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STUDY OF T53 ENGINE VIBRATION

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FOREWORD

Work outlined in this report was accomplished by Mechanical Technology Incorporated (MTI) under NASA Contract NAS3-20609. Dr. David P. Fleming was the NASA Project Manager and Mr. Thomas Pojeta was the point of contact within the U.S. Army Aviation Research and Development Command. Technical advice and guidance at MTI was provided by Mr. Richard A. Rio, Manager of Balancing Systems and Dr. Robert H. Badgley, Manager of the Mechanical Dynamics Department. A special expression of appreciation is extended to cognizant personnel at the Corpus Christi Army Depot, whose knowledge, interest and assistance in many key areas of effort have made possible the findings outlined in this report.

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SUMMARY

The objective of the present study has been to obtain a clear understanding of the present and past vibration record of T53 engines overhauled at Corpus Christi Army Depot. This understanding includes an evaluation of the vibration rejection rate, principal sources of vibration, and normal procedures taken by the overhaul center to reduce engine vibration. Documented data for 1977 indicate that 23 percent of overhauled engines tested fail acceptance test criteria because of excessive vibration. At times, this figure has been reported to exceed 40 percent of all T53 engines tested. Resulting cost penalties to the overhaul center are estimated at \$430,000 - \$750,000 per year.

Analytical and experimental data have been compared to determine the engine's dynamic response to unbalance force. Results show that the engine operates through bending critical speeds. One of these critical speeds is in close proximity to a specified vibration test speed. Flexible rotor unbalance forces excite response to this criti al speed and result in excessive vibration levels. Present rigid rotor balancing techniques are incapable of reducing this flexible rotor unbalance. A combination of high and low speed rotor module balancing techniques is proposed for evaluation and implementation.

Engine vibration levels measured in the test cells differed significantly from those measured in aircraft installations. These differences were caused by differences in engine mounting configurations. It is proposed that the present engine support structures in the test cells be modified to better reflect the dynamic response of aircraft installations.

Analysis of engine test cell data has indicated that the spline connection at the joint between the power turbine and power shaft offsets radially during engine operation, thus increasing the effective amount of engine unbalance. This phenomenon, if representative of a significant portion of T53 engines, will affect any low and/or high speed balance efforts. This spline connection should be evaluated in any future effort to reduce T53 engine vibration levels.

Work accomplished under the present contract has resulted in an understanding of the fundamental causes of T53 vibration levels. A number of excitation sources exist, which should be explored in future efforts to reduce the maintenance and overhaul costs associated with vibration related problems.

SECTION I

INTRODUCTION

In March, 1977, Mechanical Technology Incorporated (MTI) contracted with NASA to perform a twelve month study of T53 engine vibration characteristics at the Corpus Christi Army Depot (CCAD), Corpus Christi, Texas. The scope of the present work has included an evaluation of vibration rejection rates, principal sources of vibration, and normal procedures taken by the overhaul center to reduce engine vibration.

In recent years, this type of expertise in the area of high speed rotating machinery has increasingly been applied in many aspects of gas turbine design and analysis. In addition to the present effort, MTI is currently under NASA contracts NAS3-19408 and NAS3-18520. These contracts involve analytical and experimental work on other gas turbine engines for helicopter applications.

Of particular interest for these engines is the potential application of new multiplane balancing techniques, which have been developed and refined in the last several years. These techniques allow flexible rotors to be balanced at high speeds, thereby achieving a degree of balance that cannot be attained with low speed rigid rotor balancing methods. Analytical studies have revealed that the T53 power turbine rotor module does operate through bending critical speeds. Multiplane multispeed balancing techniques are applicable, and can have significant effect in reducing engine vibration levels and resulting overhaul costs.

SECTION II

ANALYSIS OF T53 ENGINE VIBRATION

A. T53 Engine Vibration Analysis

1. Computer Simulation

The T53 engine contains two rotor modules. The high speed rotor is comprised of the compressor and gas generator and the low speed rotor is comprised of the power turbine and power shaft. Figure 1 shows a cross section of the T53 engine. Figure 2 also shows an engine cross section, but with the rotors extracted from the engine case for greater clarity.

As Figure 2 shows, the compressor rotor consists of 5 axial stages and 1 centrifugal stage. The compressor is bolted to a two stage gas generator turbine which drives the rotor. This rotor is supported by two rolling element bearings, one at the front of the compressor and one between the compressor and the turbine. The power turbine rotor consists of a two stage power turbine which is splined to the power output shaft. This rotor is supported by three rolling element bearings, two located adjacent to the second stage turbine wheel and a small steady bearing at the opposite end of the power shaft.

A computer simulation of the T53 engine was constructed to analytically determine the engine's critical operating speeds and response to unbalance forces. The engine was modeled for use in an existing multilevel rotordynamic computer program. The program included gyroscopic effects and assumed both rotors to be at the same speed. Distributed unbalance forces were added to each compressor and turbine stage. From past turbomachinery manufacturing experience, the unbalance distribution was selected to provide typical forces for the T53 engine's design and assembly procedures. Total weight additions for the analysis presented below were 31g (1.1 oz) in the compressor rotor and 17g (0.6 oz) in the power turbine. These are nominal values, given the present balancing and assembly methods at CCAD.

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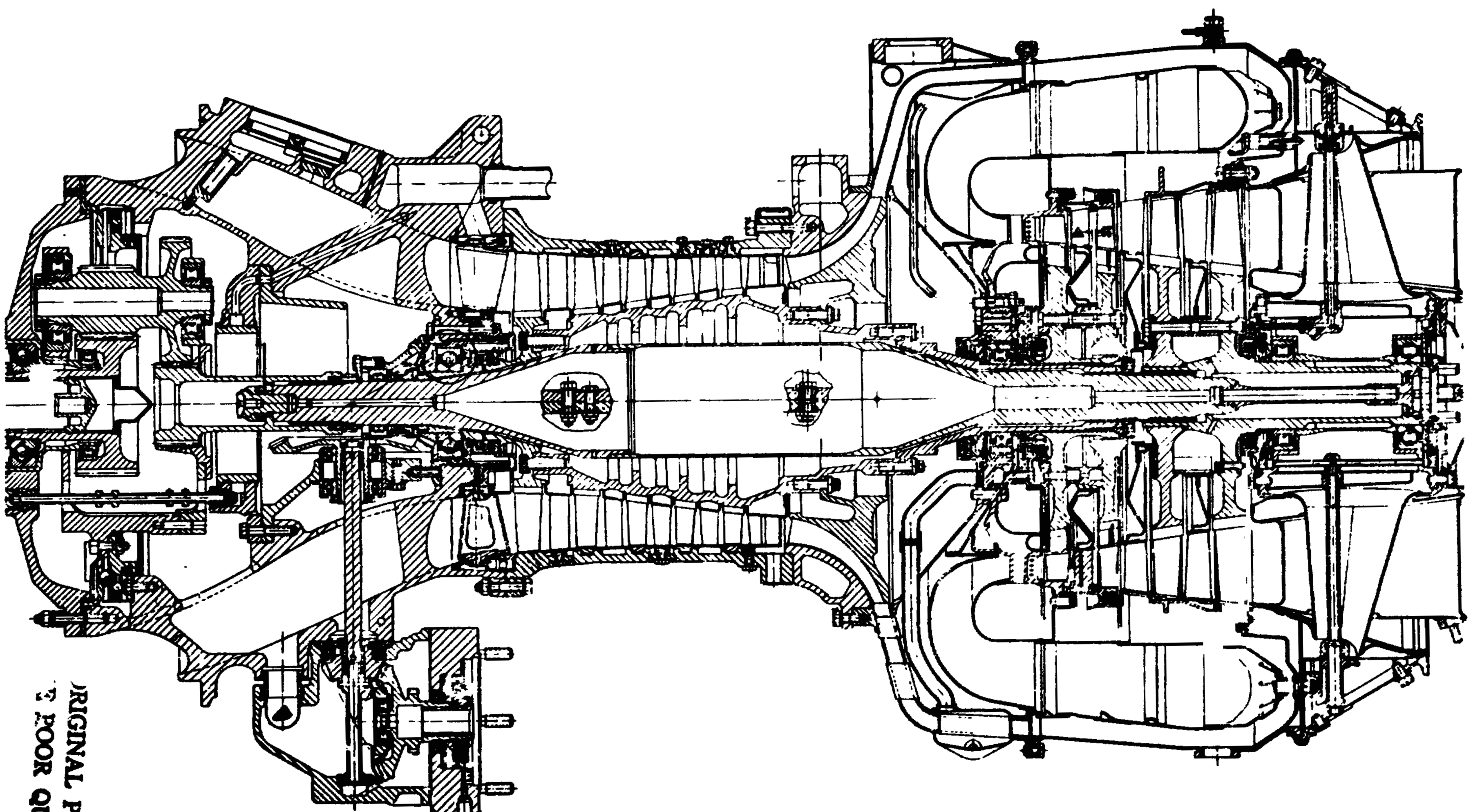

Figure 1 Typical T53 Engine Cross-Section

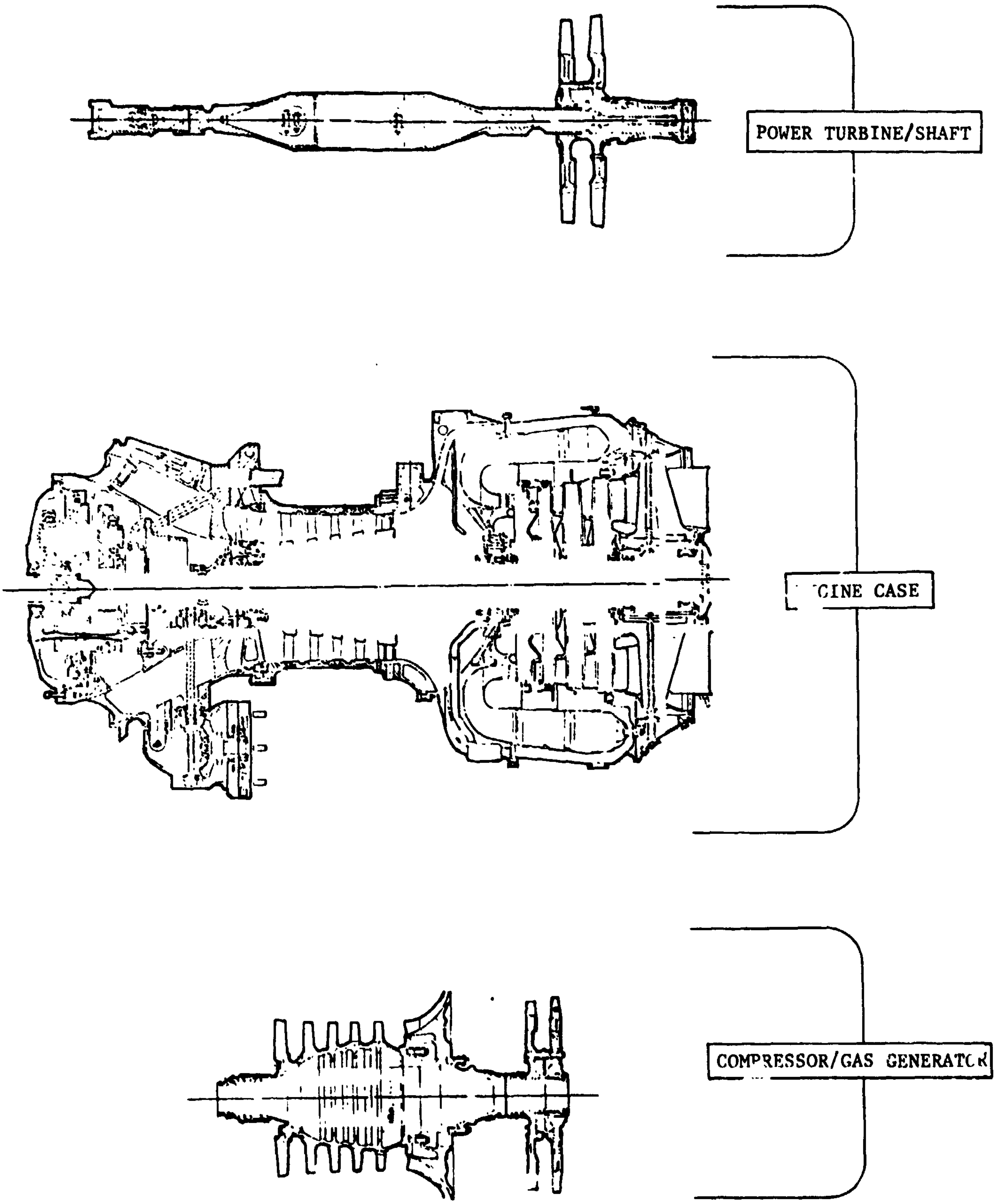

Figure 2 Expanded View Of Typical T53 Engine Cross-Section

2. Mode Shapes at Critical Speeds

The critical speeds calculated for the T53 engine model agree closely with actual engine test data. Because of significant dissimilarities in engine mounting configurations, critical speeds and mode shapes have been calculated for both test cell and typical aircraft installations.

The critical speeds calculated for both installations are shown in Table 1:

TABLE 1

T53 CRITICAL SPEED ANALYSIS

Installation	Critical Speed (rpm) 1st	2nd	3rd	4th	5th
Test Cell	3,500	10,700	13,100	17,600	23,200
Aircraft	3,100	8,900	12,700	19,600	28,000*

Figures 3 through 7 present the engine's dynamic response for each critical speed. The test cell model (lower curve on each figure) includes the dynamic effect of the water brake (used to load the engine) and an engine exhaust cone. These devices are not present in aircraft installations. Engines in test cells are mounted only at the gear case. A detailed discussion of engine mountings is presented in the following section.

The water brake is bolted to the output shaft end of the engine and weighs nearly 160 kg (350 lb), or approximately 60 percent of the weight of the T53 engine. This large cantilevered mass restricts the motion of the output shaft end of the engine and generally causes the engine to pivot around that end. The exhaust cone weighs approximately 11 kg (25 lb). This mass has little dynamic effect on the engine and generally acts as a rigid body extension of the engine case.

The first two critical speeds (Figures 3 and 4) are essentially the engine's cylindrical and conical rigid body modes although the power turbine does

*This speed is above the maximum engine speed for T53 engines (maximum operating speed: power turbine - 21,080 rpm; compressor - 25,150 rpm).

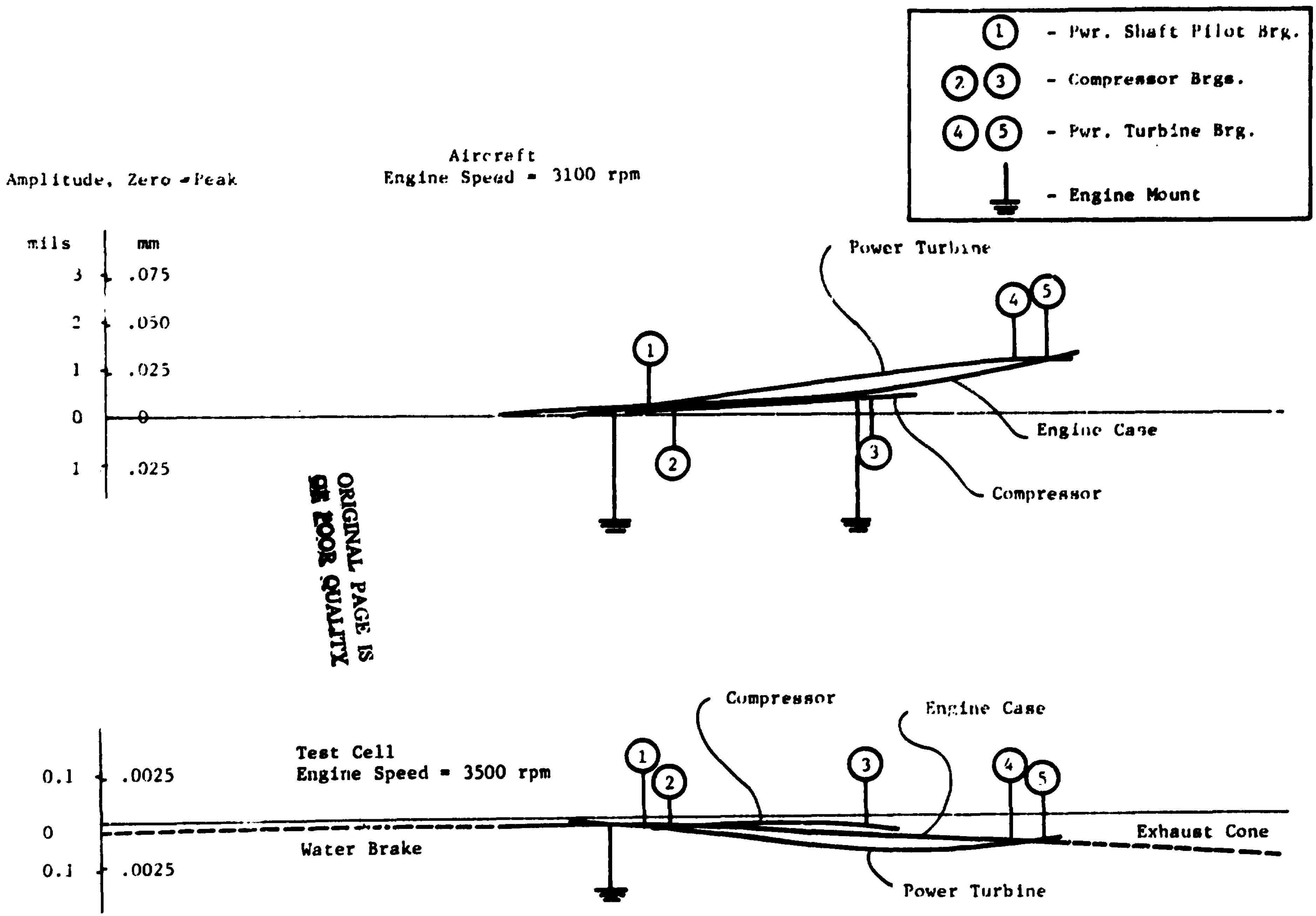

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Figure 3 First Critical Speed - T53 Engine

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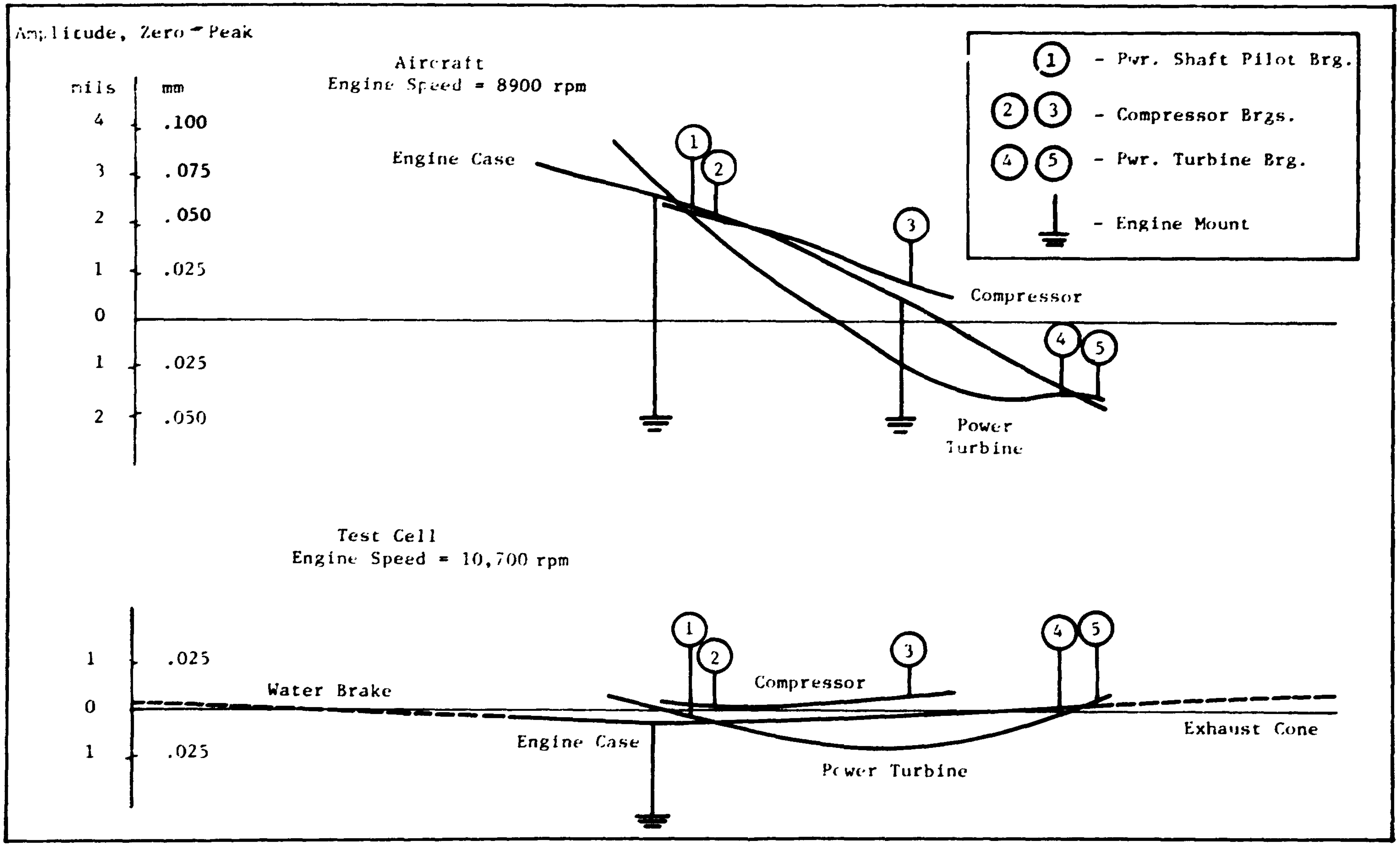

Figure 4 Second Critical Speed - T53 Engine

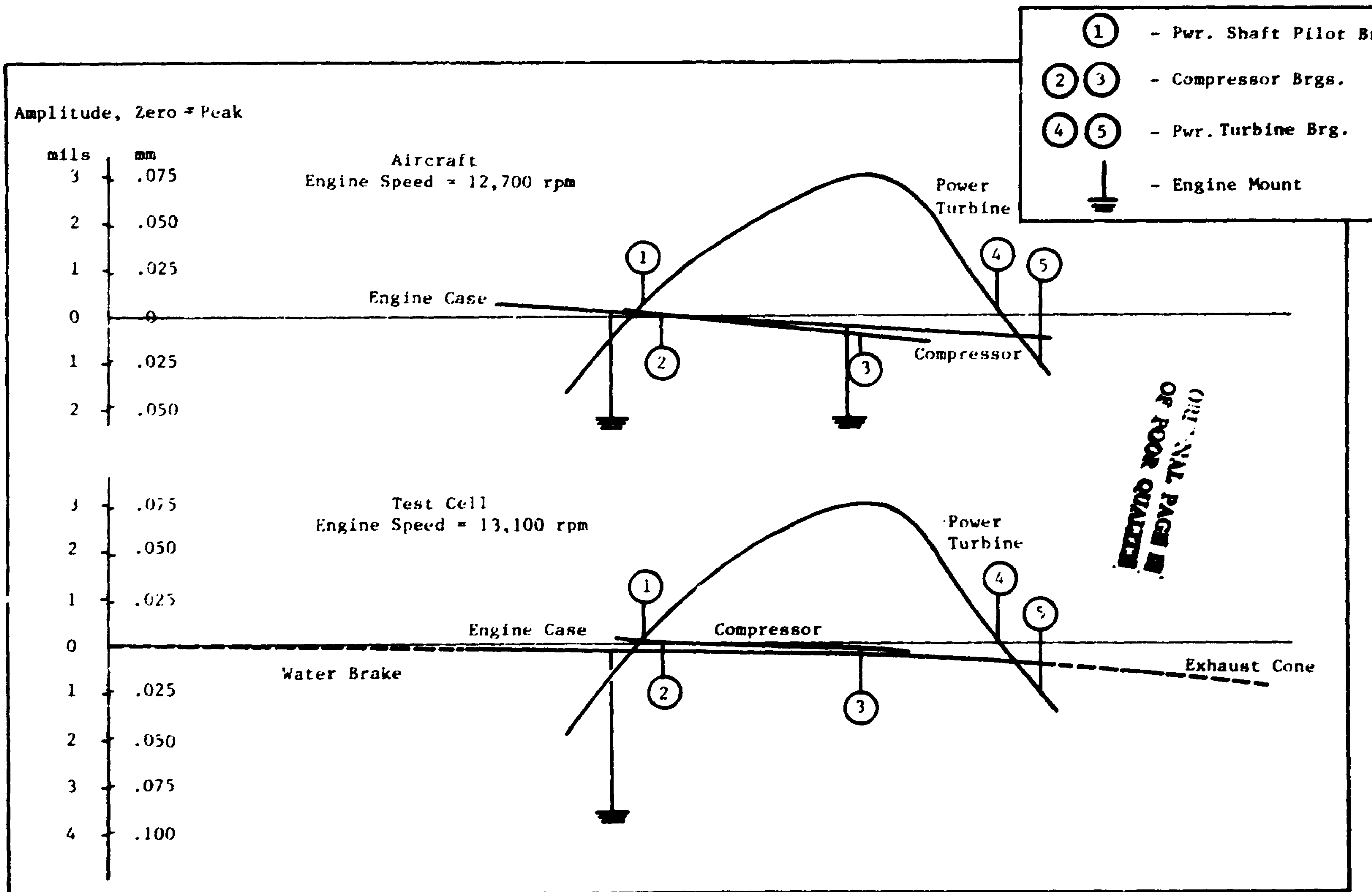

Figure 5 Third Critical Speed - T53 Engine

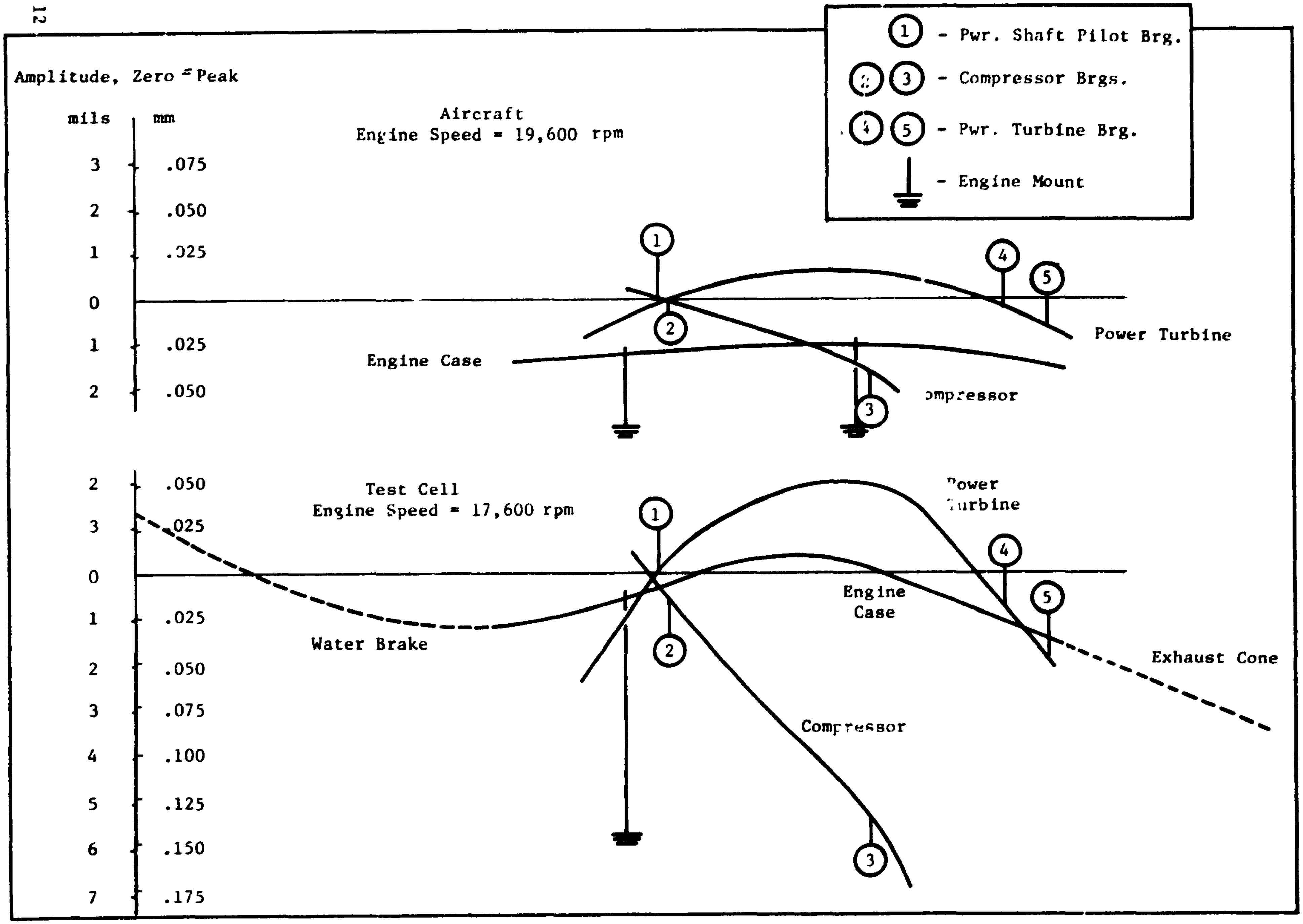

Figure 6 Fourth Critical Speed - T53 Engine

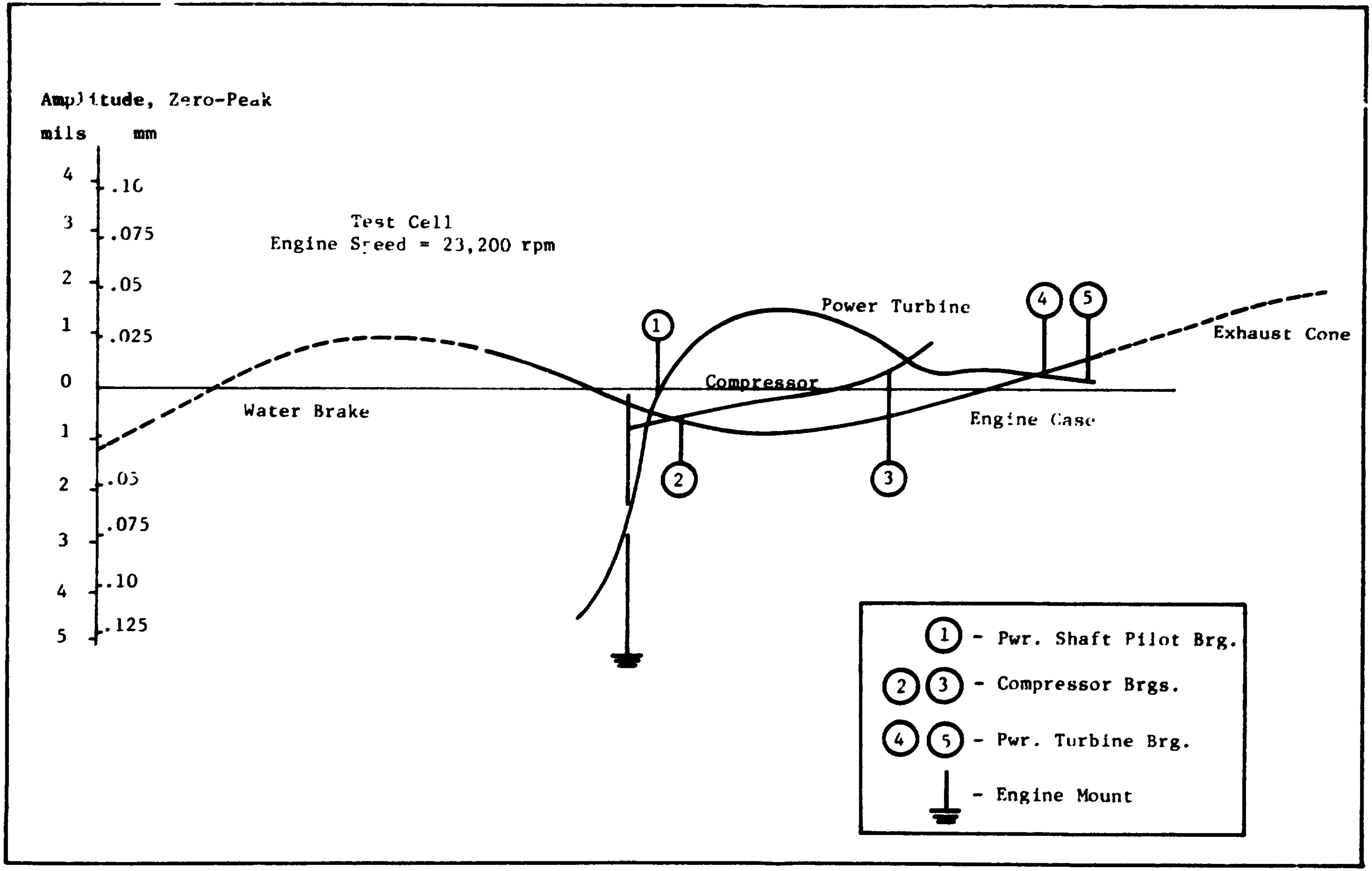

Figure 7 Fifth Critical Speed - T53 Engine

exhibit some minor bending at these speeds. Figure 4 also shows the restraining effect of the water brake. Conical mode vibration of the engine case is much less than in the unrestrained aircraft installation. At the third critical speed (Figure 5), the power turbine shows major bending, while the compressor and engine case remain relatively rigid. The test cell and aircraft mode shapes are very similar. The most significant bending occurs at the spline joint between the power turbine and power shaft.

The largest difference between aircraft and test cell mode shapes is at the fourth critical speed (Figure 6). In both installations, the compressor, and to a lesser extent the engine case, respond in the first compressor mode, which could be described as a conical "rigid rotor" mode. However, the presence of the water brake in the test cell configuration restricts the displacement of one end of the engine and causes a "whipping" action and larger displacements at the opposite end. Maximum test cell displacements of both the compressor and power turbine rotors are nearly three and a half times larger than in the aircraft model. The large displacements at the No. 3 bearing are accommodated by the relatively soft pedestal bearing support.

It is important to note that the fourth test cell critical speed is in close approximation to the 80 percent power turbine speed specified as a vibration test speed. This test speed has historically caused many vibration-related rejections in the test cell. Rotor unbalance plays a crucial role in engine response at critical speeds. This problem is discussed in detail in Section IV.

Figure 7 presents the mode shape for the fifth critical engine speed in the test cell. This speed is greater than the maximum power turbine speed. The compressor remains relatively rigid while the engine case shows a "whipping" reaction to the water brake. The fifth critical speed in the aircraft was calculated to be approximately 28,000 rpm, well above the maximum operating speed for the engine.

B. Experimental Analysis

Verification of the critical speeds predicted by the computer analysis was accomplished by measuring actual vibration levels of engines operating in both test cells and aircraft. Two engines were operated in a test cell and a third engine was tested in a helicopter. A block diagram of the instrumentation used to record and analyze engine vibration data is presented in Figure 8. Figure 9 shows the engine locations of vibration pick-ups. For test cell measurements, a fourth sensor is used to measure radial vibration and is located on the casing of the water brake.

Figures 10 through 13 present engine test cell vibration. These figures show the effect of power turbine unbalance on the engine case. These plots show only vibration that is synchronous with the power turbine's rotating speed (i.e., once per revolution). Through use of a synthesizer and tracking filter (see Figure 8), influence from other frequencies has been eliminated. These figures present vibration in units of velocity. Several points on each figure have been converted to displacement. Although each engine shows individual characteristics, the T53 power turbine generally shows peak responses at approximately 11,000, 13,000 and 17,000 rpm. These speeds compare well with the critical speeds predicted by the computer model used to simulate the T53 engine (see Table 1).

Figures 10 and 11 also show the effect of simultaneously operating both engine rotors at the same approximate speed. The large peak at 10,800 rpm for sensor No. 3 is the result of rotational excitation from both the power turbine and the compressor at an engine critical speed. As Figure 10 shows, vibration amplitude at sensor No. 3 exceeded 0.25 mm (0.01 inch) at 10,800 rpm power turbine speed. Compressor rotor speed was constant at 12,800 rpm. As power turbine speed increased to 11,000 rpm, the combined excitation from both rotors electrically saturated the sensor and the signal was lost. At this point, an inspection of the operating engine disclosed that sensor No. 3 was shaking radially approximately $\pm$ 1.5 mm (1/16 inch). This excessive vibration may have been intensified by the method of attachment of sensor No. 3. This problem is discussed in detail in Section IV.

Figure 11 also shows a phenomenon that may impact any attempt to reduce T53 vibration rejection rates. The instantaneous jump in vibration amplitude at 19,500 rpm is usually indicative of either a shift in rotor components or dynamic offset of a spline connection. The fact that the rotor did not maintain its high level of vibration in subsequent engine accels (see Figure 13) essentially eliminates shifted rotor components as the source of

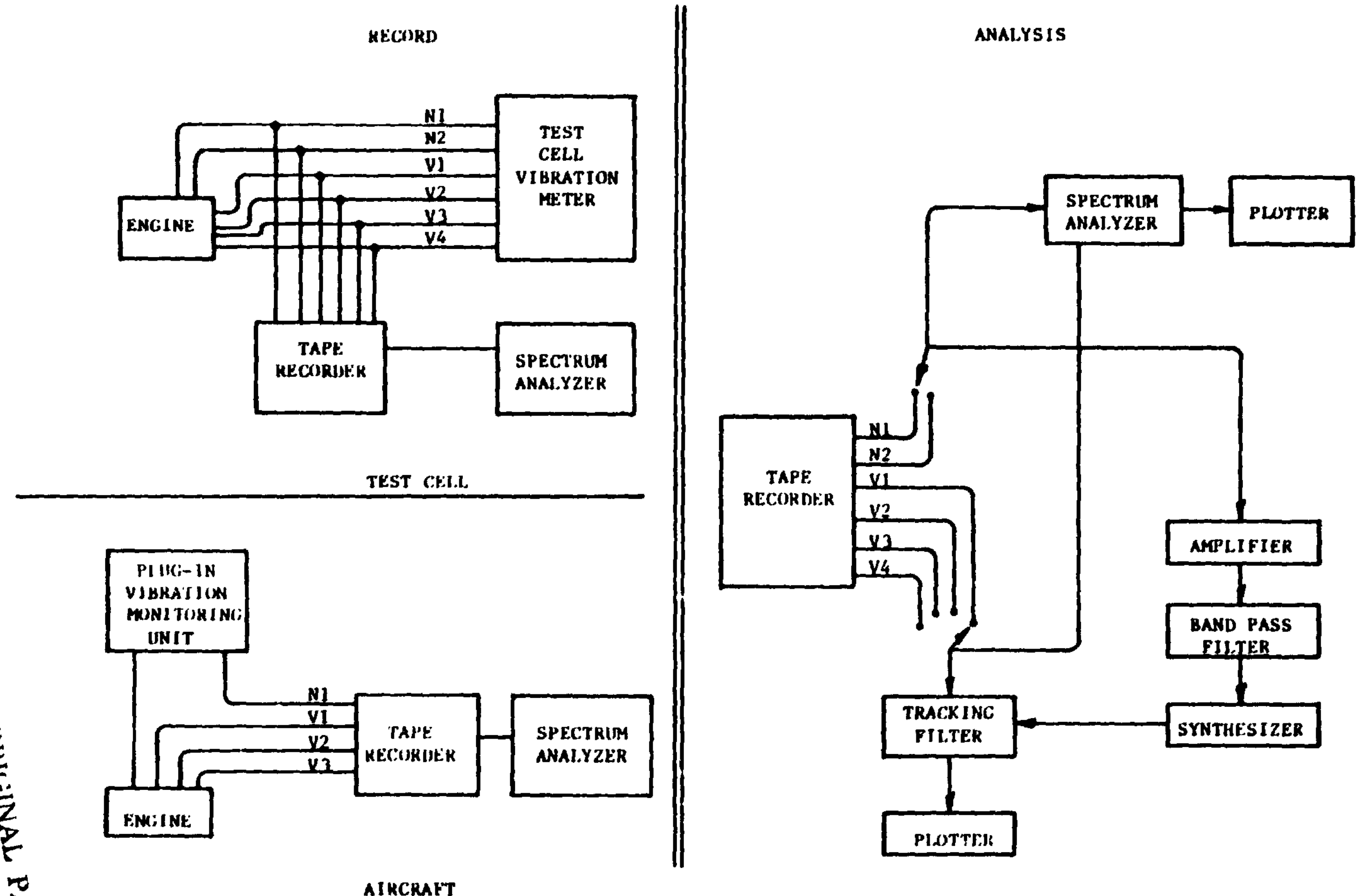

Figure 8 Block Diagram of T53 Engine Vibration Recording & Analysis Instrumentation

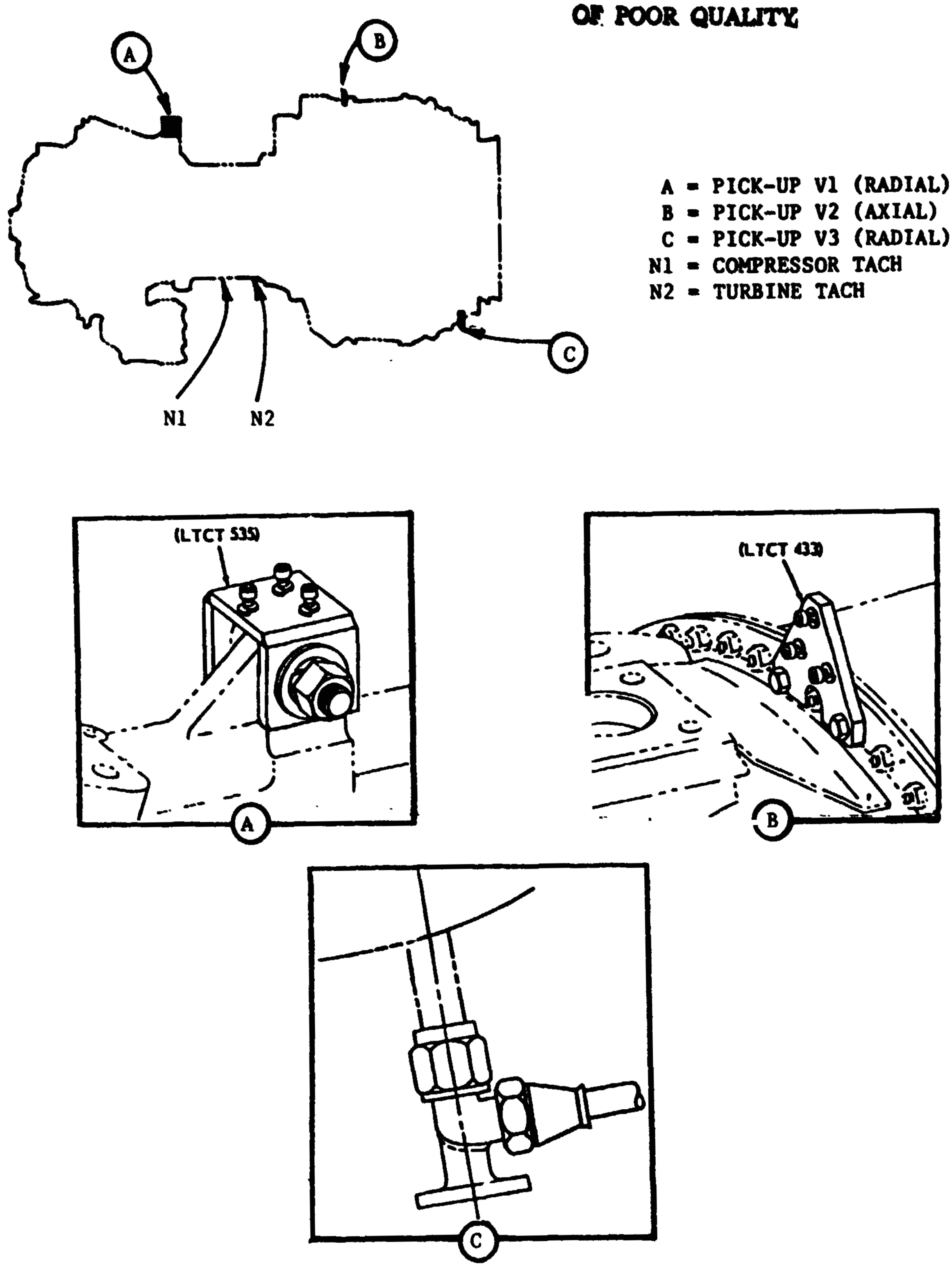

Figure 9 Locations of Engine Vibration Pick-ups - T53 - L - 13 Engine

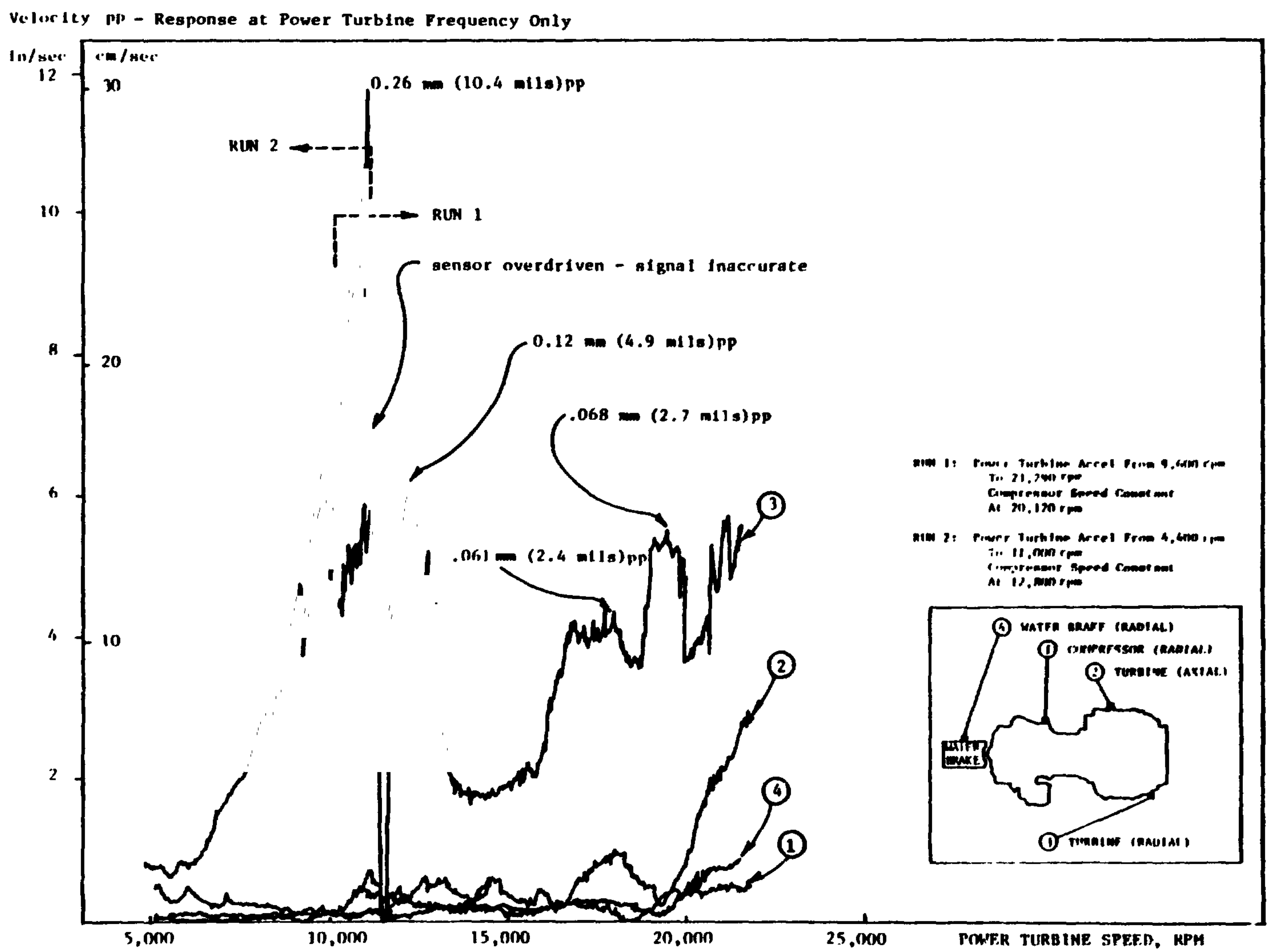

Figure 10 T53 Test Cell Vibration - Engine No. 1

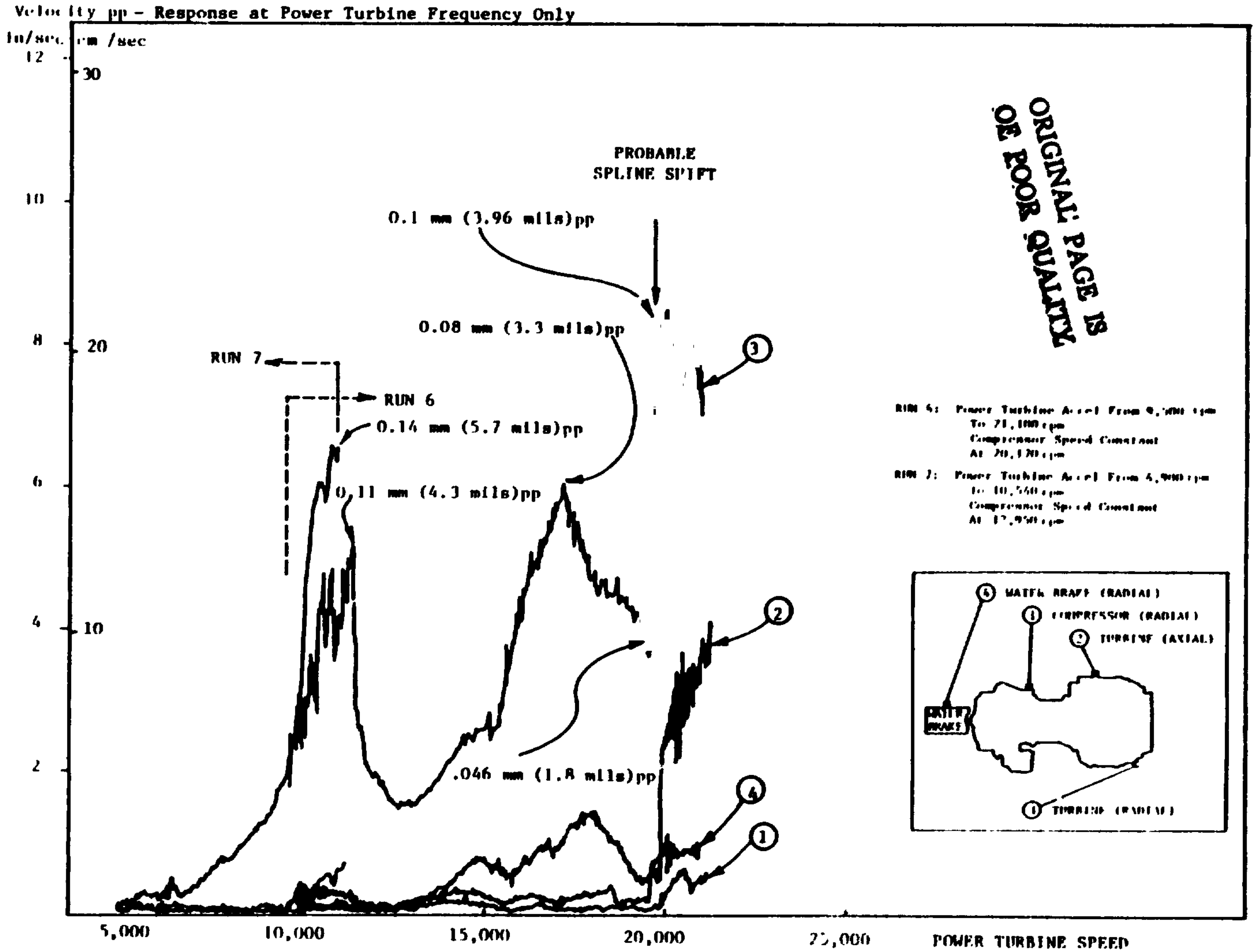

Figure 11 T53 Test Cell Vibration - Engine No. 2

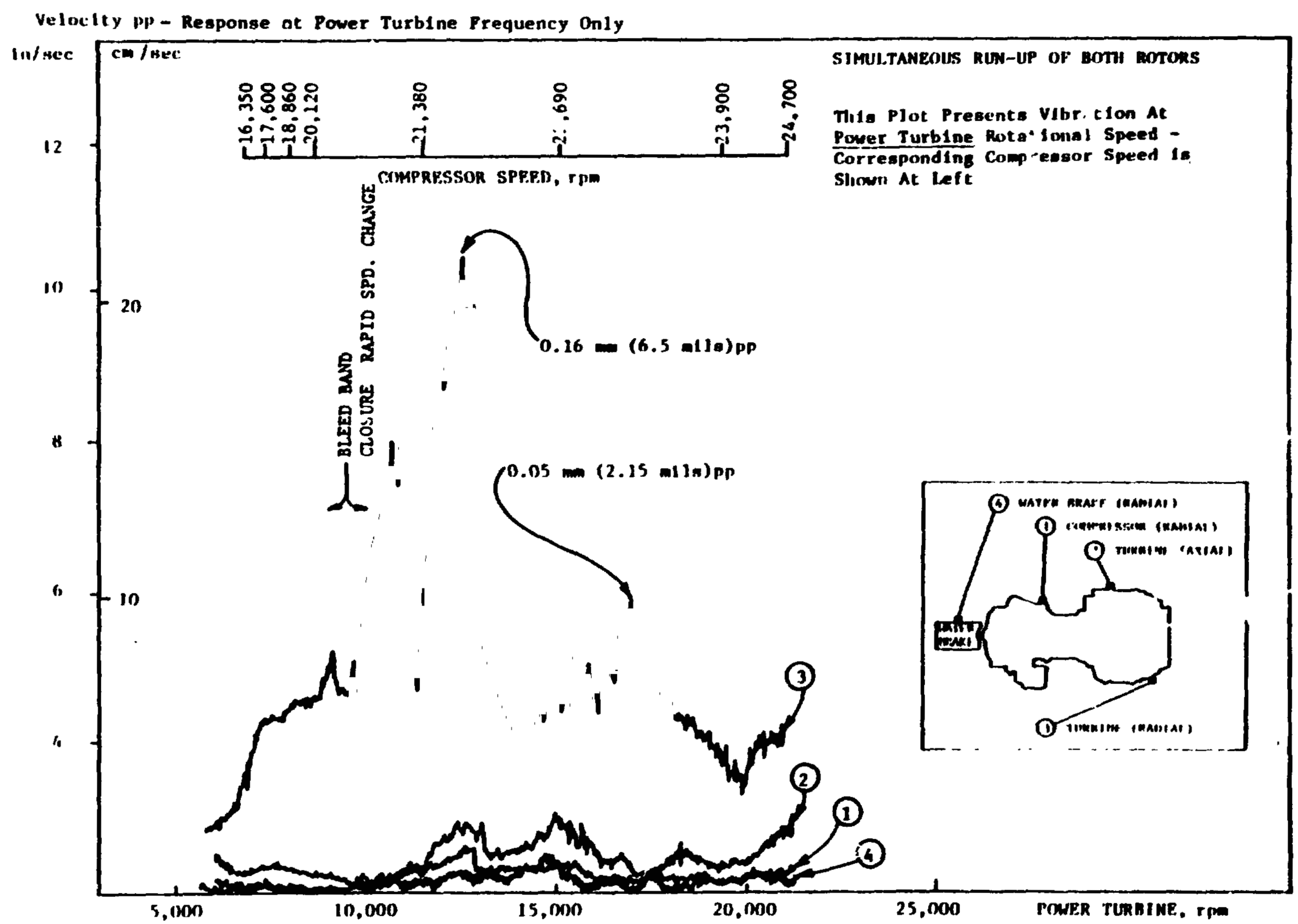

Figure 12 T53 Test Cell Vibration - Engine No. 1

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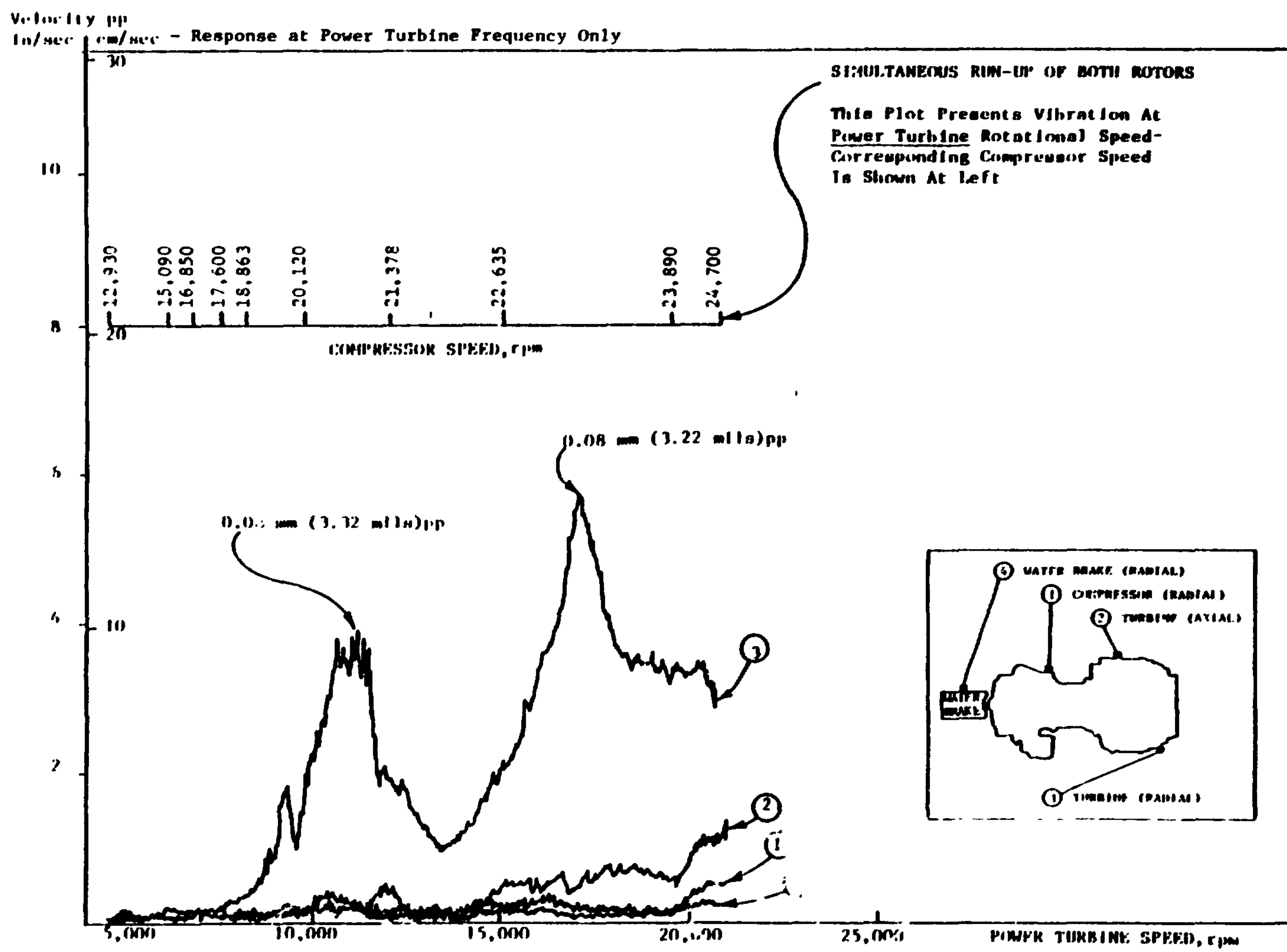

Figure 13 T53 Test Cell Vibration - Engine No. 2

the jump. At low speeds, the load centering capability of the spline resists the unbalance force of the rotor. However, the unbalance force increases with speed until at some point (19,500 rpm in this case) the unbalance force exceeds the load centering capacity of the spline and the spline connection shifts radially to an out-of-center position. This action increases the unbalance force, resulting in more engine case vibration. An engine decel allows the spline to again center the load. Subsequent accels may position the spline in alternate locations and the resulting response may not be repeatable from one accel to another.

The probability of an offsetting spline in the T53 engine should be considered as part of any follow-on effort. If representative of a significant portion of overhauled T53 engines, this problem will impact any other attempt to reduce engine vibration rejection rates.

Figures 12 (Engine No. 1) and 13 (Engine No. 2) also show the effect of power turbine unbalance. However, the compressor rotor speed was also changing and at no time were both rotors operating at the same speed (actual compressor rotor speed is also noted on these figures).

Figure 14 shows the dominance of power turbine rotation in the noise signature. The rather dominant "mountain range" running from the lower left-hand corner is the rotating speed of the power turbine. This path also gives general indications of engine critical speeds.

Figure 15 presents T53 vibration with an engine mounted in an aircraft. Vibration levels on this figure are shown as a function of compressor rotor speed. Principally because of mounting dissimilarities, the overall vibration levels were generally lower than those measured in the test cell. Stiffer mountings in the aircraft also affected the regions of peak response. As Figure 15 shows, there are few well defined peaks in the data. Computer simulation predicted critical speeds at 8,900, 12,700 and 19,600 rpm. Although some peaking of the data occurs in those speed ranges, the basic result of these measurements is the significant difference in overall vibration levels when compared to test cell operation. Compressor rotor speeds above 21,500 rpm could not be attained because of incipient

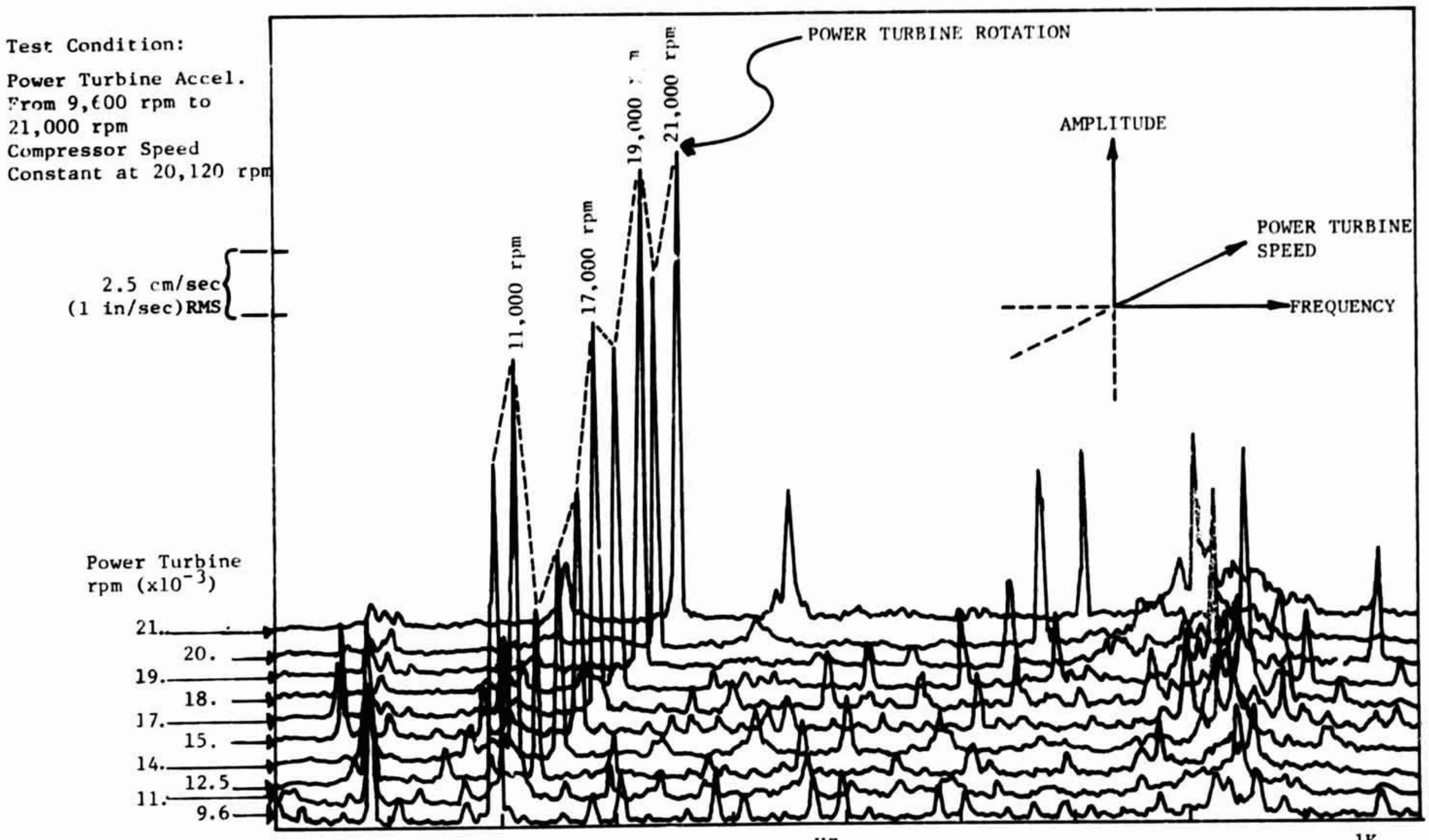

Figure 14 T53 Test Cell Vibration - Engine No. 1, Sensor No. 3

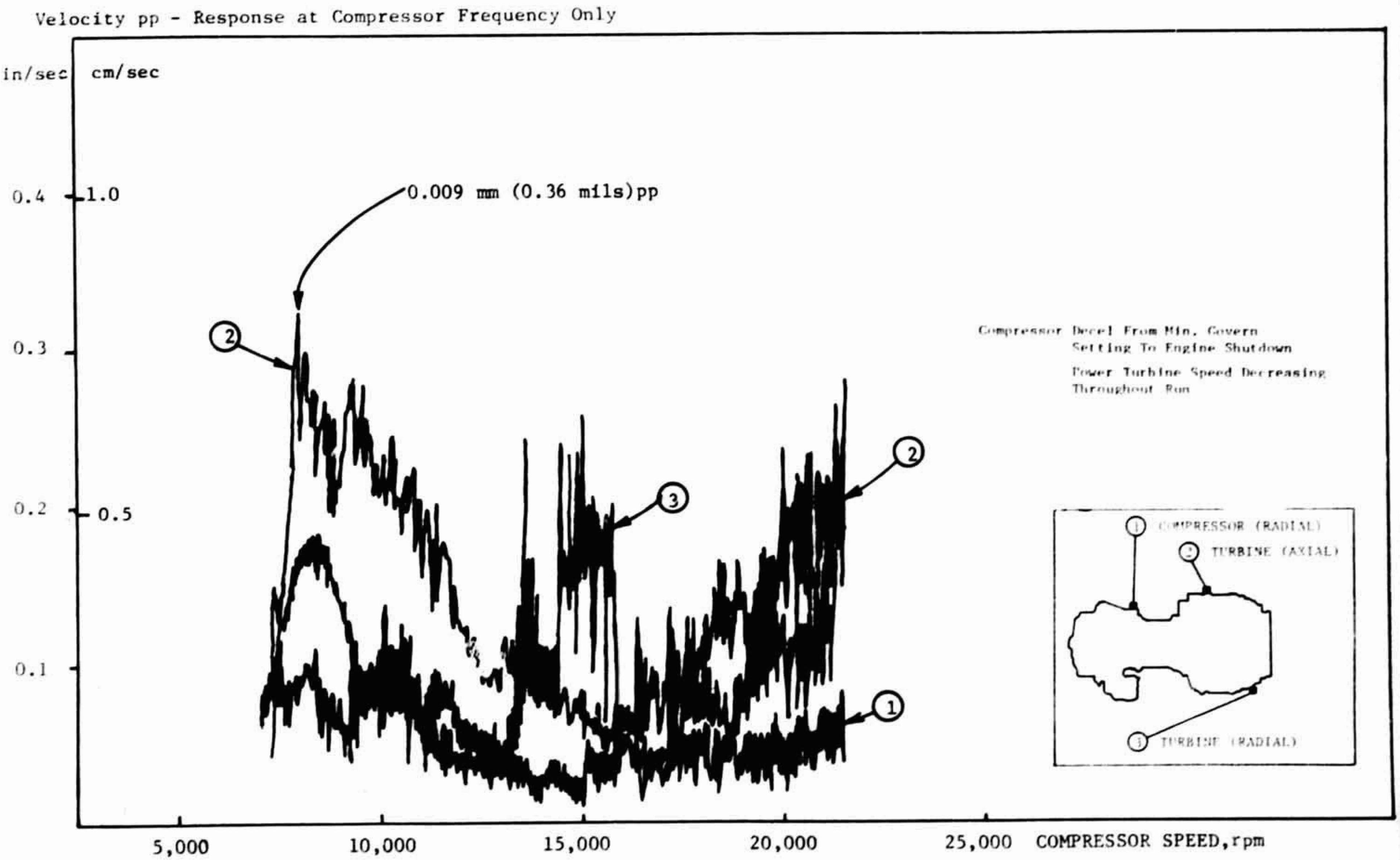

Figure 15 T53 Aircraft Vibration

lift-off. Neither in-flight nor tie-down testing was feasible because of a lack of 60Hz power for the recording instrumentation. It should be noted that the aircraft engine tested was not a particularly well balanced engine. Records of test cell vibration data for this engine showed it to be comparable to other accepted engines. CCAD personnel also indicated that few engines that meet the test cell vibration criteria have unacceptable vibration in the airframe. Comparing actual test data with amplitudes predicted by the computer simulation shows that test cell vibrations were somewhat higher than predicted while aircraft levels were lower than predicted. These differences are most likely caused by larger distributed unbalances in engine components than in the model and by aircraft support stiffnesses and damping that were not modeled into the computer program As developed, the computer model provided a key insight to understanding the dynamics of the T53 engine. "Fine tuning" of a computer model to mirror actual test data can be a time-consuming and expensive effort and was not considered to be of major importance in this program.

Test cell vibration levels were higher than aircraft levels because of installation differences between the test cell and aircraft. Figures 16 and 17 show typical aircraft mounting arrangements. The left side (Figure 16) consists of a single leg strut at the compressor housing and a double support at the gear case. The right side (Figure 17) consists of one double support at the power turbine casing. The aircraft structure to which these supports mount is made of a lightweight but rigid honeycombed metal.

Figures 18 through 20 show typical test cell mounting arrangements. When mounted in the test cell, the engine is supported in three places around the gear case by a trunnion-type yoke (see Figure 18). A rubber pad in these supports facilitates assembly and isolates engine vibration. The mounting yoke is solidly attached to a support structure. This structure is as shown in Figure 19 for the old style test cells. In modernized test cells, the structure is supported from above and positioned by an overhead crane. Figure 20 shows this support structure.

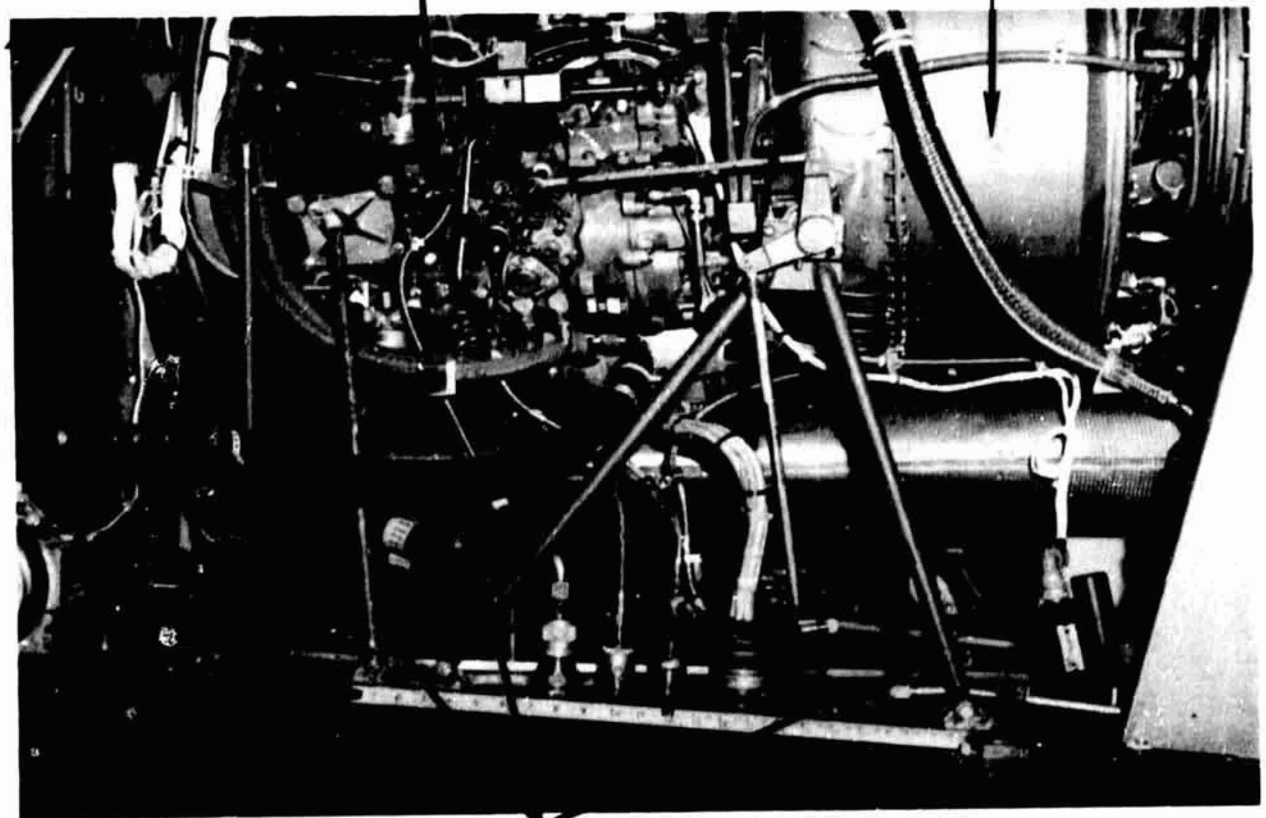

Figure 16 Typical T53 Aircraft Installation-Left Side

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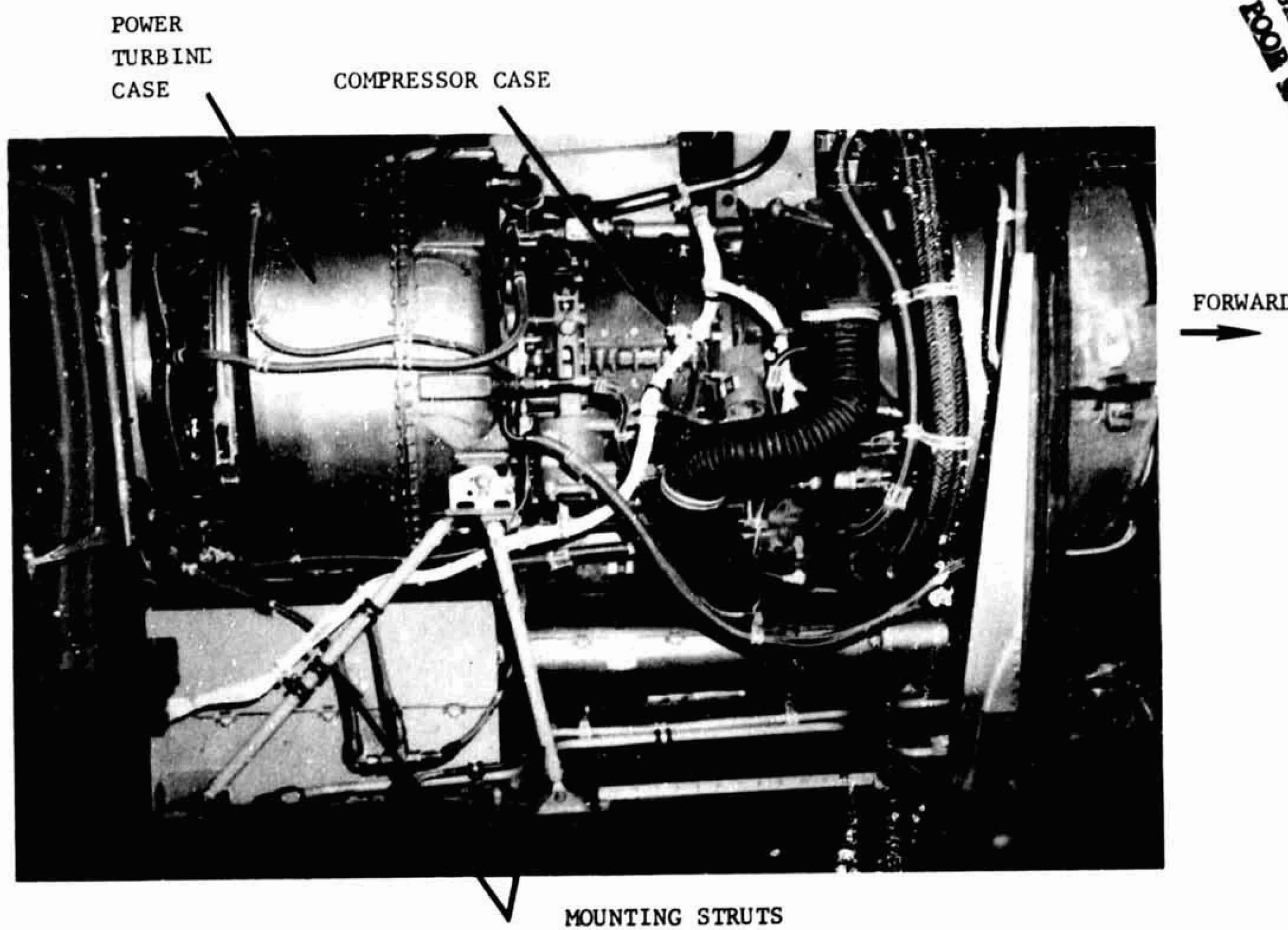

Figure 17 Typical T53 Aircraft Installation-Right Side

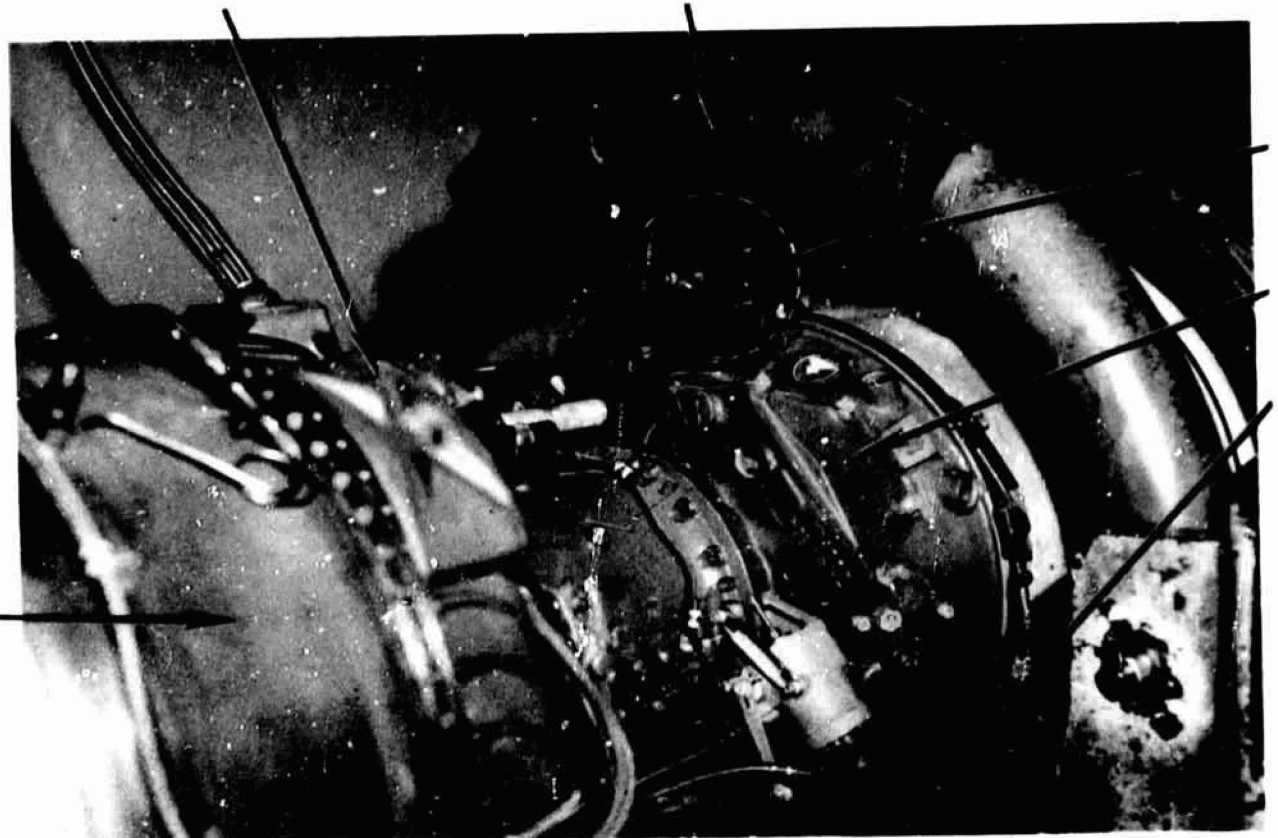

Figure 18 Typical T53 Test Cell Installation

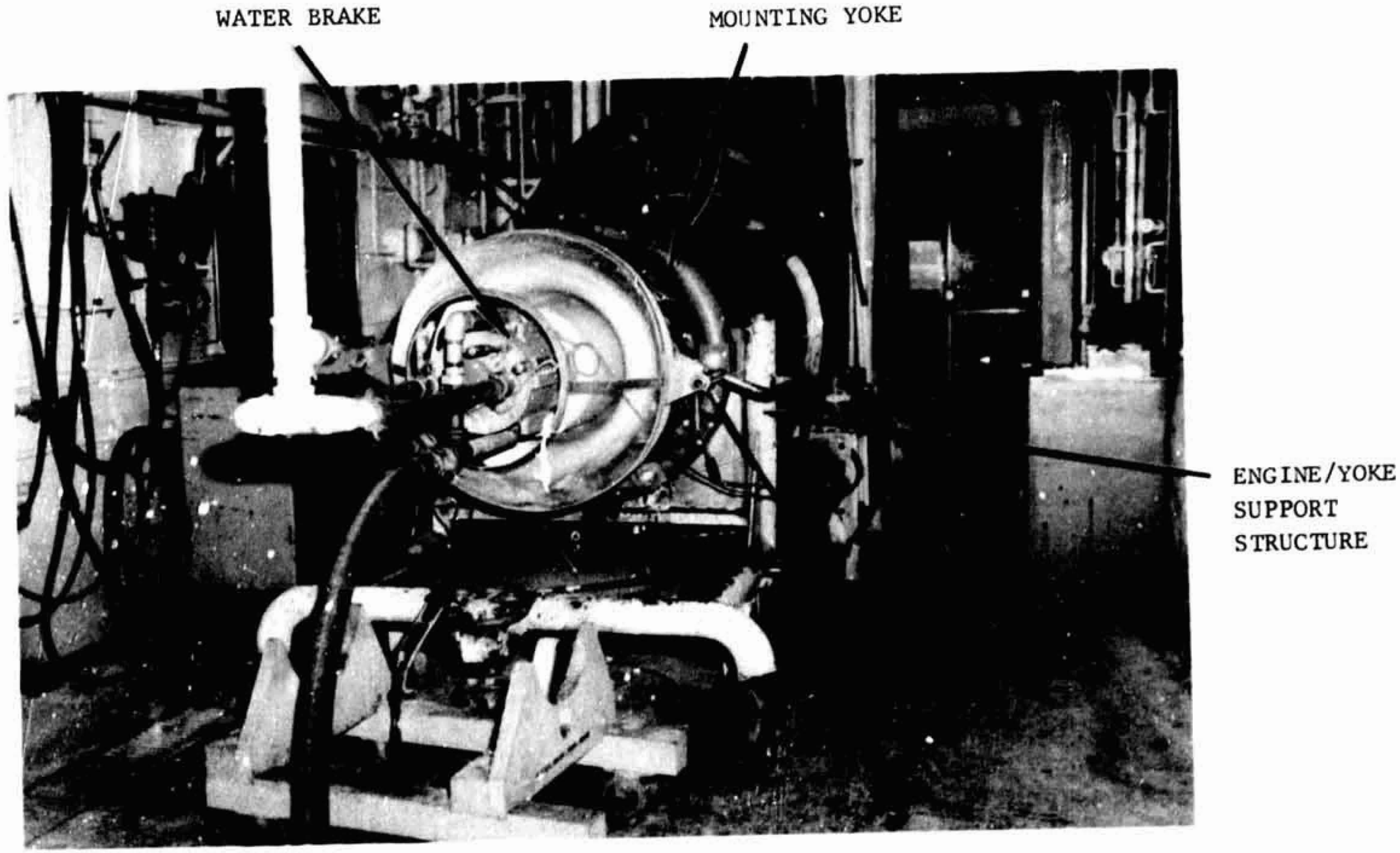

Figure 19 Typical T53 Test Cell Installation

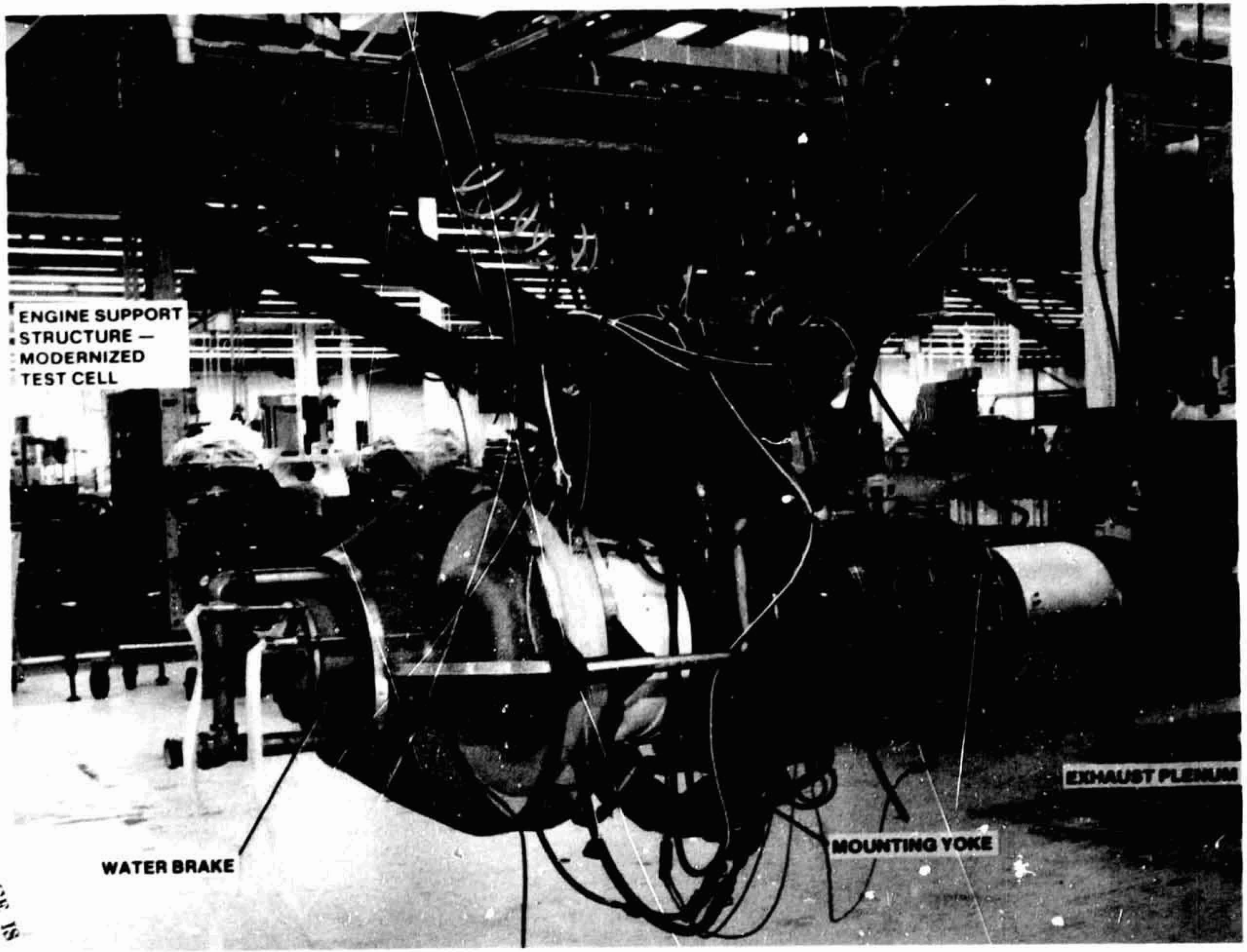

Figure 20 Typical Test Cell Engine Configuration

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Other installation differences that may affect comparability of test cell and aircraft vibration levels include the water brake and exhaust plenum used only in the test cell. (See Figures 19 and 20). In the aircraft, a torque tube and drive train take the place of the water brake. Also, the engine case is in contact with (but theoretically not supported by) fire walls and blast protectors at both the compressor and power turbine ends.

The influence of these differences between test cell and aircraft installations should be evaluated in any proposed effort to make these two test configurations dynamically similar.

SECTION III

T53 ROTOR BALANCING AND ASSEMBLY PROCEDURE AT CCAD

A. Present Balance Procedures

Figures 21 and 22 show the component parts of T53 engine rotors. The high speed rotor (Figure 21)consists of the compressor and gas generator. The low speed rotor (Figure 22) consists of the power turbine and power shaft.

Present balancing methods entail low speed subassembly balancing on Vee blocks, arbors and/or precision bearings. Wax is used to determine a "light" area 180° from the "heavy" area where grinding is done.

When balancing the power turbine, precision grade ABEC-7 ball bearings are installed on the shaft portion of the second stage power turbine rotor using aircraft spacers, locking cup, and lock nut. (Note: Both bearings are on the same side of the disc portion of the turbine rotor, thus making it an "overhung" rotor.) The "force" (i.e., static) unbalance is removed from the second stage power turbine leaving in the "couple" (i.e., dynamic) unbalance. The first stage-second stage spacer is then installed and rotated as required to obtain minimum force unbalance. If more than 2.0 gram-inches of unbalance remains, material is ground off the spacer to bring it and the second stage into less than this tolerance. The first stage turbine rotor is then bolted to the second stage rotor with the spacer serving as the locating and centering device. Then the first stage-second stage turbine assembly is balanced by removing the force unbalance, this time using the single correction plane designed into the first stage turbine rotor.

The power shaft is also balanced on a balancing machine. Vee blocks support the shaft during this two plane balance. One Vee block rides on the journal for the power shaft's steady bearing while the other Vee block rides on the pilot diameter next to the power shaft-power turbine spline. Balance corrections are made at each end of the large central diameter. Figure 22 shows areas where metal has been removed. When this shaft is supported by Vee blocks during balancing, it rotates about some center that is not the shaft's center of rotation

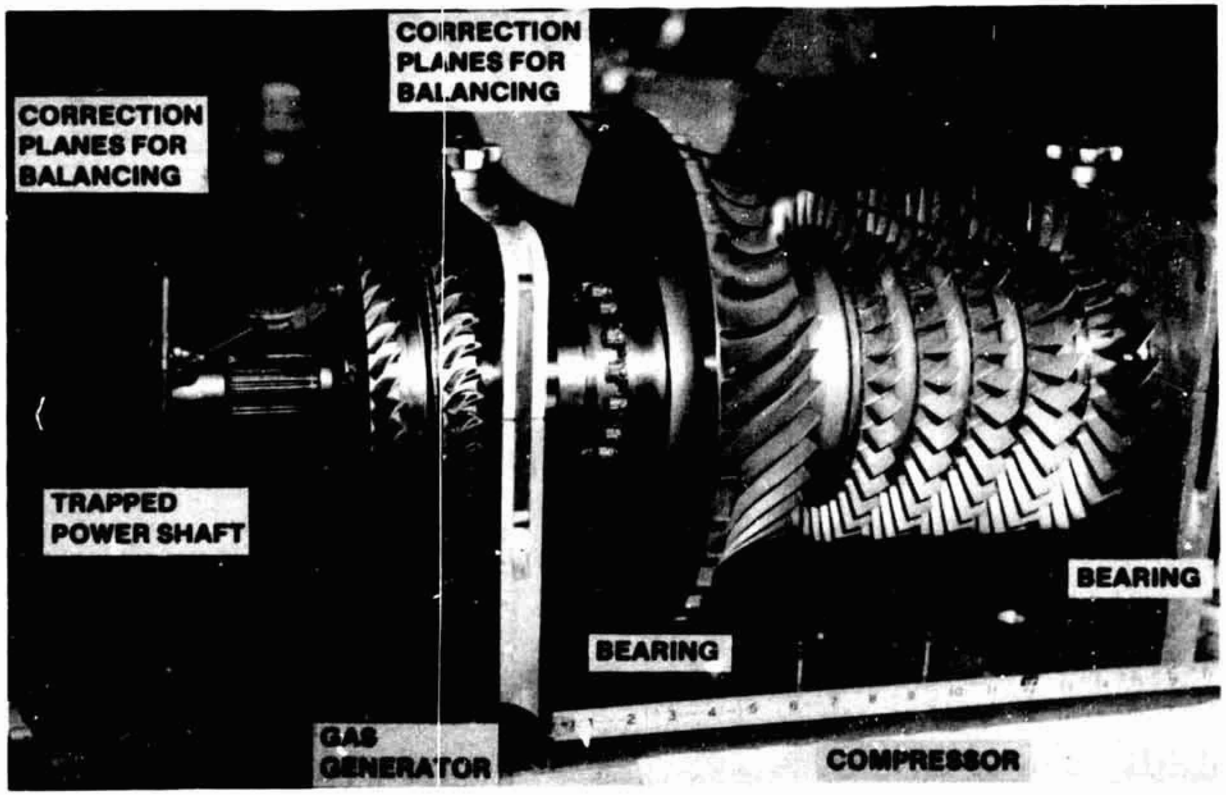
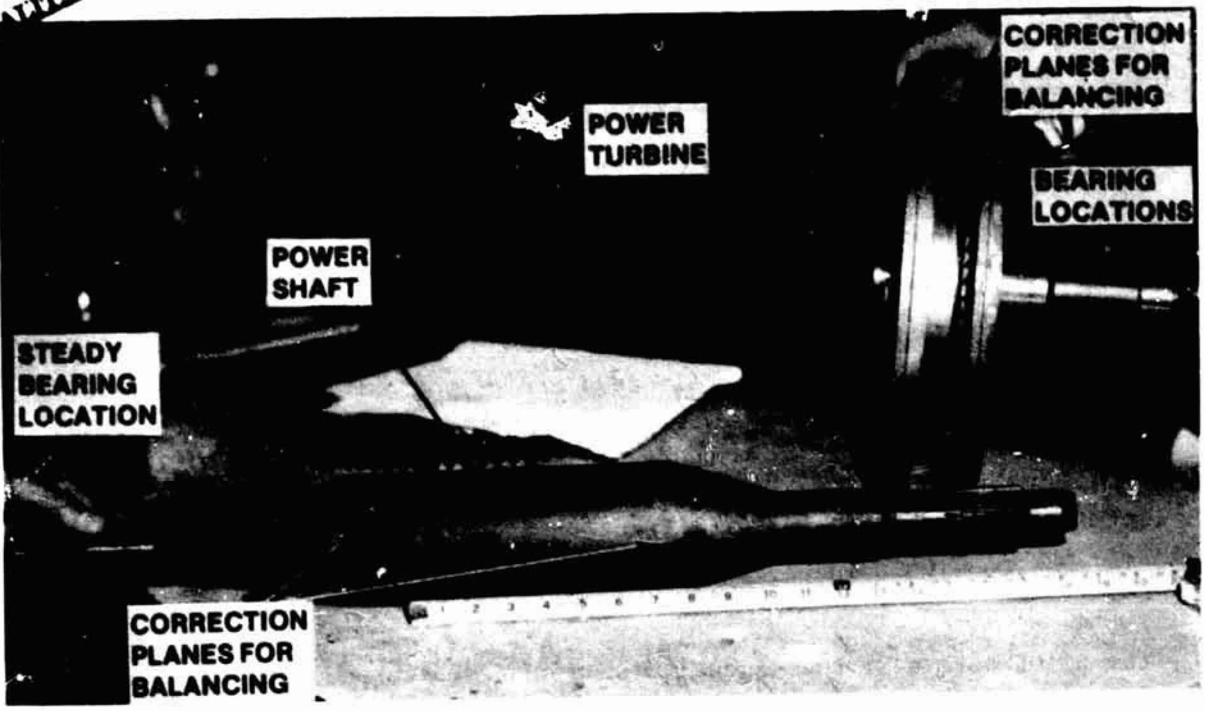


Figure 21 T53 Compressor/Gas Generator Rotor



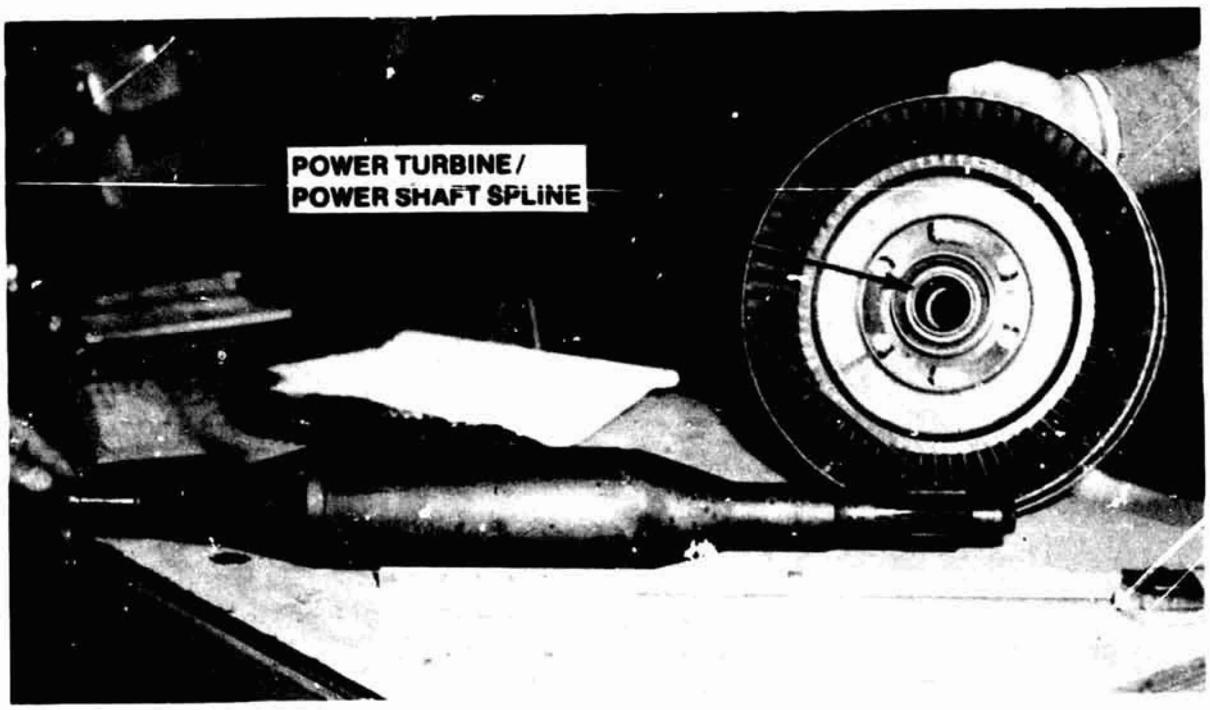


Figure 22 T53 Power Turbine/Power Shaft Rotor

in the assembled engine. Therefore, any shaft balanced by this method will not have the same degree of balance in the assembled engine.

It should be noted that since January, 1978, CCAD personnel have measured the dimensions of many overhauled power shaft-power turbine splines. These measurements have shown that very few of the splines are within the pitch diameter tolerance and print allowable run-out. Random measurements of the power shaft's pilot diameter (located aft of the external spline on the power shaft) have also shown this pilot worn undersize from .025 to .250 mm (.001 to .010 inches). When both the pilot and bore (located aft of the internal spline in the power turbine rotor) are new, there is only .075 - .14 mm (.003-.0055 inches) of diametrical clearance.

The pilot/bore wear and excessive clearances measured during overhaul could only result from shifting and wearing power shaft-power turbine splines. This phenomena is aggrevated by residual unbalance in the rotor's components. As described above, the force resulting from these unbalances increase with speed and cause the spline connection to shift radially, causing increased engine vibration and wear.

The compressor is balanced after assembly of all five axial stages, the centrifugal stage and stub shaft. This balance is accomplished in two planes on a balancing machine (using precision bearings).

The first stage gas generator turbine is balanced on a vertical balancing machine. The first stage-second stage spacer is then bolted to the first stage. The balance is then checked and the spacer rotated if required. Finally the second stage is added and the complete assembly rebalanced as necessary by grinding the second stage turbine wheel.

The balancing machines used at CCAD are relatively old and appeared to be in need of continual maintenance. Although the machines were not examined closely, it is not clear that the precision balance required of high speed machinery such as the T53 rotors can be consistently attained on these machines. Trial-and-error balancing techniques with wax trial weights in the high volume production environment at CCAD can easily result in delays and costs that may be reduced by newer balancing procedures and equipment. Approximately 3.5 manhours are presently required to balance the four rotor subassemblies - one hour each for the compressor, gas generator and power turbine, and one half hour for the power shaft. This is a creditable performance by CCAD personnel. The repetitious nature of subassembly build-up and balance, and the large volume of rebalancing work caused by test cell vibration rejections, permit CCAD personnel to become highly proficient in this area.

B. Present Assembly Procedures

Assembly of T53 engines at CCAD takes place in various departments within the plant, with the several interim balancing steps described above. Although parts of an incoming engine to be overhauled are inventoried together, there is no assurance that these parts will be reassembled on the same engine during overhaul. No pre-overhaul operational test is made. A technical evaluation of components to be overhauled is made by personnel during engine teardown.

Because of the power shaft's large central diameter, it must be placed inside the intermediate stages of the compressor (stages 2 through 5) prior to assembly of remaining compressor and gas producer components. Stages 2 through 5 are a welded unit. The first stage and stub shaft bolt to the ends of this unit. The centrifugal stage is fit and bolted to the intermediate stages. As shown in Figure 21, the power shaft is trapped inside the assembled compressor. This assembly is then installed in the engine case.

Because of the stacked nature of the engine's hot section, both the gas generator and power turbine must be disassembled for installation in the engine. After the compressor (and trapped power shaft) are installed in the engine case, the gas generator is added stage by stage with intervening stationary vanes. The first stage gas generator is located on the compressor's rear stub shaft by two bronze cones that are press fit to the stub shaft. These two cones become the first stage gas generator's aligning device. The rear face of the aft cone has pressure transmitted to it by a lock nut. After the intervening nozzle is installed, the second stage gas generator is bolted to the first stage gas generator using a spacer as a centering and locating device. The gas generator stages are permitted to have 0.05 mm (0.002 inch) runout when assembled to the compressor and no assembly balancing is done.

The engine rotors assembled thus far are as shown in Figure 21. The end of the trapped power shaft protrudes from the ends of the compressor/gas generator rotor. The power turbine is added to the aft end of the power shaft stage by stage with intervening stationary vanes. It is splined to the separately balanced power shaft and secured in place by a retainer bolt that extends through the power turbine shaft into the threaded aft end of the power shaft.

The present balancing and assembly techniques used at CCAD would, at best, be marginally acceptable for operation in a low speed engine in which rotors operate below bending critical speeds and truly remain rigid bodies. However, as results of the analytical and experimental studies discussed in Section II show, the T53 power turbine and shaft in particular traverse several bending modes in the normal operating speed range of the engine. It is recommended that the power turbine rotor module be high speed balanced as an assembly through all of the critical speeds within the engine's operating speed range. Correction planes should include both ends of the power shaft's large central diameter and the second stage power turbine wheel. Because the compressor rotor module remains rigid throughout the engine's operating speed range, a quality low speed balance should result in satisfactory operation at speed. It is recommended that the compressor-gas generator rotor module be low speed balanced as an assembly before installation into the engine. Correction planes should be on the first stage compressor and first stage of the gas generator turbine.

SECTION IV

EVALUATION OF PRESENT ENGINE VIBRATION TEST PROCEDURE AND DIAGNOSTIC CAPABILITY AT CCAD

A. Present Engine Vibration Test Procedure

1. Test Setup

Engine vibration tests at CCAD are initially performed in conjunction with other performance calibrations. If the engine is rejected because of excessive vibration, subsequent retests are solely vibration checks. Vibration measurements are taken manually by reading overall levels displayed by a vibration meter. The vibration sensors used in the test cell generate an electrical signal proportional to the velocity of the engine case at the pick-up locations. The vibration meter integrates this signal and displays displacement levels in mils, peak to peak. Figure 9 shows the locations of the vibration sensors.

Of particular interest is the location of sensor No. 3. It is attached to the end of a three-inch long oil drain pipe that extends from the bottom of the turbine case. This mounting arrangement is very flexible and can easily distort true indications of engine case vibration*. Recent unpublished studies by the Department of the Navy at their Philadelphia Engineering Center have also shown that velocity sensors are susceptible to erroneous readings in the presence of vibration in axes not parallel to the principal axis of the velocity sensor.

It is recommended that alternative mounting locations for sensor No. 3 be evaluated. The effectiveness of the other three sensors to depict rotating unbalance forces should also be evaluated. Consideration should also be given to replacing the velocity sensors now used at CCAD. Accelerometer technology has made many advances in recent years. This type of vibration sensor is probably more ideally suited to both test cell and aircraft vibration monitoring.

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*An inspection of an operating engine at CCAD disclosed visual sensor vibrations of approximately 1.5 mm (1/16 inch). See Section II.

2. Test Procedure

Specified engine speeds for vibration :ts are shown below:

TABLE 2

T53 ENGINE SPEEDS FOR VIBRATION TESTS

Test Speed No.	Compressor Rotor Speed	Power Turbine Rotor Speed
1	75.0% (18,860 rpm)	97.3% (20,510 rpm)
2	80.0% (20,120 rpm)	80.0% (16,860 rpm)
3	85.0% (21,380 rpm)	97.3% (20,510 rpm)
4	90.0% (22,635 rpm)	94.8% (19,980 rpm)
5	90.0% (22,635 rpm)	97.3% (20,510 rpm)
6	90.0% (22,635 rpm)	100.4% (21,160 rpm)
7	95.0% (23,890 rpm)	97.3% (20,510 rpm)

Figure 23 presents the maximum acceptable vibration levels for T53 engines in the test cell. Figure 24 presents the results of vibration tests for a random sample of 20 of the 1,200 T53 engines tested in 1977. The shaded area on the curve for each sensor shows the envelope of measured vibration levels. The average vibration level and the applicable vibration limit are also shown on each curve. These sample data indicate the long term trend of engine vibration levels at CCAD: very low vibration levels at sensors Nos. 1 and 4, somewhat higher levels at sensor No. 2, and highest levels at sensor No. 3. Sensor No. 3 (attached to the turbine case) shows both the highest average and the greatest variance in vibration levels. Figure 24 shows that the maximum levels at sensors Nos. 1 and 4 and the average levels at sensor No. 2 did not exceed 0.025 mm (1 mil). Sensor No. 3, however, had average levels that were in excess of 0.05 mm (2 mils) at all vibration test speeds. It is important to note that the highest average vibration levels detected were those at sensor No. 3 for 80 percent power turbine speed. This speed (16,860 rpm) excites response to the engine's fourth critical speed, calculated to occur at 17,600 rpm and experimentally detected at approximately 17,200 rpm*. CCAD test cell operators have confirmed that engine operation at "80-80" (80 percent speed of both the compressor rotor and power turbine rotor - see Table 2) is the most common speed at which the vibration limit is exceeded.

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* See earlier discussion of analytical and experimental data, Section II.

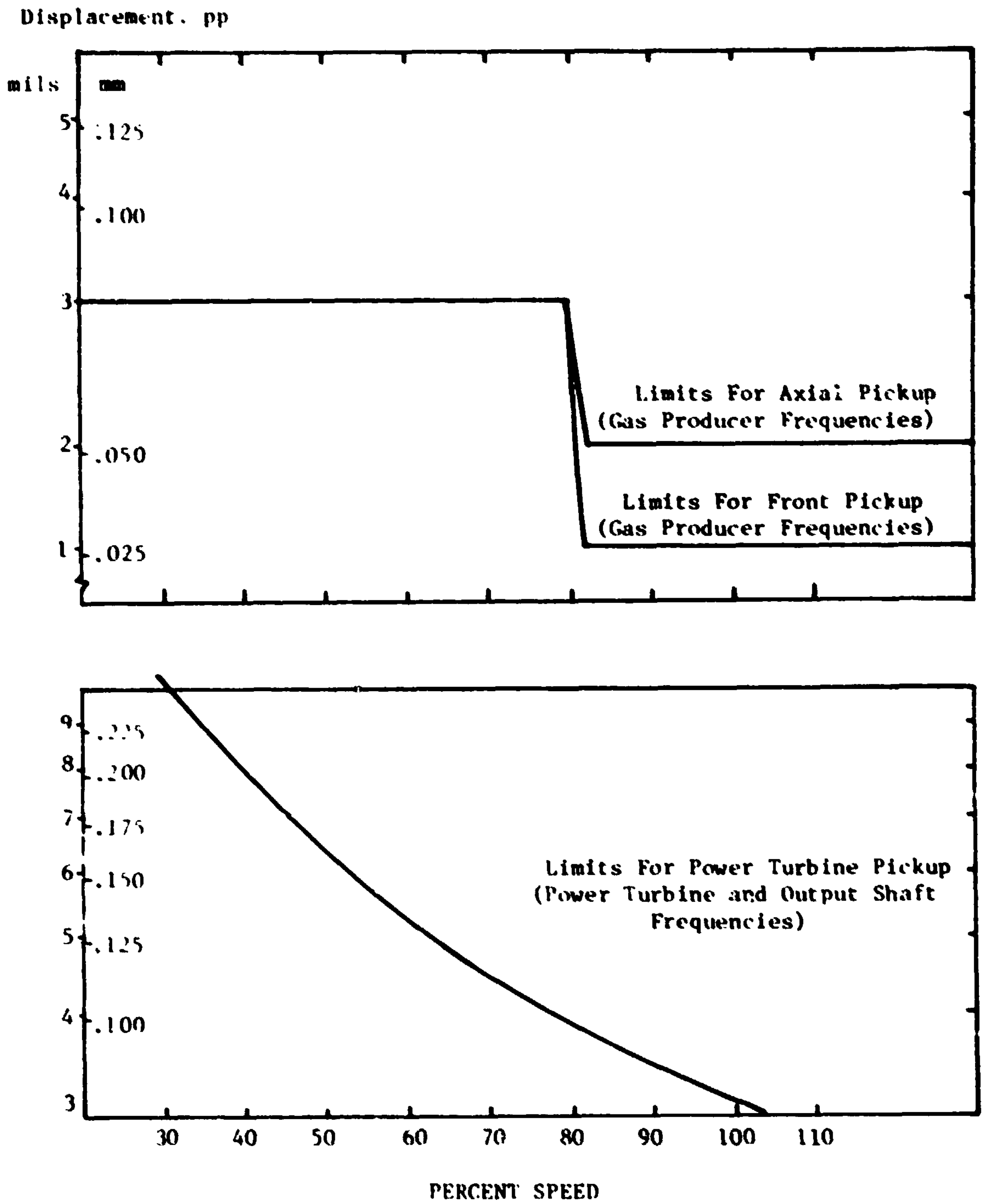

Figure 23 T53 Test Cell Vibration Limits

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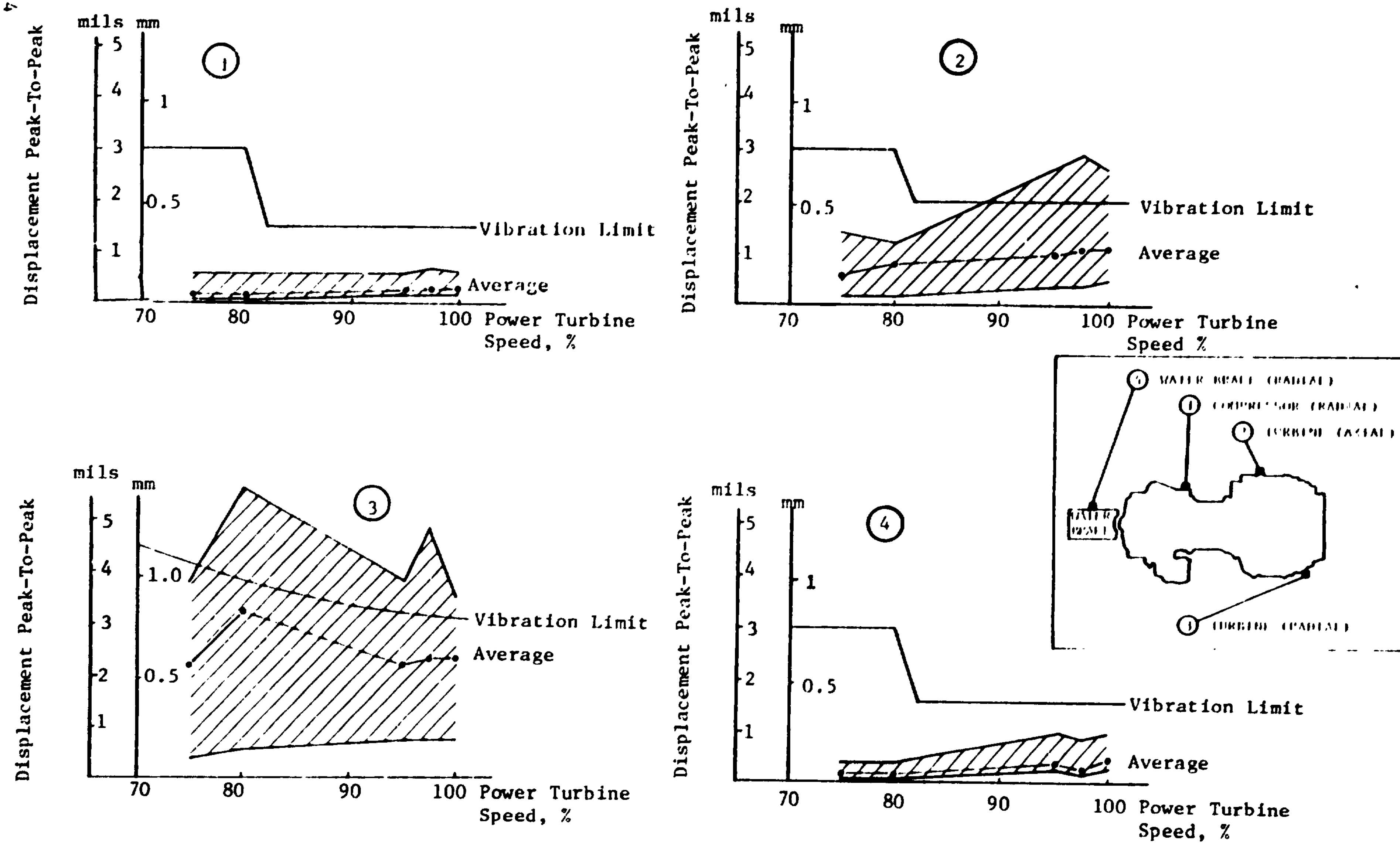

Figure 24 Test Cell Vibration Performance - T53 Engines

The following table summarizes the 1977 CCAD repair records for the T53-L13 engine.

TABLE 3

T53 VIBRATION REJECTION RATE - 1977

Total Engines Tested--------1198

Total Engines Rejected------ 469 (39% of Total Tested)

Total Engines Rejected
Because of Vibration-------- 278 (60% of Total Rejected)
(23% of Total Tested)

Sensor Causing Vibration Rejection:

Sensor 1 -	5	(2% of Vibration Rejects)
Sensor 2 -	55	(20% of Vibration Rejects)
Sensor 3 -	218	(78% of Vibration Rejects)
Sensor 4 -	0	

Table 3 shows that vibration at sensor No. 3 causes the vast majority of the vibration rejections.

As shown in Figure 9, sensor No. 3 measures radial vibration at the power turbine case. Remedial action to correct excessive vibration usually involves the power turbine. A review of test data for the three month period of February - April, 1977 disclosed that 85 percent of the vibration failures were caused by excessive vibration at the power turbine. The remaining 15 percent were caused by excessive axial vibration (sensor No. 2). The following table presents remedial actions takento correct engines rejected because of excessive vibration. As the table shows, more than 75 percent of the radial vibration problems were corrected by action involving the power turbine. Axial vibration problems exhibited no significant tendency toward any particular corrective action.

TABLE 4

REMEDIAL ACTIONS TO CORRECT T53 VIBRATION
FEBRUARY - APRIL 1977

Remedial Action	Effectiveness of Remedial Action in Correcting Excessive Vibration: Radial Vibration		Effectiveness of Remedial Action in Correcting Excessive Vibration: Axial Vibration
Replace Power Turbine Assembly	41.7%	76.7%	0.0%
Index Power Turbine Assembly	18.3%		18.2%
Rebalance Power Turbine Assembly	16.7%		18.2%
Replace Bearings	15.0%		27.2%
Replace Accessories	3.3%		18.2%
Replace or Balance Compressor	3.3%		18.2%
Replace Power Shaft	1.7%		0.0%
	100.0%		100.0%

Recommendations outlined above concerning modified assembly and balance techniques and evaluation of the power turbine-power shaft spline will reduce test cell vibration levels. These recommendations are discussed in detail in Section VI.

B. Present Engine Vibration Diagnostic Capability

Many of the test cells at CCAD are being automated to permit faster turnaround for engine testing. However, the vibration monitoring aspects of the acceptance tests are conducted using just one vibration meter with several high-pass filters. The meter reads overall vibration levels in mils, peak to peak. The filters are used to screen out low frequency mount resonances and water brake-related frequencies. The only diagnostic tool in use is a portable frequency meter. This meter can be used to individually determine vibration levels as a function of engine tachometer speed. Lists containing the ratio of the rotating speeds of major engine components to the tachometer signal are available to assist in determining the frequency and cause of excessive vibration levels. A key source of diagnostic analysis is the test cell operator. Engines may be reworked based on a decision by the operator that an engine is "too rough" to test.

It should be noted that, as part of the test cell modernization effort, a centralized analog tape secondary and spectrum analysis system has been partially installed at CCAD. However, interface problems between this system and the data acquisition hardware in the test cells have prevented this equipment from being fully utilized.

Recent advances in both vibration instrumentation and diagnostic capabilities could be effectively implemented at CCAD. Automated vibration data acquisition, display, and diagnostics could be configured to interface with the computerized capabilities available in modernized test cells. Capabilities of this kind would permit instantaneous display of vibration signatures, comparison to both the vibration limit and other accumulated engine data, determination of the cause of excessive vibration and remedial action to be taken. Overall test cell time rework cycle and the amount of retesting required could be significantly reduced. It is recommended that an automated vibration data acquisition, display, and diagnostic system be incorporated into the modernization program for CCAD test cells. The partially installed equipment now at CCAD could form the central core upon which an automated system could be built.

SECTION V

ECONOMIC IMPACT OF PRESENT T53 ROTOR BALANCE, ASSEMBLY, AND ENGINE VIBRATION TEST PROCEDURE AT CCAD

The total cost to overhaul a T53 engine is approximately $19,600. Direct labor and overhead maintenance costs are approximately $27 per hour. As noted in Table 3, 278 of the 1.198 T53 engines tested at CCAD in 1977 were rejected because of excessive vibration. The following table presents a breakdown of manhours required to rework a T53 engine rejected in the test cell because of excessive vibration:

TABLE 5

AVERAGE CCAD MANHOURS REQUIRED TO REWORK A T53 ENGINE

Work Center	Average Manhours
Test Cell (Troubleshooting)	5.7
Hot Section Shops	5.8
Small Parts and Housing Repair	2.0
Rework of Reject Parts	15.0
Balance Shops	8.7
Engine Assembly	9.2
Total	46.4

In addition to the 46.4 manhours estimated above, additional materials and parts are required for each engine rework. Each reworked engine must also be reinstalled and retested in the test cell. The following information can be used to calculate the total cost of reworking and retesting an engine rejected because of excessive vibration:

$$\left[\frac{\$27}{\text{manhour}} \times \frac{46.4\text{ manhours}}{\text{engine}}\right] + \left[\frac{\$150}{\text{test cell hour}} \times \frac{1.25\text{ test cell hours}}{\text{engine}}\right] + \frac{\$100\text{ materials and parts}}{\text{engine}} = \$1{,}540/\text{engine}$$

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During 1977, 278 T53 engines at CCAD were rejected because of excessive vibration. Thus, the total cost associated with vibration rejection at CCAD is equal to $1,540/engine x 278 engines, or approximately $430,000.

Because of the large volume of T53 engines tested at CCAD, any corrective action taken to reduce the vibration rejection rate can have a major impact on overall cost savings. Implementation of the recommendations made in this report will result in significant cost savings by accomplishing the following:

- Increasing the "first pass" acceptance rate of T53 engines. Unbalance vibration levels reduced because of better quality rotor balancing.
- Decreasing the number of times a rejected engine must be retested and reworked to achieve required vibration levels. Diagnostic and symptom fault information pinpoint causes of excessive vibration and outline remedial action.
- Testing engines in a manner which reflects their aircraft usage. Potential vibration rejections identified before they surface in the field, and vibration rejections caused by nonrelated test cell equipment eliminated.

SECTION VI

CONCLUSIONS

A. General

Work conducted under the present contract has resulted in a basic understanding of the rotordynamic characteristics and how they interact with the overhaul environment of T53 engines at CCAD. The high vibration rejection rates at CCAD are caused by several factors, which are discussed below. Corrective action should be taken in these areas to assure satisfactory vibration reduction in test cells that are representative of aircraft installations.

B. Computer Simulation

Computer simulation of the T53 has disclosed that several critical speeds are present within the engine's operating speed range. The analytical mode shapes at these critical speeds have shown that significant component bending is encountered, particularly in the power turbine rotor module. The critical speeds and resulting mode shapes depend upon whether the engine is tested while installed in an aircraft or in a test cell. The predicted critical speeds agree closely with experimental data.

Verification of critical speeds predicted by the computer model was accomplished by measuring actual vibration levels of operating engines in both test cells and aircraft. The effect of power turbine unbalance dominates the engine's vibration signature.

C. Spline Shift

Analysis of the experimental data has also disclosed a probable spline shift at the power turbine-power shaft joint. This phenomenon, if representative of a significant portion of overhauled T53 engines, will affect other attempts to reduce engine vibration rejection rates.

D. Test Cell-Airframe Mounting

Typical test cell vibration levels are higher than those measured on engines installed in aircraft. These differences are caused by several installation differences in test cell and aircraft installations, including engine mounting configurations and test cell accessories (i.e., a water brake used to load the engine and an exhaust plenum). Test cell engine mounting configurations should be modified to dynamically reflect actual aircraft installations. Such modifications will allow more realisitic engine tests, detection of potential problems before they surface in the field, and reduction of premature engine removals.

E. Assembly Balance of Power Turbine

Balancing techniques presently used at CCAD m.. allow residual unbalance in an assembled engine. The power shaft is supported by Vee blocks while being balanced. This allows the shaft to rotate about a center that is not the shaft's center of rotation in the assembled engine. A shaft balanced by this method will not have the same degree of balance in the assembled engine. This inaccuracy could be eliminated by performing a high speed balance of the assembled power turbine and power shaft rotor. High speed balancing is particularly applicable because this rotor module passes through bending modes during normal operation. High speed balancing can reduce unbalance forces to levels that cannot be attained with conventional low speed rigid rotor balancing.

F. Assembly Balance of Compressor

Although the dynamic operating characteristics (bending modes) of the power turbine-power shaft rotor indicate a need for high speed balancing, the analytical mode shapes of the compressor rotor indicate that this rotor remains essentially rigid throughout its operating speed range. Therefore, a quality low speed balance should result in satisfactory operation at speed. The compressor-gas generator rotor should be low speed balanced as an assembly before installation into the engine.

G. Vibration Monitoring Locations

The general location of the sensor that measures radial vibration at the power turbine case is good from a dynamic standpoint (i.e., the engine case in this area is sensitive to rotor vibration). However, the mounting bracket for this sensor is very flexible and at some frequencies may distort force indications of engine vibration. Alternate locations for this sensor include an assembly flange or other bolted component in the vicinity with the need for thermal isolation. An adapter similar to that and for pick-up No. 1 or 2 (see Figure 9) could then by used. The present locations for sensor No. 2 is considered adequate. The present location for sensor No. 1 is considered adequate but its capability to sense unbalance forces is questionable. The use of sensor No. 4 (located on the water brake) is of questionable value in monitoring engine vibration, but may be of some value in determining water brake included vibrations.

Attempting to reconfigure/relocate the test cell instrumentation to better reflect engine performance conditions in the aircraft is considered to be infeasible because of the dissimilarities between test cell and aircraft mountings. Stiffness and damping differences, when coupled with required test cell accessories, significantly change engine case vibration response in the two installations. Test cell mounting configurations should be modified to better reflect aircraft installations.

H. Vibration Sensors

Recent studies by the Department of the Navy have also shown that velocity sensors are susceptible to erroneous readings in the presence of vibration in axes not parallel to the principal axis of the velocity sensor. Consideration should be given to replacement of the velocity sensors now used at CCAD. Accelerometer technology has made many advances in recent years and may be more ideally suited to this application. Attaching an accelerometer next to a velocity sensor on an operating engine would help determine the accuracy of the present sensors.

I. Test Procedure and Vibration Rejections in the Test Cell

The large majority of vibration rejections are caused by excessive vibration at sensor No. 3 (see Table 3). These rejections are primarily caused by unbalance response of the power turbine rotor module. This unbalance force causes vibration problems in three distinct ways. First, the unbalance in the rotor causes higher synchronous vibration levels at all engine speeds. Second, rotor unbalance excites unacceptable response near engine critical speeds. Third, rotor unbalance forces may excite sensor support structures, thereby causing erroneous indications of vibration levels.

These three reactions to power turbine rotor unbalance may be viewed as separate sources of vibration. Action may be taken to reduce or eliminate several of these sources (i.e., move engine vibration test speeds away from engine critical speeds, alter the support structure for sensor No. 3 or modify test cell engine mounting to reflect aircraft installations, thereby changing critical engine speeds). This approach, however, is treating the symptoms of the problem and not the cause. The primary cause of a large portion of T53 vibration rejections is unbalance of the power turbine-power shaft rotor module. The bending characteristics of the rotor necessitate high speed balancing. Effective high speed balancing of the power turbine rotor module will eliminate much of the T53 vibration problem at CCAD.

The present engine speeds used to check test cell vibration are considered to be adequate. Indeed, from a rotordynamic standpoint, it is desirable to check vibration at speeds close to shaft criticals, for it is at these speeds that shaft unbalance is most pronounced.

Historical data indicates that few engines which have been accepted in the test cell perform unacceptably in the aircraft. Lowering test cell vibration limits would, therefore, tend to increase the already excessive test cell vibration rejection rate without any noticeable improvement in the field. An investigation of raising test cell vibration limits was hampered by unavailability of in-field data for critical parts failure rates and vibration degradation in service. The present vibration acceptance criteria used in test cells at CCAD is, therefore, considered to be adequate for an interim period of time until the test cell procedure and equipment more closely reflect the aircraft installation.

J. Diagnostics

Operable test cell diagnostic capabilities at CCAD consist of a frequency meter and the experience of test personnel. Modernized vibration data acquisition, display, and diagnostics should be implemented in modernized test cells at CCAD. Overall test cell time and the amount of retesting required would be significantly reduced. The partially installed centralized analog tape recording and spectrum analysis system now at CCAD could form the central core upon which an automated system could be built.

K. Economic Impact

Each T53 engine rejected because of excessive test cell vibration results in an additional $1,540 cost for rework and retest. Based on the reported test volume and vibration rejection rate for 1977, the total cost associated with T53 vibration rejection at CCAD was approximately $430,000. Based on often reported vibration rejection rates of 40 percent, this figure is approximately $750,000 per year. Implementation of the recommendations set forth in this report will result in significant cost savings by increasing the "first pass" acceptance of tested engines, decreasing the number of times a rejected engine must be reworked and retested, identifying potential vibration rejections before they surface in the field, and eliminating vibration rejections caused by nonrelated test cell equipment.

SECTION VII

RECOMMENDATIONS

In 1977, T53 engine vibration rejections cost the overhaul center approximately \$430,000. This figure has at times been reported to reach approximately \$750,000 per year. Implementation of the recommendations summarized below will result in substantial cost savings. This would be accomplished by increasing the "first pass" acceptance of tested engines, decreasing the number of times a rejected engine must be reworked and retested, identifying potential vibration rejections before they surface in the field, and eliminating vibration rejections caused by nonrelated test cell equipment.

A. Optimum Assembly and Balance of Power Turbine

Results of the computer simulation of the T53 engine have indicated that there is significant power turbine rotor module bending at speeds within the engine's operating speed range. High speed balancing techniques should be applied to this assembled rotor module. Unbalance-caused vibration of this rotor is the primary cause of the present T53 vibration rejection rate.

Assembled power turbine rotors (with intermediate static structure) and power shaft should be balanced as an assembly using multiplane multispeed balancing methods. The high speed balancing machine should be of a special design that will permit safe, repeatable and timely balancing of assembled rotors. Balancing could be accomplished in situ by removing metal at the ends of the power shaft's large central diameter and on one of the power turbine stages. Automatic metal removal (laser) and adaptability to other types of power turbine modules would provide additional advantages. The power shaft could then be match marked, removed and reassembled to the power turbine stages in the engine case without further disassembly.

B. Optimum Assembly and Balance of Compressor

The compressor-gas generator rotor module remains essentially rigid at all engine speeds. However, a quality low speed assembly balance would also reduce vibration levels. Low speed assembly balancing techniques should be applied to this assmebled rotor.

Assembled gas generators (with intermediate static structure) and compressor rotors should be low speed balanced as a module by removing metal at both the first compressor stage and the first gas generator stage. If balanced with an entrapped power shaft, the balanced compressor could then be installed in the engine case without disassembly.

C. Evaluation of Power Turbine - Power Shaft Spline

Analysis of the experimental data gathered during this study has disclosed a probable spline shift at the power turbine-power shaft joint. Field measurements by CCAD personnel confirm that significant component wear has occurred in most overhauled engines. This phenomenon, if representative of a significant portion of overhauled T53 engines, will impact other attempts to reduce engine vibration rejection rates. It is recommended that the measurements and replating efforts begun at CCAD be continued. It is also recommended that the dynamic effect of the spline shift and how it is effected and controlled by unbalance be experimentally evaluated in any follow-on effort.

D. Modification of Test Cell Engine Mounting

Because of differences in test cell and aircraft installations, typical test cell vibration levels are higher than typical aircraft vibration levels. Test cell engine mounting configurations should be modified to dynamically reflect actual aircraft installations. Such modifications will allow engines to be tested in a manner similar to their aircraft usage, as well as permit detection of potential problems before they surface in the field.

E. Incorporation of Test Cell Diagnostics

Automated vibration data acquisition, display, and diagnostics should be configured to interface with the computerized capabilities available in modernized test cells at CCAD. Capabilities of this kind will permit instantaneous display of vibration spectrums, comparison to both the vibration limit and other accumulated engine signatures, and determination of the cause of excessive vibration and remedial action to be taken. Overall test cell time and the amount of retesting required would be significantly reduced.

F. Evaluation of Velocity Sensors

Recent studies have shown that velocity sensors are susceptible to erroneous readings in the presence of vibration in axes not parallel to the principal axis of the velocity sensor. Consideration should be given to replacing the velocity sensors presently used at the overhaul center.

The mounting arrangement for the sensor that measures power turbine case vibration is very flexible and may, at certain speeds, distort true indications of engine vibration. Alternate sensor mountings should be evaluated.

G. Optimum Production Tests, Instrumentation and Repair Procedures

Evaluation and implementation of the recommendations outlined above will result in an optimum method for evaluating the vibration characteristics of overhauled T53 engines at CCAD. An optimum test cell production test would only involve engines with shafts that had been assembly balanced at high speed (power turbine) or low speed (compressor-gas generator). An automated vibration data acquisition system would acquire and display on-line vibration data and compare it to a data base of similar engines and any previous runs of the engine being tested. If the engine's vibration levels are acceptable, its vibration spectrum can be added to the data base as well as filed for future comparisons at CCAD or to in-field vibration checks. Engines with unacceptable vibration levels will be automatically diagnosed and the likely causes of the over spec vibration and most effective repair procedures identified.

Incorporation of these optimum procedures will increase the first pass acceptance of tested engines, decrease the number of times a rejected engine must be reworked and retested, identify potential vibration rejections before they occur in the field and eliminate vibration rejections caused by nonrelated test cell equipment.