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DEVELOPMENT OF A MULTIPLANE-MULTISPEED BALANCING SYSTEM FOR TURBINE ENGINES

by Michael R. Martin

MECHANICAL TECHNOLOGY INCORPORATED

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16. Abstract

A prototype high speed balancing system was developed under this contract for assembled gas turbine engine modules at the Corpus Christi Army Depot (CCAD). The system permits fully assembled gas turbine modules to be operated and balanced at selected speeds up to full turbine speed. The balancing system is a complete stand-alone system providing all necessary lubrication and support hardware for full speed operation. A variable speed motor provides the drive power. A drive belt and gearbox provide rotational speeds up to 21,000 rpm inside a vacuum chamber. The heart of the system is a dedicated minicomputer with attendant data acquisition, storage and I/O devices. The computer is programmed to be completely interactive with the operator.

The system was installed at CCAD and evaluated by testing 20 T55 power turbines and 20 T53 power turbines. Engine test results verified the performance of the high speed balanced turbines.

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1.0 SUMMARY

Rotordynamic analysis of the T53-L-13B engine showed that the power turbine passes through a bending critical speed. A similar analysis performed on the T55-L-11D indicated that its power turbine responds to a bending critical speed and is no longer rigid at its maximum speed. Based on these findings a high-speed balancing machine to handle both rotor types was designed and constructed. The system was designed for use at the overhaul facility at Corpus Christi Army Depot (CCAD).

The balancing system is a stand-alone system including a drive motor, speed increaser, vacuum chamber, and bearing support structure as well as the associated auxiliary systems such as the lubrication system and vacuum pump. The control console allows the operator to remotely control the rotor speed and also includes monitoring of vital test parameters. A dedicated minicomputer handles data acquisition and determines the necessary balance correction for a precision high-speed balance using the stored influence coefficient technique[1]. The system software was designed for use by a semiskilled operator and requires no previous computer experience. Expandability to T700 power turbines and automated metal removal was anticipated and is provided in the design of the hardware.

After development of the system at Mechanical Technology Incorporated (MTI), it was shipped to CCAD and evaluated over the period of a year. During the testing period a total of 40 power turbines were tested in the high speed balancing system (HSBS), and the corresponding 40 engines were analyzed in the engine test cell to correlate the vibration levels recorded in the HSBS with the power turbine's contribution to the overall engine test cell vibration levels. The test program clearly demonstrated the ability of multiplane/multispeed balancing procedures to balance production hardware in the overhaul cycle.

The successful correlation of HSBS vibration levels with engine test cell vibration levels allowed CCAD personnel to use the HSBS to evaluate the effect of other changes to the T53 power turbine balance procedure. Using this feature of the HSBS, a change in the current low-speed balance procedure for T53 power turbines was evaluated and found to produce significantly smoother running engines than those currently being overhauled. Using the HSBS not to balance but to test turbines to their full operating speed demonstrates an important benefit derived from the ability to take meaningful data representative of behavior in an engine throughout the full operating speed range of the power turbine.

2.0 INTRODUCTION

In March 1977, Mechanical Technology Incorporated contracted to perform a rotordynamic evaluation of the T53-L-13B helicopter engine and a review of the test cell and balancing procedures being used in the overhaul procedure. The analysis predicted multiple critical speeds in the operating range of the power turbine. A review of T53 vibration test data from test cells and from the airframe was concluded in 1978 and confirmed the results of the analysis. NASA CR-135449 is the final report for the three-task effort.[2]

A companion project, conducted under NASA Contract NAS3-19408, demonstrated that the T55-L-11D power turbine operated in the flexible regime due to a bending critical speed above its maximum operating speed.[3]

The scope of this project, NAS3-20609, was then extended from the three original tasks discussed in [2] to include the design, manufacture, and testing of a multispeed/multiplane balancing system for use on both T55 and T53 power turbines.

The following sections summarize the program history for the five-task design and implementation effort. Detailed results for these tasks are presented in Section 3.

2.1 Program History - Balancing Facility Tasks

2.1.1 Detail Design of High-Speed Balance Equipment

The preliminary design was begun in September 1978. The first coordination meeting was held at CCAD October 11, 1978. Incorporation of the input from CCAD personnel and input from potential hardware suppliers into the preliminary design was accomplished by December 12, 1978 and the preliminary design was submitted to CCAD on this date. Ordering long lead items was authorized by NASA at this time. A complete set of drawings was forwarded to NASA in January 1979. Design approval from NASA was received in March.

2.1.2 Fabrication and Checkout of High-Speed Balance Equipment

Deliveries of all mechanical hardware and computer equipment were completed by April 1979. Fabrication and assembly was completed in August. The initial system debugging was performed with the T53 support structure installed. A thrust bearing failure in October 1979 damaged some of the T53 support structure, as well as the rotor being tested. The balance machine support structure was changed to T55 hardware and the development of the T55 balancing plan continued. In January 1980 the balancing system was switched back over to T53 hardware incorporating design changes based on analysis of the bearing failure. Balancing of the T53 power turbines was halted in April 1980 due to instabilities caused by a bad spline fit in the power turbine. After high speed balancing five T55 power turbines, the system was demonstrated to NASA and AVRADCOM in May 1980. The T55 balance plan was finalized in August. In July and August 1980, components of the HSBS damaged during the bearing failure were redesigned and new hardware installed. Additional changes were made to improve the operator-balance-machine interface. The T53 balancing experiments were concluded and the balancing system was prepared for shipping to Corpus Christi.

2.1.3 Preproduction Evaluation of High-Speed Balancing System

Work under this task was begun during December 1980. Two T55 power turbine modules (previously high-speed balanced at MTI) were run in the HSBS after installation in Test Cell 2B at CCAD. A trim balance was performed on one of the power turbines. After inspection by CCAD personnel, the power turbine modules were assembled into engines in the overhaul cycle and evaluated in the engine test cell during the month of April 1981.

2.1.4 Site Modifications and Rig Installation

Mechanical and electrical preparations were completed in Test Cell 2B at CCAD during October and November 1980. Operational tests were completed in December.

2.1.5 Vibration Analysis of Low- and High-Speed Balanced Turbines

The testing phase during which the HSBS was evaluated in the overhaul facility was accomplished from July 1982 to November 1983. A total of 40 power turbine rotors were operated in the HSBS and then tracked through the engine assembly process and analyzed in the engine test cell.

2.1.6 Documentation and Training

Based on the experience gained in the testing phase, the operator and systems manuals were revised as necessary. The training was accomplished throughout the testing thanks to the active participation of CCAD personnel.

3.0 DISCUSSION

This section relates the results of work accomplished in the technical tasks authorized under this contract with the exception of the three-task analytical effort discussed in [2].

3.1 Detail Design of High-Speed Balancing Equipment

In this task, a detailed design of a high-speed balancing facility was prepared. The facility was designed to establish high-speed balance criteria for T53 and T55 power turbines, evaluate power shaft splines, and develop procedures for production high-speed balancing at the Corpus Christi Army Depot overhaul center. Other design criteria included: meeting OSHA safety requirements, supplying bearing lubrication, providing for vacuum operation and ready access to proposed correction planes. Vibration sensors were located to allow determination of rotor mode shapes and evaluation of correction plane sensitivities. The design was required to be free of disruptive resonances and to use as much actual engine hardware as practical. The design efforts were coordinated with appropriate CCAD personnel to ensure that the requirements of the ultimate user would be met. Expansion capabilities for the high-speed balancing of T700 power turbines and automated metal removal were provided for in the design.

All major components of the balancing system are standard commercial products in order to maximize system reliability. Figure 1 shows the complete balancing system. Each of the major components are summarized in Table 1 and discussed in the following subsections.

3.1.1 Prime Mover

The power turbine rotors to be balanced required an end drive arrangement. Sufficient torque was required to drive at a constant speed and also to accelerate against the drive train and power turbine rotor inertias. Meeting these power needs required a maximum torque of 27 Nm (20 lb ft) and a speed range of 600 - 21,000 rpm. Speed regulation within ±1% was necessary for the acquisition of accurate vibration data. After reviewing several potential prime movers such as ac motor/clutch, air/steam turbine, dc motor and variable speed ac motor, it was decided that the variable speed ac motor best met the program objectives.

3.1.2 Speed Increaser

The motor speed of 1725 rpm needed to be stepped up to provide the 16,000 rpm operating speed for T55 turbines and the 21,000 rpm operating speed for the T53 power turbines. A low loss, low inertia speed increaser capable of smooth operation throughout the entire speed range was required. After reviewing various approaches a combination V belt and spur gear increaser was selected.

3.1.3 Vacuum Chamber

The vacuum chamber design serves two important functions. It provides a low hp loss environment which reduces the drive requirements of the balancing system and it also provides for the operator's safety. Reducing the horsepower required to drive the turbines to full operating speed is accomplished by

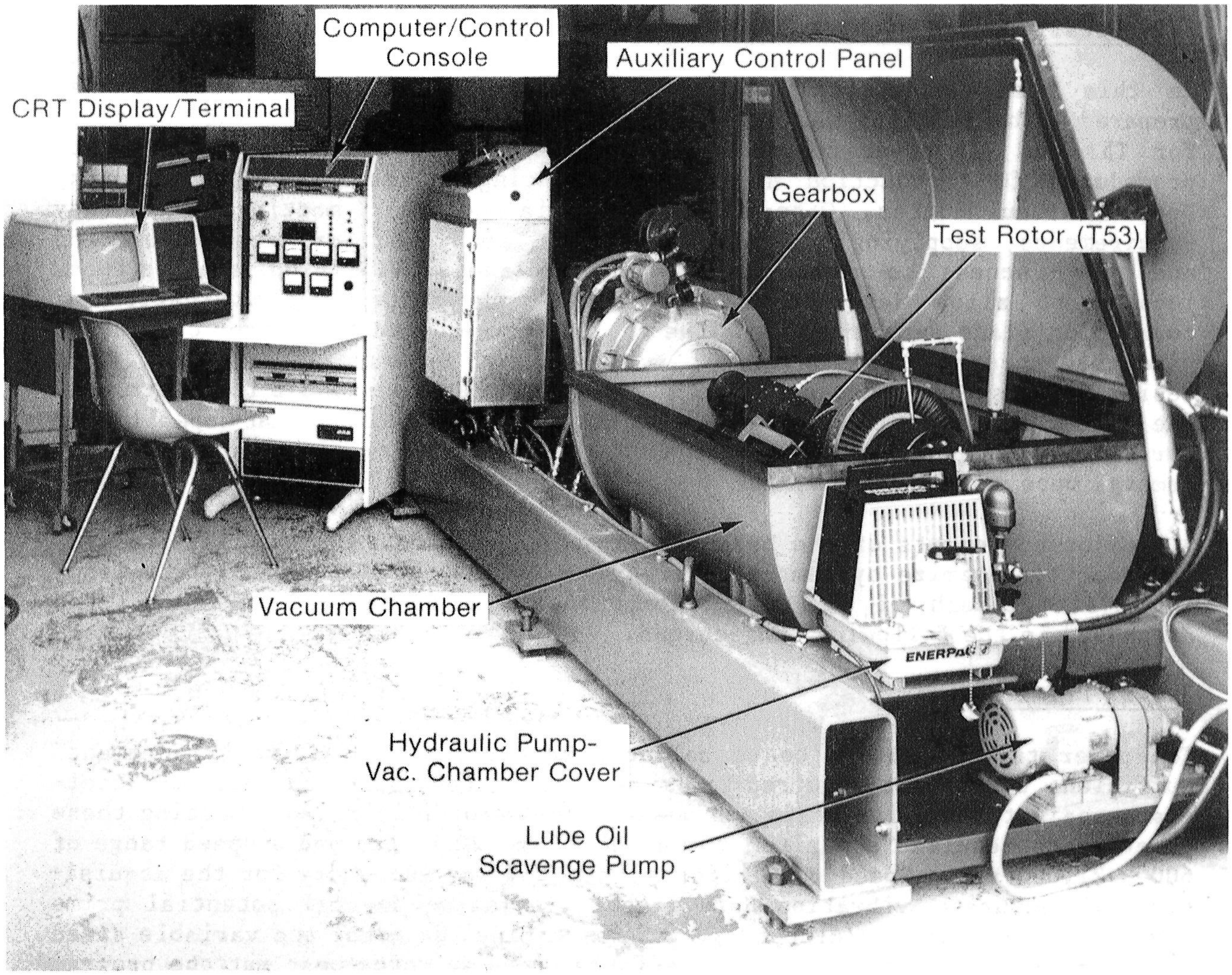

Figure 1 T53/T55 High Speed Balancing System

TABLE 1

MAJOR MECHANICAL COMPONENTS

ITEM	VENDOR	SI SPECIFICATION	ENGLISH SPECIFICATION
Variable Speed Drive (REGADYNE)	Control Products Corp. Chicago, Ill.	• 45 kW @ 181/rad/s • 5 - 180 rad/s continuously variable speed with 0.1% regulation. • 22 N•m torque @ 75% speed • 8 N•m torque @ 100% speed • accelerate 0.5 kg m^2 inertia @ 10 rad/s^2	• 60 hp @ 1725 rpm • 50 - 1725 rpm continuously variable speed with 0.1% regulation. • 16 lb-ft torque @ 75% speed • 6 lb-ft torque @ 100% speed • accelerate 1775 lb-in.^{2} inertia @ 100 rpm/s
Gearbox (SUNDYNE)	Sloan and Bischaff Inc. Springfield, N.J.	• 5.803:1 ratio increase • less than 5.2 kW loss • ball bearing on input shaft • journal tilting pad bearings on output shaft • axial hydrodynamic thrust bearing on output shaft • 150 kW @ 2160 rad/s • lubricate with Type A ATF	• 5.803:1 ratio increase • less than 7 hp loss • ball bearings on input shaft • journal tilting pad bearings on output shaft • axial hydrodynamic thrust bearing on output shaft • 200 hp @ 20,600 rpm • lubricate with Type A ATF
Vacuum Pump (KDH 80)	Kinney Vacuum Co. Boston, Mass.	• maintain 1.3 kPa absolute pressure • discharge effluent to have 99.95% filter efficiency	• maintain 10 torr absolute pressure • discharge effluent to have 99.95% filter efficiency
Gearbox Auxiliary Lube Pump (IMO C313A-106)	Delaval IMO Pump Div. Trenton, N.J.	• 190 rad/s drive provides 0.3 l/s @ 340 kPa • handle type A ATF from 15 to 93 °C	• 1800 rpm drive provides 5 gpm @ 50 psig handle Type A ATF from 60 to 200 °F
Bearing Lubrication System	Tri-Line Corp. Syracuse, N.Y.	• 114 l reservoir • deliver 38 ml @ 140 to 1400 kPa • pump MIL-L-23699C from 15 to 50 °C	• 30 gal reservoir • deliver 0.6 gpm @ 20 to 200 psig • pump MIL-L-23699C from 60 to 120 °F

reducing the air pressure inside the chamber to 1.3 kPa (10 torr). Safety is provided by the 1.3-cm (1/2-in.) thick chamber wall that is capable of containing a blade loss and most disk failures. Maximum energy disk burst containment is provided by 2.5-cm (1-in.) aluminum plate near the disks.

The chamber is cylindrical in shape with a horizontal cut that is hinged allowing convenient access to the interior of the chamber. A double acting hydraulic cylinder on each end of the chamber provides a cover lift in 90 seconds as well as providing a mechanical clamp to hold the cover shut. With the chamber evacuated an additional 12.2 kN (27,000 lb) force due to atmospheric pressure assures the cover remains closed and seated properly. After opening the chamber the cover is held open by both a mechanical and hydraulic system. A safety rod normally stored in the cover is swung into a vertical position and locked in a detent on the chamber base. The double acting hydraulic cylinder requires 7 MPa (1000 psi) to allow the cover to descend. Without the pump being turned on, this pressure is not available. An automatic and a manual locking valve are provided in the hydraulic circuit to ensure that the cover remain open.

3.1.4 Rotor Bearing Structure

The primary objectives of the bearing pedestal design were threefold. First, and most important, was the requirement that the balance machine support structure accurately simulate the characteristics of the engine supports, resulting in dynamic behavior of the power turbine in the HSBS indicative of the power turbine behavior in an assembled engine. The second design objective was to allow for the operation of both T55 and T53 power turbines as well as expandability to handle T700 power turbines. The final objective of the bearing structure design was ease of installation and removal of the power turbines so as to provide for production high-speed balancing capabilities.

The design simulates the characterstics of the engine supports while maintaining interchangeability between rotor types by using a substructure which remains constant for all engine types and a superstructure tailored to the particular engine type that uses an optimum amount of actual engine hardware. The superstructure design provides stiffness and damping values equivalent to the values for the engine supports. The T53 forward roller bearing and housing are slave items from engine stock that bolt up to the T53 superstructure. The T53 aft engine bearing fits into the T53 superstructure which contains flexures simulating hot section stiffness and damping. The T55 forward roller bearing and housing are also slave items from engine stock which bolt directly to the front T55 superstructure. The T55 engine power turbine module contains the aft engine bearings and supports and bolts directly to the aft T55 superstructure.

Preload is supplied to the engine bearings by a 15° angular contact thrust bearing loaded by wavy washers. Variable thrust loads from 0 - 530 N (0 - 120 lb) can be applied by a screw adjustment that varies the compression of the wavy washers.

3.1.5 Lubrication System

Oil is supplied to all power turbine bearings and the gearbox. The ac drive motor and all pumps have permanently lubricated bearings. The gearbox is supplied with automatic transmission fluid in a closed loop. The engine bearings for both T55 and T53 require MIL-L-23699C synthetic oil in flow rates of 4.5 kg/h (10 lb/hr) forward and 23 kg/h (50 lb/hr) aft. The thrust bearing used to preload the turbine bearings is supplied with a synthetic oil/air mist.

In order to operate in a vacuum the oil system required a reservoir external to the vacuum operation. When the vacuum is released, the oil in the internal sump is automatically returned to the external reservoir.

3.1.6 Drive Coupling

A miniature flexible disc coupling was selected as the means to connect the drive system to the power turbine shaft. The model selected was a Rexnord style CD size 62 with a torque capability of 34 Nm (300-lb/in.).

The 20.3-cm (8-in.) long, 1.6-cm (5/8-in.) diameter shaft that connects the hub flexures permits the required dynamic misalignment capabilities and provides the pass through the vacuum seal.

3.1.7 Vacuum Seal

The dynamic motion of the coupling spool had to be taken into account in the seal design since a stable spindle arrangement using carbon seals was prohibitively expensive to design and implement. A floating seal cartridge with known leakage was selected because the vacuum pump capacity easily offset the leakage. The cartridge contains three viton lip seals with silicone vacuum grease as the lubricant.

3.1.8 Vibration Transducers

Noncontacting displacement probes by Bently Nevada were selected to monitor shaft vibration. Piezoelectric load cells by Kistler were selected to monitor bearing loads. The location of the vibration probes was selected based on the predicted mode shapes from the rotordynamic analysis. For both rotor types, the probe locations are near the front bearing, shaft midspan, and near the first turbine disc. A temporary probe was also installed in the disc seal surface near the spline on the T53 turbine to study spline effects. Two probes are located at each location to allow observation of shaft orbits on an oscilloscope. The load cells are located in the bearing support structure.

3.1.9 Balance Planes

The location of the balance planes was determined by the appropriate Department Maintenance Work Requirement (DMWR) and accessability in an assembled power turbine module. The analysis had predicted that the existing balance planes were adequate for high-speed balancing. A shaft midspan balance plane was also evaluated for each rotor type. The accessible existing balance planes for the T53 power turbine are at either end of the enlarged section on the shaft and on the aft side of the second stage turbine disc. The T55 power turbine balance planes are at the forward end of the shaft and on the forward turbine disc, forward side. Two planes specified in the DMWR were not available due to accessability problems; these are the forward turbine stage on the T53 power turbine and the aft turbine disc on the T55 power turbine assembly.

These two planes cannot be reached with a grinder in the assembled configuration.

3.1.10 Signal Monitoring and Control

The operating condition of the balancing system can be monitored at either the test stand or the operator console which is located remotely from the test stand for safety reasons. The console contains meters for display of rotor vibration, lubricant pressure, bearing temperatures, and other indicators of the test stand status. A speed control potentiometer permits the operator to control rotor speed while observing a digitial speed display. The test stand controls duplicate most of the functions of the console allowing the operator to check for proper operation at low turbine speeds with the vacuum chamber cover open. The design, fabrication, and operating procedures of the high speed balancing system emphasize safety. Since any type of high-speed rotating machinery is potentially dangerous, the operator must recognize the limitations of the hardware and his ability to control it. It is, therefore, the operator that determines the ultimate safety of the system. The following system parameters are monitored:

- Speed - A Spectral Dynamics fiber-optic tachometer provides a once-per-revolution signal that is used for both digital speed readout and to determine the phase angle associated with the vibration readings.

- Thrust Bearing Temperature - A thermocouple is attached to the outer ring of the thrust bearing. A 135°C (275°F) meter trip is recommended. A temperature above the trip point activates a warning light and an alarm.

- Oil Temperature - A thermocouple senses the aft bearing drain oil temperature. A 149°C (300°F) meter trip is recommended. A temperature above the trip point activates a light and an alarm.

- Vacuum - A pressure sensor monitors chamber pressure. The meter trip point activates the warning light and an alarm. 14 kPa (2 psia) is recommended.

- Lubrication Oil Pressure - Lube oil pressure is monitored at the pump outlet. The meter on the operator's console has a low trip capability. A 140 kPa (20 psig) trip is recommended. The trip deactivates the drive motor in addition to the warning light and alarm.

- Power Turbine Vibration - Analog meters display overall vibration from the forward and aft displacement probes.

3.1.11 Inherent Safety Features

A number of safety measures are built into the system to remove drive power from the motor when a potential hazard exists. If low oil pressure exists the motor cannot be started. In addition, if low oil pressure occurs during a run, drive power automatically is removed from the motor. Low vacuum removes drive power from the motor if the console controls are in use to run high speed tests. In addition, depressing the emergency stop button automatically removes drive power from the motor allowing the rotor to gradually decelerate.

The test stand speed control limits the maximum rotor speed to 1000 rpm which permits adequate identification of mechanical readiness for high speed operation while the operator is in the test cell.

There are several features that prevent operating the rotor from the console when there is someone in the test cell. Both the console and the test stand speed control selector switches must be in the same position in order to start the drive motor. When they are both in the "Test Stand" position, the console speed control is disconnected. The "Power On" switch on the Regadyne motor control cabinet can be turned off while working in the test cell.

3.1.12 Expansion Capabilities

Throughout the design of the high speed balancing system, it was acknowledged that the purpose behind the development of the system was not just to high speed balance T53 and T55 power turbines but to develop a capability, deliverable to the overhaul facility, that will enable the facility to handle the balancing of flexible rotors in general. Specifically, the General Electric T700 helicopter engine, whose power turbine operates above two bending critical speeds, was chosen as a future capability of the HSBS. The substructure and superstructure of the bearing supports provides the means by which the balancing system's capabilities can be expanded to accomodate any rotor smaller than the T55 power turbine rotor.

Additional expandability to a laser-based, automated, metal removal system was also provided for in the design. The internal configuration of the chamber allows straight line access to each balance plane. The vacuum chamber cover can be readily modified to carry laser optics.

3.1.13 CCAD Design Review

Three design reviews were held at CCAD so the design could benefit from the experience of the overhaul facility personnel regarding production oriented use in the engine overhaul cycle. Characteristics of the engine build and engine test processes were also discussed.

CCAD personnel recommended locking out the drive spline with a four-jaw collet mechanism that eliminates spline clearance for both T55 and T53 power turbines due to potential problems from variations in fits of worn engine hardware. The T53 shaft to turbine disc spline with a four-jaw collet mechanism that eliminates spline clearance had already been shown to affect the dynamics of the engine. Therefore, plans to observe the effect of this spline on vibration in the HSBS were outlined.

3.1.14 Operating Procedures

The balancing system operating procedures are based on user friendly interactive software that allows operation by personnel with no previous computer experience. Mechanical experience is required as is the case for operation of any high speed equipment.

The steps required to install a rotor in the HSBS are given in detail in the operators manual (MTI report no. 81M66). The rotor module is clamped into place and the drive connection made. Lubrication, electric drive, and vacuum systems are checked for proper operation. The installation procedure takes approximately 15 minutes. If there are no audio or visual alarms, the system is ready to run the computerized data acquisition routine.

Monitoring the operating condition of the system can be accomplished at either the test stand or the operator console. For safety reasons, the operator console's location is remote from the test stand. Oil temperatures and pressures, rotor speed, and vacuum and bearing temperature are monitored. Visual and audible alarms signal parameters exceeding adjustable preset limits. The only instance in which the system cannot be operated by the operator is in the event of loss of oil pressure. If this problem occurs, interlocks prevent system operation and the drive system automatically shuts down.

Low speed operation of the system (less than 1000 rpm) can be accomplished from the auxilliary control panel located on the test stand. This capability permits the operator to check proper operation of all auxilliary systems before leaving the test stand. Low speed operation can be accomplished with the vacuum chamber cover open, thus allowing the operator to conduct a visual check of the rotating turbine. Interlocks prevent high speed operation with the cover open.

Figure 2 is a software block diagram, showing the main communication paths that exist among the three major files, the four main software subsystems, and the various utilities for listing the contents of the system files. The interaction of the files, subsystem, and utilities will be outlined in the subsequent discussion. The functions of each component are listed below.

- Files:
  - Configuration File - Defines how rotor is to be balanced
  - Run File - Contains vibration data for a run
  - IC File - Contains influence coefficients

- Main Subsystems:
  - CFIGEN - Generates configuration file
  - DATA - Acquires rotor vibration data
  - ICMAIN - Generates influence coefficients
  - BALSYS - Calculates correction weights

- Utilities:
  - CLIST - Lists a configuration file
  - LISTIC - Lists part or all of IC file
  - DATLST - Lists run file.

These definitions show the clean organization and modularity of the system. The configuration file is set up by the balancing engineer to control the balancing process for a particular type of rotor. Whenever data are acquired by an operator, synchronous amplitude and phase information is stored as a function of probe number and speed in the run file. Each run has a separate file and the system keeps track of file numbers. The influence coefficient file is filled by the influence coefficient subsystem and is accessed by the balancing subsystem to perform balancing calculations.

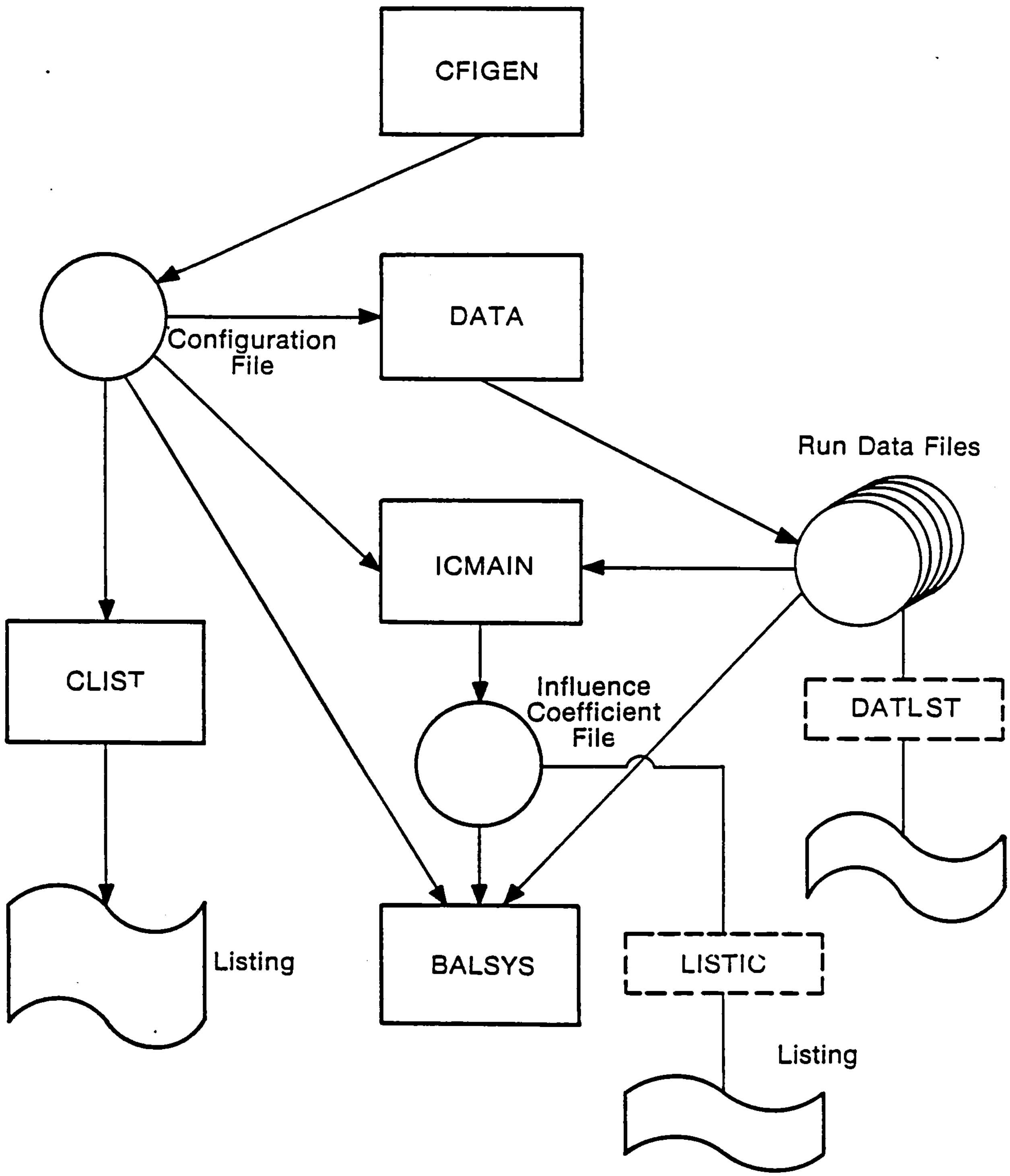

Figure 2 MTI Balancing System - Software
Data File Overview

Subsystem CFIGEN creates the configuration file which includes information such as the number of correction planes, number of sensors, balancing speeds, speed tolerance, angular relationships between sensors, etc. Subsystem DATA acquires the vibration data and fills the run file. Subsystem ICMAIN processes run file data to generate influence coefficients and subsystem BALSYS uses configuration file, run file, and influence coefficient file to determine correction weights to reduce the vibrations measured in a particular run.

Various utilities are provided to allow file contents to be reviewed: CLIST shows configuration file contents; LISTIC provides a printout of influence coefficients; and DATLST provides a printout of run file vibration data.

Flow charts for the two programs, DATA and BALSYS, used during production balancing are shown in Figures 3 and 4, respectively. For a detailed description of the system software, refer to the Balancing Systems Software Manual (MTI Report No. 81M13).

3.2 Fabrication and Checkout of High-Speed Balance Equipment

3.2.1 Fabrication

The fabrication of the mechanical hardware was accomplished as follows:

- Modify Sundyne gearbox output shaft
- Mount vacuum chamber base to machine base
- Mount driver on base
- Mount gearbox on base
- Install vacuum chamber cover
- Assemble and test chamber cover hydraulics
- Install T55 adaptive hardware and align
- Mount auxilliary control box on base
- Install gearbox lubrication system
- Remove T55 adaptive hardware and install and align T53 hardware
- Build and install drive shaft vacuum seal
- Assemble hub adaptors for both rotor types
- Install lube oil scavenge system
- Install variable speed drive controller, vacuum pump, and bearing lubrication system
- Connect computer and CRT
- Install instrumentation, temperature, vacuum, vibration
- Calibrate lube oil flow rates to specification
- Install preload bearing assembly
- Install T53 rotor
- Conduct operational tests to 900 rpm with chamber cover open and to 1500 rpm with chamber evacuated
- Debug operating systems.

The assembled hardware is shown in Figures 5a and 5b.

3.2.2 Checkout of T53-L-13B

Figure 6 represents synchronous vibration amplitude for the first T53 power turbine operated in the HSBS. The as-installed amplitudes for the turbine and

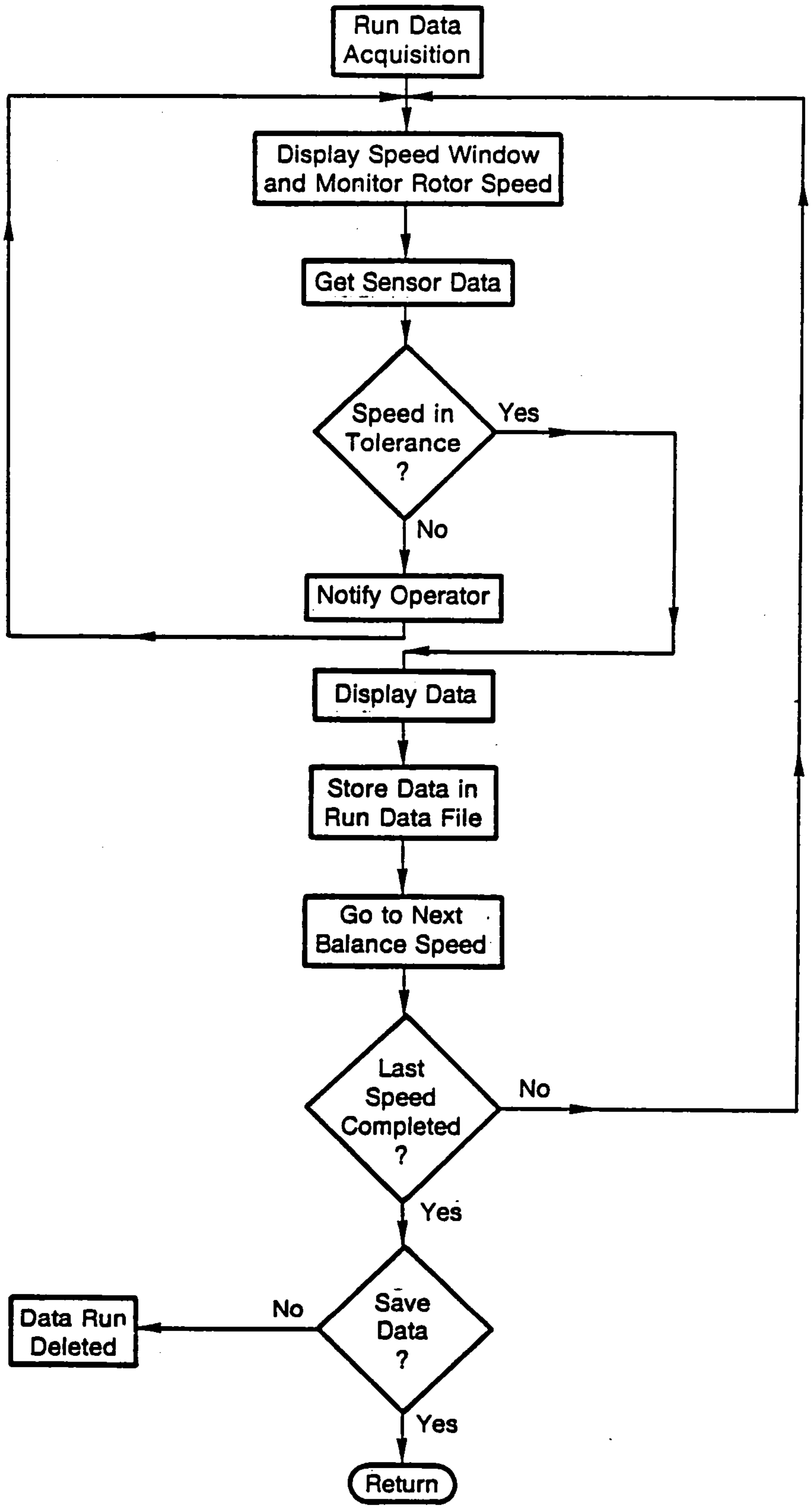

Figure 3 Data Acquisition Flow Chart

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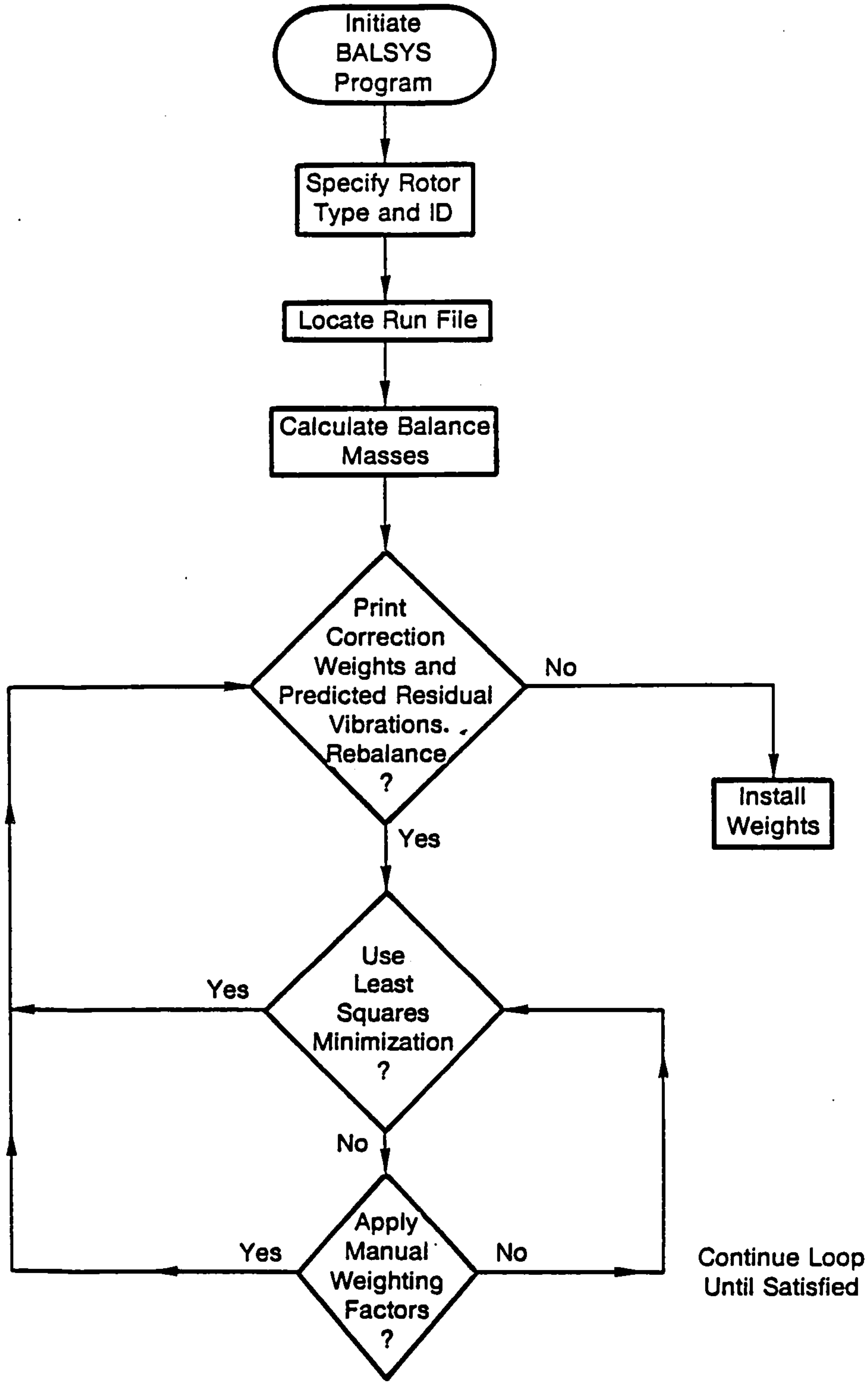

Figure 4 Balance Weight Calculation Flow Chart

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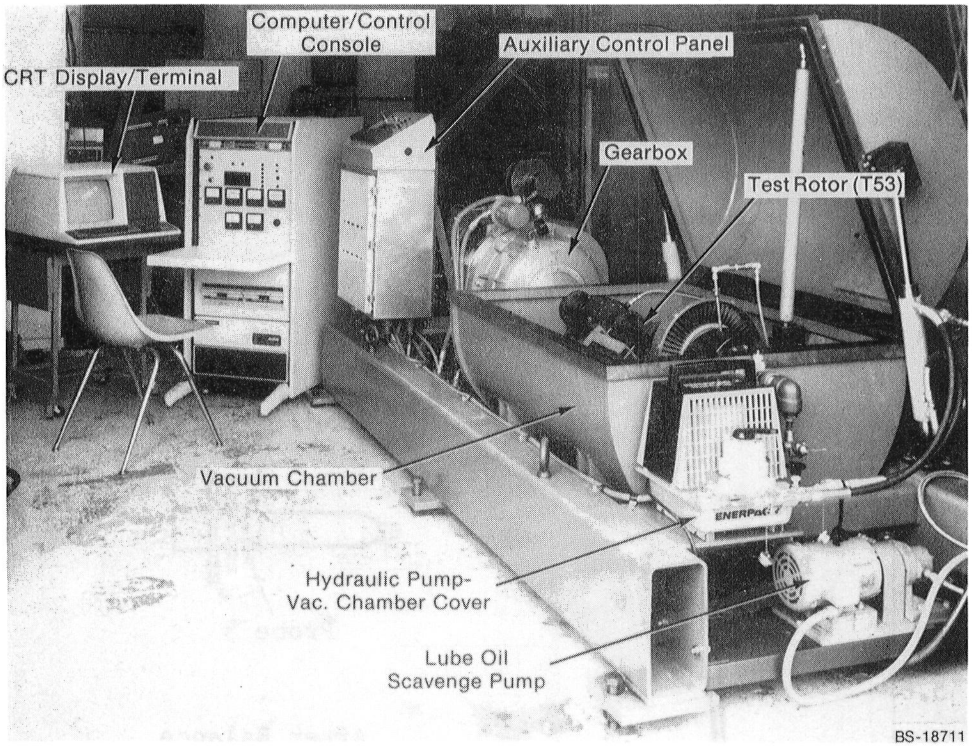
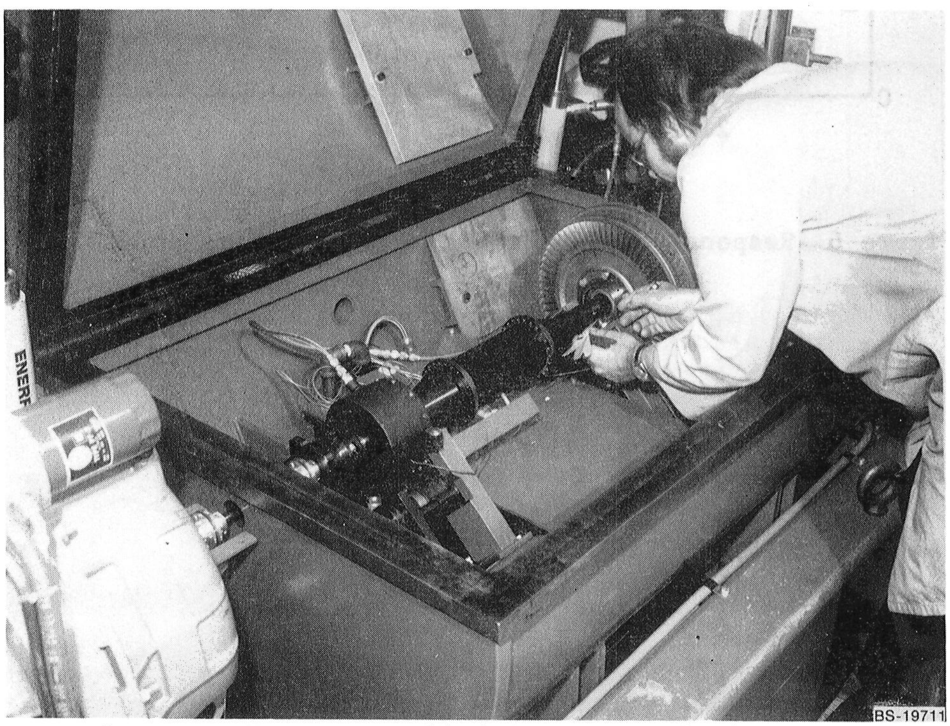


a. Test Equipment and Computer Support System



b. Test Rotor Installed in Vacuum Chamber

Figure 5 T53/T55 High Speed Balancing System

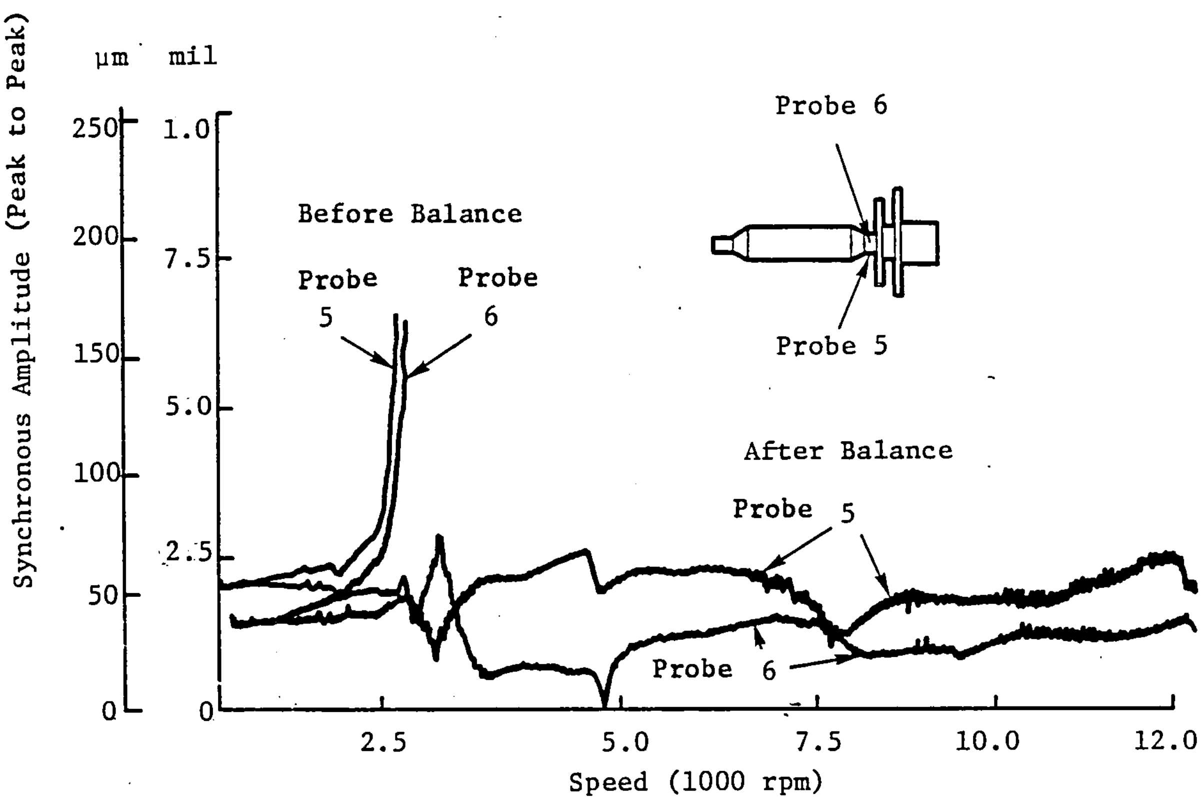


Figure 6 Response of T53 Before and After Balance at 2750 rpm

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sensors show a rapid increase that starts at approximately 2500 rpm that agrees well with the predicted response to the critical speed at 3000 rpm. At this early stage of the balancing tests conservative limits had been established for maximum amplitude, consequently this initial run was stopped at 2750 rpm and balance weights were calculated. Later in the tests, it was learned that this peak can easily be passed through and that the amplitudes drop as fast as they rise in a very well defined peak. Influence coefficients were developed at 2750 rpm and a successful balance resulted in the amplitudes labeled "after balance" shown in Figure 6.

After this balance run, the power turbine rotor was removed for inspection. The inspection revealed that the nozzle had rubbed against the rotating disc spacer. Tolerance stackup of the nozzle fixture was the cause. The nozzle and fixture were eliminated from the remainder of the tests. Further inspection of the balance system revealed wear on the shaft passing through the vacuum seal. The seals were replaced and a new shaft with Rockwell C60 material was installed.

Figure 7 shows the as-installed condition from the test run following the rebuild in which the first critical was traversed. The buildup in amplitude at 13,000 rpm also agrees with the analytical predictions. The peak evident at 8000 rpm is structural and does not respond to balancing. At this point in the testing, a considerable number of trial weight and balance runs were made to develop influence coefficients and to develop the balance plan. Figure 8 shows the results of a two-plane, two-speed balance which allowed operation to 17,400 rpm. Synchronous amplitudes became increasingly unstable in the upper speed range, making it impossible to acquire consistent high speed data. Further testing demonstrated the problem to be twofold, with probe bracket resonance and motion of the machine base both contributing to the oscillating levels. After modifying the probe bracket, the vibration levels were more stable and influence coefficients were developed for 17,200 rpm. Development of the influence coefficients at the higher speeds was hampered by vibration amplitude variations caused by shifting of the spline.

Figure 9 represents vibration amplitude versus speed (measured by the turbine end probes) in a test run that clearly showed a spline shift. Shifts were occurring at both high and low amplitudes and at various rotor speeds indicating the possibility that a destabilizing mechanism might be involved. Instabilities caused by problems with the thrust bearing might provide sufficient excitation to cause a spline shift. Consequently, the grease packed thrust bearing was replaced with an oil fed bearing that could provide more preload. The new thrust bearing resulted in no change in response at the same preload used in the earlier runs. A series of tests was conducted in which the preload on the aft bearing was varied by making changes in the axial load on the thrust bearing.

A lower than nominal preload resulted in no change in response. A higher than normal preload raised the rotor speed at which the shift occurred. The measured vibration at the higher speeds was the lowest observed at this time. The thrust bearing failed at 20,500 rpm due to lack of lubrication (additionally, 18,240 rpm was later determined to be the bearing manufacturers recommended maximum speed). With the sudden loss of preload, rotor amplitudes became excessive and the forward roller bearing was overloaded to failure. Damage was limited to the bearings and bearing support structure.

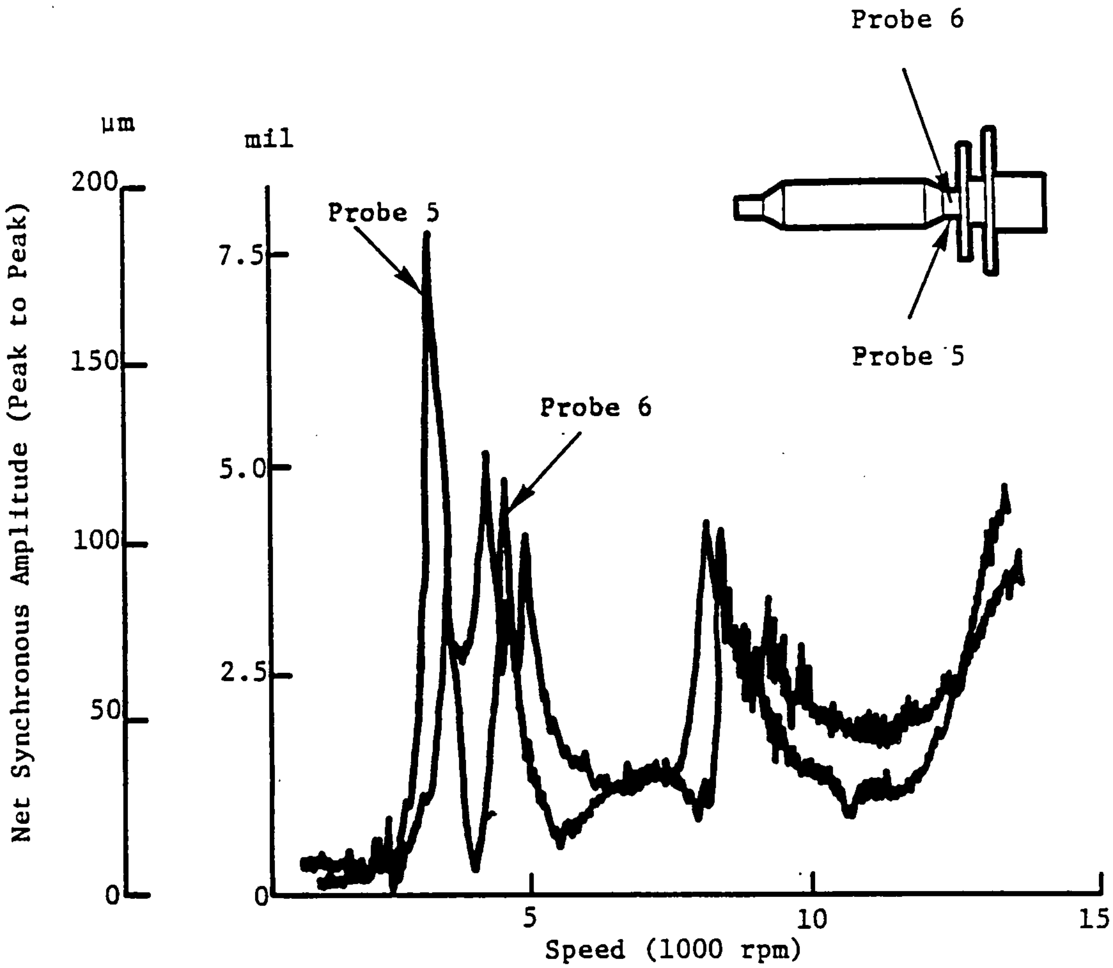

Figure 7 T53 Rotor Amplitudes for As-Installed Condition

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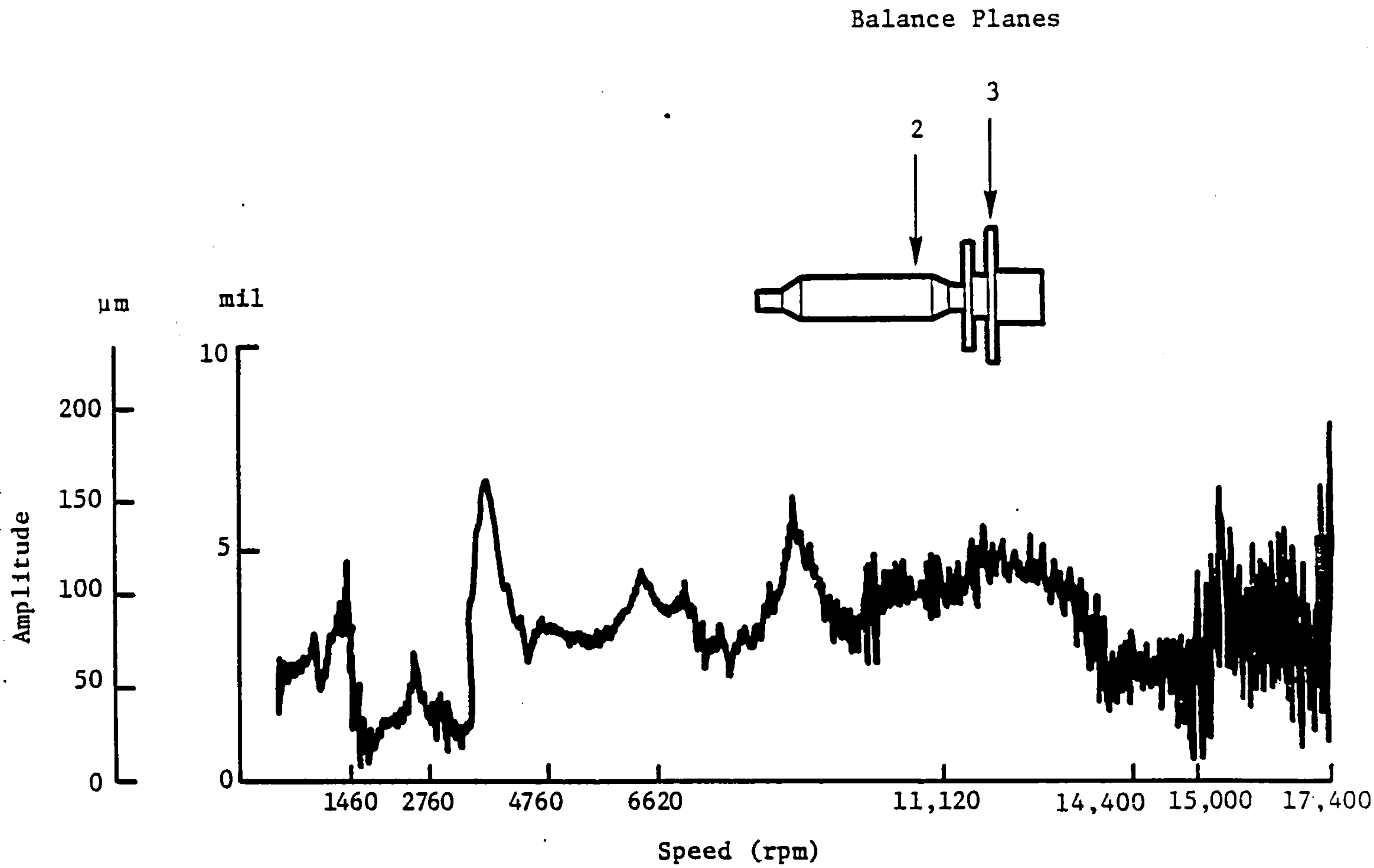

Figure 8 First T53 Rotor Vibration at Probe 5 After Balance on Planes 2 and 3

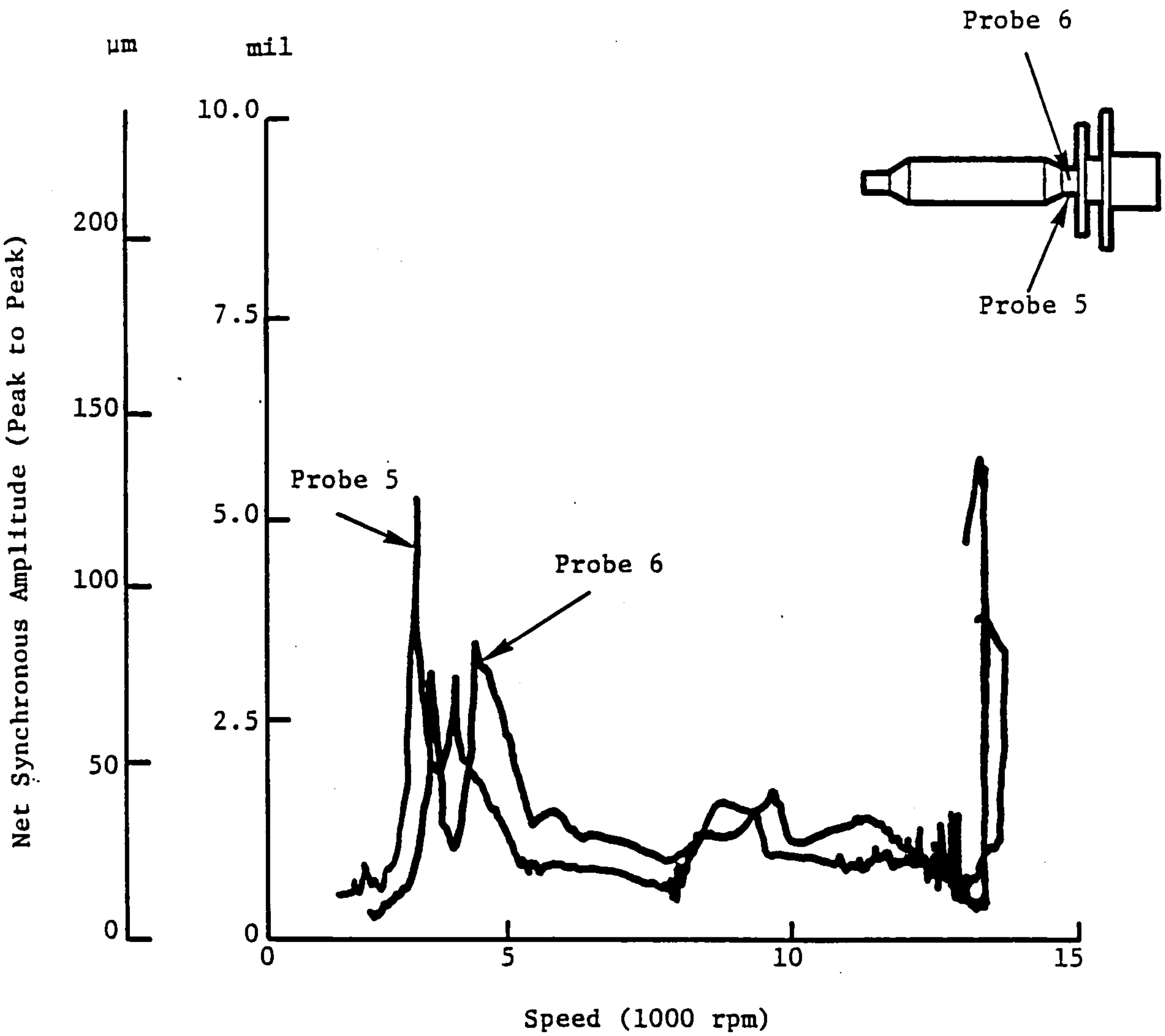

Figure 9 T53 Rotor Response Showing Spline Shift

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A survey of the T53 spline measurements was made to determine the optimum fit for future runs. The inspection revealed that all of the T53 rotors at MTI were worn such that clearance in the spline pilot exceeded blueprint tolerance. Additionally, two out of three exceeded the CCAD overhaul procedure wear specification which was used to determine when chrome plating was required as corrective action. The tabulated values for each of the rotor assemblies are shown in Table 2.

3.2.3 Checkout of T55-L-110

The T53 supports were removed and replaced by the T55 superstructure. The T55 probe bracket, which permits displacement orbit determination in three planes along the power turbine shaft, was installed. Rotor serial number 265503 was installed for system checkout. After checking the mechanical runout, calibrating the oil flow rates and setting the preload of the grease-packed thrust bearing at 120 N (27 lb), the balancing system was ready to run.

Early balancing runs of the T55 power turbines were hampered due to a slow acceleration rate and problems with cooling the thrust bearing. The number of seals in the drive shaft seal cartridge was reduced from 3 to 1 and air cooling was added to the thrust bearing. Measurements of drive current were made to analyze the slow acceleration problem. It was discovered that the full 44.7 kW (60 hp) was being used. The direction of rotation was reversed, reducing windage losses and providing an acceptable acceleration rate. The thrust bearing temperature, however, still limited the duration of a test run.

A representative plot of vibration amplitude versus speed for the first T55 power turbine is shown in Figure 10. The large peaks in amplitude at approximately 4000 and 8000 rpm were diagnosed as structurally related resonances. One of the tests performed to determine this was the use of an independently mounted vibration probe to assess whether the rotor itself was vibrating at the large amplitude or if relative motion of the probe bracket was contributing to the peak readings. The data taken with this probe was much lower, suggesting a structurally related problem driven by rotor vibration. Impact tests on the probe bracket revealed natural frequencies present in the range of the large indicated amplitudes. Various durometer elastomers were used to attempt to isolate the probe bracket. The peak amplitudes were not reduced but the repeatability of the vibration data was improved at the higher speeds.

3.2.4 Additional Testing of the T53-L-13B Power Turbines

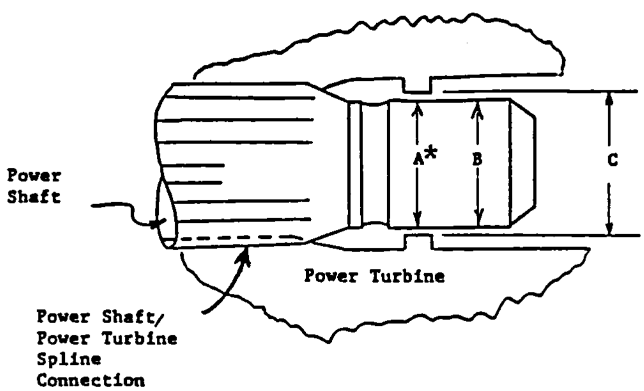
The deep groove thrust bearing used for thrust loading the T53 power turbine was replaced with an angular contact ball bearing lubricated by an oil mist. The new bearing could operate with a preload in excess of 450 N (100 lb). The modifications to the design are summarized in the following list:

- provisions for adjustable thrust load
- increased number of wave springs
- oil mist lubrication
- common thrust bearing size for both the T53 and T55 engines
- replace the ABEC 25 bearing with an ABEC 9 bearing
- more rigid thrust pad assembly incorporating precision machining to eliminate alignment problems.

TABLE 2

RESULTS OF INSPECTION & MEASUREMENT OF T53
PILOT BORES AND PILOT DIAMETERS

ITEM	A	B	C	FINDINGS
SHAFT No. 1 (SN P6621)	2.995 cm Min (1.179 in. Min) 2.997 cm Max (1.180 in. Max)	2.999 cm Min (1.181 in. Min) 3.002 cm Max (1.182 in. Max)	-	Severe fretting at A* Conclusions • Out of Print • Needed Plating • In Rig at Failure
SHAFT No. 2 (SN 8707)	3.003 cm Min (1.1824 in. Min) 3.004 cm Max (1.1826 in. Max)	3.004 cm Min (1.1828 in. Min) 3.005 cm Max (1.1830 in. Max)	-	Conclusions: • Out of Print only at one point Status • Usable (Best Candidate)
SHAFT No. 3 (SN 1F010)	3.001 cm (1.1815 in.)	3.003 cm (1.1822 in.)	-	Conclusions: • Out of Print • Should be plated
TURBINE No. 1	-	-	3.021 cm (1.1895 in.)	• Out of Print • Worse Case (Largest Bore) • In Rig at Failure
TURBINE No. 2	-	-	3.020 cm (1.1890 in.)	• Out of Print • Usable
TURBINE No. 3	-	-	3.007 cm (1.1840 in.)	• Out of Print (Too small) • Usable - (Best Candidate)



Print Dimensions
A = B = 3.004 to 3.005 cm
(1.1825 to 1.830 in.)
C = 3.012 to 3.018 cm
(1.1860 to 1.1880 in.)

CCAD Overhaul Procedure
Chrome Plate if:
A or B < 3.002 cm
(1.1820 in.)

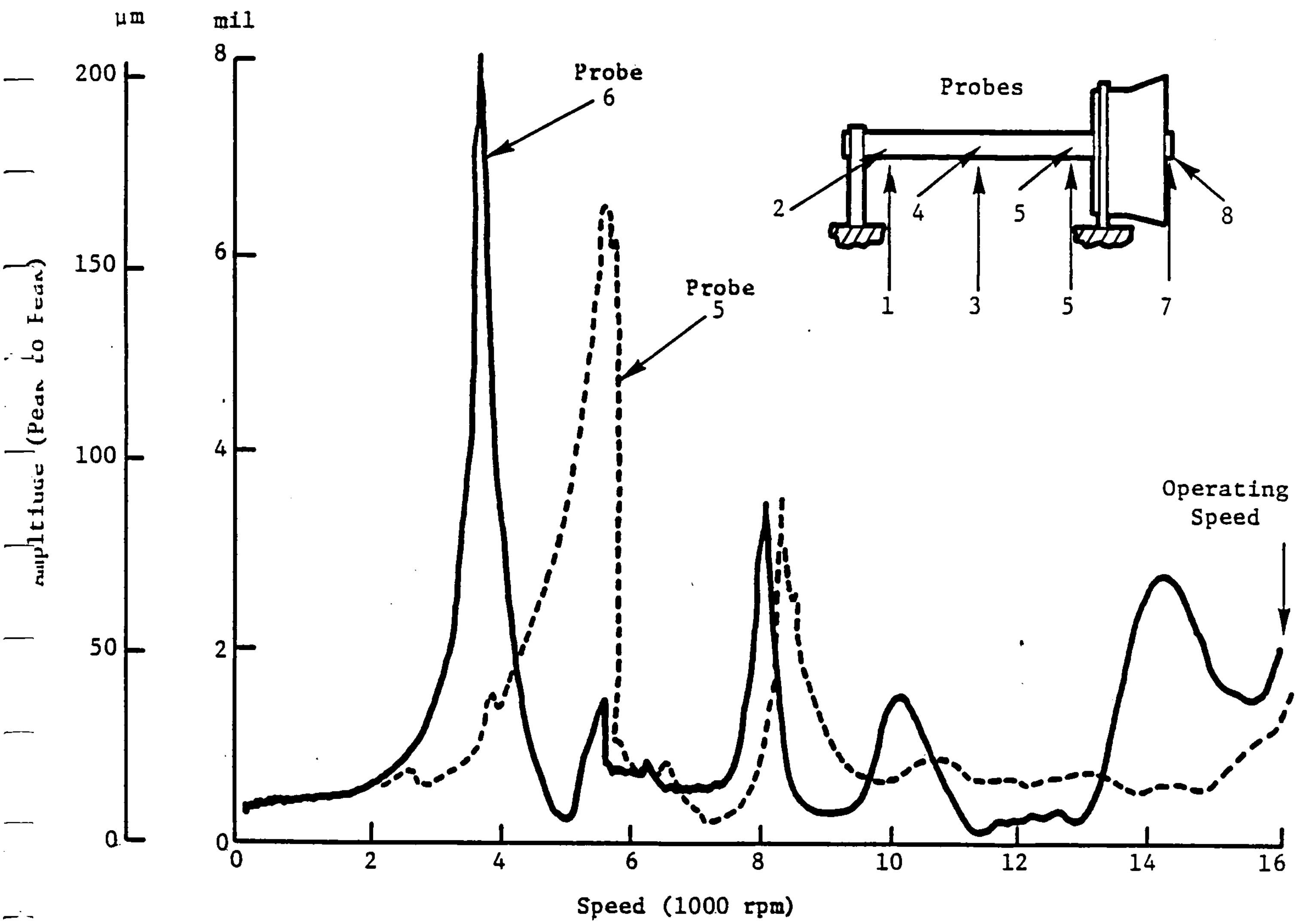


Figure 10 T55 Vibration Data Through 16,000 rpm (Serial No. 265503)

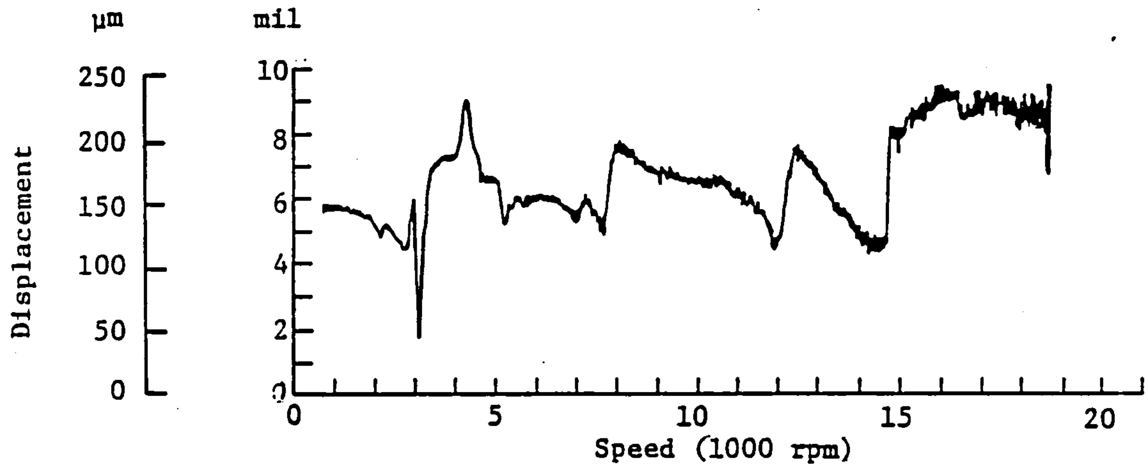
A T53 rotor was assembled from the shaft and disks that resulted in the tightest pilot fit. The preload was set at 220 N (50 lb). A single speed balance at 2750 rpm allowed operation to 12,400 rpm. Spline shifts prevented acquisition of consistent data at higher speeds. Test runs with incremental increases in preload were conducted to study the effect on the spline. The speed at which the spline shift occurred increased with each increase in preload, thus allowing the gathering of vibration data at the higher speeds. After multispeed balancing at 2,750, 12,750, and 14,400 rpm, the maximum speed of 21,000 rpm was reached. Figure 11 represents the effect of increasing preload on the vibration data from the turbine end probe for this balance condition.

The successful operation of the HSBS up to the maximum speed of 21,000 rpm permitted the beginning of the development of a comprehensive T53 balance plan. Rotor sensitivity was evaluated at each of the balance planes and influence coefficients were calculated at 2,750, 13,000, 17,200, and 21,000 rpm. During these tests it was observed that the rotor repeatability was deteriorating. Testing was halted and the rotor removed for a complete inspection. The turbine tie bolt was found to be loose and significant spline and pilot wear was evident.

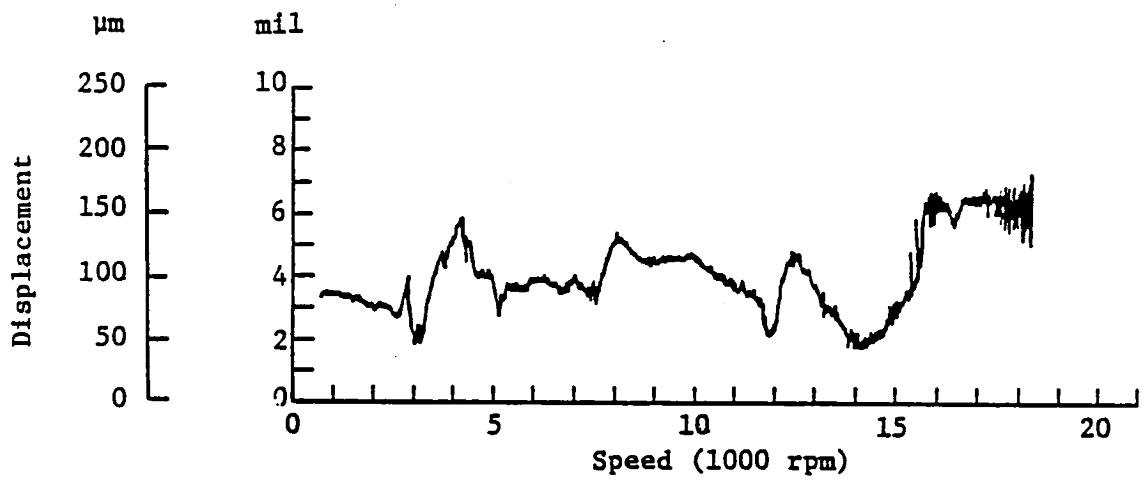
The spline teeth on the third test shaft were in better condition so its pilot diameter was chrome-plated to give a 50 μm (2 mil) clearance with the second test disks. The rotor was operated to 13,000 rpm in the as-installed condition. A three plane balance at 2750 and 13,000 rpm allowed operation to 19,500 rpm. Oscillating synchronous amplitude hampered data acquisition at the upper speeds. Using accelerometers mounted on the balancing system base, it was discovered that increased vibration of the base coincided with the onset of the oscillations in synchronous amplitude of the power turbine. Bolting the frame to the floor eliminated the oscillations. Vibration levels were acceptable to 21,000 rpm without further balancing.

As mentioned earlier, rubbing problems required that these balancing tests be conducted without the nozzle installed between the turbine stages. Nozzle installation requires disassembly and reassembly of the power turbine module after high speed balancing in order to install in an engine. Consequently, tests were required to assess the impact of this operation on the balance quality. The rotor was therefore removed from the HSBS and disassembled, including separation of the turbine disks. The rotor was reassembled, reinstalled in the balance machine and operated to 21,000 rpm. The vibration response was unchanged, demonstrating that the nozzle was not required as part of the module installed in the HSBS.

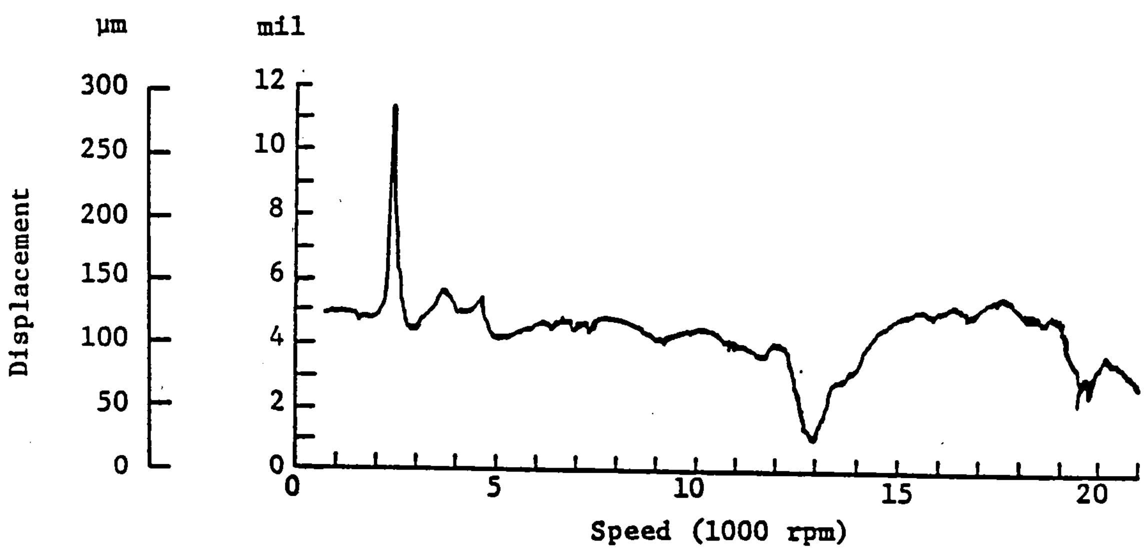
Problems with spline shifts continued to hamper the gathering of high speed vibration data. The thrust bearing was still running hot. It was decided to inspect the thrust bearing assembly and review the design. Inspection revealed that the bearing was misaligned angularly, and that the thrust pad was damaged. This damage caused the wave washers to jam thus creating an uneven preload. The angular contact thrust bearing being used was an automotive bearing. The tolerances, plastic retainer and retainer configuration were not well suited for a high speed, vacuum application. An ABEC 9 bearing of the same size, which provided direct inner race lubrication, was substituted. The thrust pad was repaired and the wave washers were replaced. No shifts were observed in the succeeding balance runs. The need for a new



a) 60 lb Preload



b) 80 lb Preload



c) 100 lb Preload

Figure 11 Effect of Preload on T53 Vibration balance speed between the ones at 12,750 and 17,200 rpm became evident as different unbalance configurations were evaluated. A series of trial weight runs were made and influence coefficients calculated for 15,500 rpm.

3.2.5 T55 Balancing

The bearing support structure was changed over to the T55 superstructure. Rotor 265503 (with no low speed balancing performed during overhaul) was installed and run to a speed of 12,000 rpm. A plot of the vibration data for this run is shown in Figure 12. A two-speed balance at 8,265 and 12,000 rpm was effective in reducing vibration levels at the upper speeds although the low speed peak was not reduced. The influence coefficients were updated by including additional data taken at 14,000 and 16,000 rpm.

Rotor 268922 was installed and operated to a speed of 14,100 rpm in the installed condition. A balance calculation based on data at 8,265 and 14,000 rpm followed by a trim calculation at 16,000 rpm reduced the levels as shown in Figure 13. The plots for this rotor show the power turbine first critical speed at approximately 9000 rpm as predicted by analysis.

Rotor U00553 was installed in the HSBS and operated to a speed of 8600 rpm. Data from this rotor was not consistently repeatable and little success was achieved in balancing. The rotor was removed from the vacuum chamber.

The next T55 power turbine installed was U00257 which ran to 16,000 rpm with extremely low vibration levels as shown in Figure 14. No balancing was required for this power turbine. The peak at 9000 rpm on Figure 14 corresponds to the power turbine's first critical speed.

Rotor U00559 (with no prebalance) was operated to 16,000 rpm as installed. The vibration amplitudes were reduced significantly as shown in Figure 15, by balancing at 8,265, 15,000 and 16,000 rpm.

Up to this point all the balancing was performed using collars for temporary weight addition on the shaft and safety wire wrapped around turbine blades. Permanent corrections needed to be made on the rotor's existing balance planes according to the overhaul center's specifications. Since the diameter of the shaft where the metal would be ground is different than the diameter of the temporary collars and the diameter of the material removal on the turbine disks is different than the blade tip diameter where the temporary weights were located, the weight removal amounts as calculated by the balancing system require scaling to give the same results.

The correction weights calculated for rotors U00559 and 268922 were replaced with permanent corrections ground into the rotor. Detailed instructions for material removal are included in the operator's manual. The synchronous amplitude versus speed plots for these two rotors with the permanent correction weights ground in are presented in Figures 16 and 17.

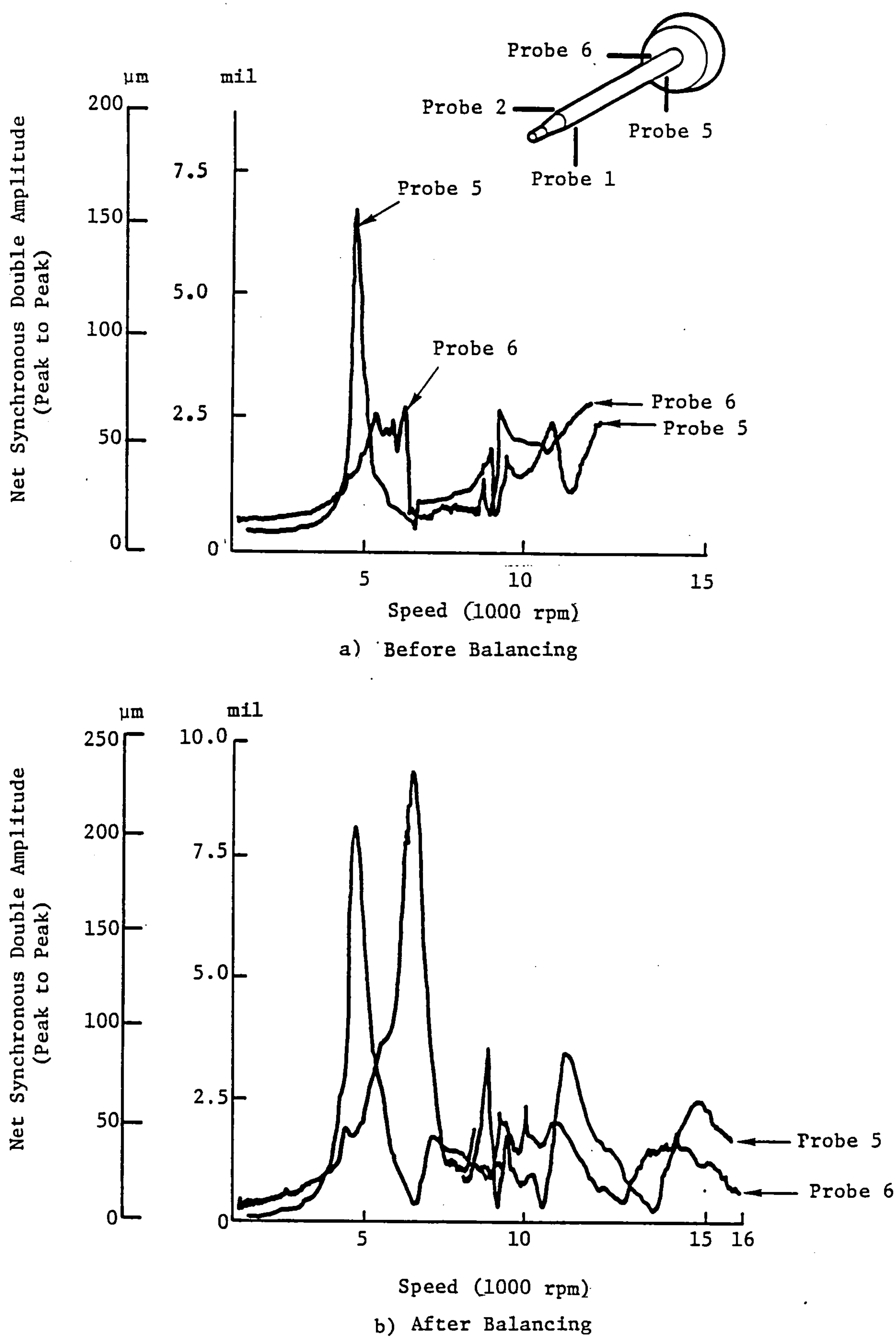


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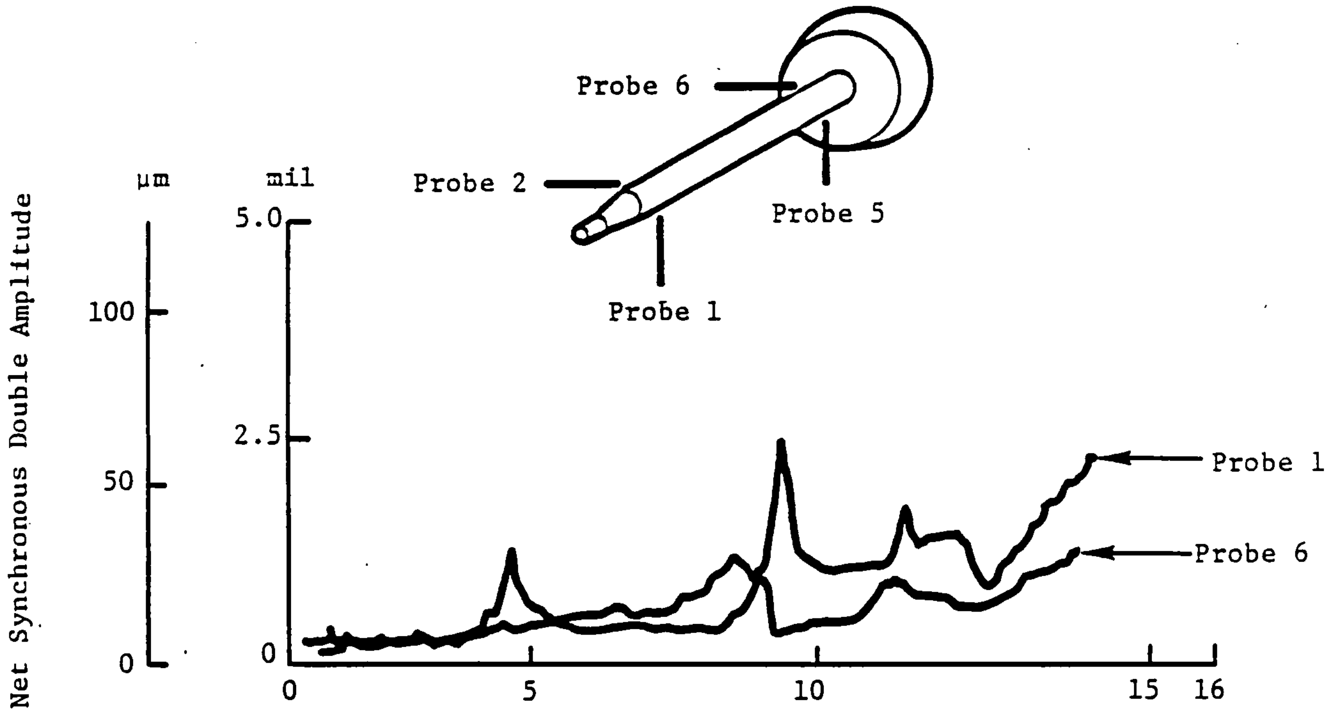
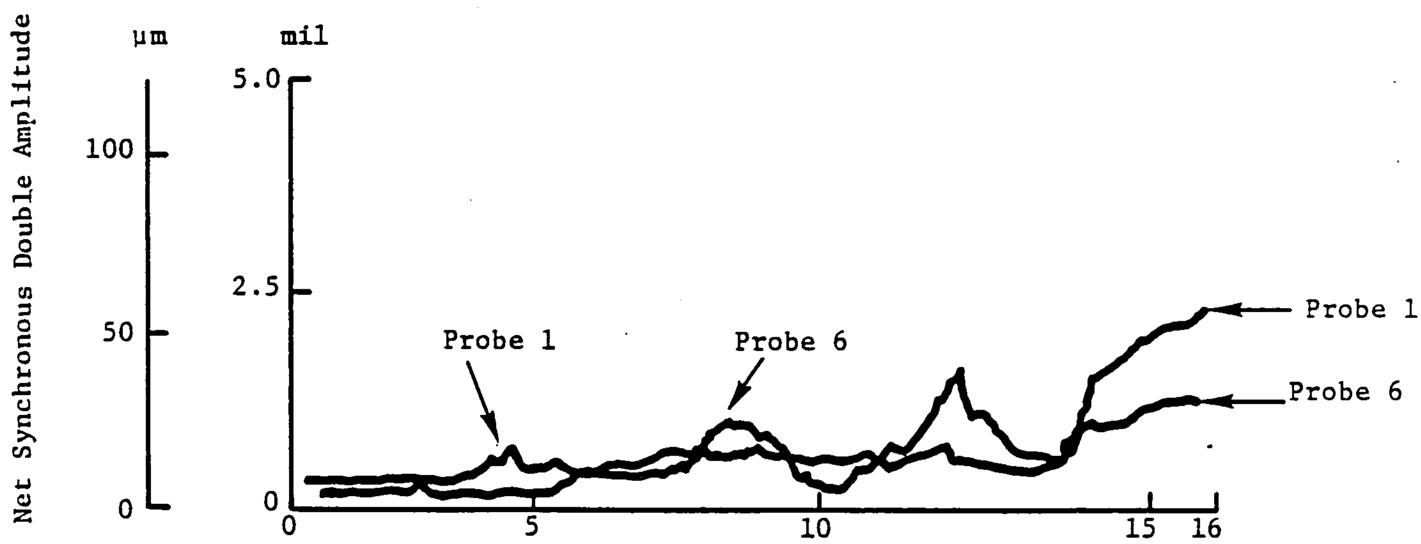
Figure 12 Power Turbine Vibration Before and After Balancing in the HSBS. T55 Serial No. 265503



a) Before Balancing



b) After Balancing

Figure 13 Power Turbine Vibration Before and After Balancing in the HSBS, Serial No. 268922

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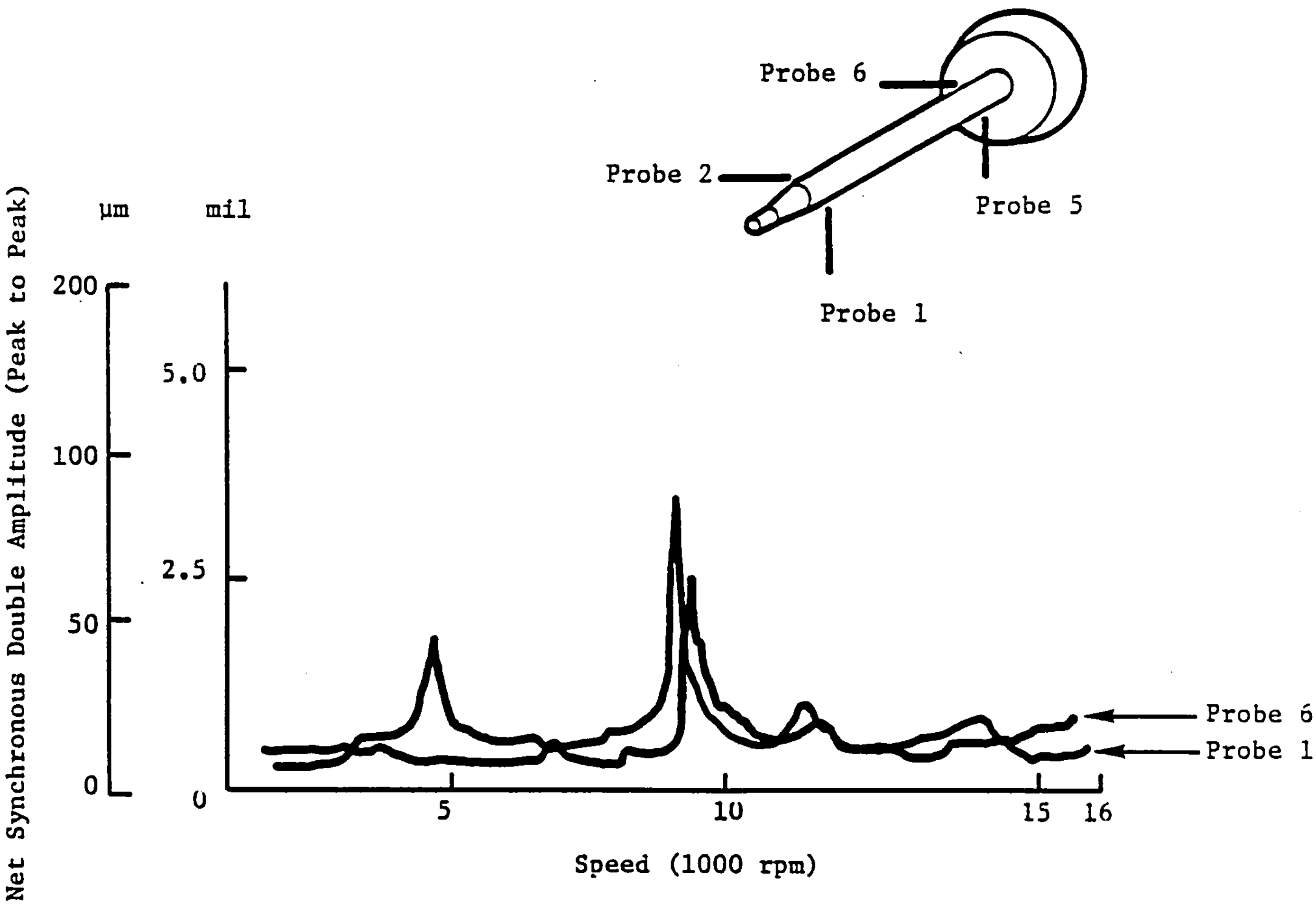

Figure 14 Initial Vibration Level in HSBS, Serial No. U00257

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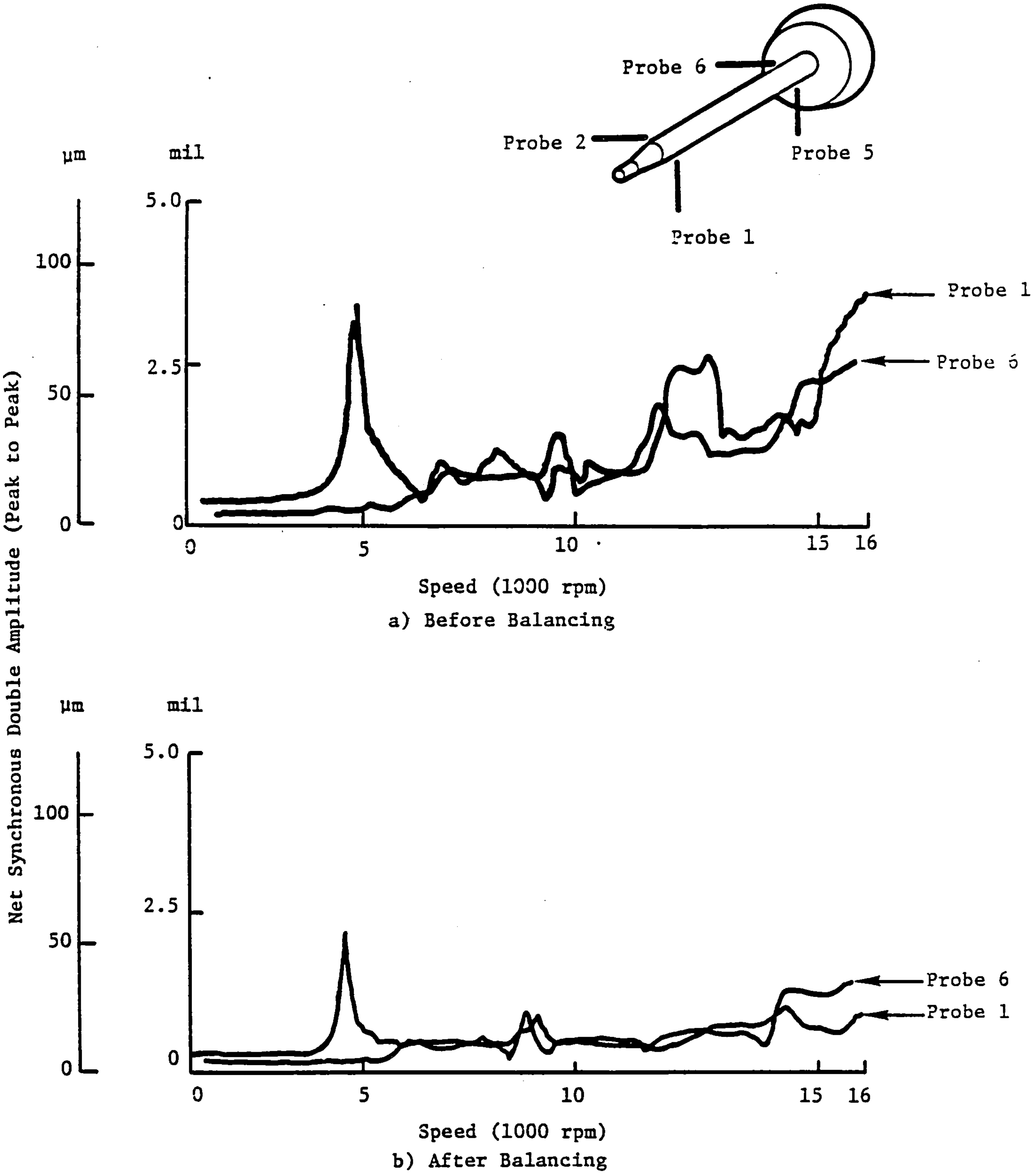

Figure 15 T55 Vibration Level in HSBS (Turbine U00559)

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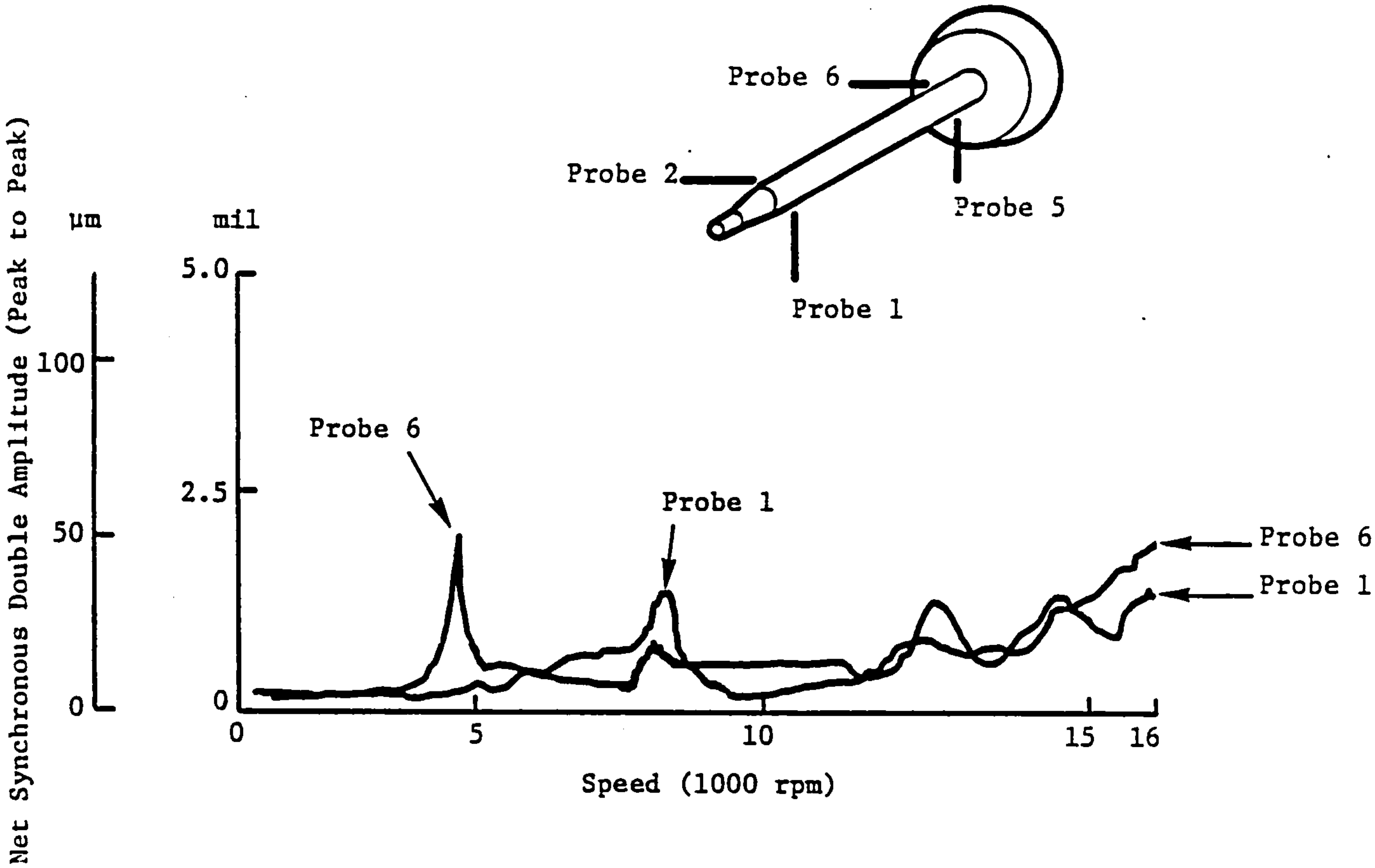

Figure 16 T55 (U00559) Vibration Level After Permanent Balance Weight Grind

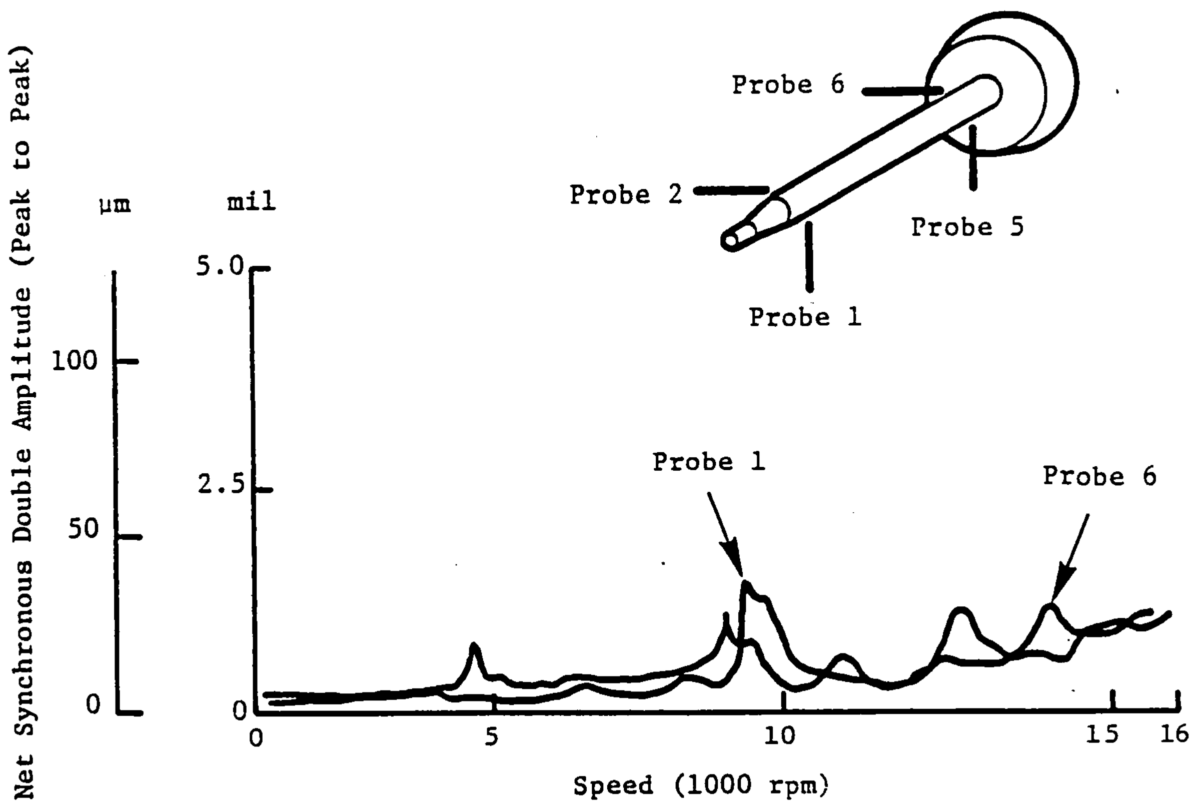

Figure 17 T55 (Serial No. 268922) After Permanent Balance Weight Grind

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3.2.6 Checkout of New Thrust Bearing Design

The redesigned thrust bearing assembly included the following new features:

- Bearing maximum speed of 22,000 rpm at 900 N (200 lb) preload
- More rigid thrust pad
- Precision manufacture to improve alignment
- A four jet oil mist aimed at the bearing's inner race.

The design was first checked out with a T55 power turbine installed in the HSBS. No change in response was noted for preload changes from 270 to 530 N (60 to 120 lb). Higher thrust was not tried.

The balancing system was switched over to T53 supports for additional testing. After a baseline run of the installed T53 power turbine, the existing influence coefficients were used to calculate the correction weights required. A two-speed 2,750 and 13,000 rpm, three-plane balance resulted in satisfactory amplitudes up to 21,000 rpm. Preload is applied by moving the thrust pad assembly until it contacts the thrust bearing housing. Further motion compresses a spring and develops a thrust. Rotor vibration response was compared from 220 to 710 N (50 to 160 lb) preload. Spline shifting problems were minimized at 620 N (140 lb) preload although any amount between 530 and 710 N (120 and 160 lb) was acceptable. The tolerance of 180 N (40 lb) allows the operator to adjust the preload by advancing the adjusting bolt 1-1/3 turns rather than the time-consuming use of a dial indicator to measure the 1.8 mm (0.070 in.) deflection. With the preload set at 530 N (120 lb), the bearing temperature was lower than the original design.

A method was established for permanent weight removal in the T53 disk. A large plastic sheet was used to cover the open vacuum chamber. Access to the area to be ground was provided by cutting a hole in the sheet and taping the edges to the disk. A backup seal for protecting the bearings was made by placing a foam strip between the disk and bearing housing. A 1.7 cm (5/8 in.) diameter grindstone ball was used in a high speed pneumatic tool to remove metal. A .10 cm (0.040 in.) deep cut over a 40° arc was calculated as the replacement for 2.93 grams of safety wire concentrated at the blade tip diameter. The permanent weight removal provided an acceptable replication of the temporary weight vibration response.

3.2.7 Shipping and Installation of the System

The HSBS was dissassembled and crated as required for shipping. The vacuum chamber base, motor, gearbox and two pumps were left attached to minimize reassembly efforts. The test cell at CCAD was only accessible through a 1.06 m (42 in.) wide doorway and a narrow corridor. Therefore, the 2270-kg (5000-lb) crate, whose dimensions were 1.36 x 1.03 x 3.51 m (53.5 x 40.5 x 138 in.), had to be handled on its side.

Two trips were made to CCAD by MTI personnel, the first to oversee the installation of the hardware, and the second to verify proper operation of the system.

3.3 Preproduction Evaluation of the High-Speed Balance Equipment

3.3.1 Engine Test Results

Two T55 power turbines that had been high-speed balanced at MTI were run in the HSBS after installation of the system in Corpus Christi. One of the two turbines, 268922, was checked in the HSBS during a demonstration given to CCAD, NASA, and AVRADCOM personnel.

After an initial run to operating speed at which the peak amplitude was 100 μm (4 mil), the system was demonstrated by using the data from this one run in conjunction with the stored influence coefficients to calculate a trim balance. The calculated weights were then installed and the power turbine was run to its operating speed to check the balance. The resultant peak amplitude was reduced to a level of 38 μm (1.5 mil). The trim weights were therefore ground into the rotor in preparation for engine testing. Figure 18 represents the vertical amplitudes at the forward and aft probes for both as installed, and after high speed trim balancing conditions.

The second T55 power turbine (U00559) was operated to its maximum speed of 16,000 rpm in the HSBS with a peak vibration reading of 50 μm (2 mil). The two turbines were inspected and approved by CCAD personnel prior to engine testing.

The first of the two power turbines was successfully run in an engine during a visit to CCAD by MTI personnel. The engine used for the test had been tested earlier in the week and then the power turbine was pulled out and replaced with the one incorporating high-speed balancing. This power turbine module (268922) had not been low-speed balanced prior to being high-speed balanced. The engine was retested and showed a decrease in overall engine vibration. Figure 19 shows the engine test cell vibration readings for the engine, before and after the power turbine changeout. The entire remainder of the engine, including harness and instrumentation was the same for both tests. Figure 20 represents a vibration spectrum plot taken from the engine test with the high-speed balanced turbine operating at 15,600 rpm (97.5%). This breakdown of the overall vibration signal into its various frequency components shows that the compressor (N1) frequency now dominates overall engine vibration. Any further reduction in the vibration levels measured by the casing mounted test cell instrumentation would require improving the balance of the compressor rotor to a level comparable to the high-speed balanced power turbine.

As mentioned previously the engine used for this test had been run earlier in the week with a different power turbine. The power turbine that was removed was tested in the HSBS to aid in further definition of vibration limits. At 16,000 rpm, the peak amplitude was 200 μm (8 mil) which indicates the limits of 130 μm (5 mil) established in the operator's manual are probably conservative since this power turbine had run acceptably in an engine.

The second high-speed balanced T55 power turbine (U00559) was engine tested at CCAD a month later. MTI personnel were not present during the test. The engine was rejected for high vibration, predominantly on the V3 sensor which is mounted on the waterbrake. CCAD personnel diagnosed the problems as a faulty waterbrake. Unfortunately, the test was not rerun with an alternate waterbrake. The test data from the run support the conclusion of a faulty

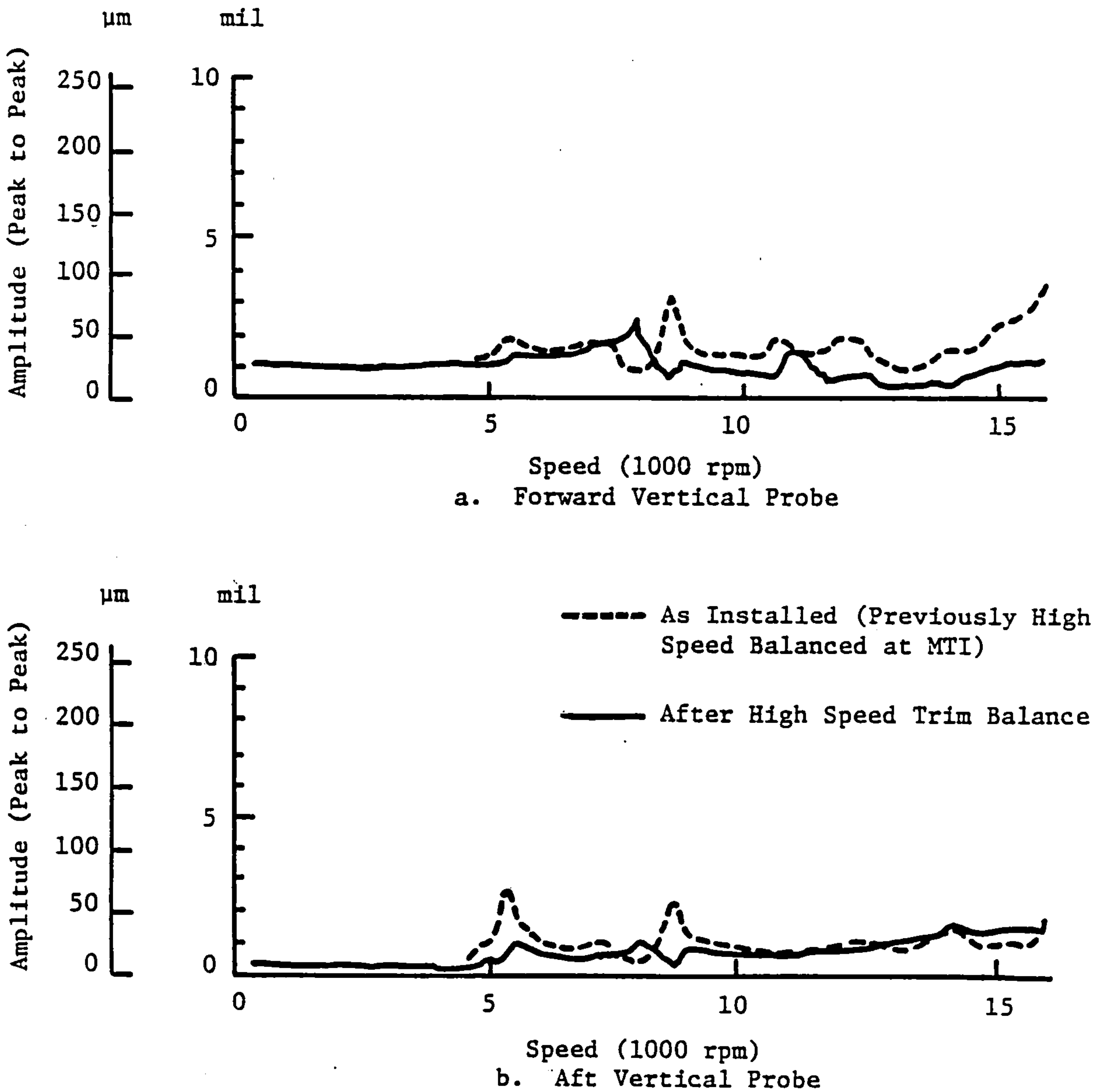

Figure 18 Vibration Level in HSBS, Turbine End (T55 Serial No. 268922)

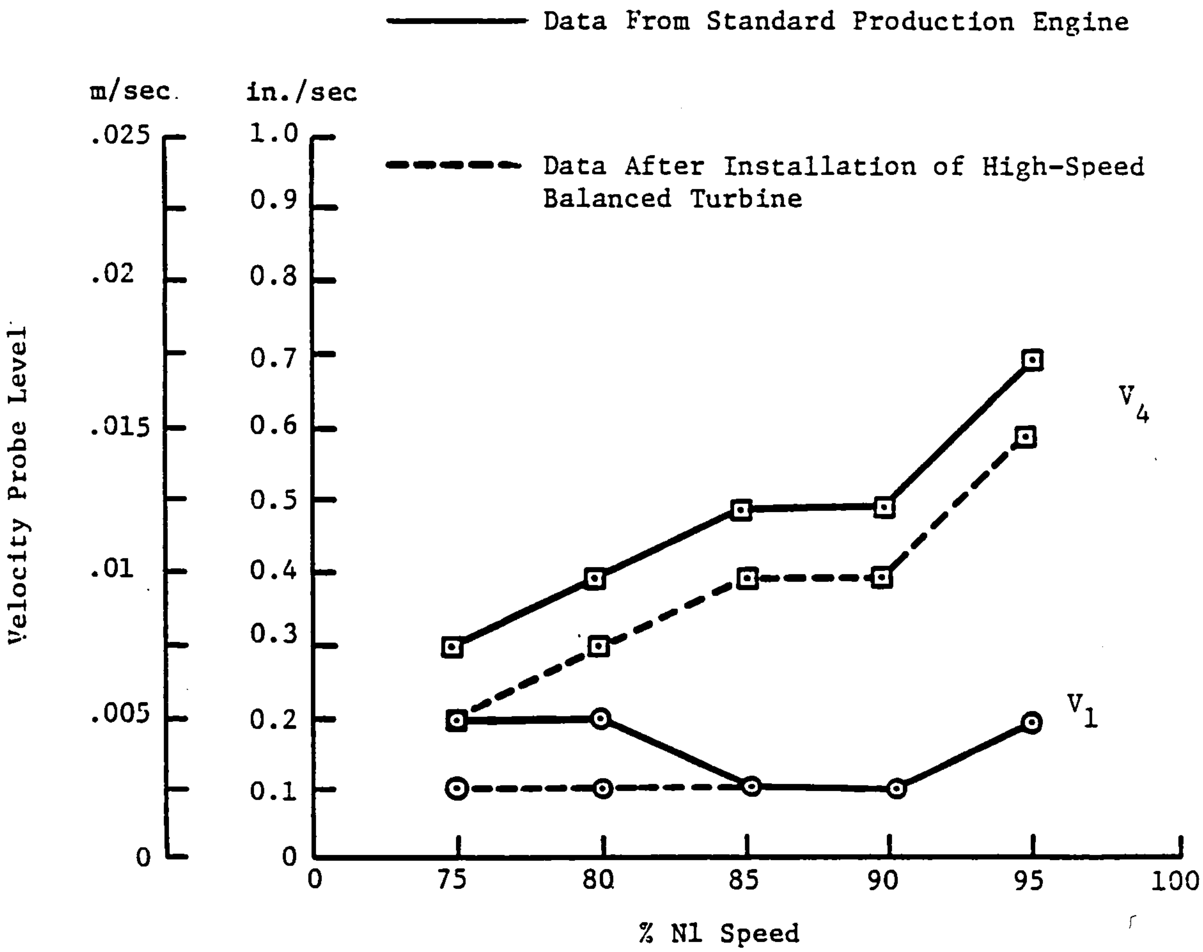

Figure 19 T55 Overall Engine Vibration

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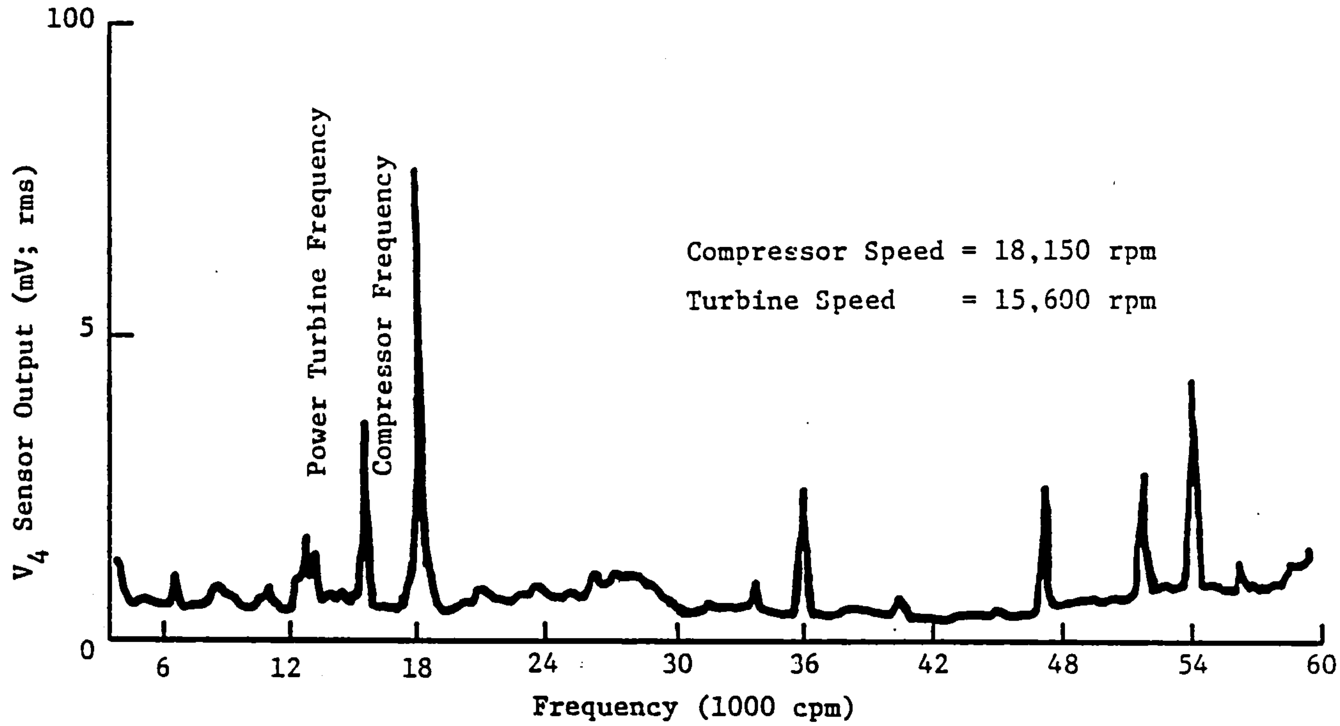

Figure 20 T55 Frequency Spectrum, Serial No. 268922

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waterbrake. Even at the lowest power engine setting the V3 sensor has an unacceptable level with all other sensors mounted on the engine reading very low. At the higher speeds there is some transfer of energy to the V4 sensor which is the closest engine sensor to the waterbrake.

3.3.2 Operations Manuals/Facility Repair Procedures

Four manuals and one set of engineering drawings are provided with the HSBS. The manuals are:

- Operators Manual - describes the basic operational procedures for: rotor installation and removal, high speed operation, computer operation, software user instructions and grinding instructions. Troubleshooting is outlined and maintenance intervals and spare parts are identified.

- System Hardware Manual - detailed information on the computer and data acquisition system, sufficient for skilled personnel to isolate faults.

- Software Manual - detailed information on the system's computer programs including listings of error messages and suggested corrective action.

- Vendor Manual - data sheets and maintenance instructions for all vendor supplied equipment.

3.4 Site Modification and Rig Installation

Subcontractors were assigned the task of preparing the former Test Cell 2B at CCAD for use as a high-speed balancing facility. The preparations in the test cell included the following items:

- preparation of floor mounting pads
- connections to adequate electrical, air and water supplies (included installation of an isolation transformer)
- reassembly of all balancing equipment components
- establishment of electrical & mechanical interconnections between components of the balancing system
- operational checkout of the entire system

Figure 21 details the layout of the hardware as installed in Test Cell 2B.

3.5 Vibration Analysis of Low and High-Speed-Balanced Power Turbines

This task involved checking and recording the HSBS-determined vibration characteristics of 20 T53 power turbines and 20 T55 power turbines. After operation in the HSBS, the power turbines were tracked through engine assembly and vibration characteristics of the assembled engine in the test cell were recorded and analyzed.

The power turbine modules of each type (T53 and T55) were divided into three lots. The power turbines in the first and second lots were low-speed balanced prior to testing in the HSBS. The power turbines in the third lot were not

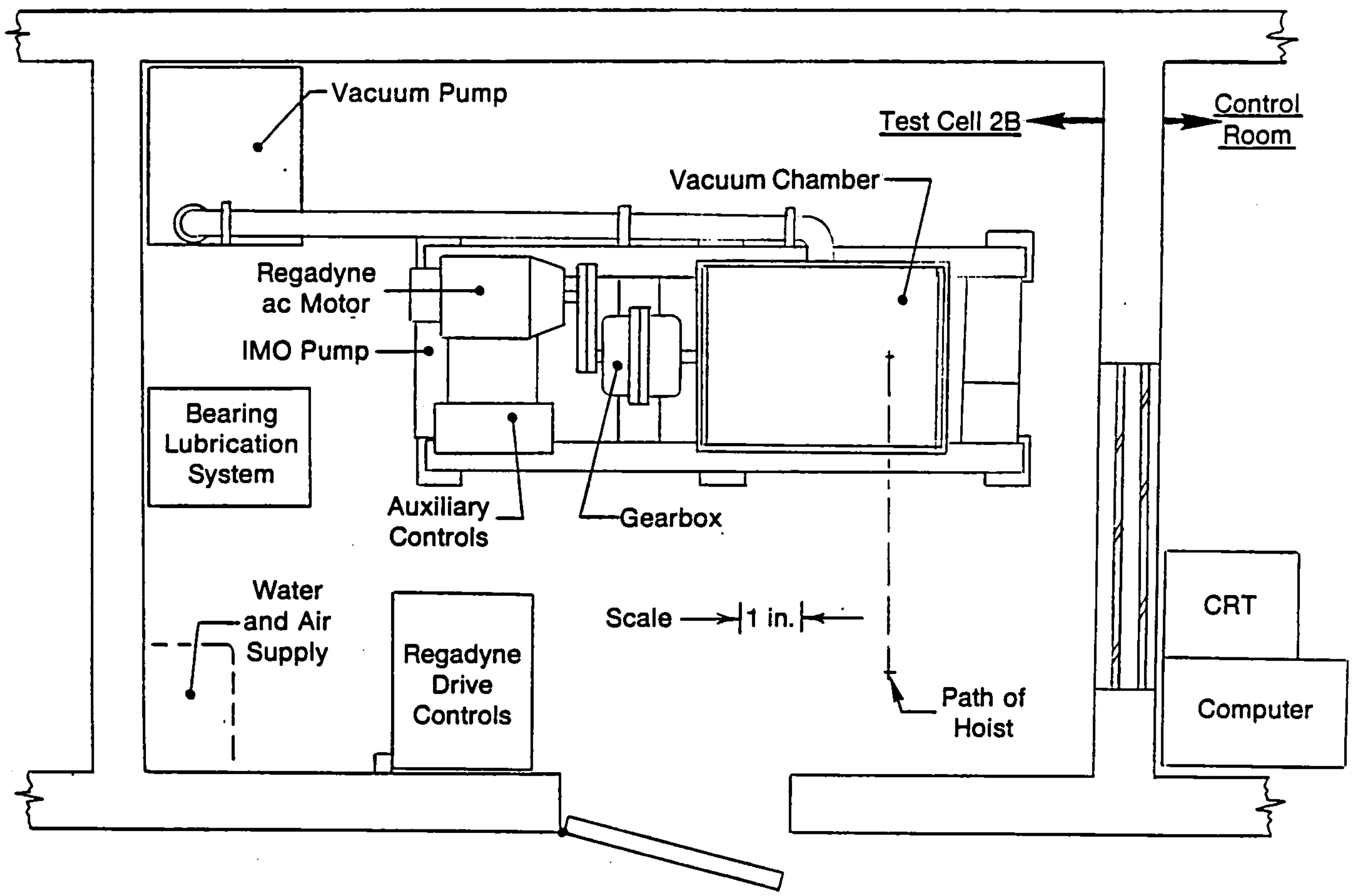

Figure 21 Layout of High Speed Balance Machine

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low-speed balanced. Lot 1 was tested in the HSBS but not high-speed balanced. These modules provided a baseline correlation between shaft vibration as measured by HSBS sensors and engine case vibration as measured by test cell sensors. The second and third lots of power turbines were high-speed balanced and tracked in a similar manner.

The large number of rotors evaluated in their task afforded the opportunity to use the HSBS to determine the influence of various power turbine parameters on engine vibration. Examples of the parameters studied that will be discussed in later sections include:

* dynamic imbalance between turbine stages
* spline fit variation
* low-speed balance results.

3.5.1 T55 Vibration Data from HSBS

The vibration data for all of the T55 power turbines tested in the HSBS is presented in Table 3. The general trend of vibration amplitudes versus rotor speed for the T55 power turbines was that of low amplitudes up to around 14,000 rpm followed by increasing levels up to the maximum speed of 16,000 rpm. All but one of the T55s was operated to its full speed in the as-installed condition. The only exception was a Lot 3 rotor that showed high amplitudes at 14,000 rpm. This agrees well with the analysis that predicts the onset of rotor bending in the mode shape of the second critical speed as the power turbine approaches it operating speed. Figure 22 represents a typical T55 amplitude versus speed plot. In all cases high speed balancing was effective in reducing the vibration levels at 16,000 rpm.

3.5.2 T53 Vibration Data from HSBS

The vibration data for all the T53 power turbines operated in the HSBS is presented in Table 4. The vibration amplitudes for the T53s showed a much higher variability than the levels measured for T55s. The low-speed balance data was recorded for the turbines of Lot 1 and was found to be of no value in predicting behavior in the HSBS. The vibration levels measured at 15,500 rpm in the HSBS for the Lot 1 turbines varied from 50 to 500 μm (2 to 20 mil) whereas the low-speed balance results were very tightly grouped.

After determining that the low-speed balance results did not correlate with the high-speed vibration levels, another explanation for the large variation in the T53 vibration levels was sought. Based on discussions with CCAD personnel concerning multiple reject engines that had been problems in the past, it was evident that the couple unbalance between the turbine stages of the T53 power turbine rotor was a potential cause of the high vibration levels measured for some of the rotors. The current production procedure at CCAD calls for a static balance on the third stage turbine, assembly of the third and fourth stages and then a static balance on the fourth stage to balance the assembly. This procedure can and does result in a large residual couple between the stages with some rotors. The low-speed balance machines at CCAD can measure the level of this 'dynamic unbalance' although it is not part of the current production procedure. For the purposes of these tests, the levels were checked for a group of 8 rotors. In addition, 3 of these rotors were dynamically balanced (couple removed) and tested in the HSBS. The vibration

TABLE 3

T55 VIBRATION DATA IN THE HSBS

Serial No.	Lot No.	Probe No.	Speed (rpm)	Amplitude As Installed Peak-to-Peak (μm)	(mil)	Amplitude After Balancing Peak-to-Peak (μm)	(mil)
U00259	1	1 3 5	16,000	114 163 71	4.5 6.4 2.8	No high-speed balancing was conducted for Lot No. 1 rotors.	
U00212	1	1 3 5	16,000	107 137 53	4.2 5.4 2.1		
268920	1	1 3 5	16,000	125 206 107	4.9 8.1 4.2		
265530	1	1 3 5	16,000	117 188 94	4.6 7.4 3.7		
261830	1	1 3 5	16,000	104 158 74	4.1 6.2 2.9		
265539	1	1 3 5	16,000	53 89 38	2.1 3.5 1.5		
95250	1	1 3 5	16,000	48 36 10	1.9 1.4 0.4		
265549	2	1 3 5	16,000	183 287 130	7.2 11.3 5.1	49 74 43	1.9 2.9 1.7
U00433	2	1 3 5	16,000	91 43 23	3.6 1.7 0.9	-- -- --	-- -- --
U00316	2	1 3 5	16,000	70 81 38	2.74 3.17 1.5	27 51 25	1.05 2.0 1.0

TABLE 3 (Cont'd)

Serial No.	Lot No.	Probe No.	Speed (rpm)	Amplitude As Installed Peak-to-Peak (μm)	(mil)	Amplitude After Balancing Peak-to-Peak (μm)	(mil)
U00589	2	1 3 5	16,000	142 218 99	5.6 8.6 3.9	104 140 69	4.1 5.5 2.1
268541	2	1 3 5	16,000	114 165 86	4.5 6.5 3.4	15 23 10	0.6 0.9 0.4
268934	2	1 3 5	16,000	208 325 158	8.2 12.8 6.2	69 84 31	2.7 3.3 1.2
265566	2	1 3 5	16,000	158 239 102	6.2 9.4 4.0	61 91 48	2.4 3.6 1.9
U00295	2	1 3 5	16,000	274 429 211	10.8 16.9 8.3	51 140 64	2.7 5.5 2.5
268922	2	1 3 5	16,000	224 318 163	8.8 12.5 6.4	140 142 59	5.5 5.6 2.3
265499	3	1 3 5	16,000	58 64 20	2.3 2.5 0.8	-- -- --	-- -- --
U00559	3	1 3 5	16,000	64 56 31	2.5 2.2 1.2	-- -- --	-- -- --
M265682	3	1 3 5	16,000	41 56 36	1.6 2.2 1.4	-- -- --	-- -- --
M254954	3	1 3 5	16,000	130 165 71	5.1 6.5 2.8	74 86 36	2.9 3.4 1.4
265604	3	1 3 5	14,000	150 249 132	5.9 9.8 5.2	81 76 19	3.2 3.0 0.75
265604	3	1 3 5	16,000	-- -- --	-- -- --	58 109 53	2.3 4.3 2.1
M261845	3	1 3 5	16,000	99 152 76	3.9 6.0 3.0	18 31 18	0.7 1.2 0.7

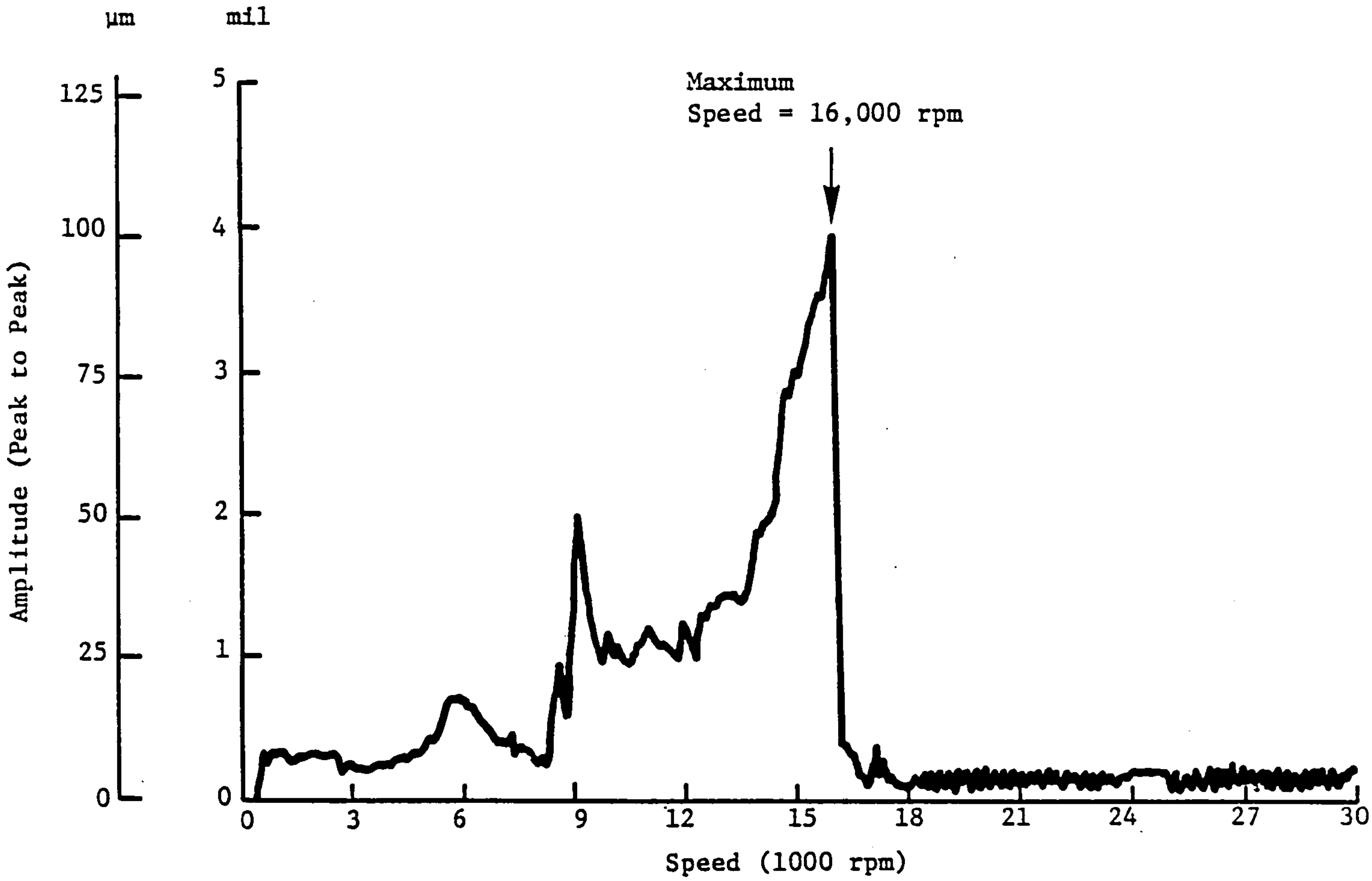


Figure 22 T55 Vibration Level in HSBS, Turbine End/Serial No. 265530

TABLE 4

T53 VIBRATION DATA IN THE HSBS

Serial No.	Lot No.	Probe No.	Speed (rpm)	Amplitude As Installed Peak-to-Peak (μm)	(mil)	Amplitude After Balancing Peak-to-Peak (μm) (mil)
1896	1	1 3 5	17,200	145 254 226	5.7 10.0 8.9	No high-speed balancing was conducted for Lot No. 1 rotors.
5247	1	1 3 5	17,200	46 79 53	1.8 3.1 2.1	
8873	1	1 3 5	21,000	53 25 25	2.1 1.0 1.0	
1377	1	1 3 5	15,500	173 180 152	6.8 7.1 6.0	
5586	1	1 3 5	15,500	81 165 234	3.2 6.5 9.2	
U00278	1	1 3 5	17,200	102 198 155	4.0 7.8 6.1	
C9M11244	1	1 3 5	17,200	152 132 102	6.0 5.2 4.0	
2484	1	1 3 5	15,500	104 61 69	4.1 2.4 2.7	
9925	1	1 3 5	17,200	53 74 86	2.1 2.9 3.4	
1541	1	1 3 5	17,200	69 53 38	2.7 2.1 1.5	

TABLE 4 (Cont'd)

Serial No.	Lot No.	Probe No.	Speed (rpm)	Amplitude As Installed Peak-to-Peak (μm)	(mil)	Amplitude After Balancing Peak-to-Peak (μm)	(mil)
LE21788	2	1 3 5	15,500	76 86 119	2.99 3.4 4.7	49 37 32	1.91 1.47 1.26
M289659	2	1 3 5	15,500	94 66 81	3.7 2.6 3.2	81 53 119	3.2 2.1 4.7
12956	2	1 3 5	17,200	142 325 246	5.6 12.8 9.7	17 43 43	.67 1.66 1.7
96441	2	1 3 5	15,500	61 76 76	2.4 3.0 3.0	33 38 25	1.3 1.5 1.0
P5132	2	1 3 5	17,200	107 295 285	4.2 11.6 11.2	41 38 69	1.6 1.5 2.7
3494	2	1 3 5	17,200	185 150 180	7.3 5.9 7.1	81 84 56	3.2 3.3 2.2
82C004	3	1 3 5	15,500	221 277 259	8.7 10.9 10.2	51 714 122	2.0 5.3 4.8
82C012	3	1 3 5	15,500	150 191 114	5.9 7.5 4.5	84 66 89	3.3 2.6 3.5
80K003P	3	1 3 5	15,500	218 274 226	8.6 10.8 8.9	69 56 36	2.7 2.2 1.4
4871	3	1 3 5	17,200	315 490 358	12.4 19.3 14.1	191 201 84	7.5 7.9 3.3
82C011	3	1 3 5	17,200	56 150 79	2.2 5.9 3.1		

levels as measured by the HSBS for these 8 rotors correlated directly to the dynamic unbalance levels measured by the CCAD low-speed balance machines. This correlation was confirmed by the engine test results for the engines containing these power turbines. The engine test results are discussed in the following section.

Dynamically balanced power turbine 8873 was operated to its maximum speed of 21,000 rpm with extremely low vibration levels. A plot of the vibration data from the turbine end probe for a run to 20,600 rpm is included as Figure 23. The relatively constant low amplitude shown from 15,000 - 21,000 rpm is only observed for dynamically balanced power turbines and the best of the high-speed balanced turbines. The other two dynamically balanced T53 power turbines, 1541 and 9925, were also tested to near 21,000 rpm. Spline shafts prevented the acquisition of data at 21,000 rpm. Plots of vibration amplitude versus speed for rotors 1541 and 9925, operated in the HSBS are included as Figures 24 and 25, respectively. The T53 turbine assemblies that were found to have the highest level of couple imbalance were 12956 and 4871. The levels for 12956 measured on the low-speed balance machine at CCAD were 15.4 gm in. at 137° in the third stage turbine and 16.8 gm in. at 318° in the fourth stage. The angles given are relative to a reference mark used by the operator. Since the unbalance levels measured are very close to equal and opposite (i.e. 180° phase difference) the assembled turbine stages meet the currently used production balance specification. The resulting amplitude for the shaft center probe in the HSBS was 325 μm (12.8 mil) at 17,200 rpm and rising rapidly with increasing speed.

Rotor 4871 was an unusual case. The third stage turbine disk was brand new and was not low speed balanced. The low-speed balance check indicated a dynamic unbalance level of 133 gm cm (52.5 gm in.) at the third stage and 27 gm cm (10.5 gm in.) at the fourth stage. The static check which is the only one used in the current procedure yields the difference between the two vectors, 98 gm cm (38.5 gm in.). The resulting amplitudes for the shaft center probe in the HSBS was 490 μm (19.3 mil) at 17,200 and also rising rapidly.

The high-speed balancing of the T53 power turbine was successful in reducing amplitudes for each rotor, although much more difficult than the T55 balancing. The sensitivity to the dynamic unbalance along with problems with spline shifts cause the extra difficulty. The T55 turbine disks are dynamically balanced. For the T53 power turbines with any significant third stage balance (only encountered in new parts rotors, Lot 3) the high-speed balance quality is compromised due to the unavailability of the third stage as a correction plane. Therefore, any correction to compensate for the third stage unbalance must be installed on the fourth stage introducing a couple.

The balancing software was effective in reducing the amplitude for these worst-case rotors but was not capable of producing the low-amplitude levels reached by eliminating the dynamic unbalance directly. The high-speed balancing of the T53 rotors in Lot 2 and those in Lot 3 that did not contain new parts was very successful and resulted in smoother running T53 engines.

3.5.3 T53 Engine Test Cell Data

The T53 engine is instrumented with three case-mounted sensors when operated in the engine test cell. The engine is set at specific test points that are

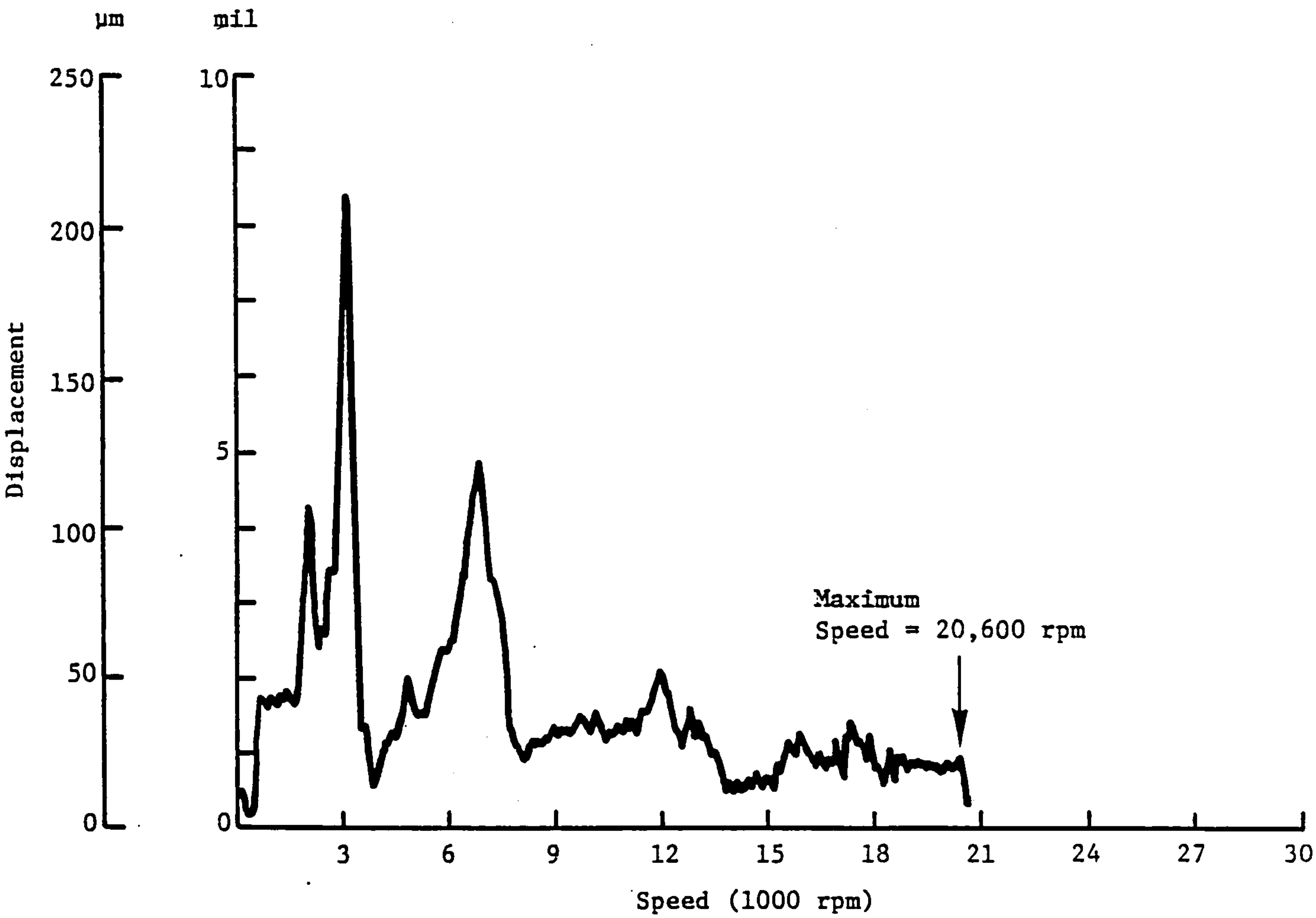


Figure 23. T53 Vibration Level in HSBS, Turbine End/Serial No. 3873

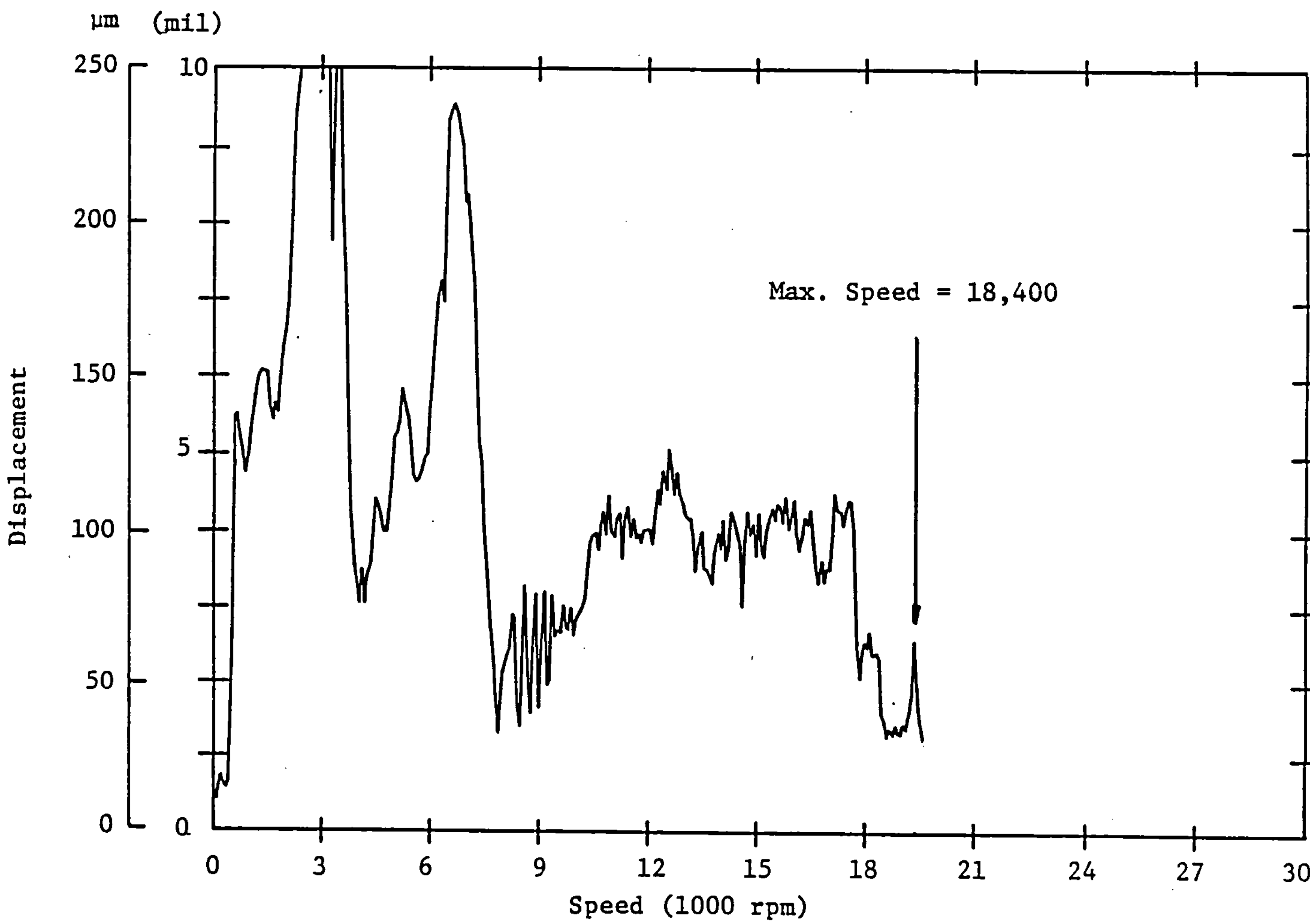


Figure 24 T53 Vibration Level in HSBS, Turbine End/Serial No. 1541

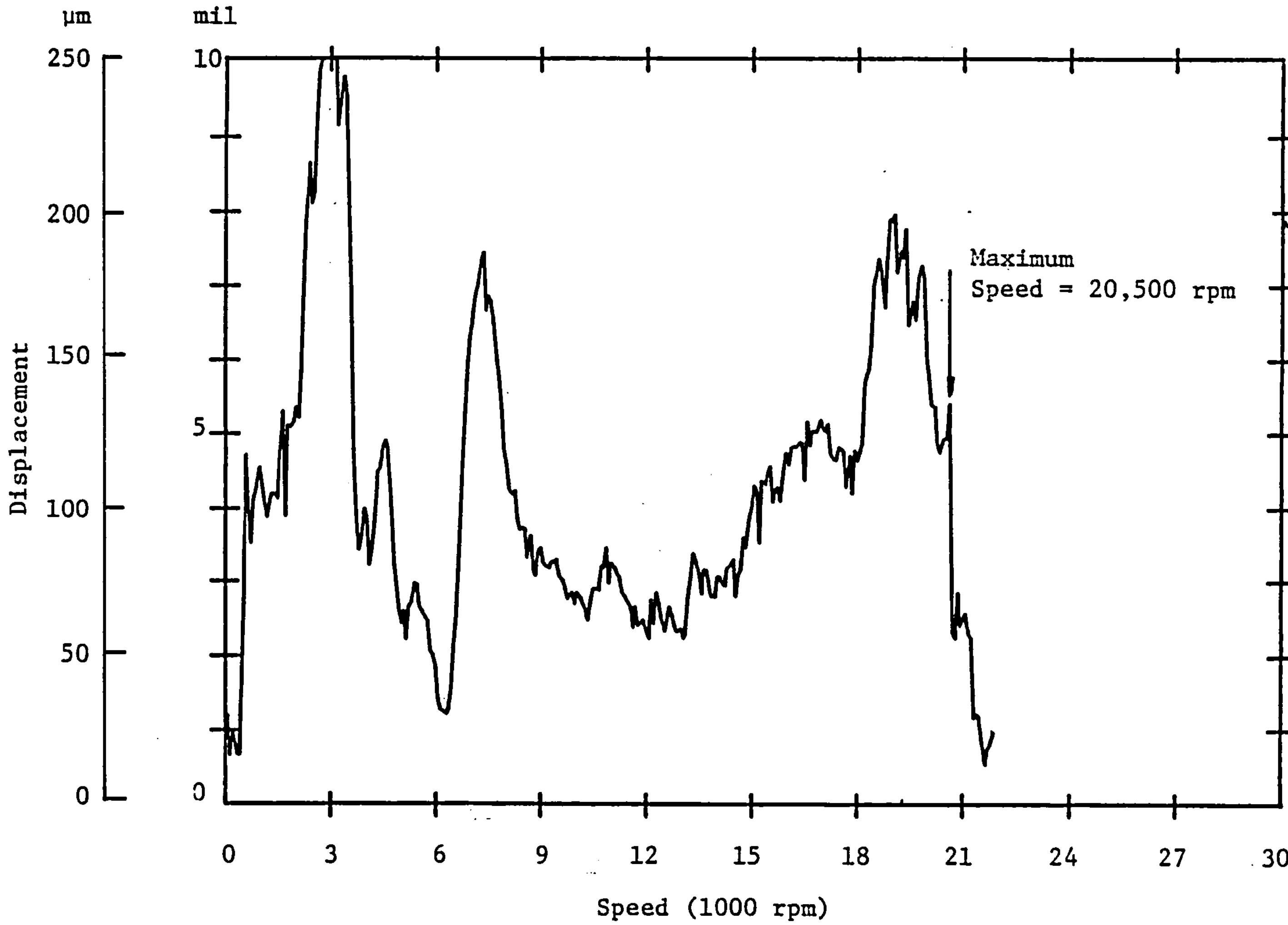


Figure 25 T53 Vibration Level in HSBS, Turbine End/Serial No. 9925

defined by the two rotor speeds (power turbine rotor and compressor rotor) and the overall vibration reading from each sensor recorded and compared to a specified limit. The term, overall vibration, refers to the total case vibration that is made up of the sum of all the various components at different frequencies. The overall vibration will therefore include numerous components at discrete frequencies such as: a contribution at the power turbine freuqency, a contribution from the compressor rotor, contributions from any of the engine auxilliaries running off one of the main rotors, and a contribution from the test cell waterbrake, among others. Table 5 presents the overall vibration reading from the test cell sensor on the turbine end of the engine (V3) for all of the T53 engines tested. This sensor is the most sensitive to the power turbine vibration. The T53 engine test cell data is characterized by a dominant power turbine frequency that makes up a large percentage of the overall vibration for most of the engines tested. For this reason, changes in the T53 power turbine balance show up clearly as changes in overall engine vibration. A frequency spectrum plot for each T53 engine tested appears in Appendix A. The plots are a breakdown of the overall vibration signal from the V3 sensor into its various frequency peaks, taken from the vibration test point with the power turbine operating at 100% speed (√21,000 rpm). The measured overall vibration at the test point is included on each plot.

The objective of running the Lot 1 power turbines was twofold. First, it determined if a correlation exists between the levels measured in the HSBS and the vibration resulting from the power turbine in the assembled engine. Second, it determined what levels in the HSBS would result in acceptable engine vibration levels. An acceptable engine is determined by the vibration specification listed in the DMWR; for the V3 sensor on the T53, the allowable limit is 81 μm (3.2 mil) above 85% power turbine speed and is determined by a curve for the lower speeds. The HSBS data showed the worst levels of Lot 1 for power turbine 1896 and the best levels for turbine 8873. The engine test results agreed with the HSBS results; turbine 1896 was very near to being a vibration reject with a peak level of 94 μm (3.7 mil) at the 80/80 check point. On the other hand, turbine 8873, which had been run to 21,000 in the HSBS, showed a peak level of only 41 μm (1.6 mil). The correlation between HSBS levels and engine test results was confirmed by the T53 turbines from Lots 2 and 3. The dynamically balanced power turbines (8873, 1541, 9925) all showed very low vibration levels in the HSBS and resulted in exceptionally smooth running engines. Power turbines P5132 and LE2178 were the best of the high speed balanced power turbines and also resulted in very smooth engines. The peak vibration levels for these two engines were 23 and 33 μm (.9 and 1.3 mil), respectively.

The T53 power turbines that resulted in the worst running engines were 1896, 5586, 80K03P, 82C012, and 4871. As previously mentioned, 1896 was the roughest running Lot 1 power turbine in the HSBS and power turbine 5586 was the second worst Lot 1 rotor. The other three power turbine rotors are Lot 3 rotors that were tested as examples of the worst case non-low-speed-balanced turbines that could possibly occur in the overhaul cycle. This required the use of completely new turbine disks and blades that had not been low-speed balanced. The three power turbines were assembled with three different angular orientations of the two turbine disk's imbalances. That is, the heavy area of the two disks was in line in one case, 180° and 90° apart in the other two cases. This was done to thoroughly test the ability of the HSBS to handle actual nonbalanced parts. The other Lot 3 rotors tested were

TABLE 5

T53 ENGINE TEST CELL DATA, V4 SENSOR

Power Turbine Serial Number		Speed - N1%/N2% 75/75	80/80	85/97	90/95	90/97	91/100	95/97	100/97
82C011	µm	51	46	31	28	28	23	28	38
	mil	2.0	1.8	1.2	1.1	1.1	0.9	1.1	1.5
4871	µm	REJECTED AT IDLE							
	mil								
LE21788	µm	33	23	13	18	18	18	20	23
	mil	1.3	0.9	0.5	0.7	0.7	0.7	0.8	0.9
M289659	µm	53	76	74	71	71	—	—	—
	mil	2.1	3.0	2.9	2.8	2.8	—	—	—
1541	µm	10	10	18	18	18	18	15	20
	mil	0.4	0.4	0.7	0.7	0.7	0.7	0.6	0.8
82C012	µm	84	79	64	79	74	69	69	74
	mil	3.3	3.1	2.5	3.1	2.9	2.7	2.7	2.9
82C004	µm	53	89	64	79	64	53	74	—
	mil	2.1	3.5	2.5	3.1	2.5	2.1	2.9	—
80K03P	µm	61	99	64	53	53	48	—	—
	mil	2.4	3.9	2.5	2.1	2.1	1.9	2.3	2.1
U00278	µm	23	25	15	15	15	13	20	15
	mil	0.9	1.0	0.6	0.6	0.6	0.5	0.8	0.6
C9M11244	µm	43	28	23	18	23	23	23	33
	mil	1.7	1.1	0.9	0.7	0.9	0.9	0.9	1.3
9925	µm	41	22	10	10	8	8	13	10
	mil	1.6	0.9	0.4	0.4	0.3	0.3	0.5	0.4
2484	µm	36	41	33	31	33	38	38	38
	mil	1.4	1.6	1.3	1.2	1.3	1.5	1.5	1.5
P5132	µm	23	18	18	18	18	18	18	18
	mil	0.9	0.7	0.7	0.7	0.7	0.7	0.7	0.7
96441	µm	25	64	41	41	41	41	41	—
	mil	1.0	2.5	1.6	1.6	1.6	1.6	1.6	—
12956	µm	56	48	43	36	33	20	46	56
	mil	2.2	1.9	1.7	1.4	1.3	0.8	1.8	2.2
1896	µm	84	94	38	41	43	43	33	—
	mil	3.3	3.7	1.5	1.6	1.7	1.7	1.3	—
5247	µm	74	25	33	25	33	36	38	23
	mil	2.9	1.0	1.3	1.0	1.3	1.4	1.5	0.9
8873	µm	41	33	36	33	31	31	36	—
	mil	1.6	1.3	1.4	1.3	1.2	1.2	1.4	—
1377	µm	53	28	28	23	28	28	28	53
	mil	2.1	1.1	1.1	0.9	1.1	1.1	1.1	2.1
5586	µm	53	81	18	15	15	15	15	18
	mil	2.1	3.2	0.7	0.6	0.6	0.6	0.6	0.7

non-low-speed-balanced as far as that particular overhaul was concerned. Since the rotors had been low speed balanced at one point they were only slightly out of the balance specification. This latter case involving the non-low-speed balanced rotor represents the typical rotor that is found in the overhaul cycle. The high-speed balancing of the three rotors involving new turbine disks was difficult due to the excessive unbalance at the inaccessible third stage disk. Power turbine 4871 had a third stage disk that was 132 gm cm (52 gm in.) out of balance. The initial run of this power turbine in the HSBS was characterized by very large vibration amplitudes measured at the lowest balance speed, 1000 μm (40 mil) at 2750 rpm, as well as high levels at the top balance speed, 482.6 μm (19 mil) at 17,200 rpm. The balance calculation based on this run gave correction weights too large to be ground in the rotor. Using the rebalance option in the balancing program to calculate correction weights based on only the upper balance speeds resulted in a more practical correction weight set. Installing this weight set in the form of temporary weights reduced the 483 μm (19 mil) amplitude at 17,200 rpm to 201 μm (7.9 mil) but did not affect the peak at 3200 rpm. These weights were therefore ground in the rotor on the assumption that the engine would not see any effect from the 3200 rpm peak due to the rapid acceleration through this speed. The engine test results refuted this assumption, as the engine rejected at idle and was not accelerated to any of the vibration check points. This test clearly demonstrated the need to retain the 2750 rpm balance speed.

The other two power turbines containing new parts that had not been low speed balanced were 80K003P and 82C012. Both of these power turbines were successfully high speed balanced up to 15,500 rpm using the same balance plan as the Lot 2 rotors. After the engine test results were reviewed, the unbalance levels measured for these rotors were simulated by the addition of weights to a well-balanced rotor to allow further testing in the HSBS. The tests indicated that for the case of a large unbalance installed on the third stage turbine the amplitudes increased much more sharply between 15,500 and 17,200 rpm than the Lot 2 rotors. Balancing the Lot 2 rotors to a top speed of 15,500 rpm had worked well and eliminated the recurrent spline shift problems. The increase in vibration between these two balance speeds demonstrated for the extreme Lot 3 imbalance levels explains why the engine test levels were higher than the HSBS results would indicate. High speed balancing up to 17,200 rpm would have improved the engine test results but would not match the low levels of the dynamic balanced rotors due to the known imbalance at the inaccessible third stage turbine balance plane.

During the period of MTI's testing at CCAD, a total of 151 T53 engines were run in the test cells out of which twenty failed for high vibration at sensor number 3 (the sensor most indicative of power turbine imbalance). Of the twenty failures, eighteen were corrected by dynamically balancing the power turbine, after which all eighteen passed testing with levels of 1.0 mil or less in the succeeding run. The other two were checked "by the book", i.e., static balanced, reassembled, and retested with the same poor results - rejection because of high vibration. They were then dynamically balanced and passed engine testing with exceptionally low levels.

It was a common occurrence a few years ago at CCAD to have engines rejected five to ten times in a row for this same reason. It is probable that the twenty rejects in this group would have resulted in a hundred or more wasted engine tests had they not been dynamically balanced. Unfortunately, a request to change the balancing procedure was turned down; the following reasons were cited:

- The number of engines failing test is too small
- The cost to change the procedure would probably exceed savings that would result.

Not only would the proposed change in procedure have eliminated the twenty rejects and the multiple time rejects, but also the decrease in overall engine vibration for all T53 engines would have extended the life of numerous engine components and provided the aircraft's crew a less fatiguing environment.

3.5.4 T55 Engine Test Results

Table 6 presents the engine test results from the twenty T55 engines tested. The data was obtained from the V4 sensor which is the hot section sensor equivalent to the V3 sensor discussed for T53 engines. The vibration limit for this sensor is .024 m/s (.94 in./sec). Generally the T55 engine test data is characterized by a large contribution to the overall vibration from the compressor rotor. Both the HSBS data and the test cell data indicates that CCAD's current T55 power turbine production balancing procedure yields smooth running power turbines. All Lot 1 power turbines were operated to the full speed of 16,000 rpm in the HSBS and all resulted in acceptable engines. Appendix B includes the frequency spectrum plots for each of the engines, operating at the 90/100 vibration check point.

The three smoothest running T55 engines tested were those containing power turbines 95250, M261845 and U00316. All of these engines during the engine test had a peak value of .0150 m/s (.6 in./sec) or less. Power turbine 95250 was the smoothest running Lot 1 rotor tested in the HSBS with a peak vibration reading of 48 μm (1.9 mil) at 16,000 rpm. Power turbine M261845, a non-low-speed-balanced unit (Lot 3), was the most successful high speed balance with a peak reading of 30 μm (1.2 mil) at 16,000 rpm in the HSBS. Turbine U00316 was a high speed balanced unit from Lot 2 that exhibited a peak vibration in the HSBS of 51 μm (2.0 mil) at 16,000 rpm. All of the Lot 2 turbines resulted in acceptable engines.

Of the five T55 engines that exhibited the highest vibration levels in the engine test cell, 265530 and U00259 were two of the roughest Lot 1 power turbines tested in the HSBS. Power turbine serial number U00295 was the least successful high-speed balanced turbine from Lot 2. The final peak amplitude in the HSBS was 140 μm (5.5 mil), a level slightly below the levels recorded for the two Lot 1 rotors previously discussed. These three rotors help establish the maximum allowable vibration specification for high-speed-balanced T55 power turbines. The other two power turbines involved in the five engines with the highest vibration levels were Lot 2 rotors. In each case, the compressor rotor contributed more to overall vibration than the power turbine.

3.5.5 T55 Gas Producer Balance

The original intent of the vibration analysis was to evaluate the power turbines in the HSBS, in the engine test cell, and in the aircraft. The aircraft tests were found to be unworkable for the overhaul center; additionally, verification of the test cell results had already been done. It was

TABLE 6

T55 ENGINE TEST CELL DATA, V3 SENSOR

Power Turbine Serial Number		Speed - N1%/N2% 75/95	80/94	85/95	89.7/90	89.7/94.5	89.7/100	93/102	94/94
M265499	m/s	.0075	.0075	.0125	.0125	.0125	.0125	.0125	.0175
	in/sec	.3	.3	.5	.5	.5	.5	.5	.7
95250	m/s	.0050	.0100	.0100	.0100	.0100	.0100	.0150	.0150
	in/sec	.2	.4	.4	.4	.4	.4	.6	.6
U00316	m/s	.0075	.0075	.0100	0.100	.0100	.0125	.0150	.0150
	in/sec	.3	.3	.4	.4	.4	.5	.6	.6
265539	m/s	.0125	.0125	.0125	.0125	.0175	.0125	.0175	.0175
	in/sec	.5	.5	.5	.5	.7	.5	.7	.7
U00589	m/s	.0125	.0125	.0125	.0125	.0125	.0125	.0175	.0175
	in/sec	.5	.5	.5	.5	.5	.5	.7	.7
265604	m/s	.0200	.0225	.0200	.0150	.0175	.0225	.0275	.0250
	in/sec	.8	.9	.8	.6	.7	.9	1.1	1.0
268922	m/s	.0075	.0075	.0075	.0075	.0125	.0125	.0125	.0175
	in/sec	.3	.3	.3	.3	.5	.5	.5	.7
268541	m/s	.0125	.0175	.0175	.0175	.0175	.0175	.0225	.0255
	in/sec	.5	.7	.7	.7	.7	.7	.9	.9
265566	m/s	.0175	.0175	.0175	.0175	.0175	.0225	.0225	.0225
	in/sec	.7	.7	.7	.7	.7	.9	.9	.9
U00295	m/s	.0125	.0125	.0175	.0175	.0175	.0225	.0225	.0225
	in/sec	.5	.5	.7	.7	.7	.9	.9	.9
U00433	m/s	.0025	.0025	.0075	.0125	.0075	.0075	.0125	.0175
	in/sec	.1	.1	.3	.5	.3	.3	.5	.7
265549	m/s	.0075	.0125	.0125	.0125	.0125	.0125	.0125	.0175
	in/sec	.3	.5	.5	.5	.5	.5	.5	.7
M265682	m/s	.0075	.0075	.0075	.0125	.0125	.0125	.0175	.0175
	in/sec	.3	.3	.3	.5	.5	.5	.7	.7
M261845	m/s	.0075	.0125	.0125	.0125	.0125	.0125	.0125	.0125
	in/sec	.3	.5	.5	.5	.5	.5	.5	.5
254954	m/s	.0050	.0100	.0100	.0100	.0150	.0150	.0100	.0150
	in/sec	.2	.4	.4	.4	.6	.6	.4	.6
U00259	m/s	.0075	.0100	.0125	.0150	.0150	.0150	.0225	.0200
	in/sec	.3	.4	.5	.6	.6	.6	.9	.8
265530	m/s	.0150	.0150	.0225	.0175	.0175	.0175	.0225	.0225
	in/sec	.6	.6	.9	.7	.7	.7	.9	.9
261830	m/s	.0050	.0050	.0050	.0075	.0075	.0100	.0125	.0125
	in/sec	.2	.2	.2	.3	.3	.4	.5	.5
268934	m/s	.0100	.0100	.0100	.0100	.0100	.0100	.0125	.0125
	in/sec	.4	.4	.4	.4	.4	.4	.5	.5
U00212	m/s								
	in/sec								

the opinion of depot personnel that if an improvement in vibration levels was demonstrated in the engine test cell, the correlation to improved airframe vibration was established. For these reasons, the effort planned for the airframe tests was refocused to investigate the reason for the large contribution from the T55 compressor rotor to the overall level of engine vibration. CCAD personnel were interested in a possible correlation between high vibration attributed to the compressor and the dynamic imbalance between the gas producer stages of the compressor rotor. The reason for the interest was the similarity between the gas producer (size and bearing configuration) and the T53 power turbine stages which had been conclusively shown to have a strong effect on engine vibration, if dynamically out of balance.'

Seven compressor rotors were studied and readings recorded using CCAD low-speed balance machines. The dynamic imbalance measurements for the gas producers of these seven compressor rotors is presented in Table 7. Using the frequency spectrum plots to compare the magnitude of vibration at the compressor frequency (for each of the seven engines), it is evident that the magnitude does not follow the level of dynamic imbalance recorded. This is illustrated by comparing Figures 26 and 27, the spectrum plots for engine sequence numbers 843-3005 and 843-2027. The corresponding dynamic imbalance recorded for the gas producers of these two engines is 14.7 gm-cm (5.8 gm in.) and 10.7 gm-cm (4.2 gm in.) for 843-3005 and 83.8 gm-cm (33 gm in.) and 71.1 gm-cm (28 gm in.) for sequence number 843-2027. The spectrum analysis plot for 843-2027 shows a significantly lower peak at the compressor frequency than the engine with the low level of dynamic imbalance on the gas producer. It is clear from these results that the cause of the vibration of the compressor rotor is imbalance located along the rotor and not dynamic imbalance of the gas producer stages.

TABLE 7

T55 GAS PRODUCER LOW-SPEED BALANCE CHECK

T55 Engine Sequence No.	Power Turbine Serial No.	Gas Producer Imbalance		
841-3004	U00433	1st stage 2nd stage	22.9 gm-cm 25.4 gm-cm	9 gm-in. @ 143° 10 gm-in. @ 302°
841-3006	265549	1st stage 2nd stage	23.6 gm-cm 19.1 gm-cm	9.3 gm-in. @ 152° 7.5 gm-in. @ 355°
841-3007	265604	1st stage 2nd stage	10.2 gm-cm 7.1 gm-cm	4.0 gm-in. @ 306° 2.8 gm-in. @ 184°
843-3005	M265682	1st stage 2nd stage	14.7 gm-cm 10.7 gm-cm	5.8 gm-in. @ 75° 4.2 gm-in. @ 298°
843-3007	M261845	1st stage 2nd stage	31.8 gm-cm 41.9 gm-cm	12.5 gm-in. @ 42° 16.5 gm-in. @ 232°
843-3001	268934	1st stage 2nd stage	30.5 gm-cm 43.2 gm-cm	12 gm-in. @ 140° 17 gm-in. @ 316°
843-2027	M265499	1st stage 2nd stage	83.8 gm-cm 71.1 gm-cm	33 gm-in. @ 64° 28 gm-in. @ 247°

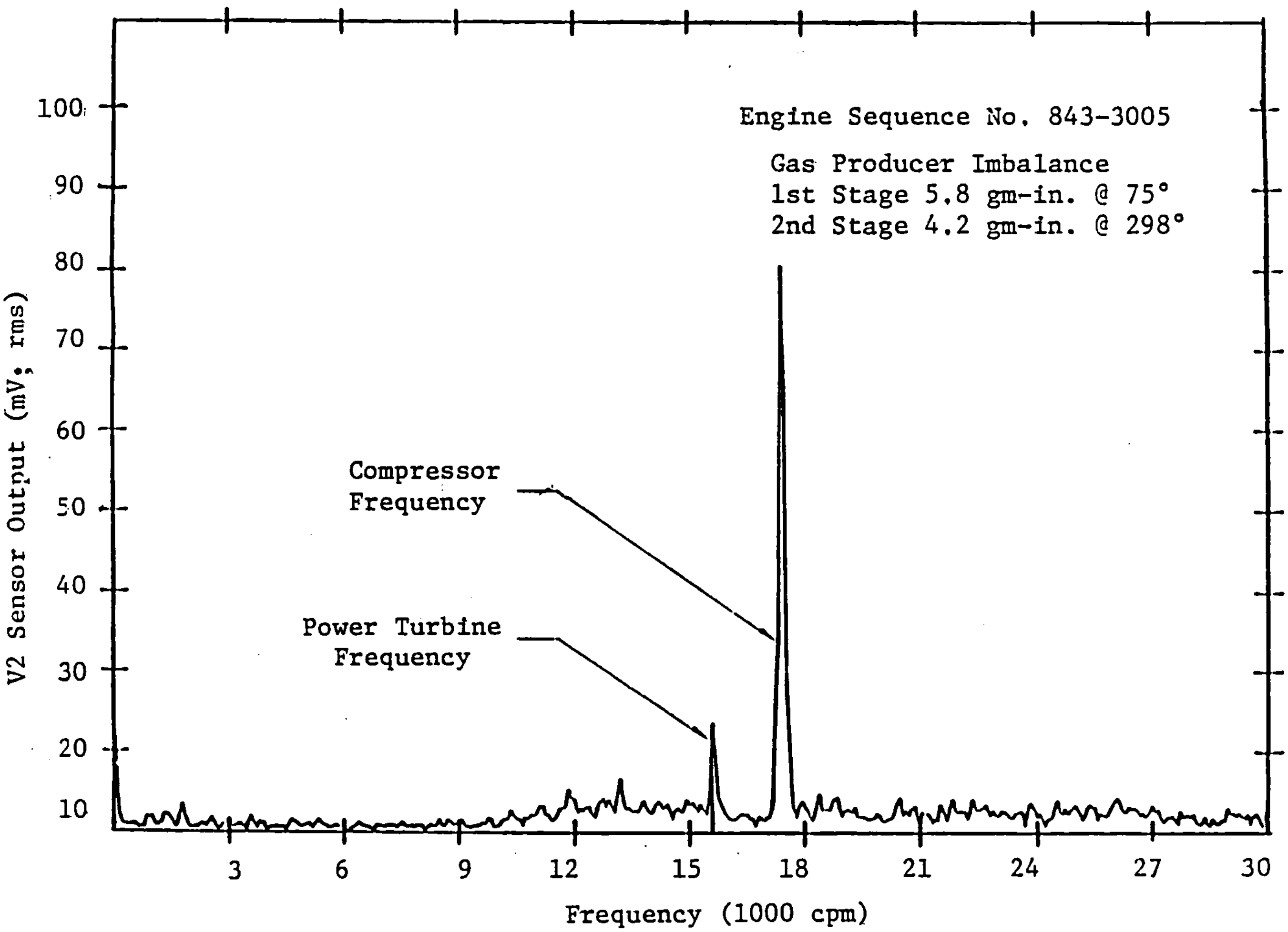


Figure 26 T55 Engine Test Cell Data - Frequency Spectrum

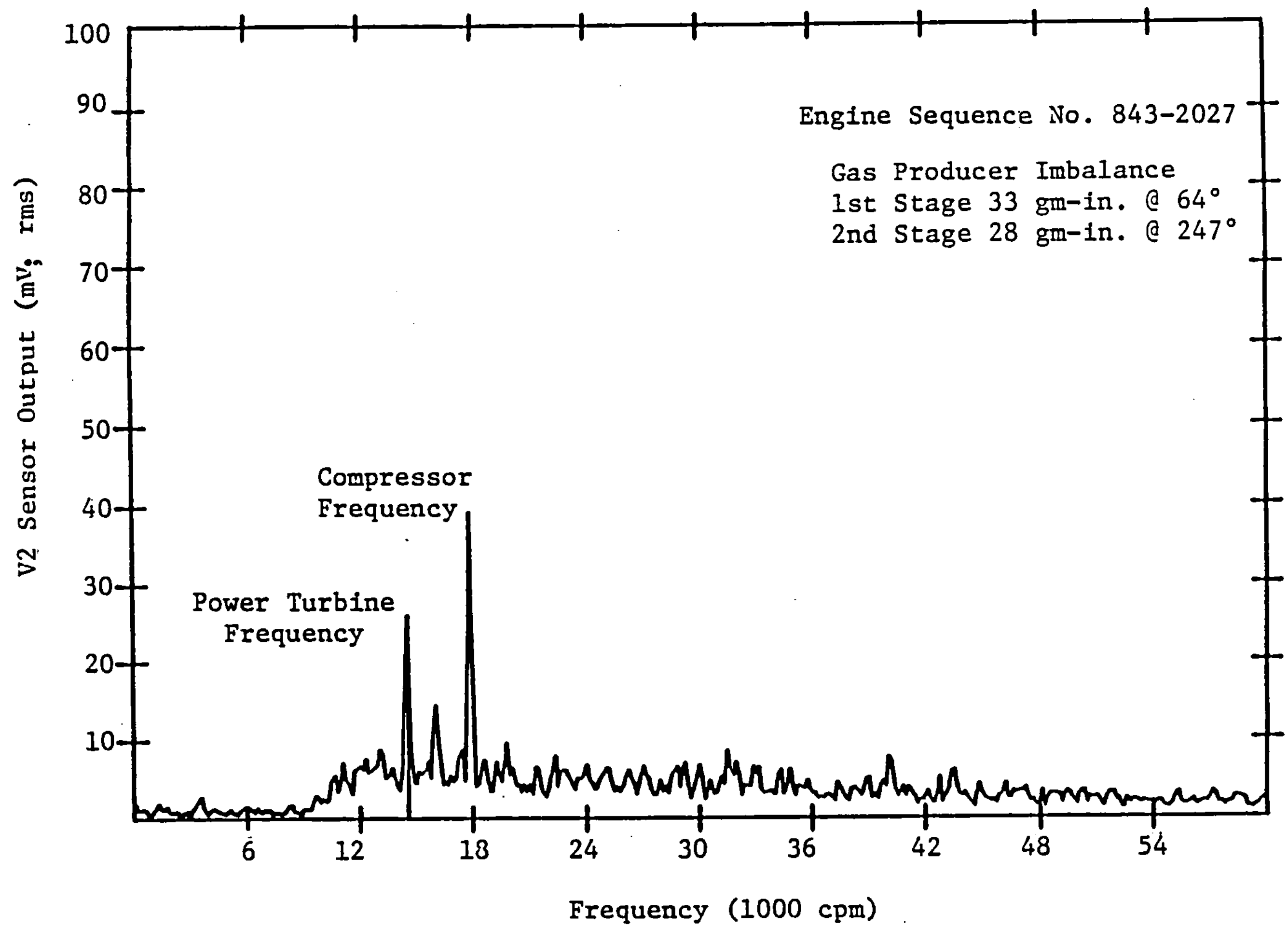


Figure 27 T55 Engine Test Cell Data - Frequency Spectrum

4.0 CONCLUSIONS AND RECOMMENDATIONS

4.1 Conclusions

The multiplane multispeed balancing system designed and constructed under this program was successfully demonstrated on production hardware in the overhaul center at CCAD. Test data taken for twenty T53 power turbines and twenty T55 power turbines demonstrated a direct correlation between the vibration level of the power turbines when tested in the HSBS and when operating in an assembled engine. High speed balancing successfully reduced the vibration levels measured by the HSBS for all turbines balanced.

The testing of T53 power turbines balanced by the current production procedure (not high speed balanced) revealed a wide range of HSBS/engine vibration levels. The T53 vibration levels were found to correlate directly with the magnitude of the dynamic imbalance between the two turbine stages. The by-the-book current production/overhaul procedures do not allow for any check of the dynamic imbalance level, and until recently it was not an uncommon occurrence for the same engine to be rejected as many as ten times. Therefore, some not-by-the-book checking was done in an effort to determine the reason for the large number of engines rejected for high vibration. Over a one-year period, the CCAD balance shop determined that the cause of rejection for 19 out of 20 high-vibration rejects was dynamic imbalance. Correcting the dynamic imbalance of the power turbines from rejected engines has resulted in smooth running engines, in all cases, on the second engine test.

Dynamically balanced T53 power turbines tested in the HSBS resulted in exceptionally low vibration levels. The engines containing these power turbines were also exceptionally smooth running. High speed operation of the power turbines in the HSBS also served as a proof test of the turbine module. In two cases, power turbines with problems were identified in the HSBS prior to engine assembly.

The T55 power turbines balanced with the current production procedure and tested in the HSBS showed much less variation in vibration levels than the T53 turbines. The corresponding T55 engines showed acceptable levels of vibration attributable to the power turbine. The dominant frequency contributing to the overall vibration of the T55 engines is generally the compressor rotor frequency. Tests were conducted to determine if the dynamic imbalance of the T55 gas generator turbine stages, which bolt onto the compressor rotor, were the cause of the large contribution from the compressor rotor. No correlation was observed between the gas generator dynamic imbalance and compressor rotor vibration levels.

The successful development of high speed balancing techniques provides the overhaul center with a capability that will enable them to handle the increasing flexible rotor designs incorporated in the modern, lightweight, high horsepower engine such as the T700.

4.2 Recommendations

The current T55 power turbine balancing procedure is adequate. Implementation of high speed balancing is not recommended for this rotor.

The current T53 power turbine balance procedure is unacceptable. Implementation of dynamic balancing of the turbine stages would result in a reduction in the average engine vibration level by a factor of two and would virtually eliminate vibration rejects. Because of the ease of implementation of dynamic balancing, high speed balancing is not recommended for T53 power turbines.

The continuing development of automated material removal techniques such as laser balancing is essential to the continuing development of precision, high speed balancing. The effectiveness of high speed balancing is dependant on the accurate removal of the required balance correction.

Analysis of engine test cell vibration data is needed at CCAD. Currently, personnel at the overhaul center have only the overall vibration level on which to base a decision as to the appropriate corrective action for a vibration reject. The overall vibration reading is a summation of the vibration components at discrete frequencies. For this reason, a vibration reject may be due to a large contribution from the waterbrake or from a system auxiliary or other problems that could be addressed directly rather than checking all major engine components before retesting. CCAD has the necessary equipment to do frequency spectrum analysis, although it has not been used since being purchased. Putting this hardware to use would provide CCAD engineering personnel with the data required to minimize the rework involved in handling vibration rejects of all engine types.

APPENDIX A

T53 ENGINE SPECTRUM PLOTS

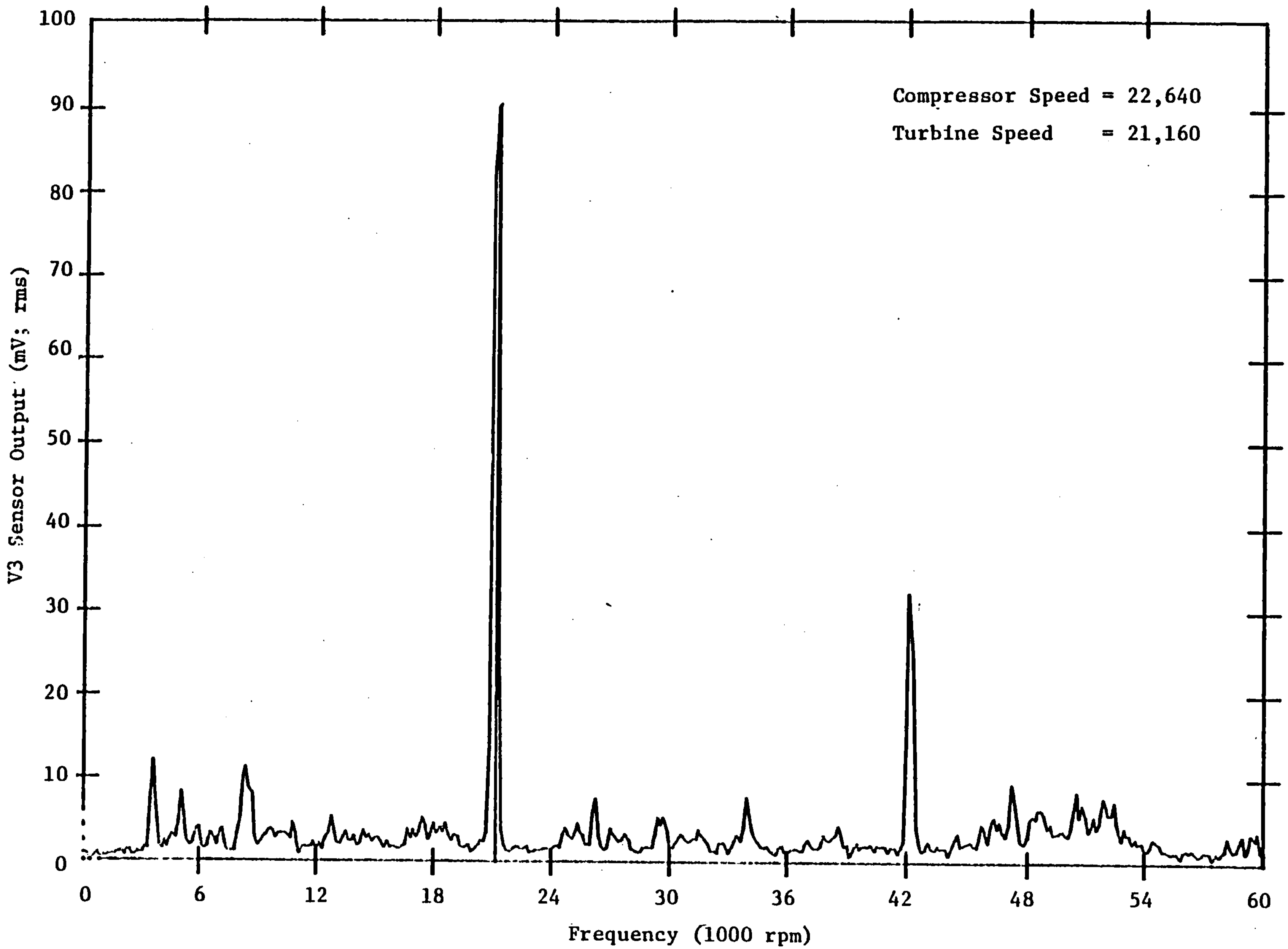


Figure A-1 T53 Power Turbine, Serial No. 1896

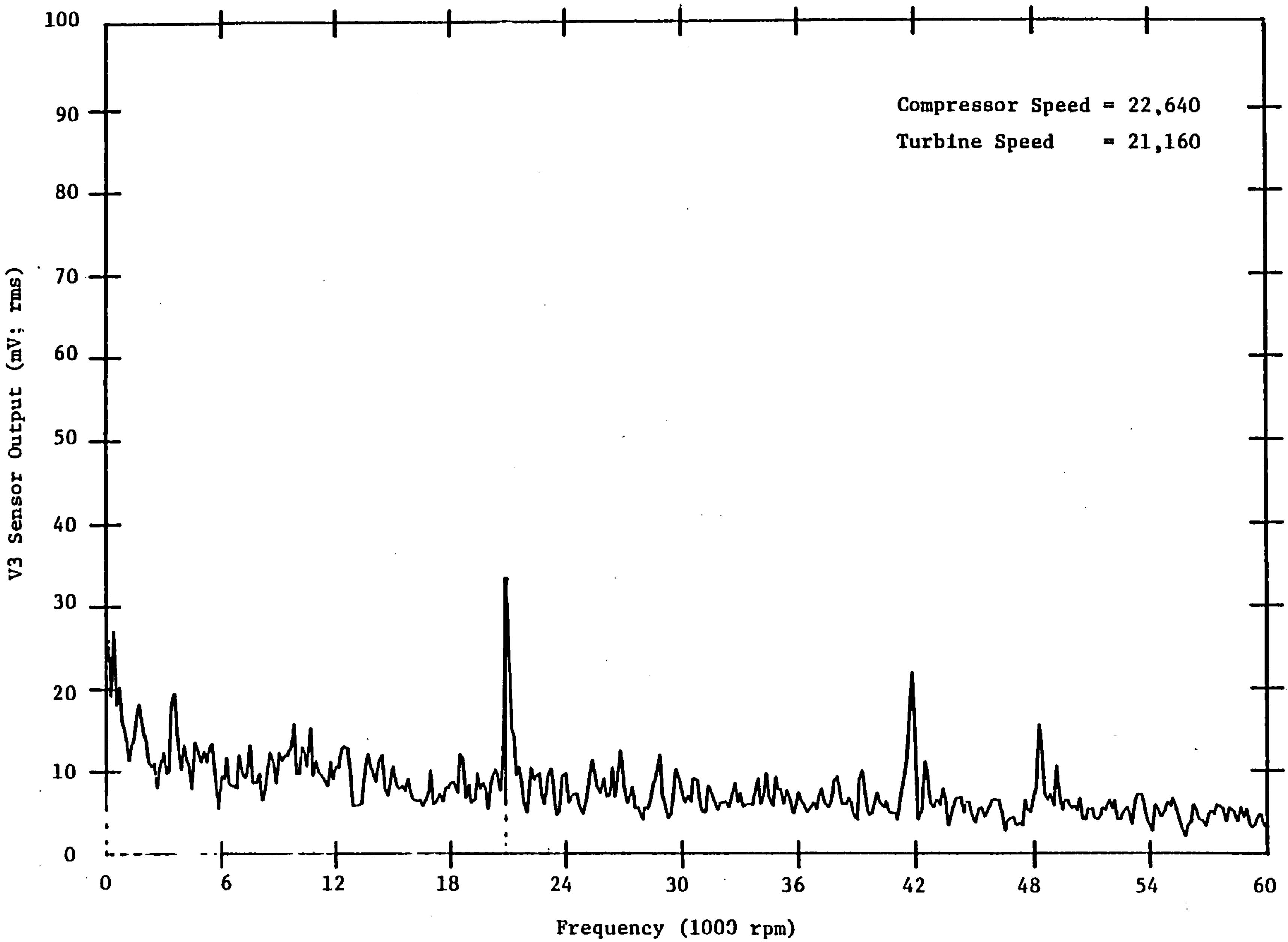


Figure A-2 T53 Power Turbine, Serial No. 5247

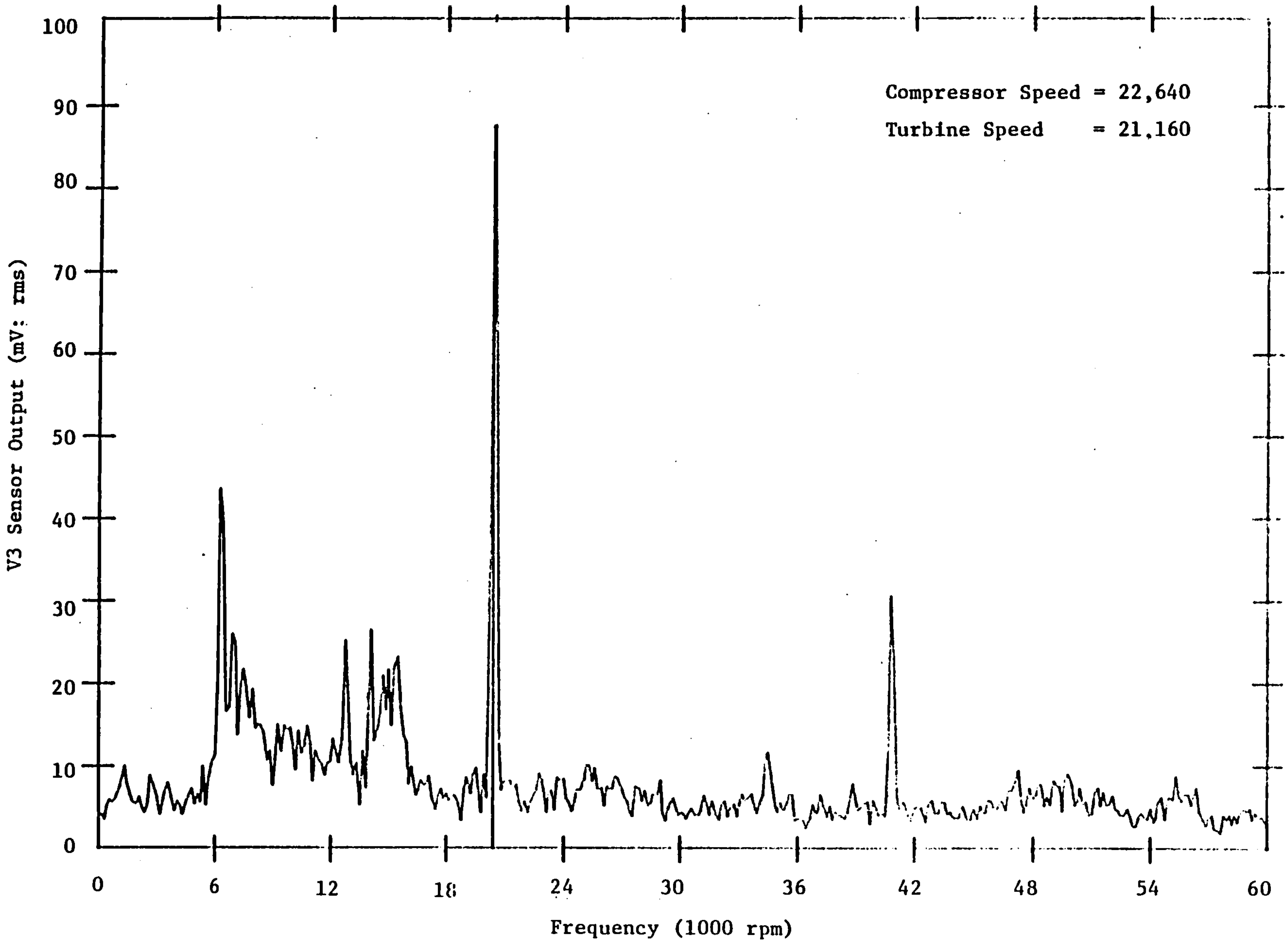


Figure A-3 T53 Power Turbine, Serial No. U12956

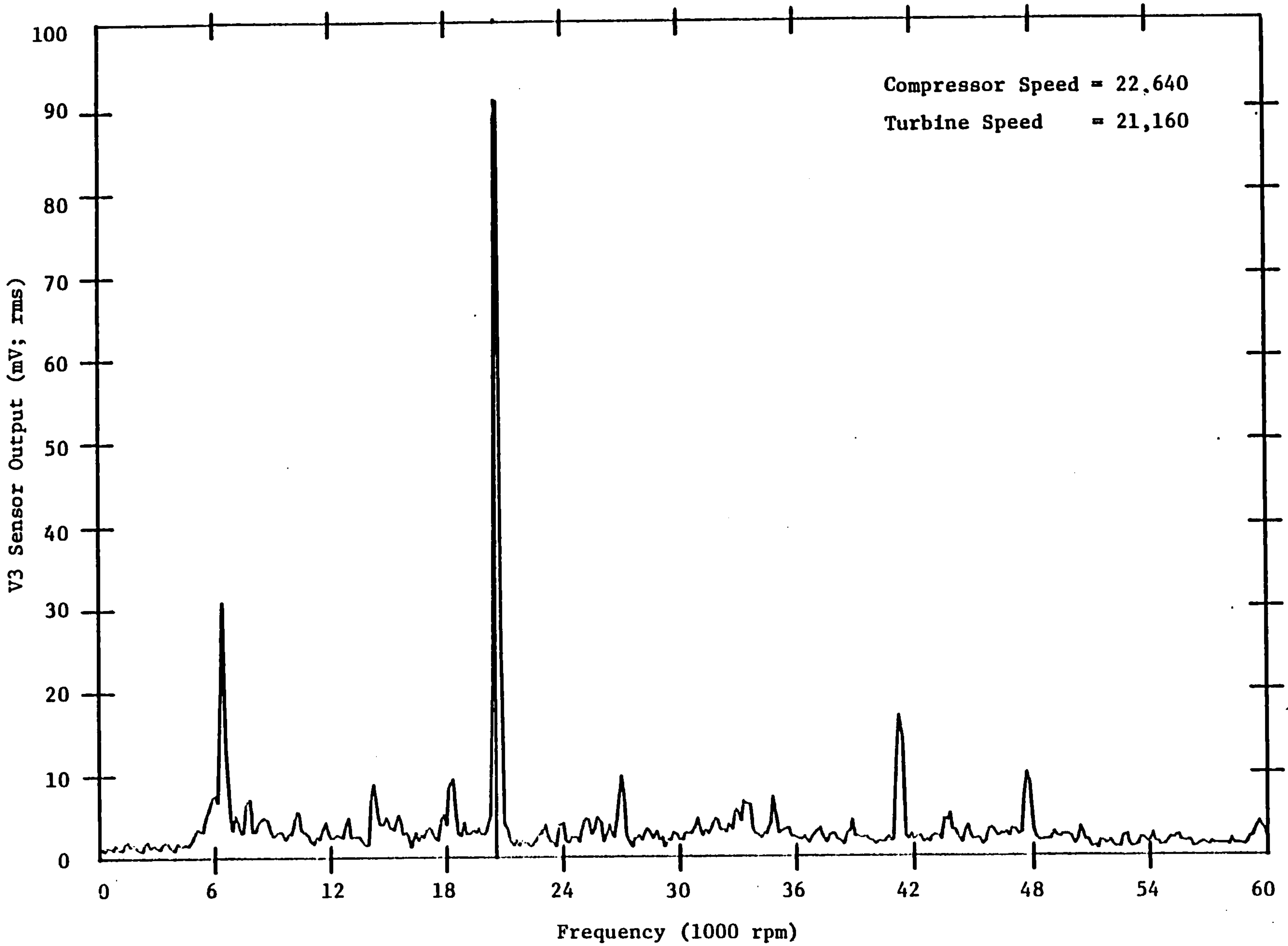


Figure A-4 T53 Power Turbine, Serial No. 96441

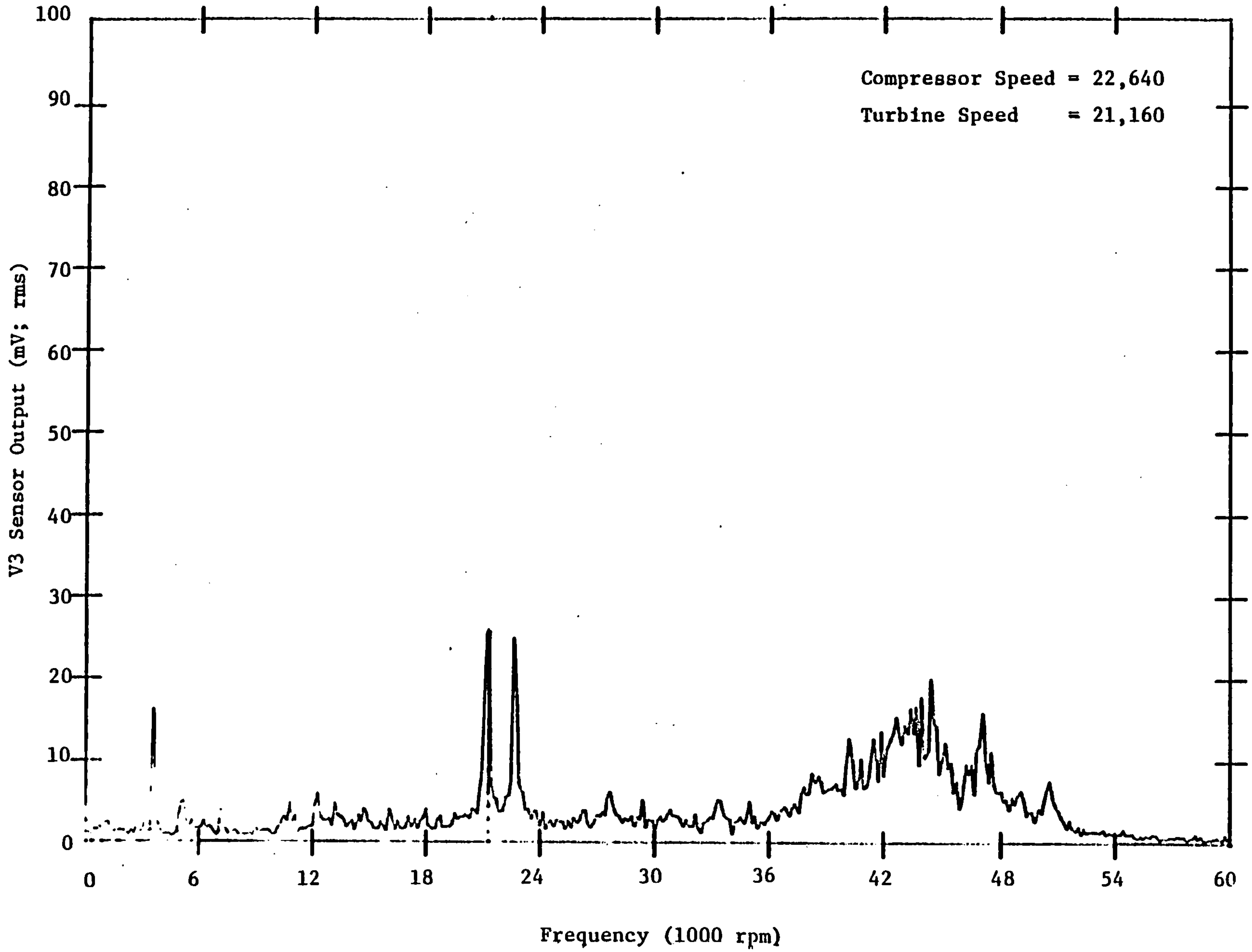


Figure A-5 T53 Power Turbine, Serial No. 8873

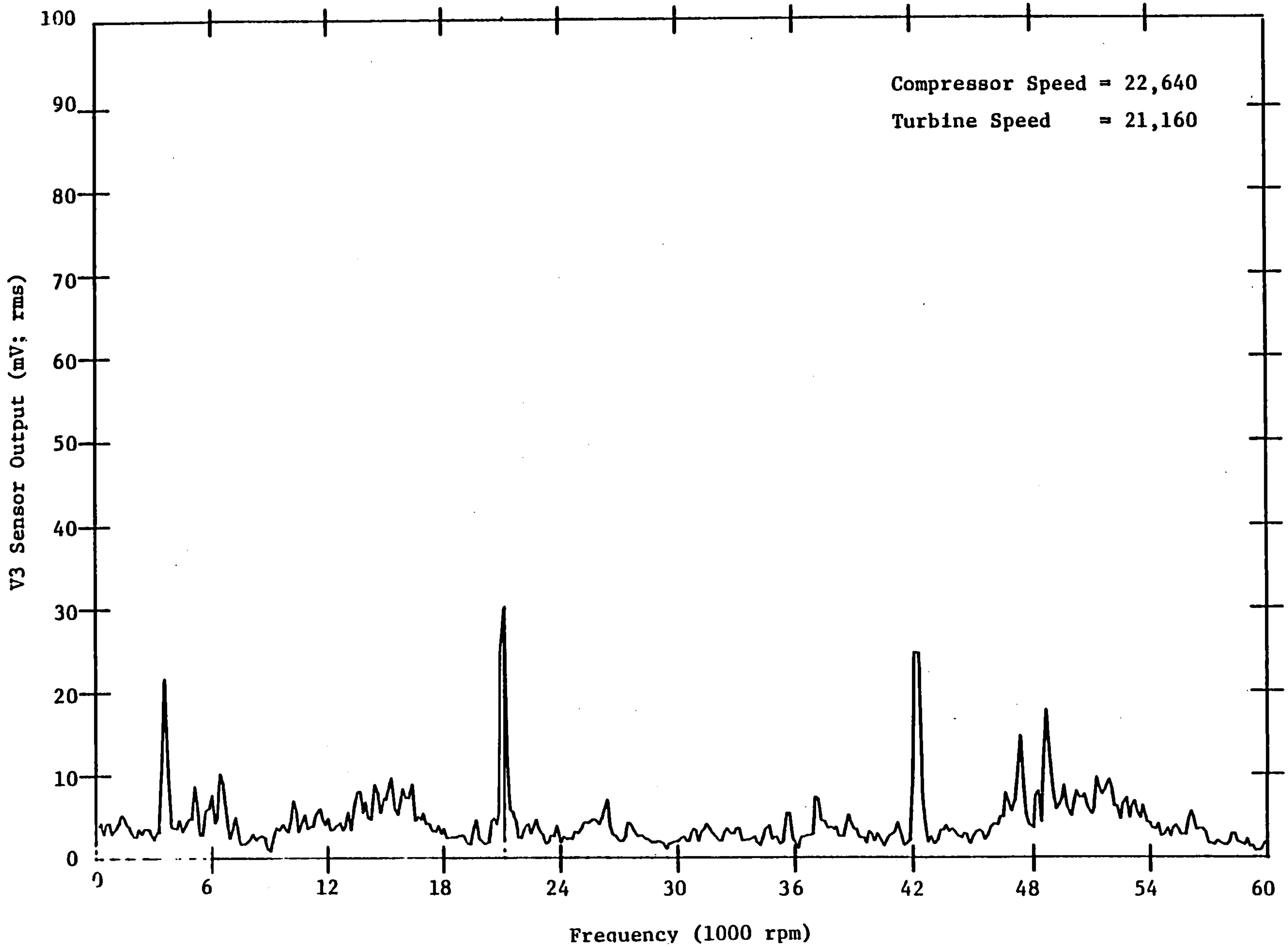


Figure A-6 T53 Power Turbine, Serial No. 1377

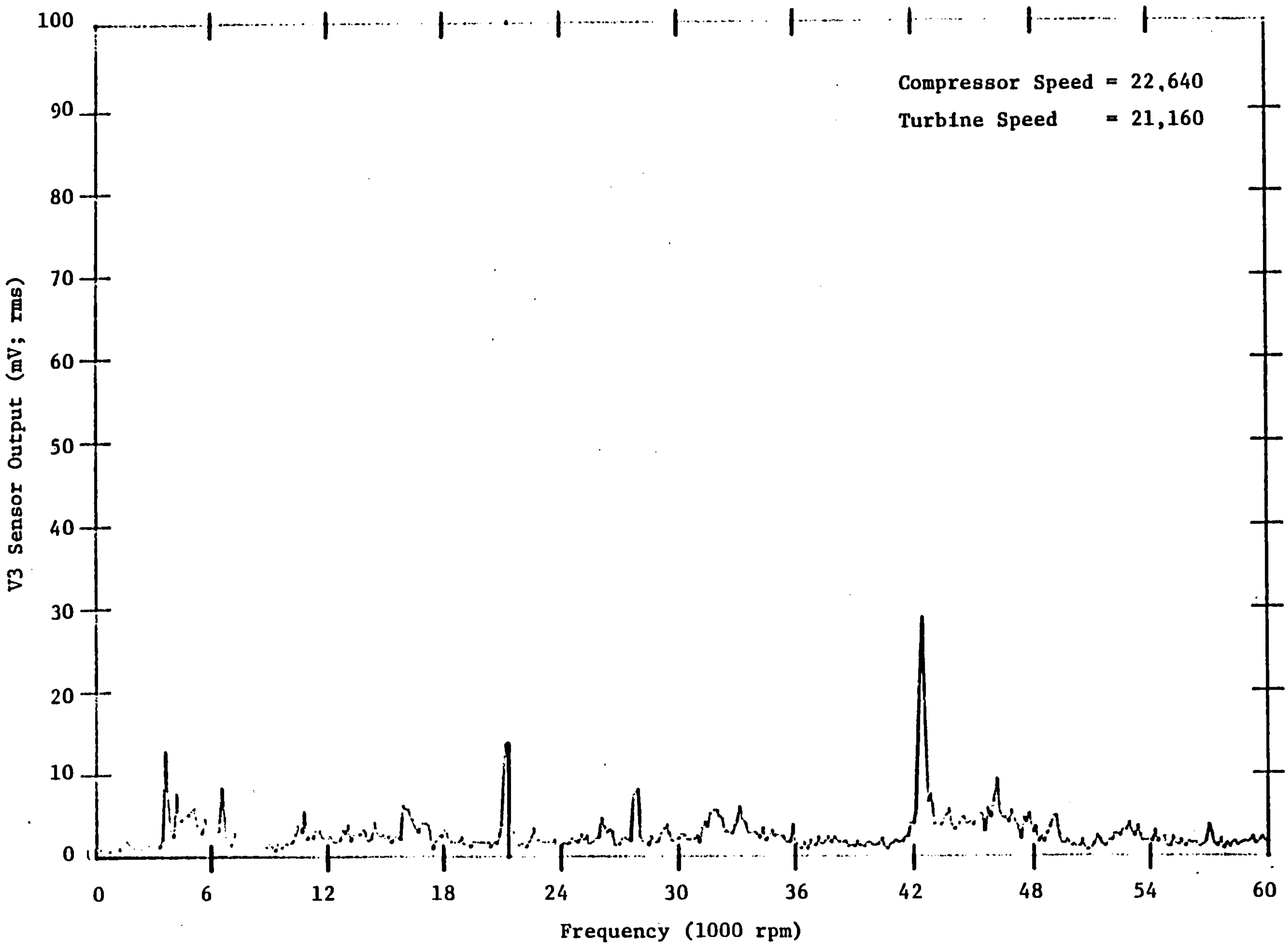


Figure A-7 T53 Power Turbine, Serial No. 5586

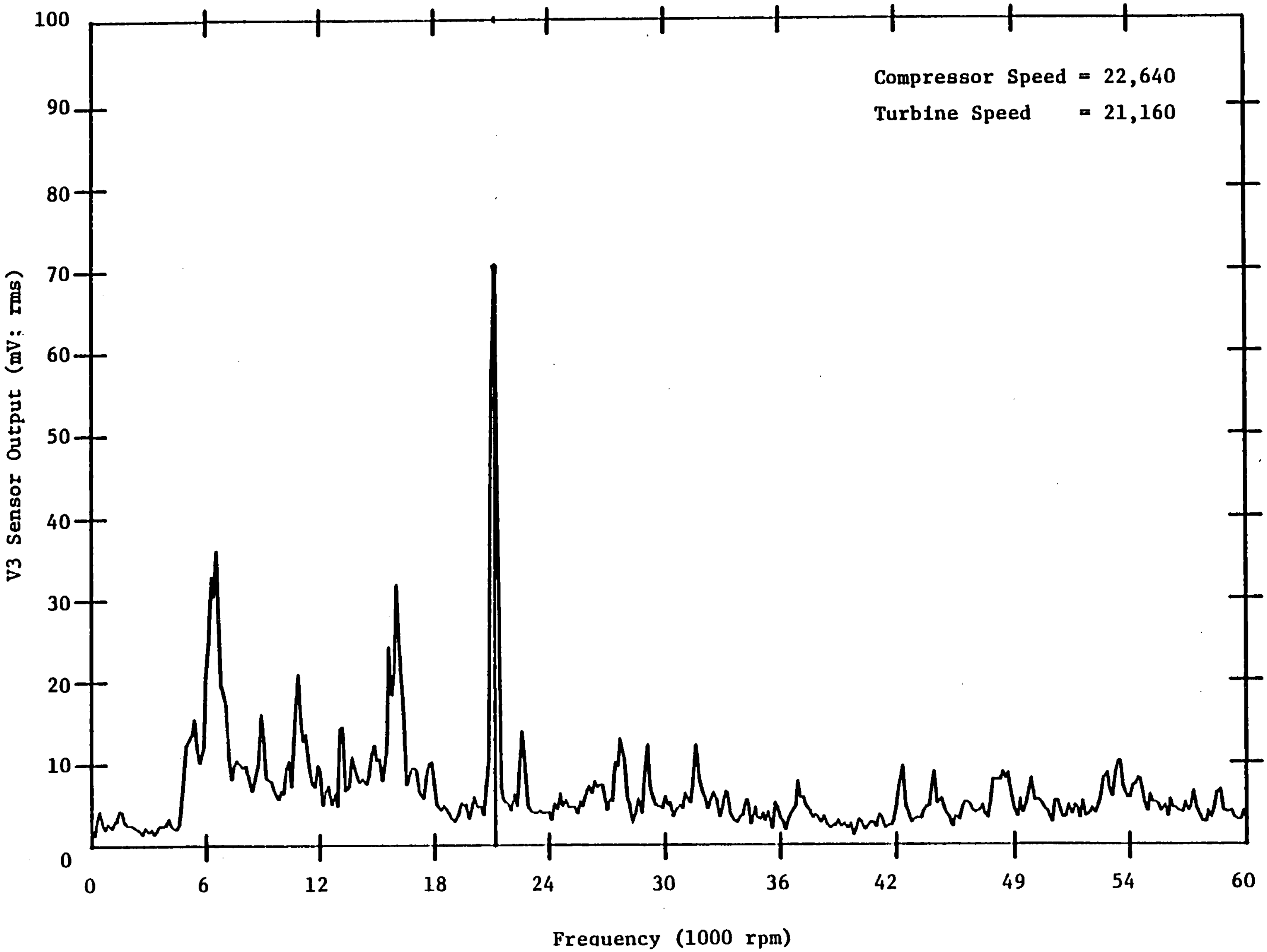


Figure A-8 T53 Power Turbine, Serial No. U00278

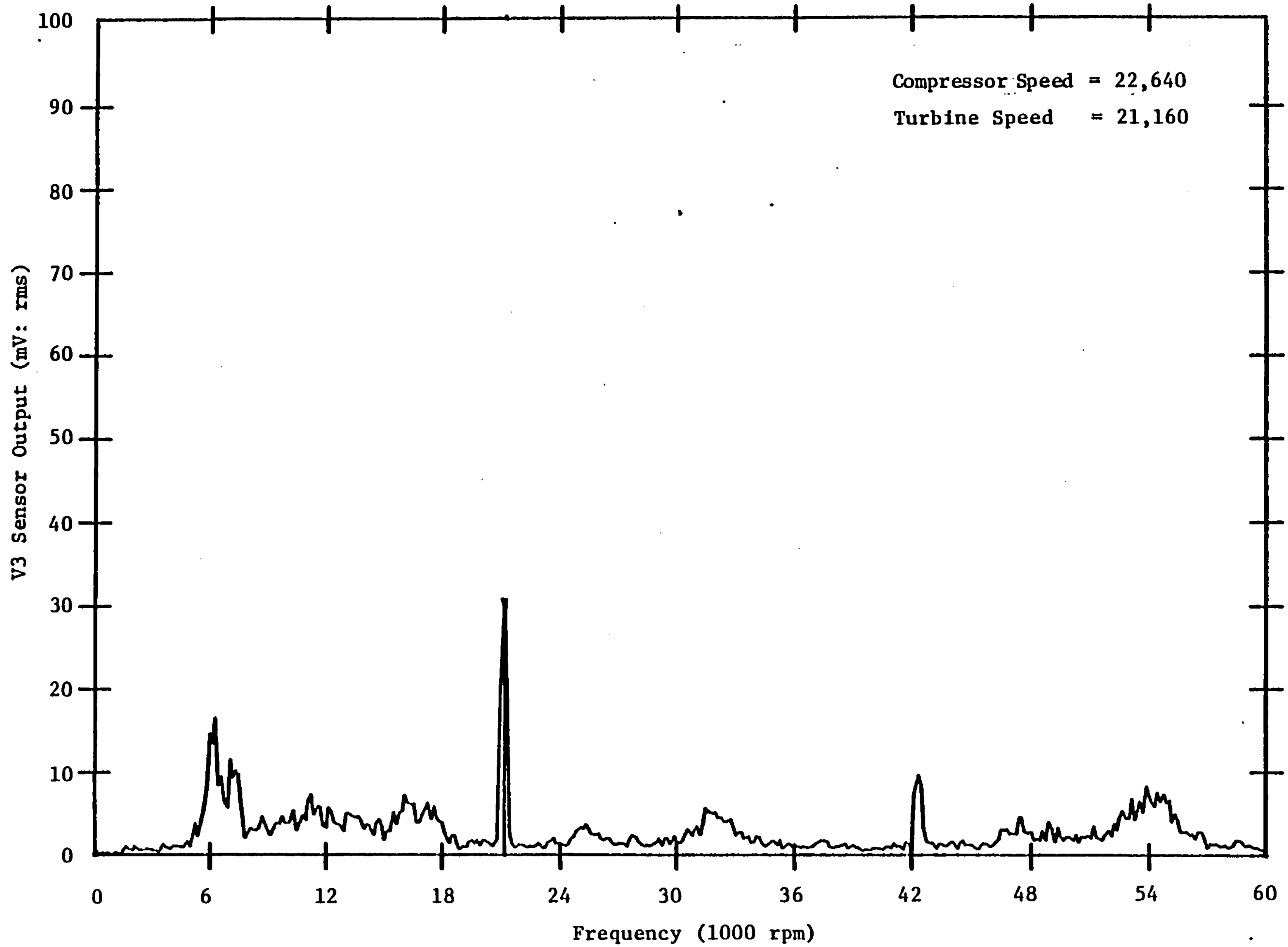


Figure A-9 T53 Power Turbine, Serial No. C9M11244

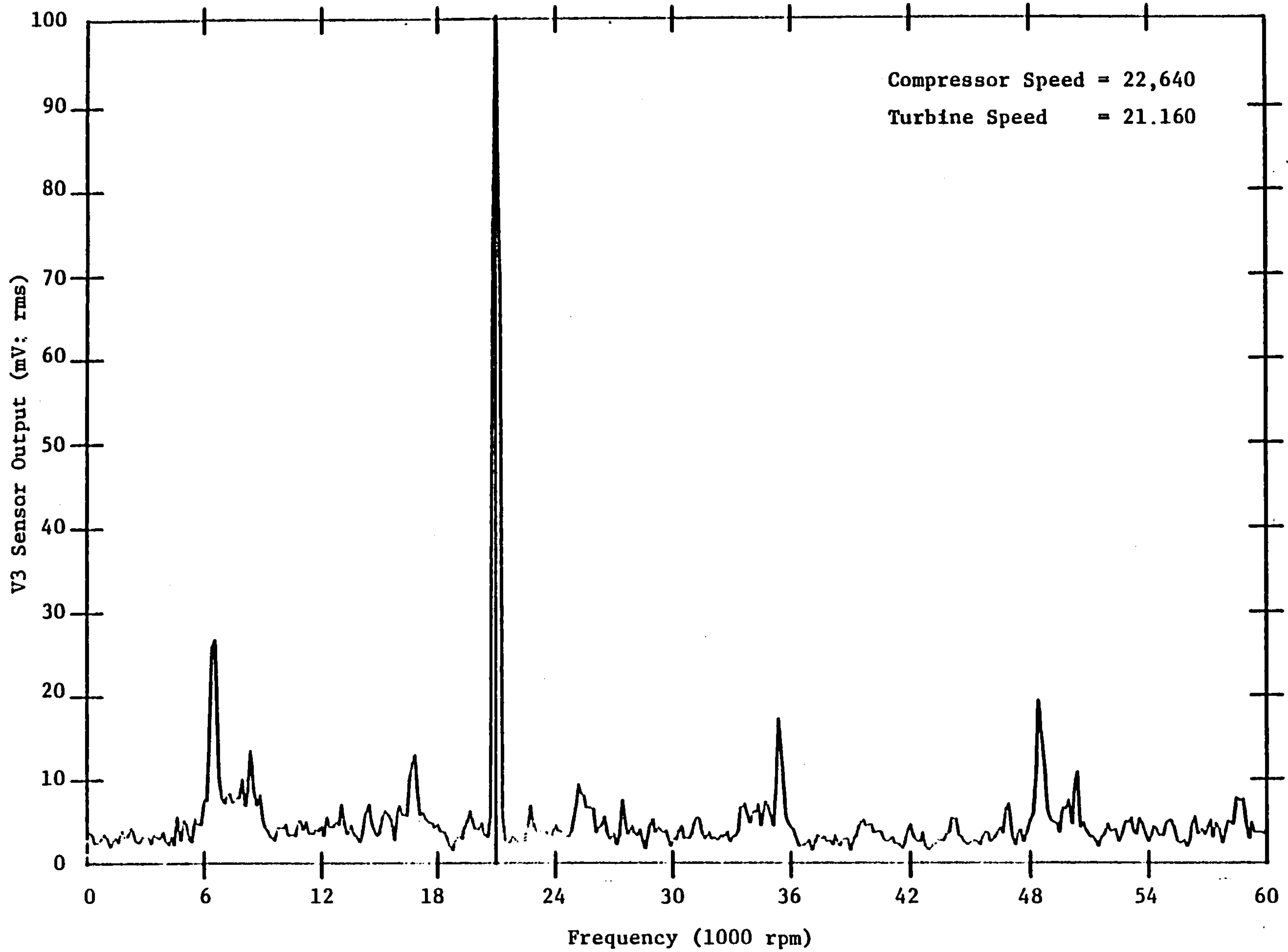
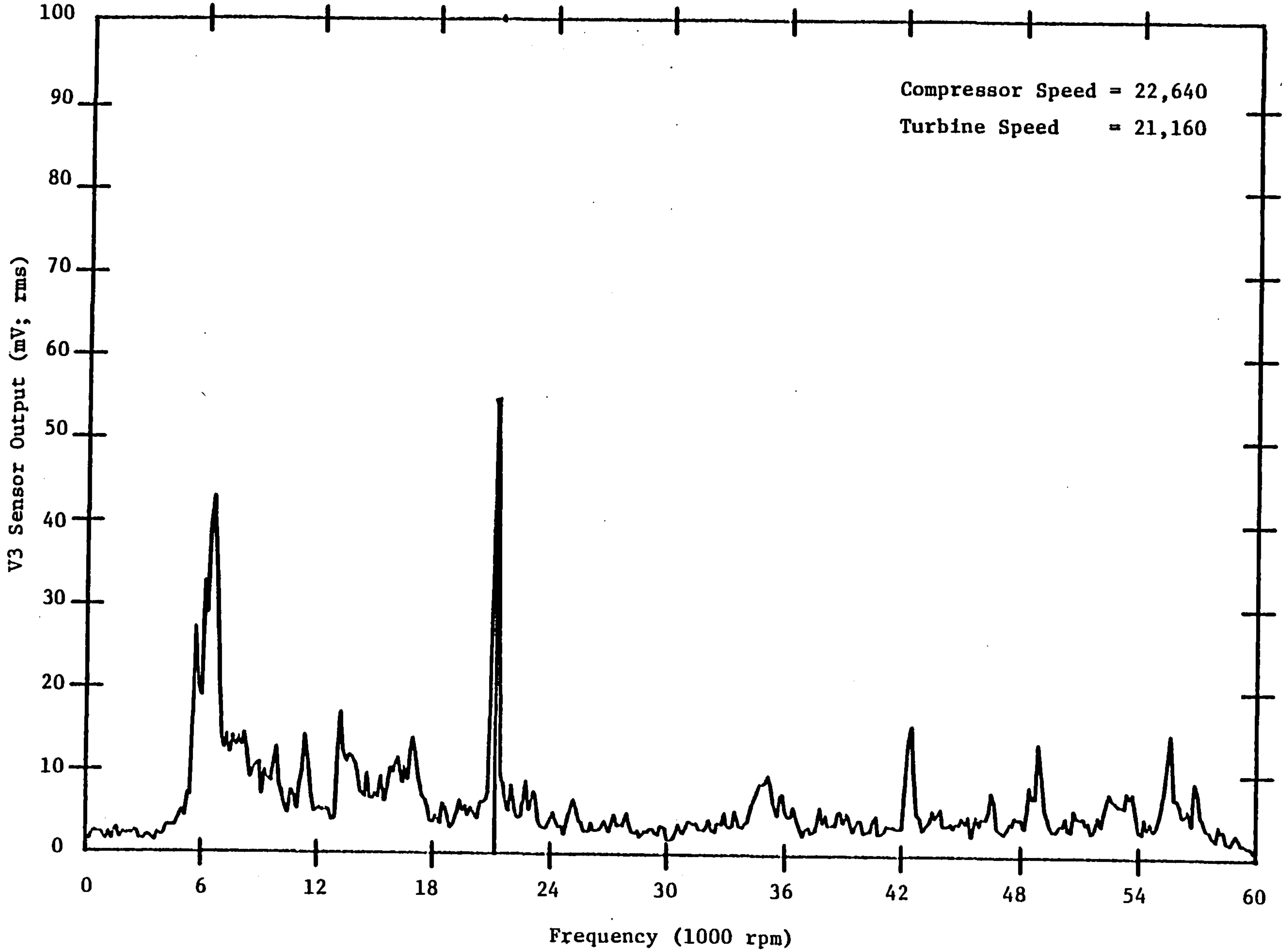


Figure A-10 T53 Power Turbine, Serial No, 2484



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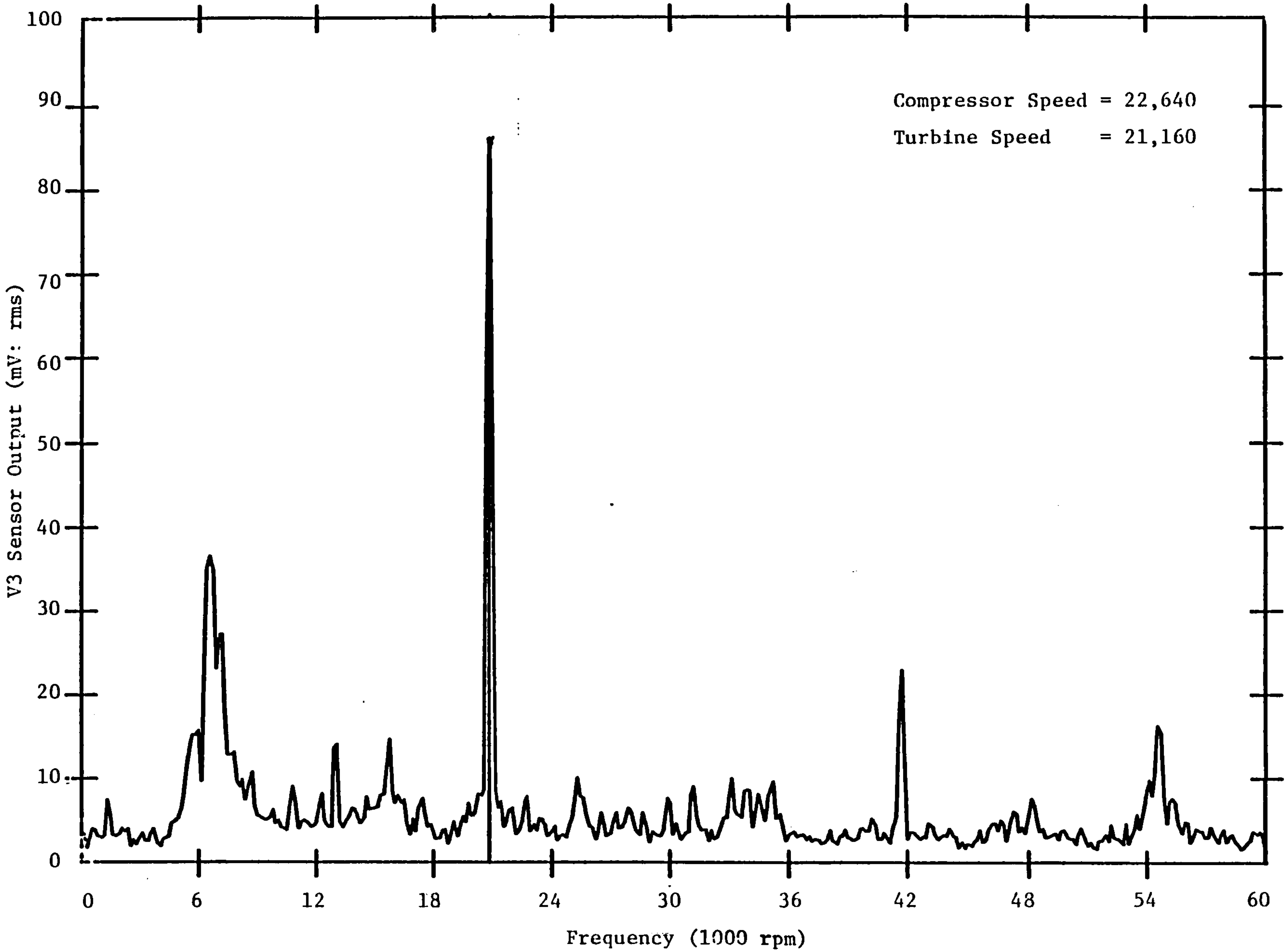

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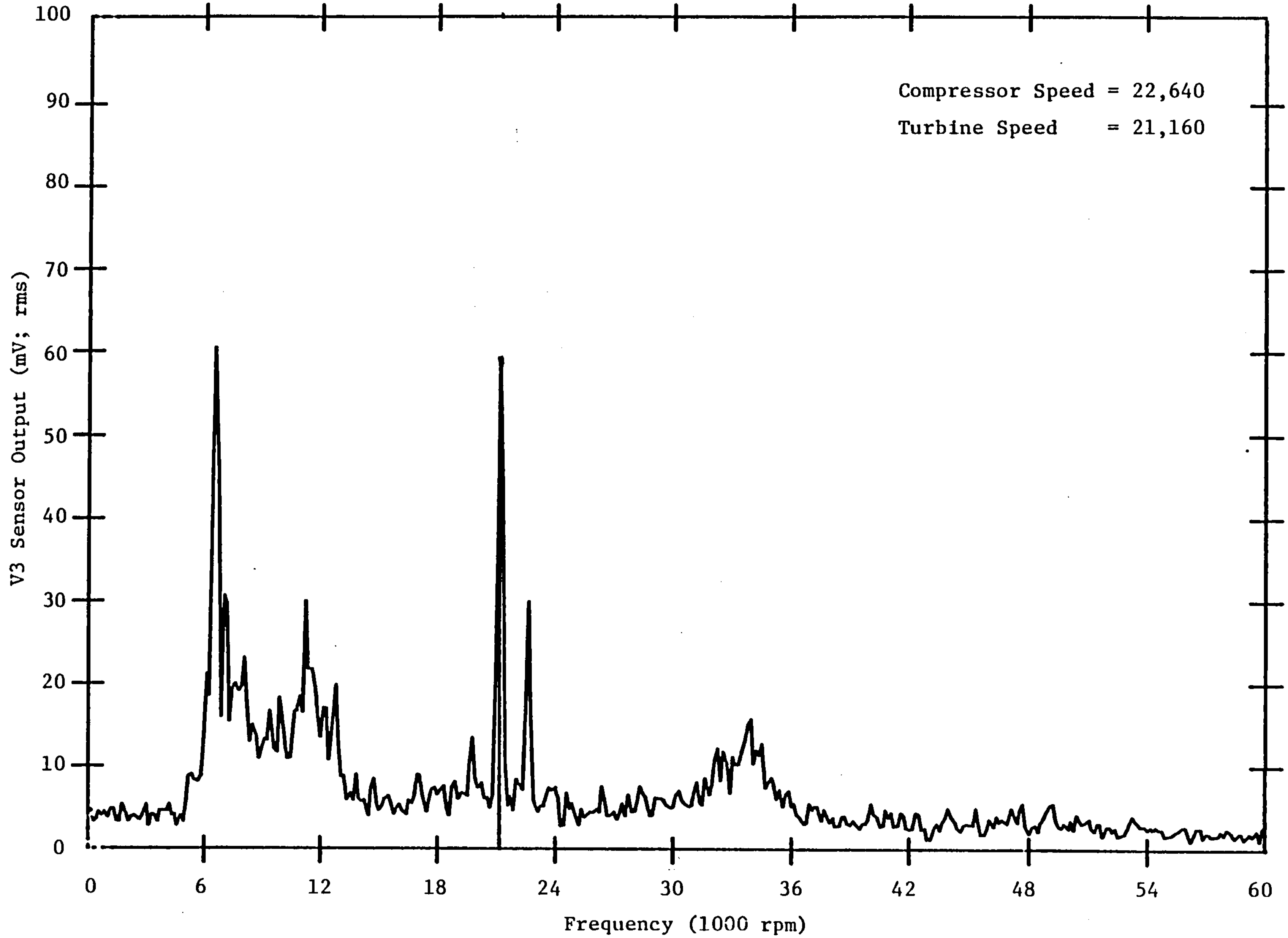


Figure A-13 T53 Power Turbine, Serial No. 21788

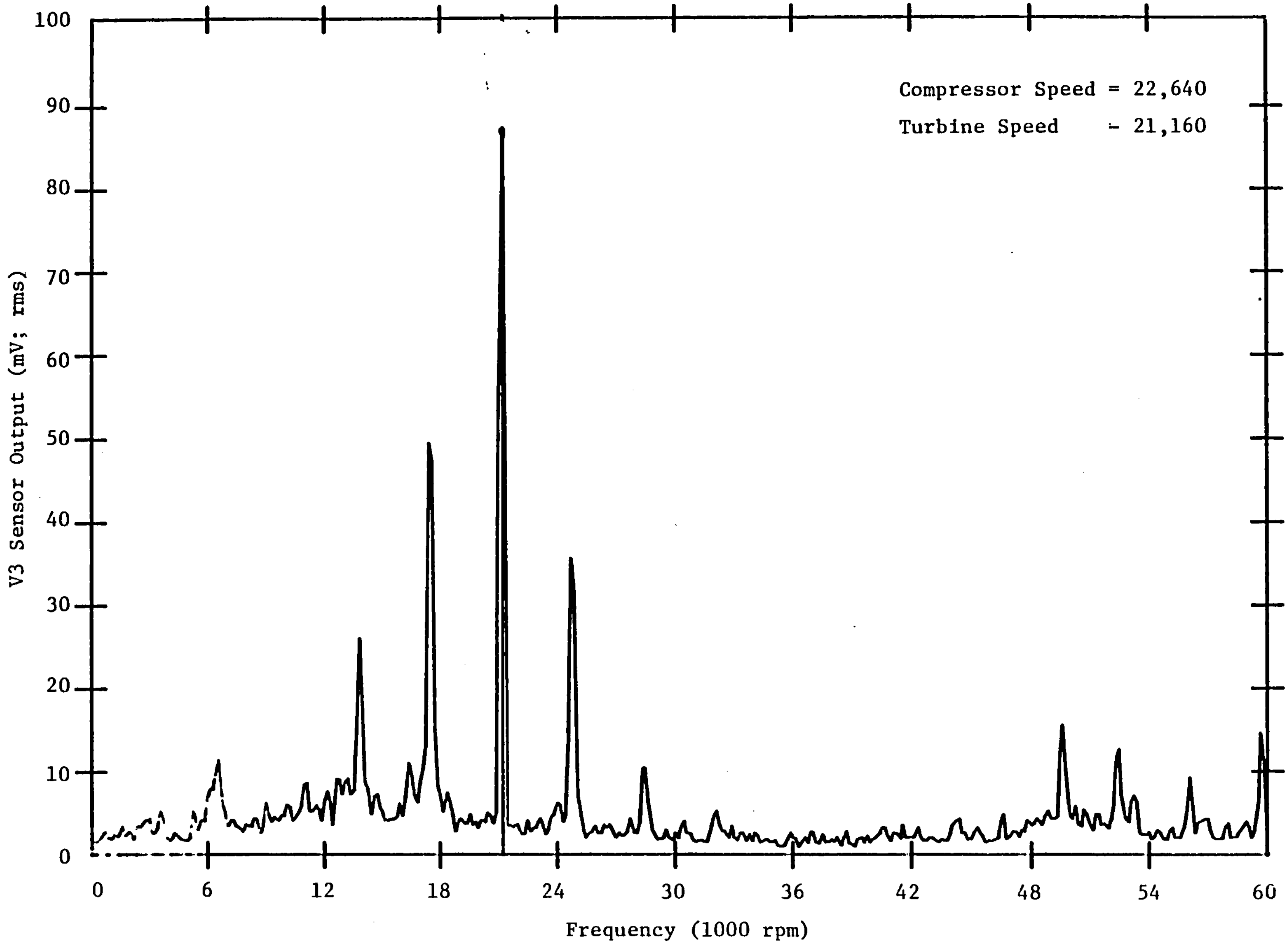
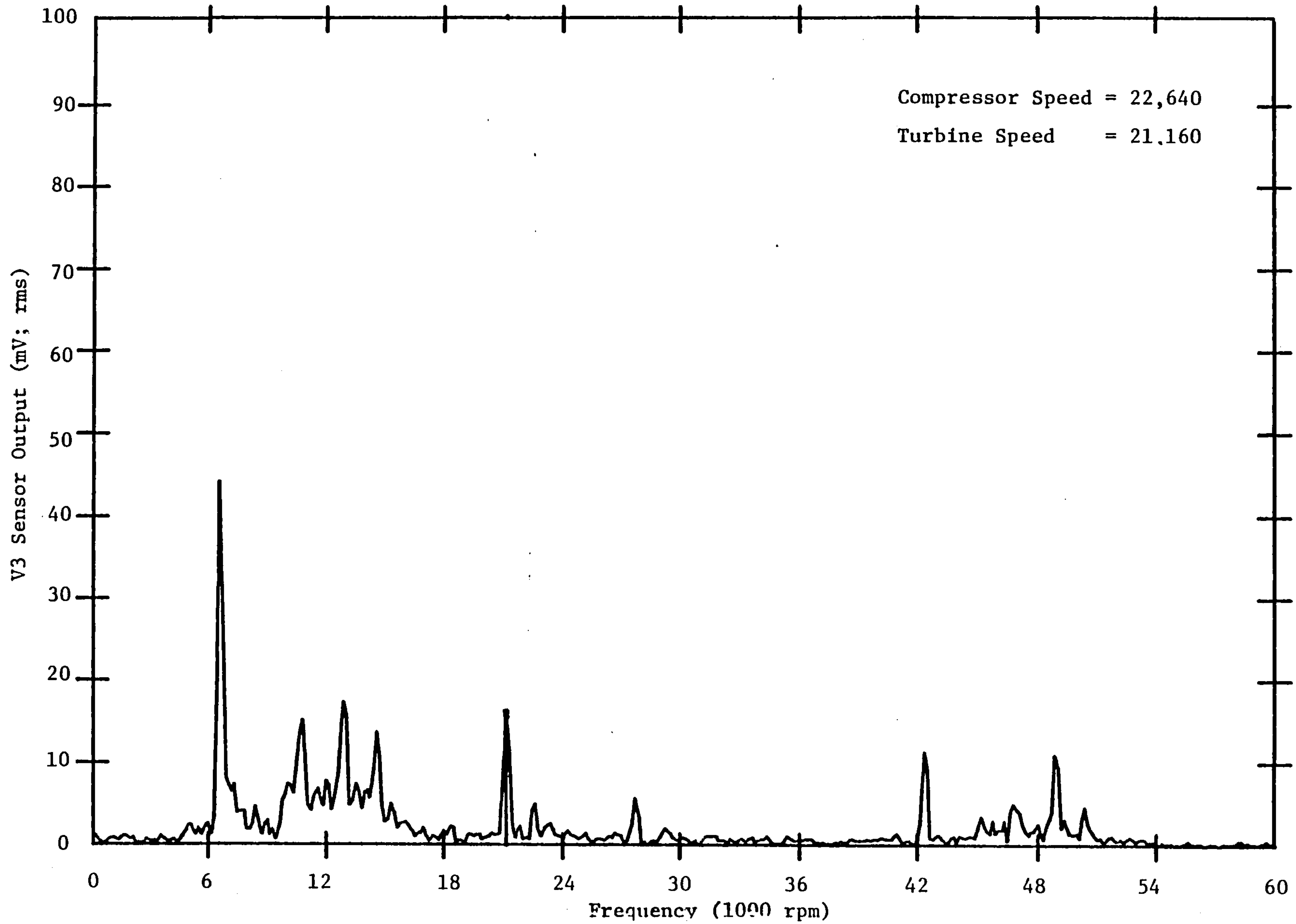


Figure A-14 T53 Power Turbine, Serial No. 289659



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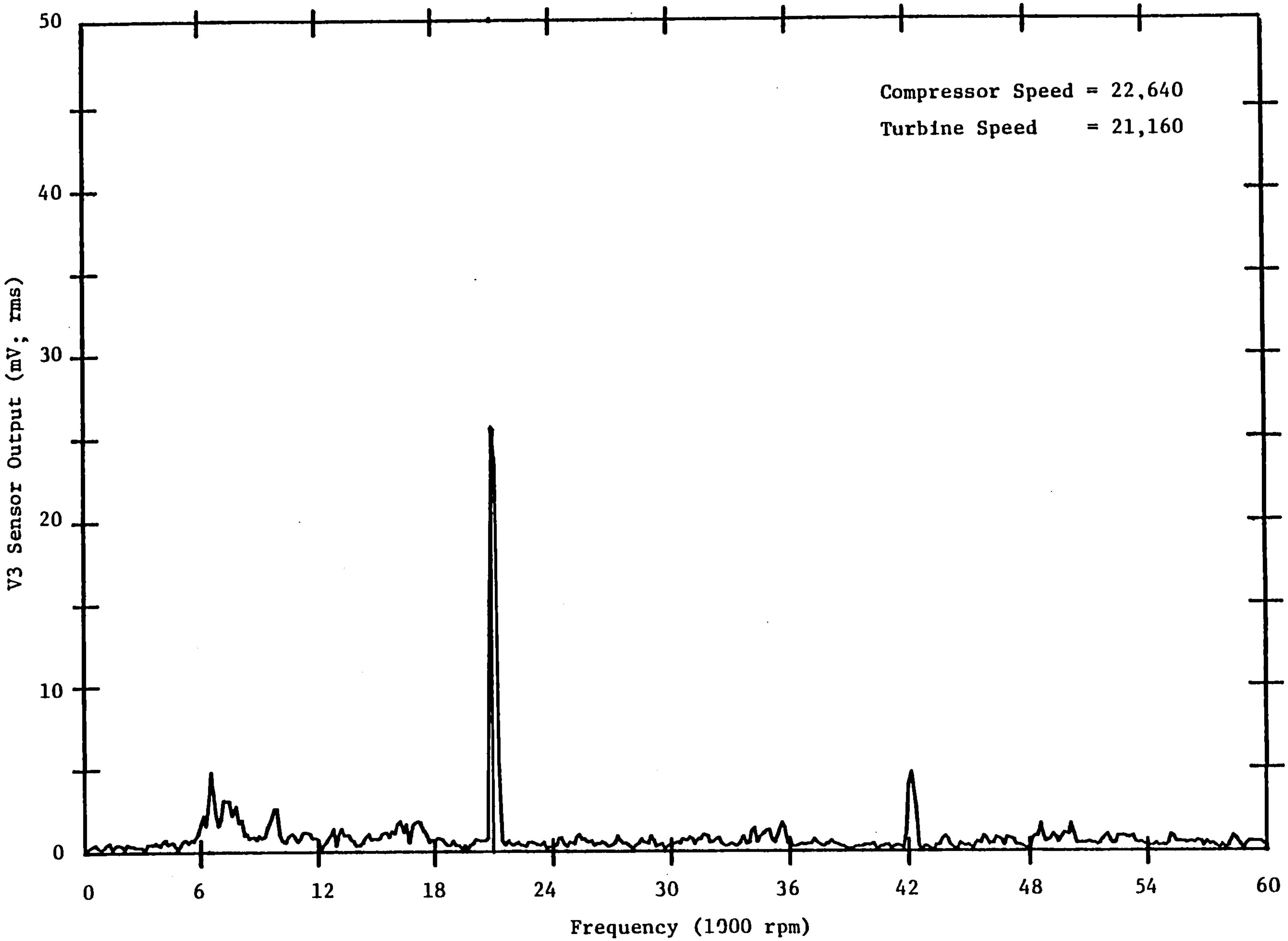


Figure A-16 T53 Power Turbine, Serial No. 82C004

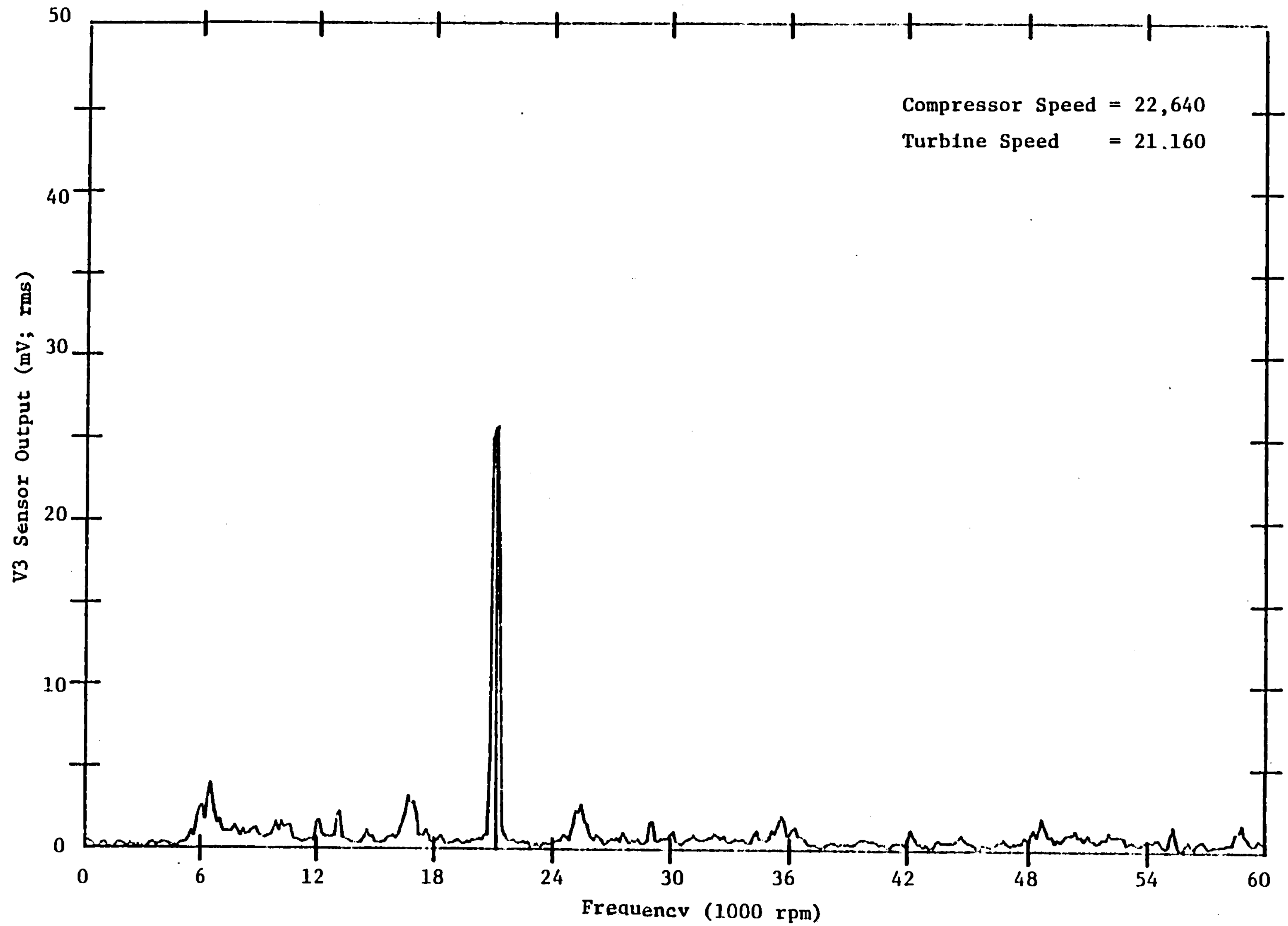


Figure A-17 T53 Power Turbine, Serial No. 82C012

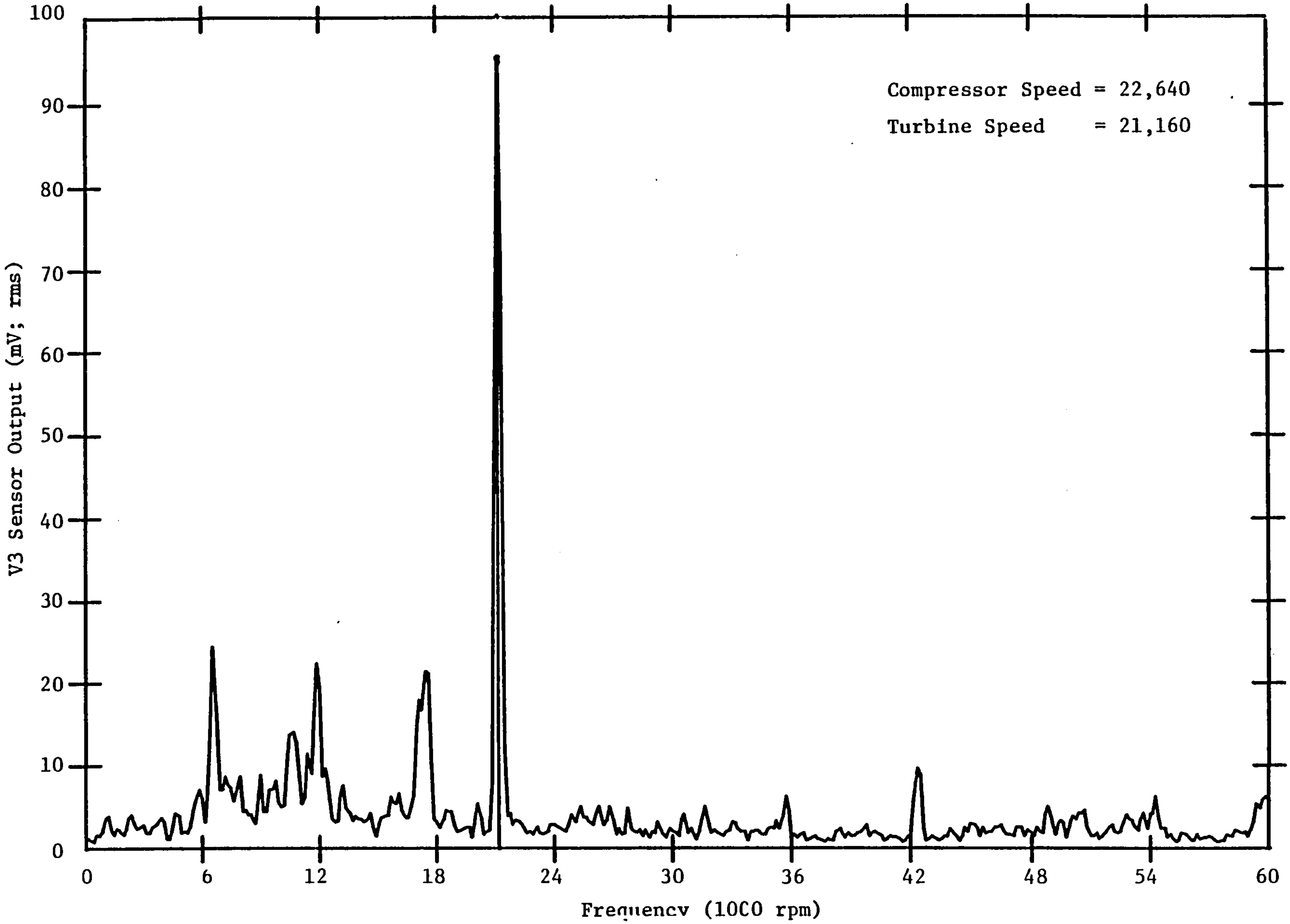
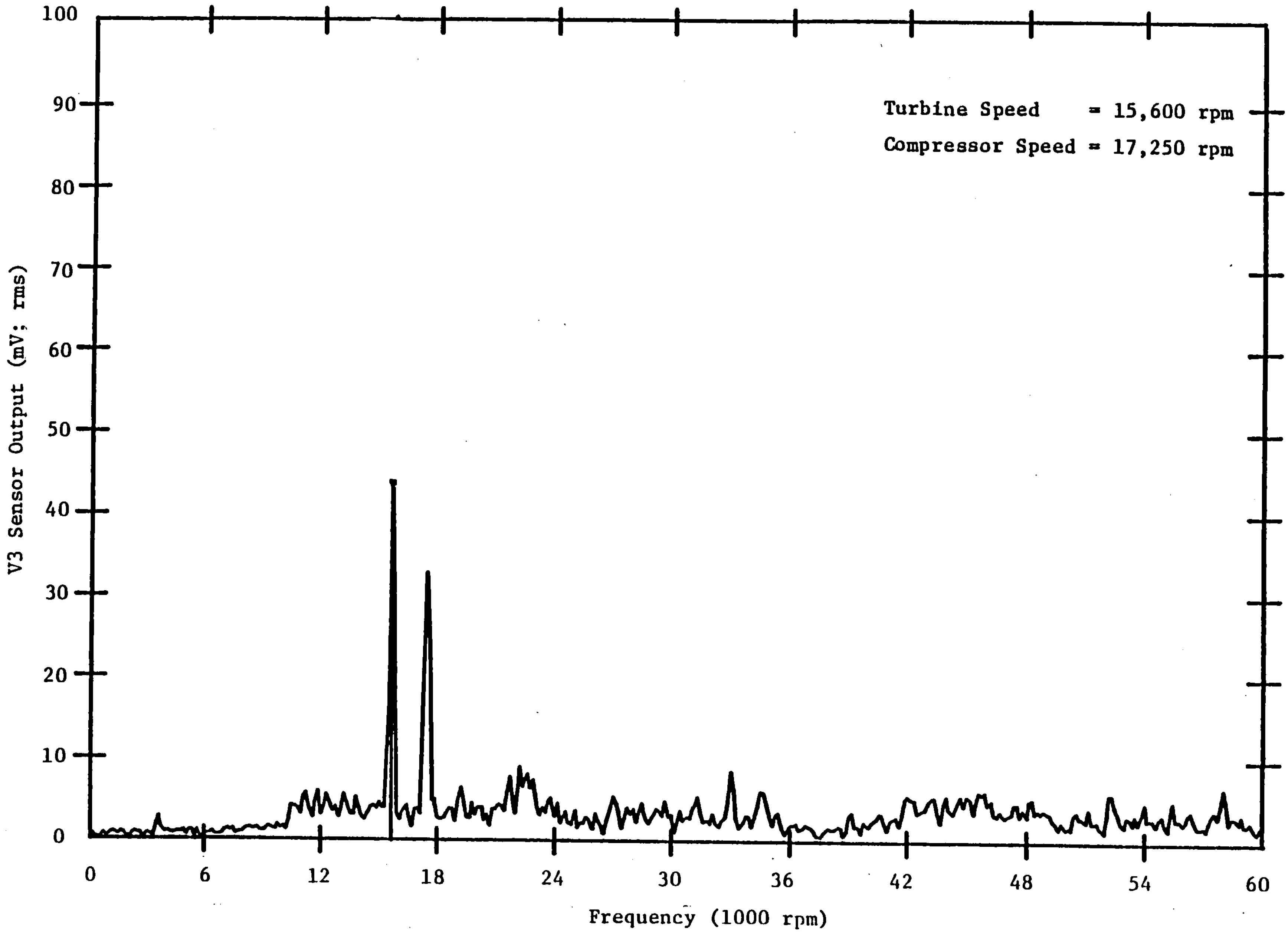


Figure A-18 T53 Power Turbine, Serial No. 80K003P

APPENDIX B

T55 ENGINE SPECTRUM PLOTS



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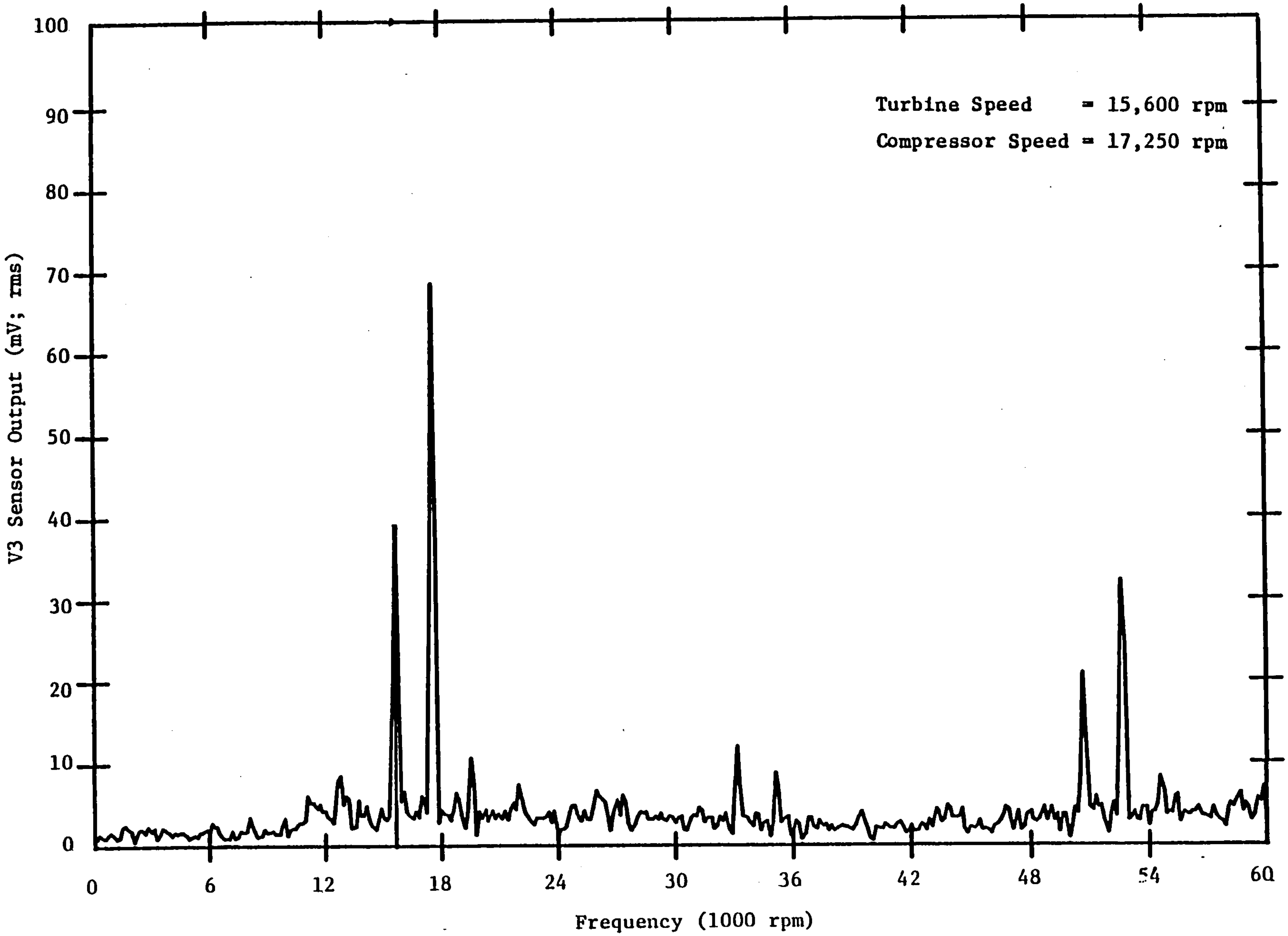

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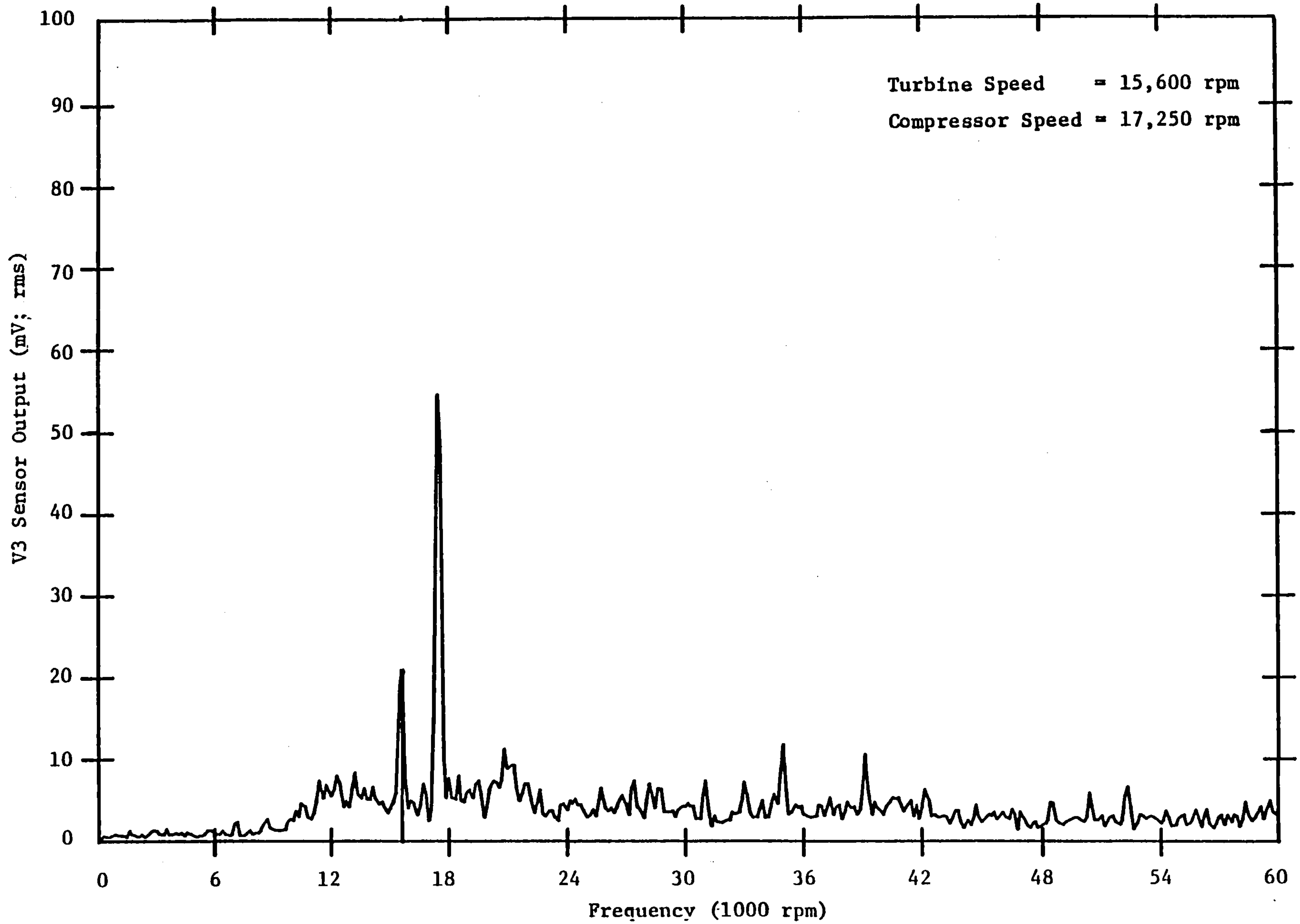

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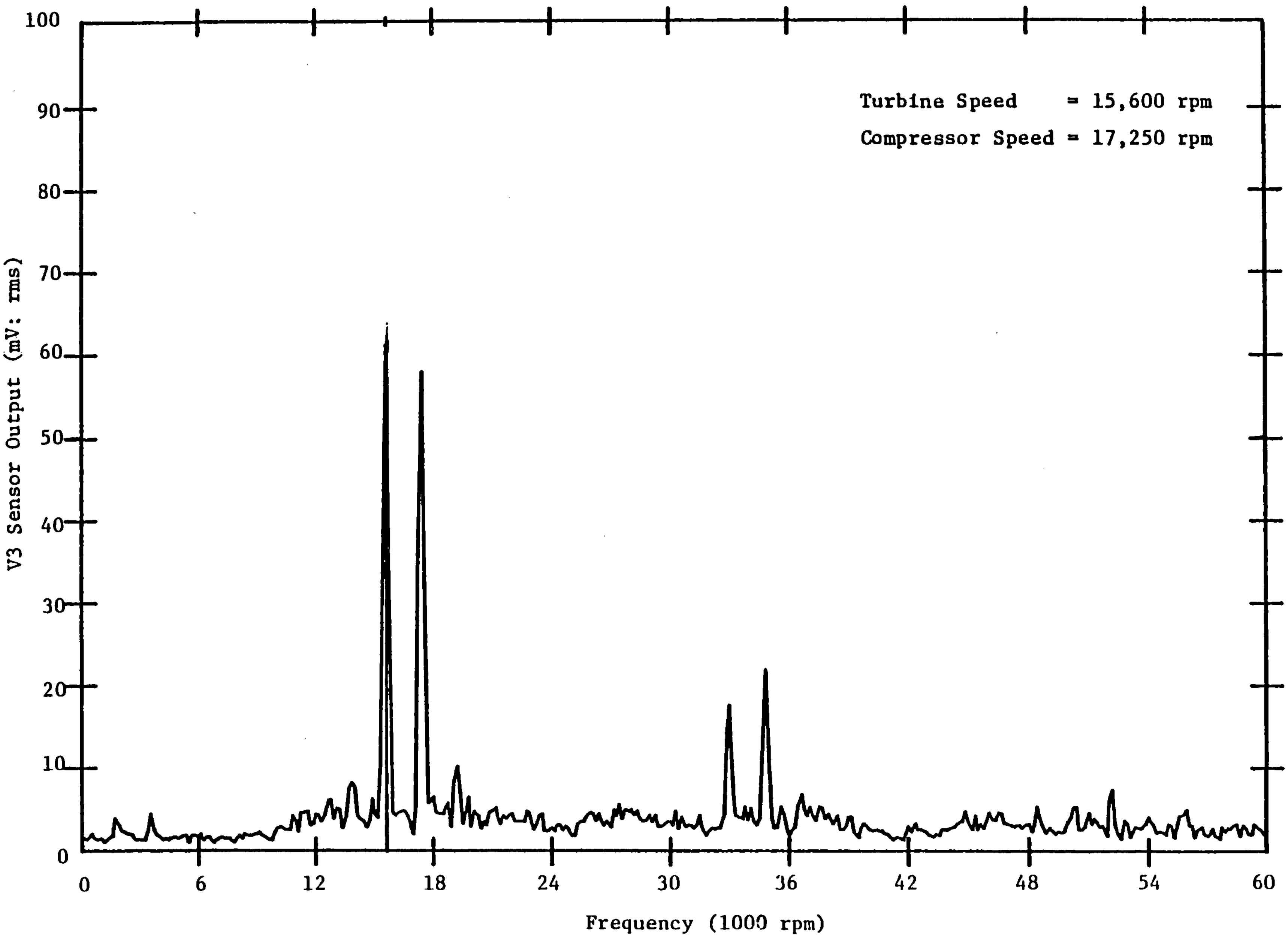

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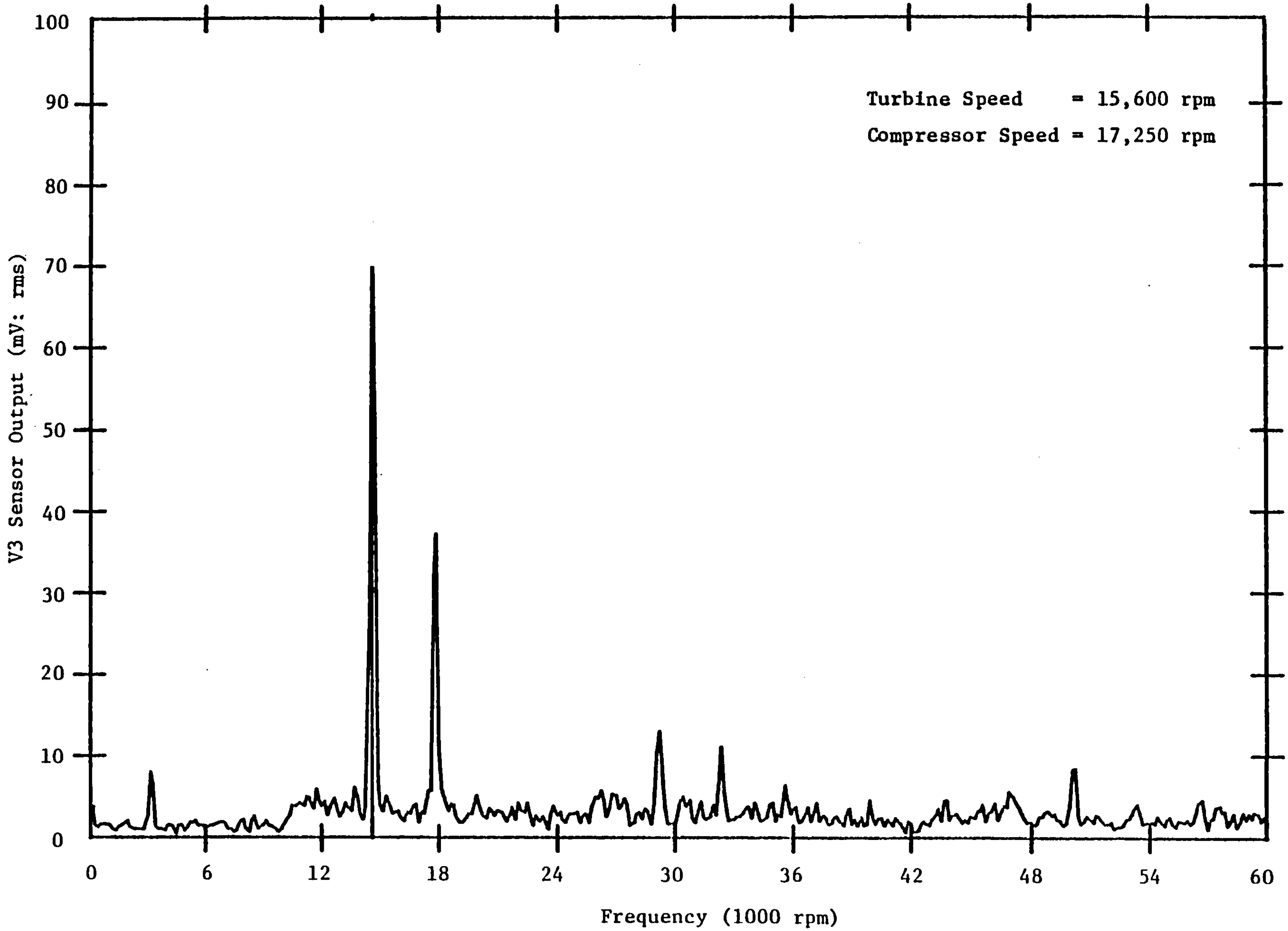

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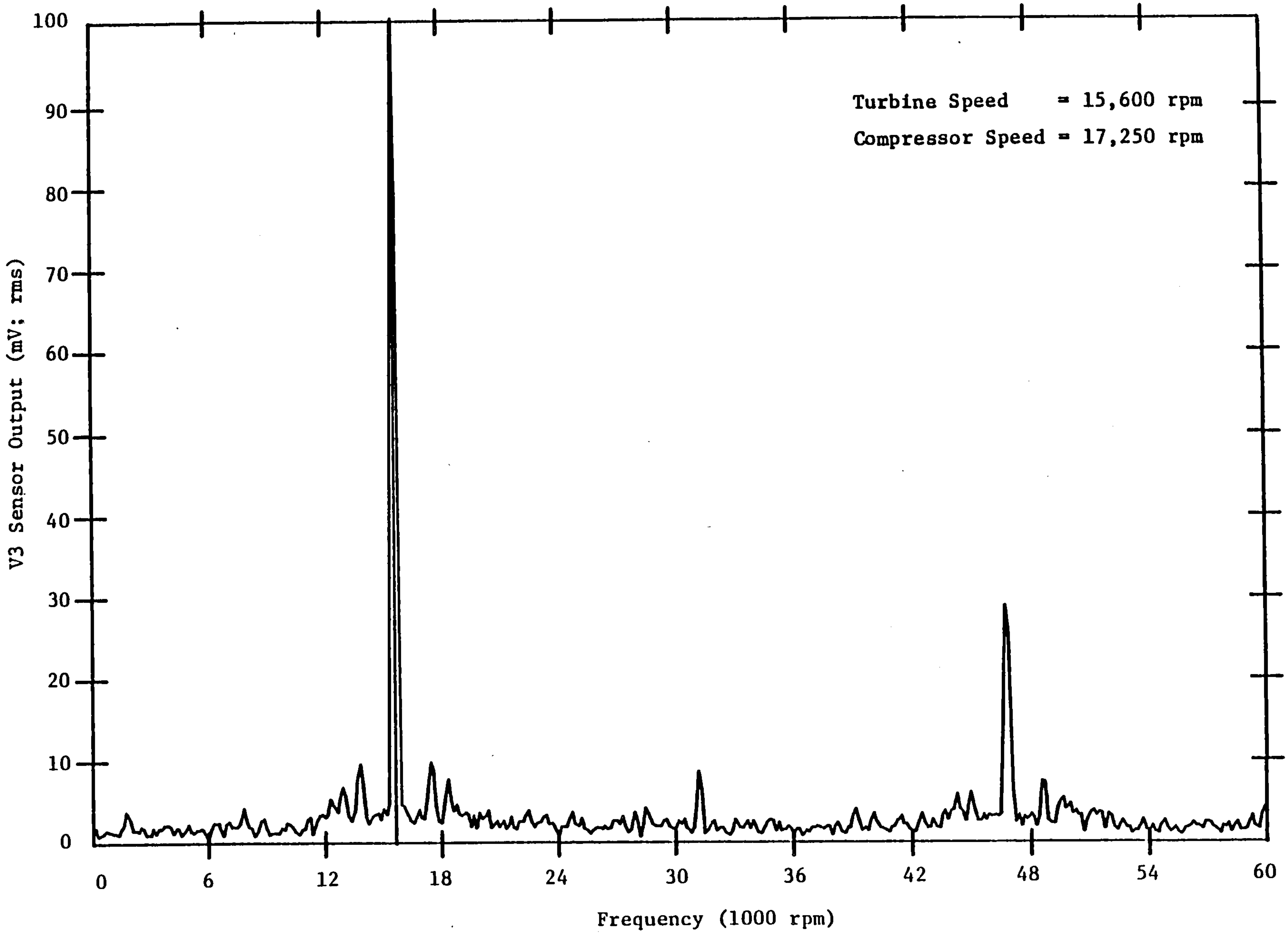

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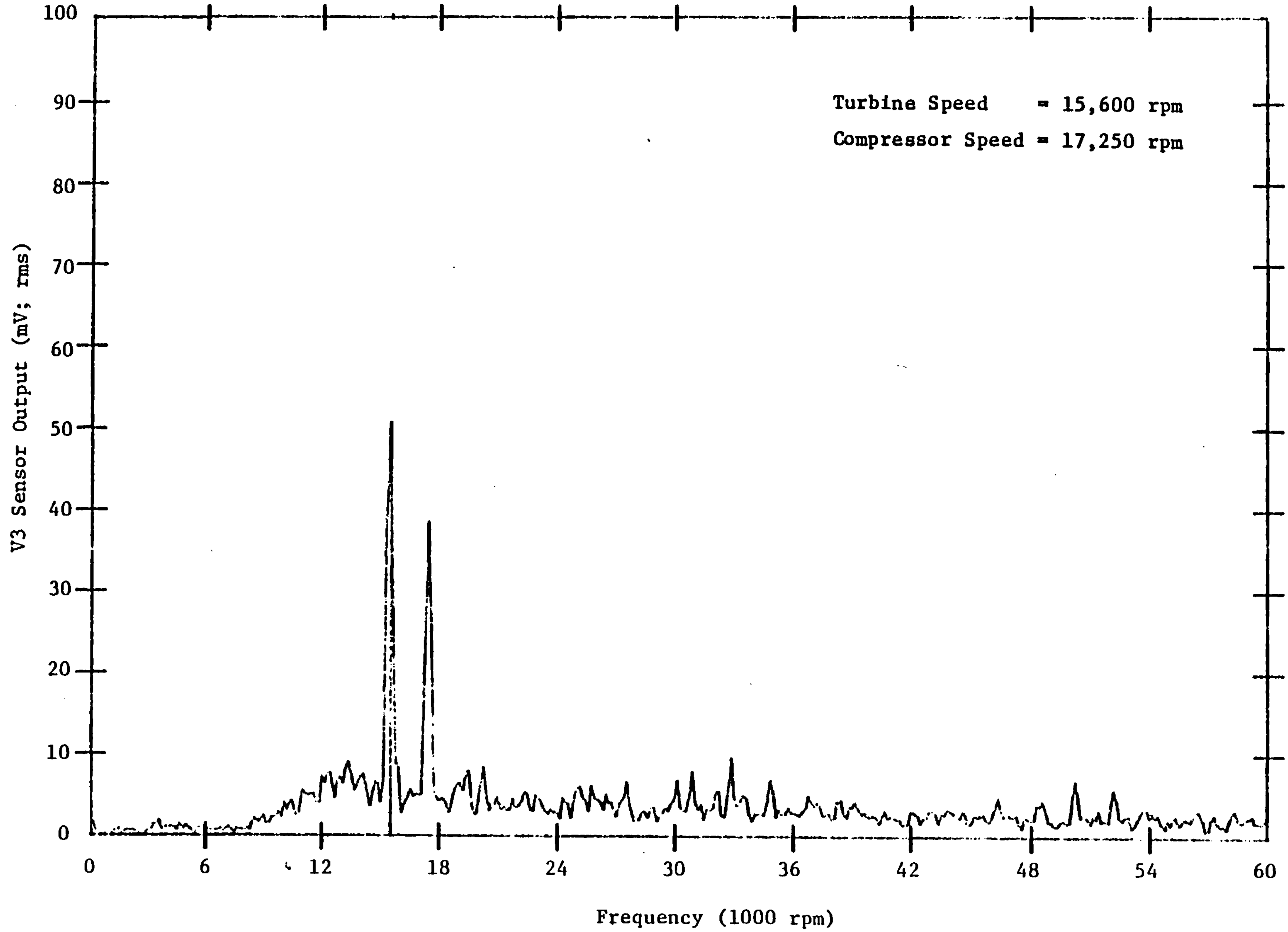

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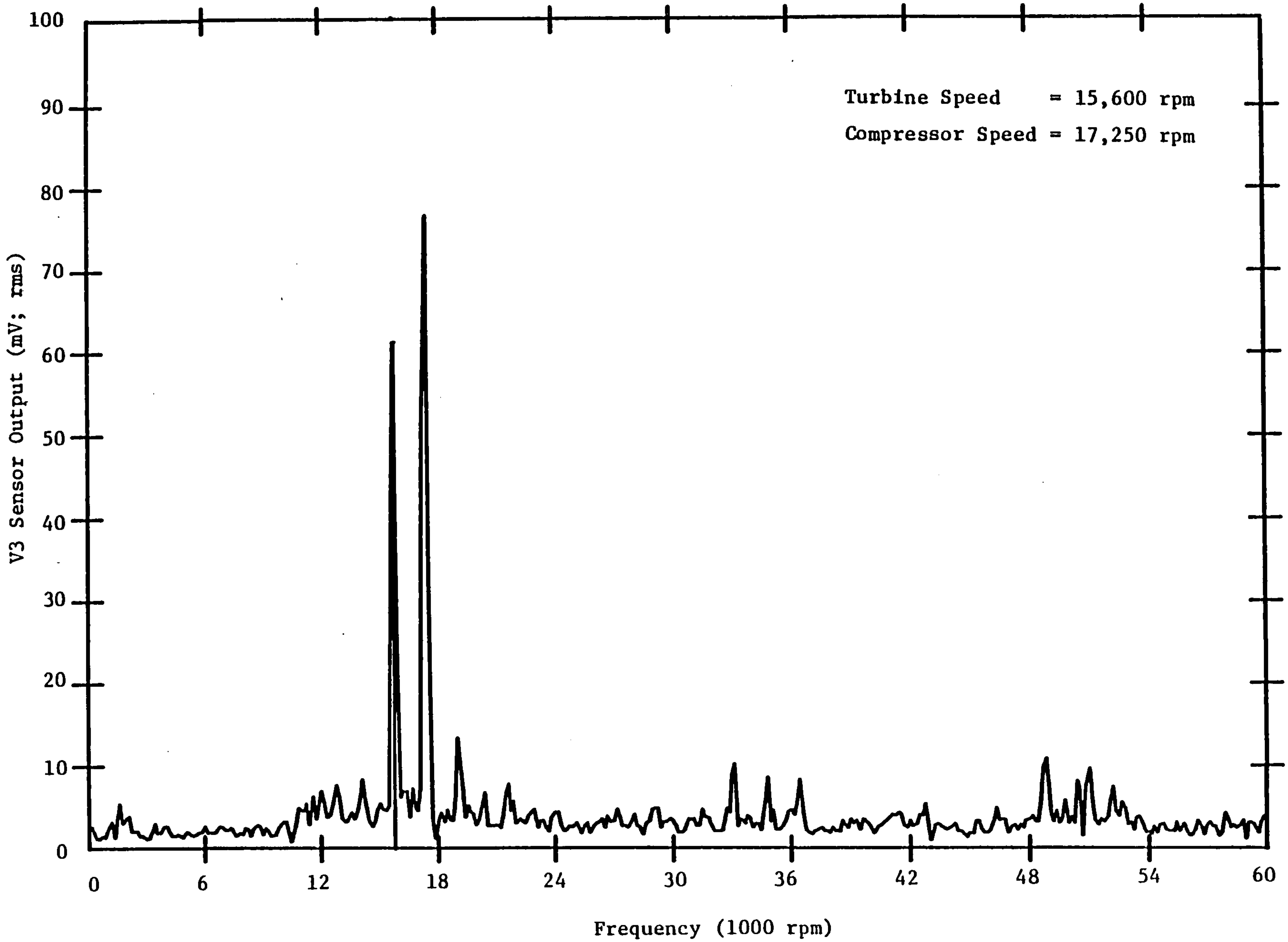

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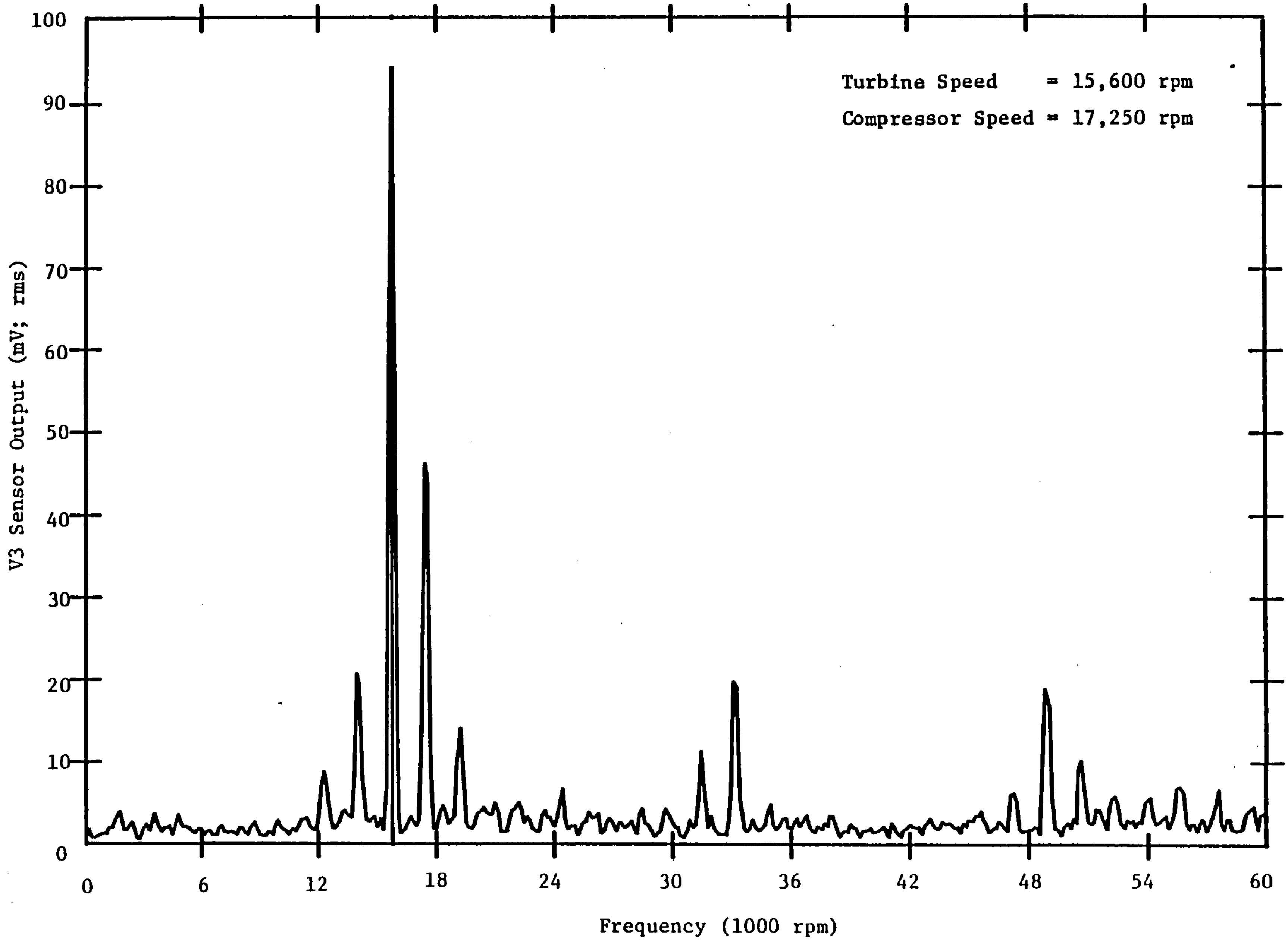

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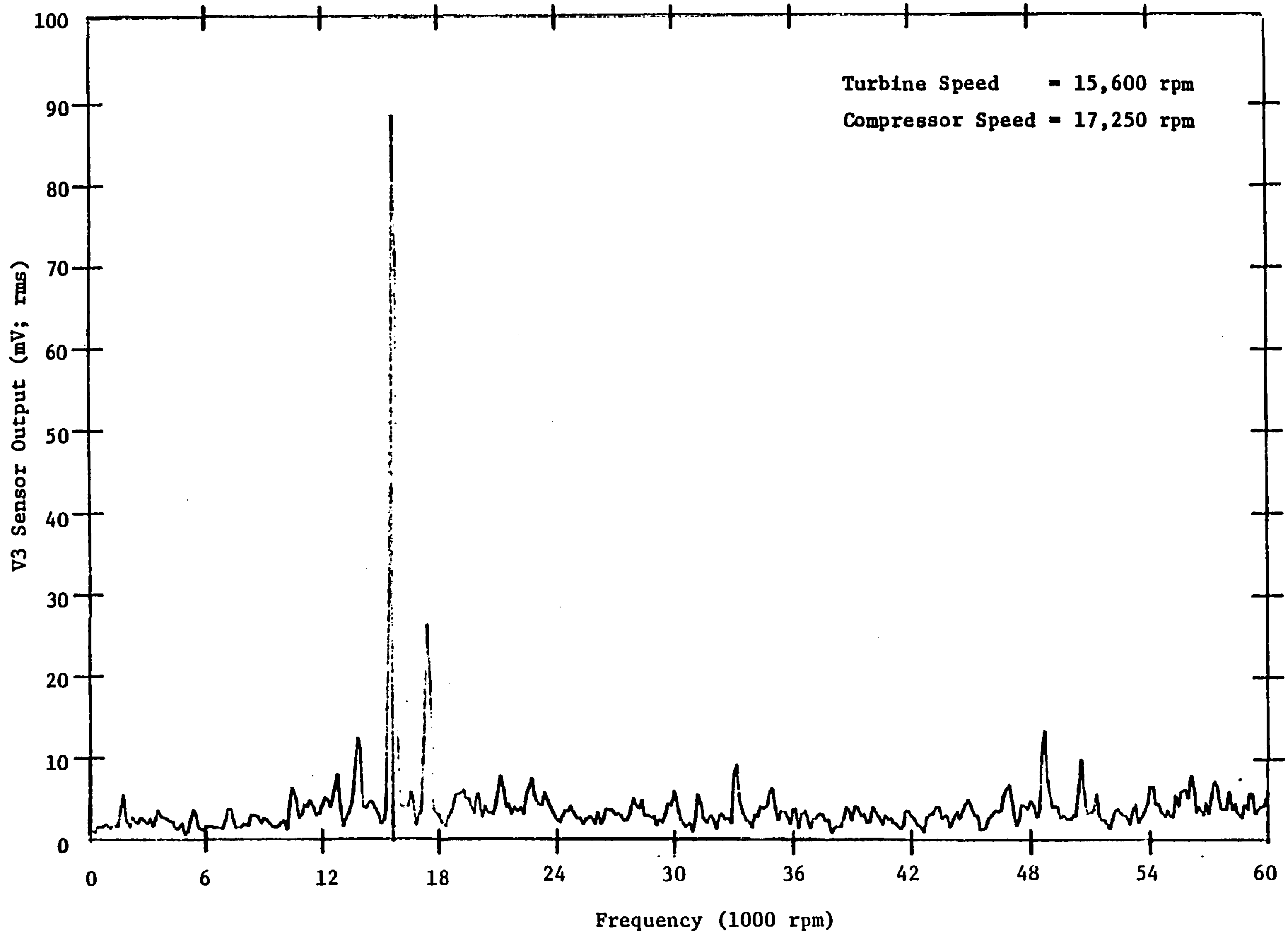

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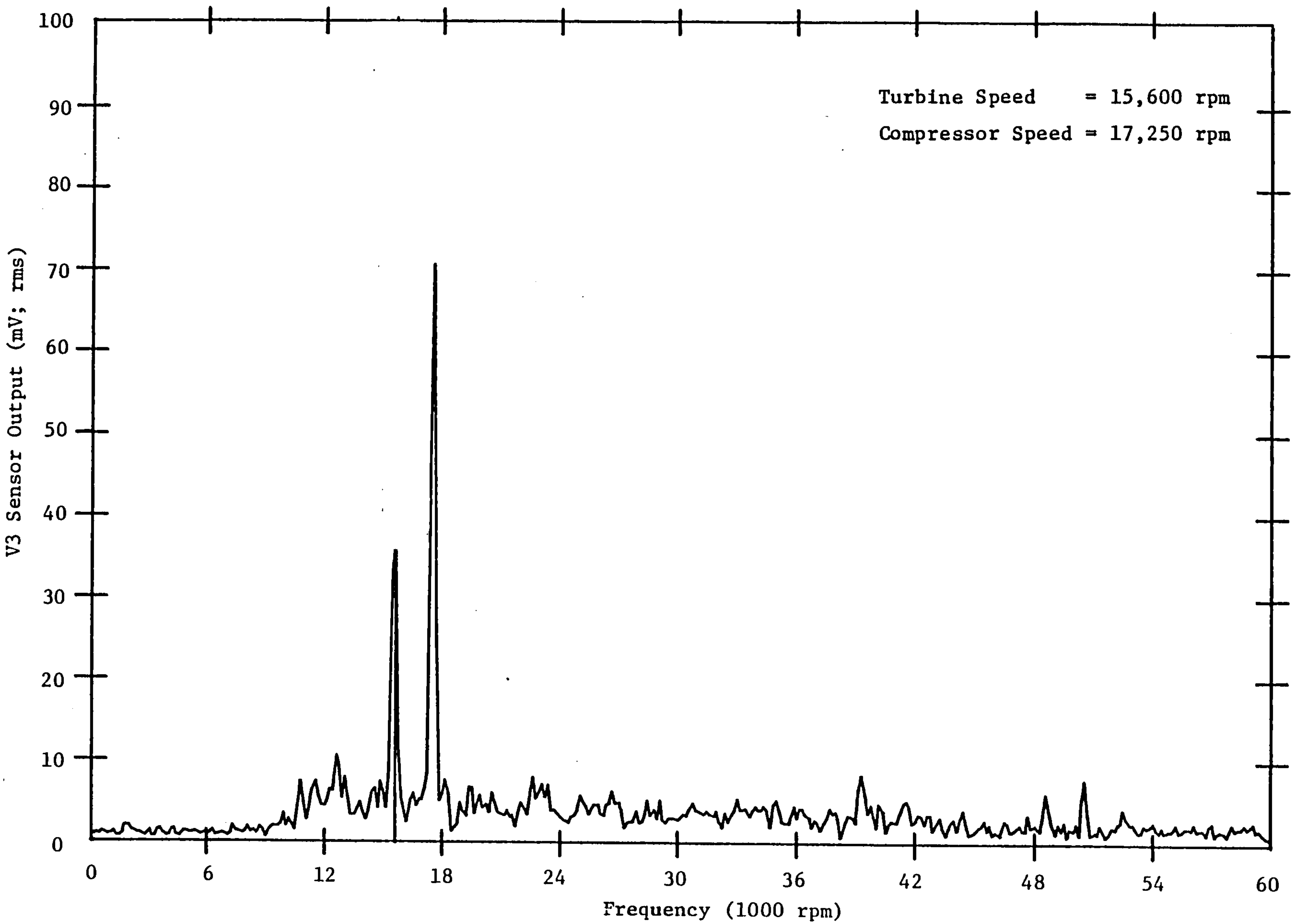

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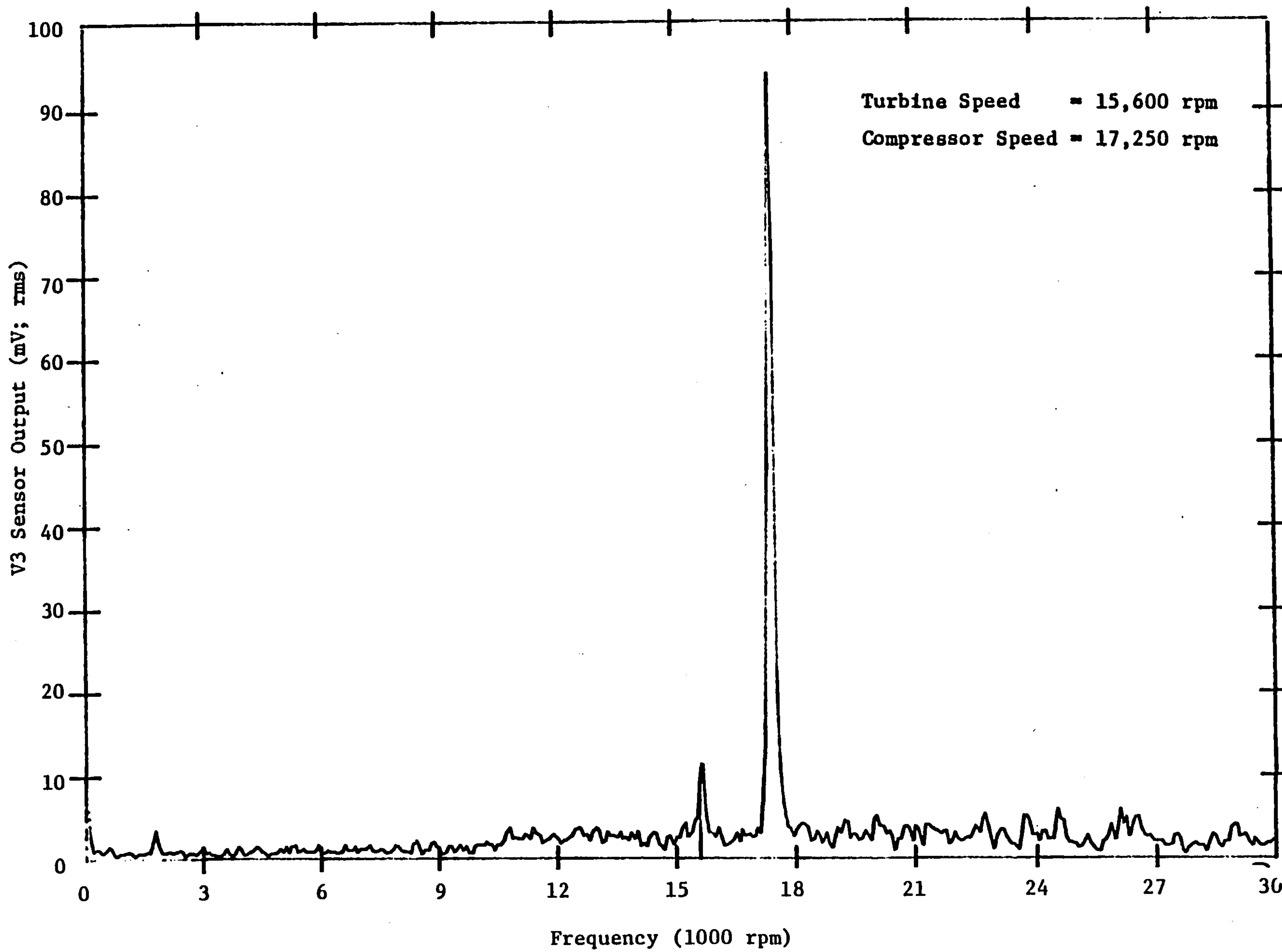

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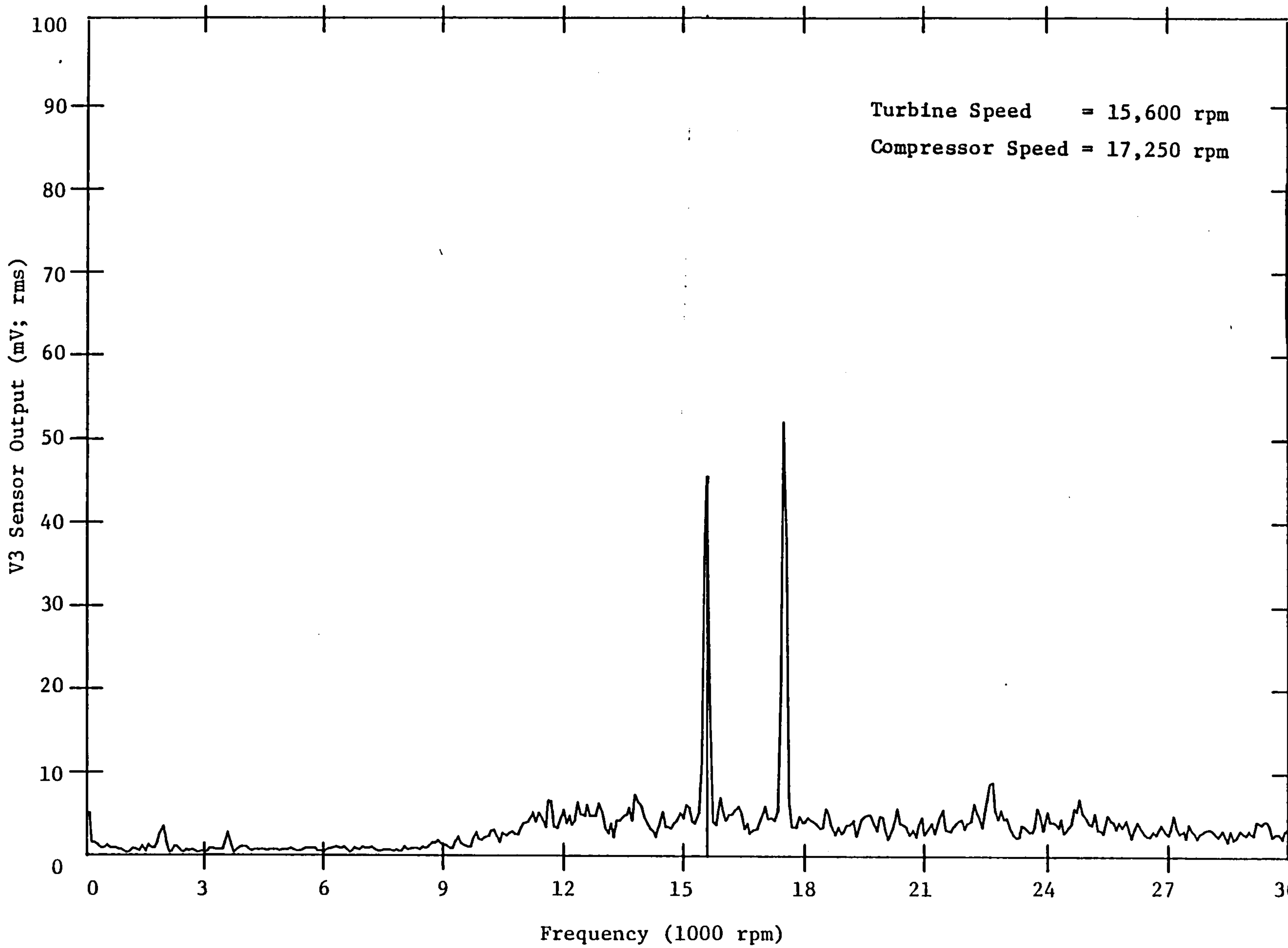

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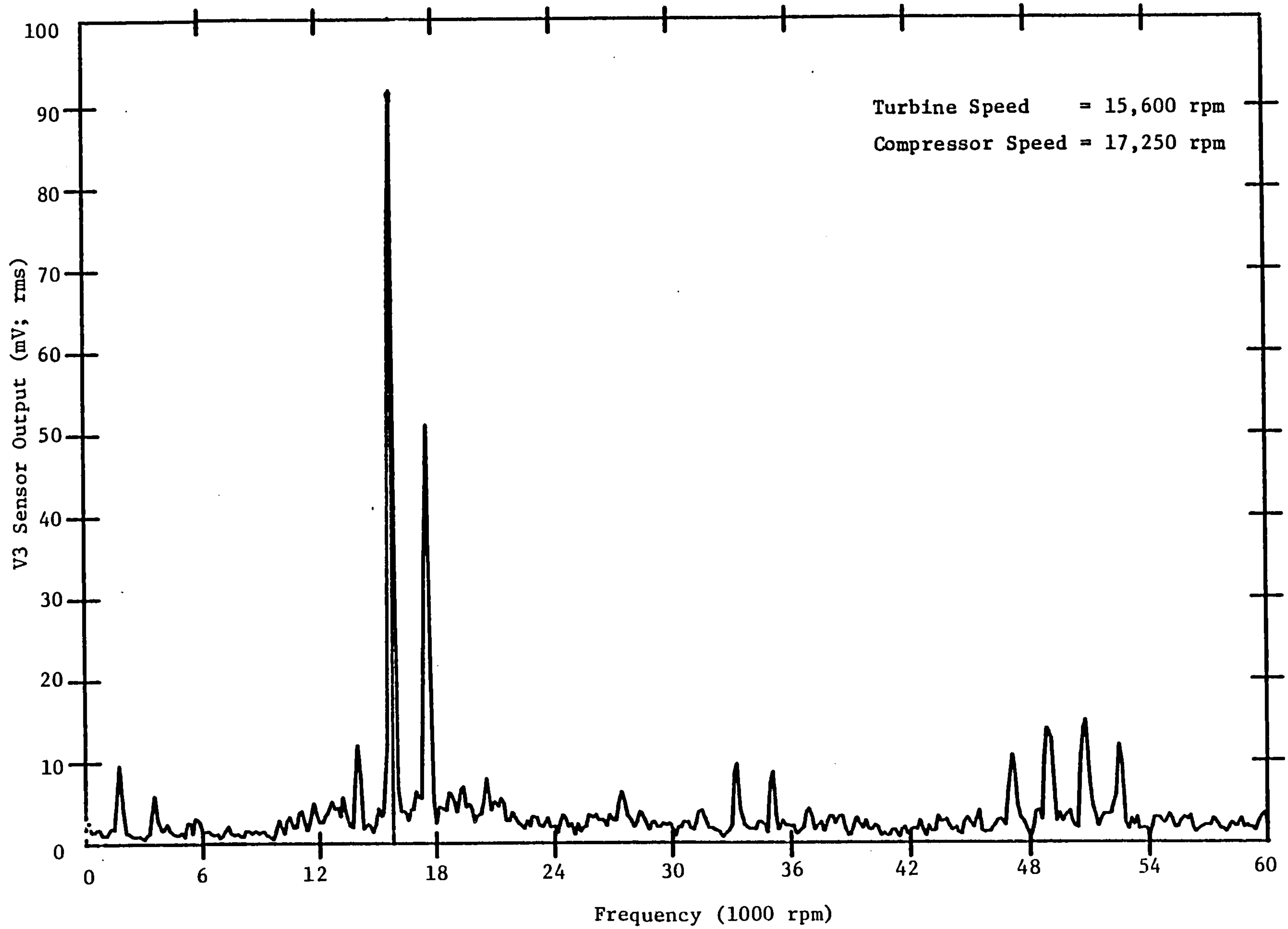

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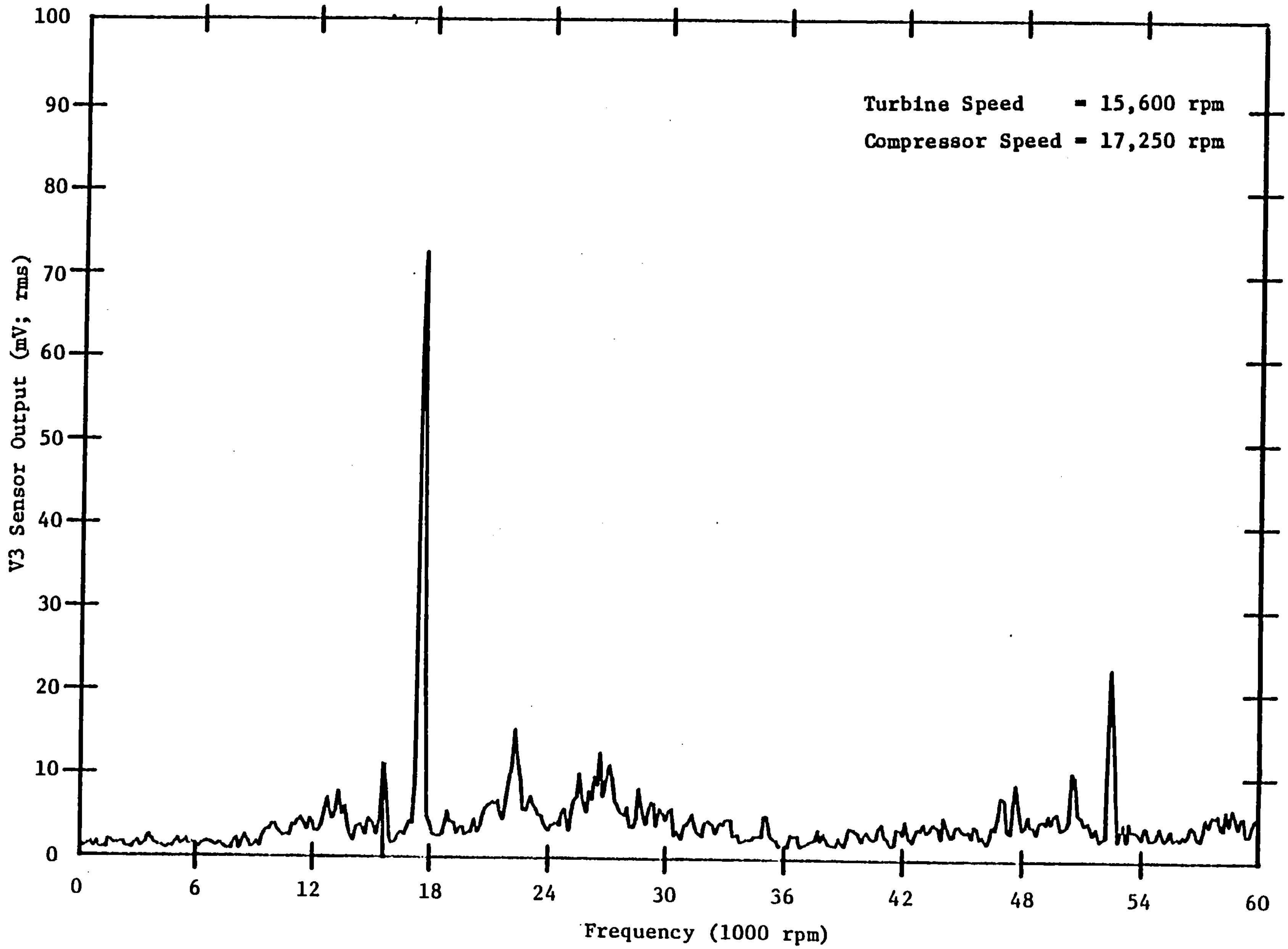

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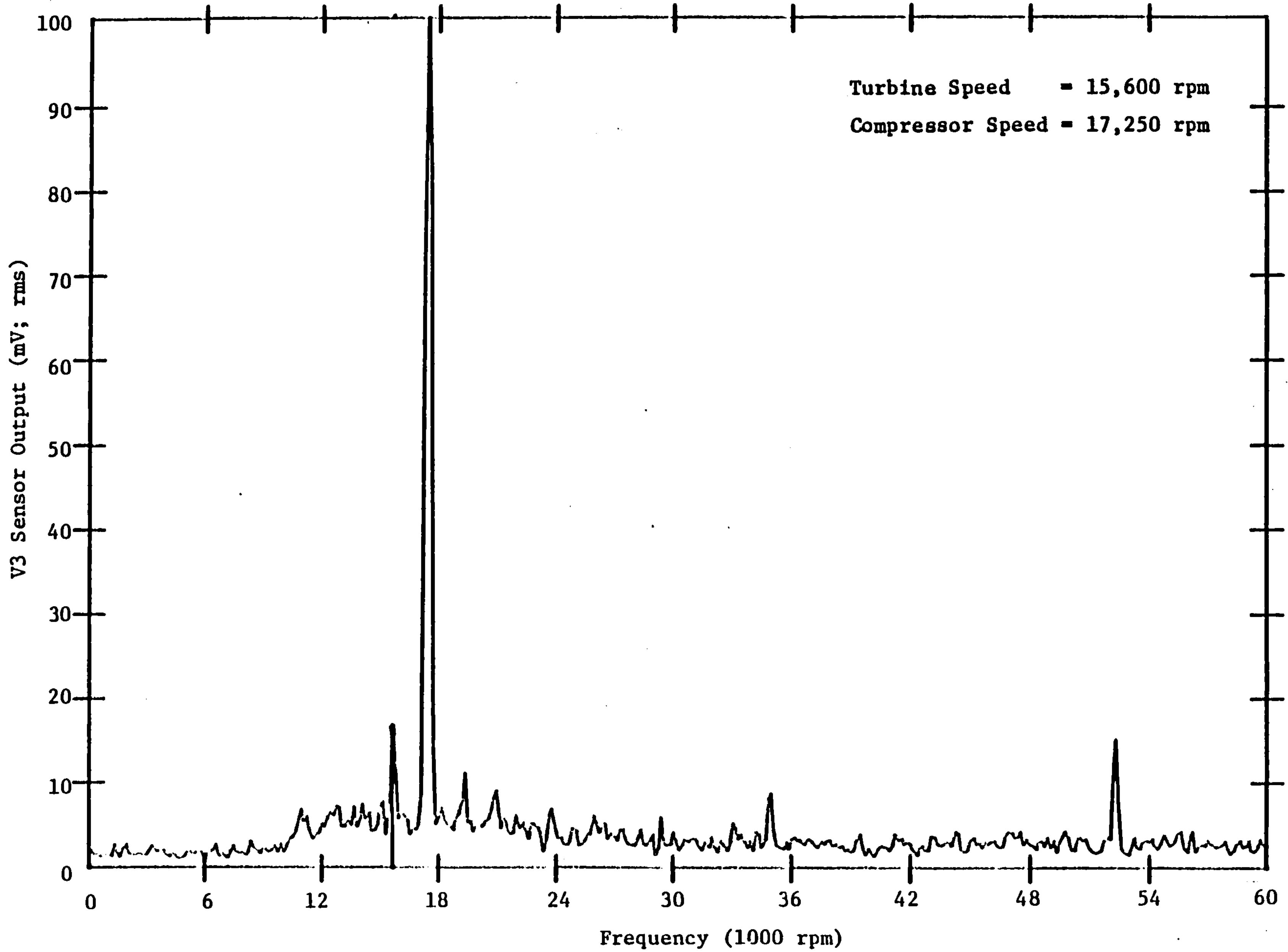

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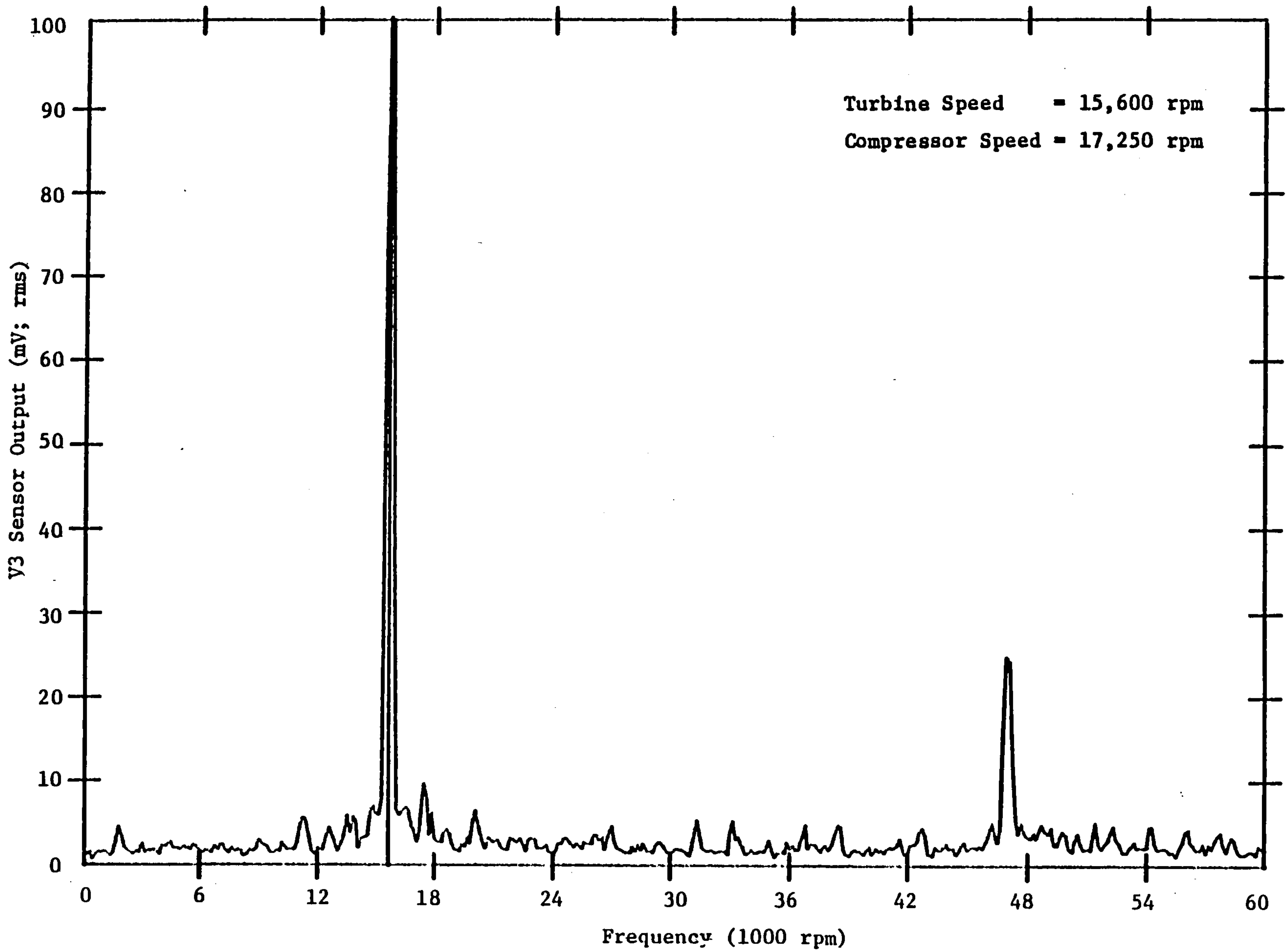

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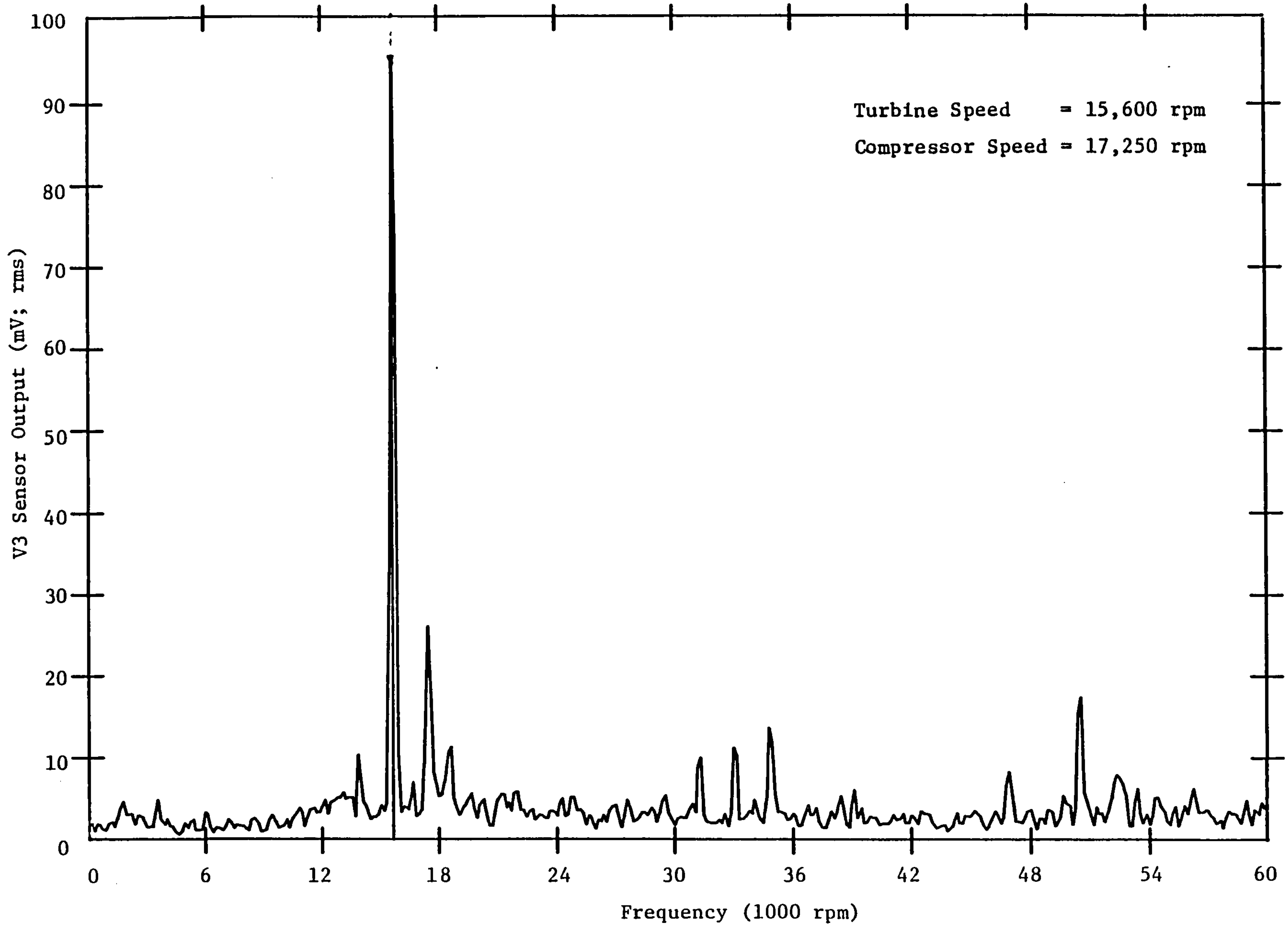

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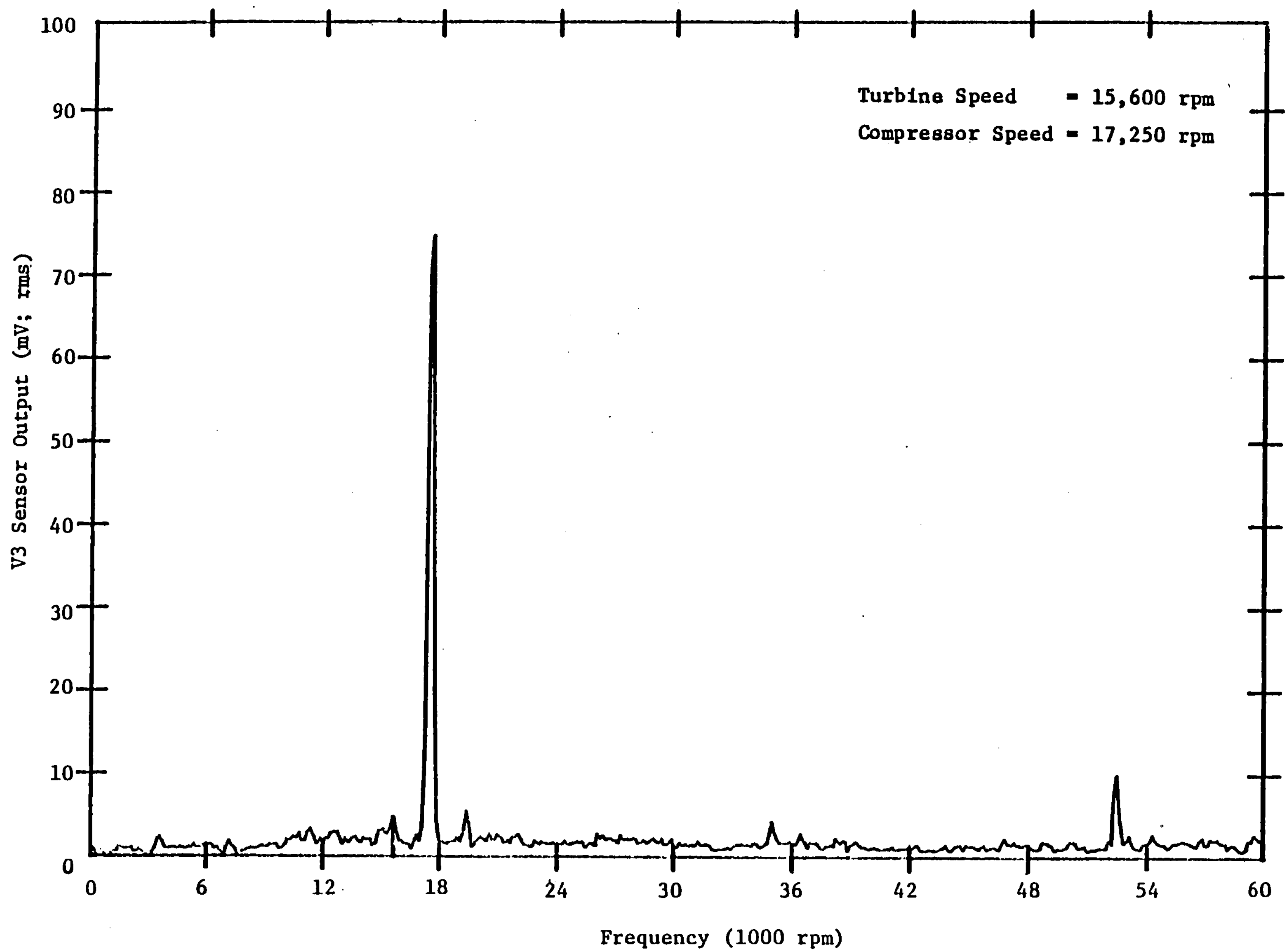

Figure B-19 T55 Power Turbine, Serial No. U00212

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2. Walter, Thomas J. "Study of T53 Engine Vibration." NASA CR-135449, 1978.

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