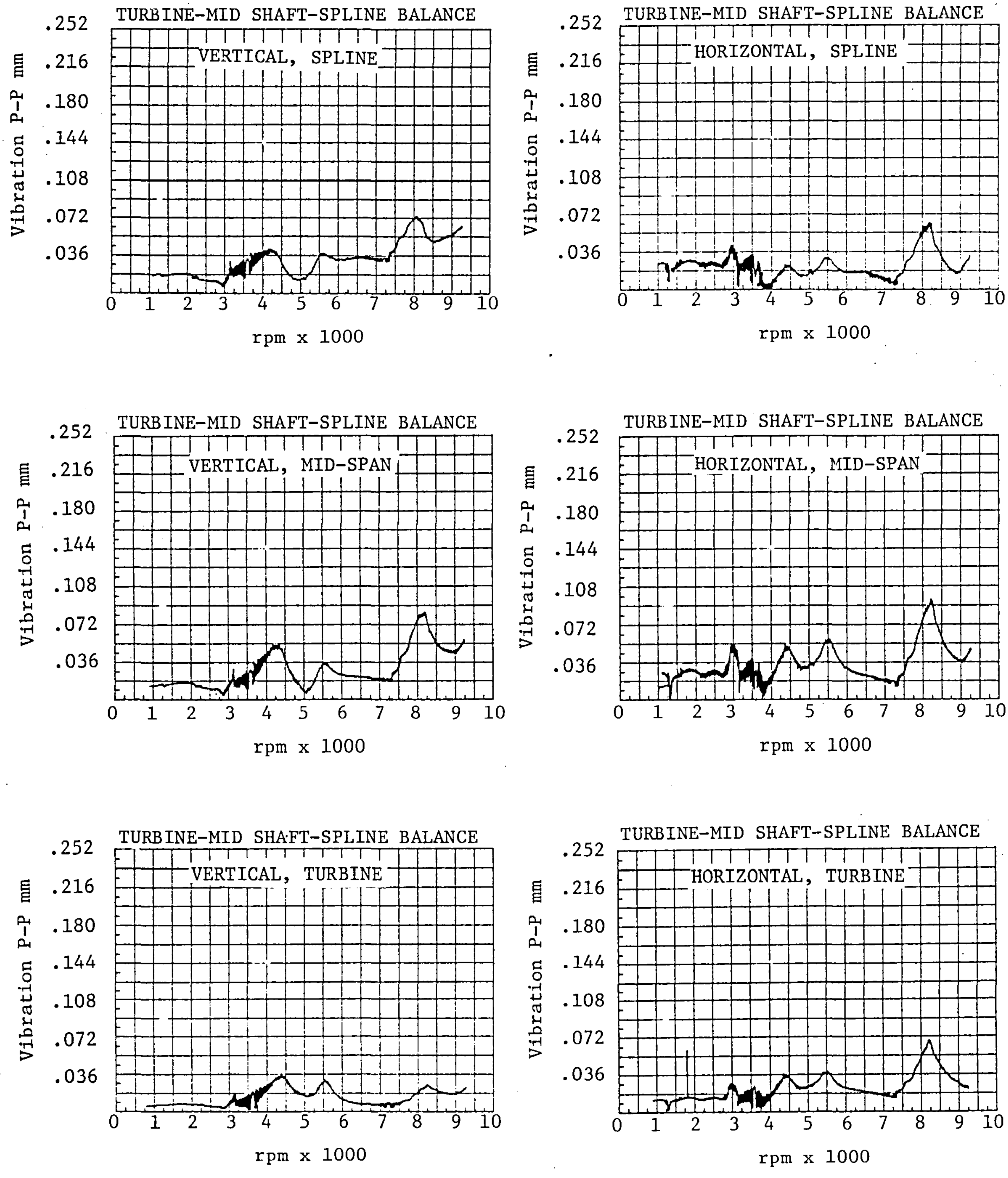


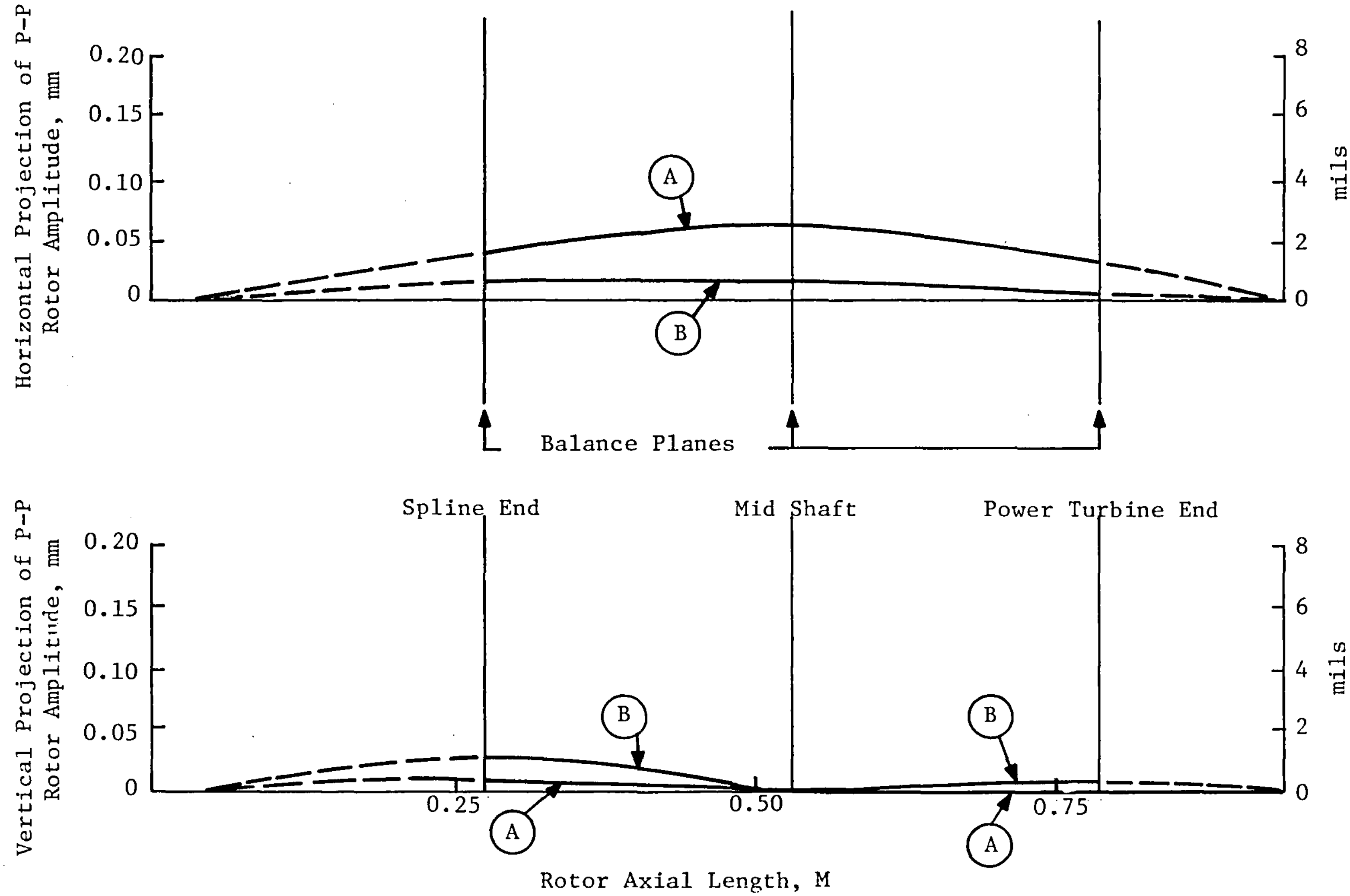
Figure 19 Condition of Balance After Correcting for Bending Critical, T700 Power Turbine

The original program goals were to demonstrate multiplane balance through the engine service speed at 16,000 rpm. The previous discussion covers the speed range up to 9500 rpm. Beyond this speed, the heavy coupling (see page 31) in the drive system exhibited large vibrational amplitudes which were undesirable from a general operating standpoint. The cross talk from coupling to turbine caused the rotor unbalance response to be masked by the coupling problem.

The conclusion was that even the reduced weight, well balanced coupling was not adequate for power turbine service speed checks. The test schedule and financial limitations did not permit the selection and installation of another type coupling.

The actual mode shape of the power turbine at 7300 rpm (see Figure 20) compares favorably with the shape and speed determined analytically. The good analytical/experimental correlation at the first critical lends support to the prediction that the second critical should be balanceable using the same technique. Since the first and second critical speed mode shapes are analytically similar, a large reduction in second critical vibration should have resulted from the first critical balance. Therefore, it is possible that the need for second critical balance may be eliminated.

It is unfortunate that the power turbine service speed could not be achieved, but the lower speed results confirm our ability to balance very flexible rotors. Indeed, the unexpected spline shaft and coupling phenomena point out the extra skills required that exceeded in difficulty the turbine requirements.



A = After 1 Plane Balance @ 7300 rpm

B = After 3 Plane Balance @ 7300 rpm

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Figure 20 T700 Main Power Shaft Mode Shape: First Lateral Critical

6.0 CONCLUSIONS AND RECOMMENDATIONS

6.1 Conclusions

- The low-speed balanced T700 power turbine exhibited high rotor vibration during operation up to 10,000 rpm. The mode shapes and response confirmed analytical predictions that the T700 power turbine is a flexible shaft which can benefit from high-speed balancing.
- The T700 power turbine shaft vibration can be controlled by three-plane balancing through the critical speeds up to 10,000 rpm. The experimental procedure showed successful use of a modular balancing technique. This technique also showed substantial promise for practical production use. Future experiments should show that no disassembly will be required before installation and successful operation in an engine.
- The front spline shifted radially during operation. This dynamic offset was caused by the radial unbalance force which exceeded the spline's self-centering capability. This phenomenon could have been caused by the poor condition of the spline as received by MTI, the low torque applied, or the inherent nature of the original design.

6.2 Recommendations

- The existing coupling in the MTI drive system only allowed for operation up to approximately 10,000 rpm. The T700 power turbine rotor should be observed up to its maximum operating speed which in this test rig would provide documentation of the T700's true rotordynamic operating characteristics.
- The T700 power turbine components had all been processed using the standard low-speed balancing techniques. New parts having not been low-speed balanced, or the present parts, purposely unbalanced, should be assembled into the power turbine module to investigate the potential of eliminating many of the low-speed balancing operations.
- The T700 power turbine shaft is a flexible rotor operating at high rotational speeds resulting in a shaft system which is very sensitive to unbalance. Normal low-speed balancing may not provide enough force

resolution to make accurate corrections. The high-speed balancing procedure takes advantage of increased centrifugal forces and amplification factors at resonant speeds. In the present experiments, balance corrections were made using collars to accurately control the correction amounts. Hand grinding, as typically used in production, is inaccurate and therefore may not be satisfactory for making corrections on a long slender shaft. Laser balancing now being experimentally investigated by NASA should be considered for application to the T700 engine. The accuracy of high-speed balancing could then be coupled with precise corrections resulting in a well-balanced module using the latest technology in an automated procedure.

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