# TURBINE AND COMPRESSOR VIBRATIONS ENCOUNTERED AT STARTUP OF A METHANOL PLANT

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## ABSTRACT

This paper discusses several types of vibration and failure problems which occurred in a steam turbine-compressor string in a methanol plant resulting in considerable downtime of the process. The vibrations which occurred are representative of those which can occur in most modern high speed installations. By using new and effective ways of presenting data utilizing signature analysis techniques the causes of the excessive vibrations and failures were established. Vibrations encountered were caused by (1) unbalance, (2) misalignment, (3) mechanical resonances of attached structural components, (4) running near critical speeds, (5) sleeves which shifted during operation, (6) nonsynchronous instabilities, and (7) intermittent faulty operation of governor control and overspeed trip. The nonsynchronous instabilities caused several compressor failures, even though tilted pad bearings were used to eliminate instabilities.

By monitoring and recording vibration data on startups, it was possible to replay and study transient vibration phenomena which typical control board instruments can not accurately display. The spectral time history plots and diagnostic techniques presented in this paper can be used to identify similar vibration problems on other units.

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## INTRODUCTION

The use of high speed turbine and compressor strings in chemical process plants has resulted in increased interest in vibration monitoring and control since the outage of any piece of equipment can cause considerable downtime of the process. With downtime sometimes costing up to $100,000 a day, the economic impact can be overwhelming.

The increased complexity of processes many times prevents the complete simulation and testing of the dynamic vibratory characteristics of turbines and compressors in the shop acceptance tests, since it is usually not possible to simulate all design parameters such as pressures, temperatures, process gas, etc., in a manufacturer's facility. In many cases, the unit startup is the first opportunity to check out extrapolated design features. Many troubles occur during startup because of this. Increased surveillance during plant startups is therefore a necessity.

This paper will present vibration data measured on a steam turbine and syn gas compressor rated at 10,600 rpm. These machines ran successfully in their shop tests and no problems were indicated; however, when installed several types of vibration problems occurred during startup and operation. Vibrations were caused by (1) unbalance, (2) misalignment, (3) mechanical resonances of attached structural components, (4) running near critical speeds, (5) sleeves that shifted during operation, (6) instabilities which excited subharmonic frequencies, and (7) intermittent faulty operation of governor control and overspeed trip. The symptoms and dynamic characteristics of vibrations, such as those caused by unbalance, misalignment, mechanical resonances, and critical speeds are well known, and rather simple frequency analysis techniques are sufficient to define them. However, detailed spectral information and analyses of the destructive instabilities that occurred in the compressor have not been previously reported. This information is valuable since it shows that the use of tilted pad bearings is not sufficient to prevent subharmonic instabilities at the shaft criticals. Additional detailed spectral information is also presented for the first time on simultaneous subharmonic excitation of the two lowest shaft criticals on the turbine. Of prime importance is a time history of an overspeed failure which was actually recorded. This data shows how quickly failures can occur and indicates the need for additional safeguards in speed control circuitry.

The problems which occurred illustrate that additional effort is still needed to insure that newly installed equipment will not cause excessive downtime. Areas in which additional improvement are needed are

(1) shop test specifications, (2) acceptable vibration limits, (3) stability calculations, (4) startup procedures, and (5) vibration monitoring. A discussion of the vibrations encountered, the types of signature analyses that were made, the significance of the spectral content of vibrations, and the results of changes that were made may help acquaint others with the types of vibration problems that can occur, how they can be recognized and analyzed on other units, and how to anticipate such problems in future purchase specifications and shop test specifications.

## DESCRIPTION OF EQUIPMENT

The steam turbine was a three-stage machine rated at 13,000 hp with a nominal speed of 10,600 rpm. Maximum continuous speed was 11,750 rpm. The overspeed trip was set at 12,700 rpm. The original rotor was equipped with pressure pad bearings at a bearing span of approximately 60 inches. These were later changed to five shoe tilted pad bearings.

The syn gas compressor had eight stages with a suction pressure of 200 psia and a discharge pressure of 700 psia. The bearings were five shoe, tilted pad bearings with the load on the pad. The bearing span was approximately 64 inches.

The response of the turbine and compressor rotors to unbalance was calculated to determine their sensitivity to unbalance magnitude and location. As pointed out in an earlier paper (1), the only way the measured

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(1)Numbers in parentheses refer to similarly numbered references in bibliography at end of paper.

vibration response of a unit can be correctly interpreted is to have accurate information on critical speeds and amplitude response. Figure 1 shows two calculated responses for the turbine rotor at the outboard bearing with only a phase change in the unbalance. This data shows that the rotor response could change considerably depending on the location of the unbalance. Vibration measurements made on the test stand indicated that the first critical was near 6900 rpm.

## VIBRATION ANALYSES

When vibrations exceed their allowable amplitudes on a machine, it is necessary to make frequency analyses of the complex vibration signal to define possible causes. At startup and after the unit has run for a short time, it is extremely important to establish baseline signature patterns for all data points so that comparisons can be made later if amplitudes increase. For high speed rotating equipment, the most valuable data is the relative shaft-to-bearing housing vibrations obtained by proximity probes. Experience has shown that to fully define the shaft motion, at least two probes 90° apart are needed at each bearing and at least one axial probe per shaft. Figure 2 gives the location of the data points on the turbine and compressor and shows schematically the instrumentation system used to analyze the vibrations. Figure 3 shows some of the instrumentation used.

## VIBRATION CASE HISTORIES

A short summary of several of the vibration problems follows. The frequency analyses and other information used to analyze causes of the problems are discussed.

## Half Speed Vibrations on the Turbine

The retainer ring on the thrust bearing on the turbine was incorrectly installed and axial movement of the shaft caused sufficient damage that the spare rotor had to be installed. New bearings were used when the spare rotor was installed. Figure 4 gives a composite peak amplitude-frequency plot using a real time analyzer for the turbine outboard vertical vibration as the turbine speed was gradually increased to 11,200 rpm. This curve gives a complete response plot which defines the critical speeds. One was at 4320 rpm and another at 6200 rpm. This is similar to the response predicted in Figure 1, Curve 2.

When the turbine speed reached 11,580 rpm, a vibration component of 1.5 mils at exactly half speed was excited which was larger than the running speed component. The bearings were found to be approximately 1.5 mils out of round. A new set of bearings was checked and installed. After balancing the new rotor, the half speed components were nearly zero. This illustrates that for this machine, even small out-of-roundness or changes in the bearing properties could cause instabilities to occur in the pressure pad bearings.

## Nonrepeatable Vibrations on Turbine

After several balancing tries on the new rotor, the vibration amplitudes were reduced to less than 1.5 mils at all four data points. The data taken at the outboard end of the turbine in the vertical direction is given in Figure 5, Curve 1. When an attempt was made to repeat this data, Curve

2 was obtained. Note that the shapes of the curves are different. This indicates that the balance condition changed. To check out the nonrepeatable data, the unit was slowly cycled from 3000 to 11,000 rpm six times and the data obtained again. Curve 3 shows that the amplitudes had increased by a factor of nearly 2:1. When the unit was brought back up, Curve 4 resulted showing that the amplitudes were not repeatable.

This evidence was sufficient to prove that the sleeves on the rotor shaft were shifting and causing the unbalance of the rotor to change. This design feature was later changed to eliminate this undesirable characteristic.

Several interesting events occurred during the time this design was being changed. First of all, in order to run the turbine, constant monitoring of the shaft vibrations was necessary since the vibration amplitudes could change quite suddenly. The allowable vibration amplitudes had been extended from 1 mil to 2 mils in order for production to continue; however, the amplitudes finally exceeded 2 mils. The turbine speed was varied to determine a speed at which the vibration amplitudes were less than 2 mils; however, none could be found. Finally in desperation, primarily due to production commitments, the turbine speed was lowered to the critical speed and kept there for a few seconds to "shake" the rotor up. When the rotor was brought back to 9500 rpm, the vibration amplitudes were less than 1 mil. This was due to the sleeves shifting to a better balance position. Although this procedure could not be recommended in every case, sometimes by monitoring the unit with proper equipment, unorthodox steps can be taken to insure that production continues.

Another important aspect was the excellent performance of the turbine rotor which ran for approximately a month with vibrations in the order of 2 mils near 9600 rpm. When opened, the internals, seals, and bearings were in excellent shape. Although established criteria and codes are useful in setting goals and establishing specifications, more information is needed to determine when it is safe to exceed allowable levels.

When the sleeve design was changed, the vertical vibration amplitudes at the outboard bearing were less than 0.5 mil (Curve 5) and repeatable after many speed cycles.

Compressor Instability Vibrations

When the speed of a rotor is greater than two times the first critical, lateral vibrations at the first critical can be excited. The classical oil whip resonance has been discussed by many investigators. Instabilities at the first critical can also be introduced due to the internal friction effects at press fits and by aerodynamic drag effect at seals and impellers. This has been discussed by Gunter (2), Trumpler (3), Tondl (4), and Ehrich (5). Historically, the solution to instability resonances has been to use tilted pad bearings, since they are highly resistant to the initiation of this phenomenon. This was one of the reasons the syn gas compressor was installed with five shoe, load on the pad, tilted pad bearings.

The compressor failed after three months of operation. The labyrinths and seals were completely wiped in a concentric circular pattern. When the compressor was inspected, it contained a large quantity of water; therefore, this failure was attributed to liquid carryover.

After installing additional knockout drums and other safety devices, a second failure was experienced. This time, the seal oil pressure dropped suddenly and the speed dropped to 4500 rpm, which was the compressor first critical, and remained there for an undetermined length of time. It was thought that this caused the second failure. A month after the compressor was rebuilt, it vibrated severely when the system tripped out. The seals and labyrinths were again wiped. Since no spare parts were available, the unit was brought back up under partial load. Data was recorded on the next trip-out to capture the severe compressor vibrations. A strip chart recording of the trip-out is given in Figure 6. Figure 7 is a time history plot of this sequence of events. Each vertical offset of the horizontal line represents an interval of time; therefore, the overlay of the instantaneous frequency analyses from the real time analyzer shows exactly what frequencies were excited as a function of time.

An analysis of the time history shows that immediately after the trip-out occurred, the component already present in the vibrations at 4000 rpm increased in amplitude (approximately 16 mils) until the bearing clearances were exceeded. The rotor then vibrated primarily at 6200 cpm for approximately 0.5 second, with an amplitude of 8 mils. The speed then locked onto the first critical at 4500 rpm with an amplitude of 15 mils. The vibrations remained at this high level for several seconds. The severe instability which excited the first critical while the speed was still above 9000 rpm apparently could be initiated by system upsets such as sudden pressure changes, fast speed drop, or overspeed trip-outs. This type of vibration problem could become more prevalent if analytical techniques are not improved so that the instabilities can be accurately predicted in the design stage.

For this system several steps were taken to reduce those elements which promote this type of instability. The wheels were undercut to reduce the frictional effects at the mating surfaces. Also, the clearances in the labyrinths and seals were increased. The five shoe tilted pad bearings were also modified by reducing the pad areas on the side and by increasing the radial clearance to force the rotor to vibrate in a horizontal elliptical orbit, which helps prevent the pure circular orbit of the first critical. The theory explaining why these changes were made is beyond the scope of this paper; however, the author and his colleagues plan to deal with this one subject in a later paper utilizing this data and similar data measured at other installations.

The changes were sufficient to remove the tendency toward instability since the unit has run satisfactorily since these modifications were installed. In addition, several unexpected trip-outs and process upsets have occurred without any harmful effects on the compressor vibrational characteristics.

Subharmonic Vibrations of Turbine

Five shoe tilted pad bearings were installed in the turbine in an attempt to eliminate the half speed problems which were discussed earlier. During the turbine startup with the new bearings, a phenomenon occurred which was recorded on tape and later replayed and analyzed. Figure 8 shows that two subharmonic criticals at 4800 and 7000 cpm and a component at approximately one-half speed were excited.

Figure 9 shows that a large amplitude component at approximately 0.3 times the running speed occurred at 2500 cpm when the speed was 9000 rpm. When the speed suddenly dropped 200 rpm, the instantaneous frequency analysis (Figure 9) shows that this moved the subharmonic component from 2500 cpm to 4400 cpm.

The exact causes of these subharmonic instabilities were difficult to define. Without proper analysis equipment, the entire sequence would have been virtually undefined. Several positive steps were taken which reduced the magnitude significantly. One was to strengthen the bearing retainer ring. Another was to make the seals nonrotating.

### Intermittent Faulty Operation of Governor and Overspeed Trip

Some maintenance and slight modifications were made to the governor speed control system. The machine was monitored on startup to check on the improvement of the subharmonic turbine and compressor vibrations. During startup, some difficulties were again experienced with the governor speed control. Since this problem eventually caused a disastrous overspeed of the turbine, the various indications of malfunction will be presented.

The turbine was being soloed and warming up at approximately 3000 rpm when suddenly the speed increased rapidly to 15,500 rpm in less than a second. Some adjustments were made and the turbine was again soloed and successfully checked on overspeed trip.

After the compressor was coupled up, measurements of the turbine and compressor vibrations showed that total amplitudes did not exceed 1 mil anywhere. The turbine tripped out suddenly while running at 10,500 rpm when seal and governor oil pressure was lost. After a short time, and without warning, the unit came back on and accelerated to 10,600 rpm in 2 seconds as shown in the time history in Figure 10. Vibrations on the turbine and compressor were in excess of 4 mils.

The turbine and compressor were uncoupled and inspected for damage. After needed adjustments were made to control circuits to prevent the valves from sticking fully open, the turbine was soloed again. Vibration amplitudes on the turbine at 10,500 rpm were less than 0.4 mil at all four probes. On the overspeed trip-check, the unit was slowly brought up to the trip speed of 12,700 rpm. Upon trip-out, the speed jumped to 19,600 rpm instantaneously and then dropped. This is documented in Figure 11. Due to the response time of the digital tachometer in the control room it did not indicate any overspeed.

After the turbine had experienced overspeed on trip-out, it was necessary to check the overspeed trip again. After slow rolling the unit, the control valve was slowly opened to increase the turbine speed. The turbine, which was under no load, suddenly jumped to 10,600 rpm in approximately 0.4 seconds. It was then shut down.

After rechecking all the governor speed controls, the turbine was brought up to 9000 rpm and vibrations were less than 0.5 mil overall. Suddenly without warning, the speed jumped from 9000 to 25,500 rpm in less than 2 seconds. The overspeed trip did not work and efforts to shut down the machine in the control room were unsuccessful. The thrust collar finally was thrown through the bearing housing, an oil line broke, the oil ignited, and a fire resulted.

These events were being recorded and monitored at the time of the failure and the spectral time history of this failure recorded on the inboard vertical probe is given in Figure 12. Notice that the amplitudes at 25,500 rpm were less than 2 mils for about 3 seconds; however, at approximately one-half speed, 12,750 cpm, the amplitude gradually increased to nearly 4 mils. The shaft then appeared to touch off and excite several low frequencies. The shaft speed decreased to 20,000 rpm and the amplitudes were greater than 12 mils. The speed decreased to 16,000 rpm where the amplitude reached 14 mils, at which time the probe was lost. All these events took place in approximately 30 seconds.

The value of monitoring rotating equipment can be easily understood when one considers the fact that none of the board instruments indicated the overspeed which caused the failure. By analyzing the taped vibration data, the exact sequence of events was documented and therefore, steps could be made to correct the speed control circuitry.

## CONCLUSIONS

This paper has described several vibration problems which resulted in major equipment failures at a methanol plant. Some of the classic problems such as unbalance, misalignment, shaft criticals, and half-speed excitation were observed; however, new signature analysis information was obtained on the problems which have not been previously detailed or documented:

(1) Instabilities of the compressor shaft at its first critical that caused repeated failures. These vibrations were unexpected since tilted pad bearings were used for stability purposes.

(2) Simultaneous instabilities of the turbine rotor were observed at two criticals below running speed when pressure pad and tilted pad bearings were used.

(3) Shifting sleeves on the turbine rotor caused drastic changes in vibration amplitudes and response characteristics.

(4) The overspeed failure of the turbine. Vibrations were recorded and documented using spectral time history plots which gave additional insight into the sequence of events.

Indications of these problems were not found when the normal shop tests were made in the manufacturer's facility. This illustrates the need for additional vibration testing and monitoring procedures for shop tests and at plant startups since state-of-the-art analytical techniques are not sufficient to accurately predict the subharmonic vibrations.

Signature analysis techniques were presented in this paper which can be used to identify these and other types of vibration problems. The use of these techniques during plant startups is desirable so that:

(1) The exact location of the critical speeds can be experimentally verified. This is important since unbalance location and

differences between shop test and field installations can shift criticals significantly.

(2) The amplitude response and dynamic balance over the entire speed range can be established.

(3) Baseline signature analyses can be established. These can be valuable in determining the causes of excessive vibrations if they occur later.

(4) Undesirable characteristics which might be potential problems can be identified. This includes the excitation of subharmonic frequencies and critical speeds, particularly on high speed trip-outs.

(5) The dynamic spectral characteristics of any excessive vibrations or design deficiencies can be documented. This is extremely important since the equipment manufacturer and the user must be in agreement as to the cause of vibration problems before steps can be taken to correct them. The complete analysis of all vibrations at plant startups thus can expedite any remedial actions that are needed.

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# LIST OF FIGURES

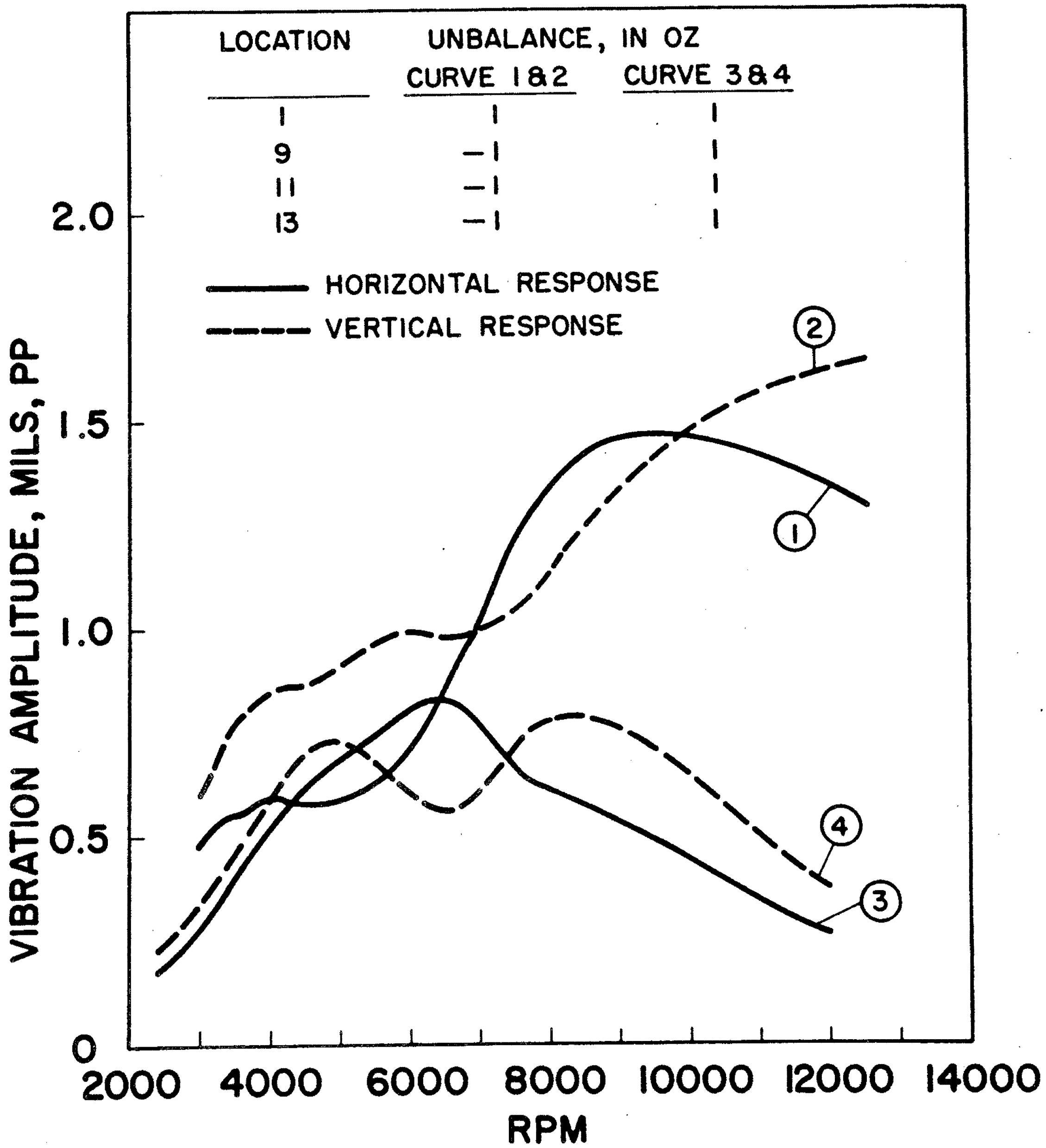


FIG. 1 PREDICTED VIBRATION RESPONSE OF TURBINE ROTOR

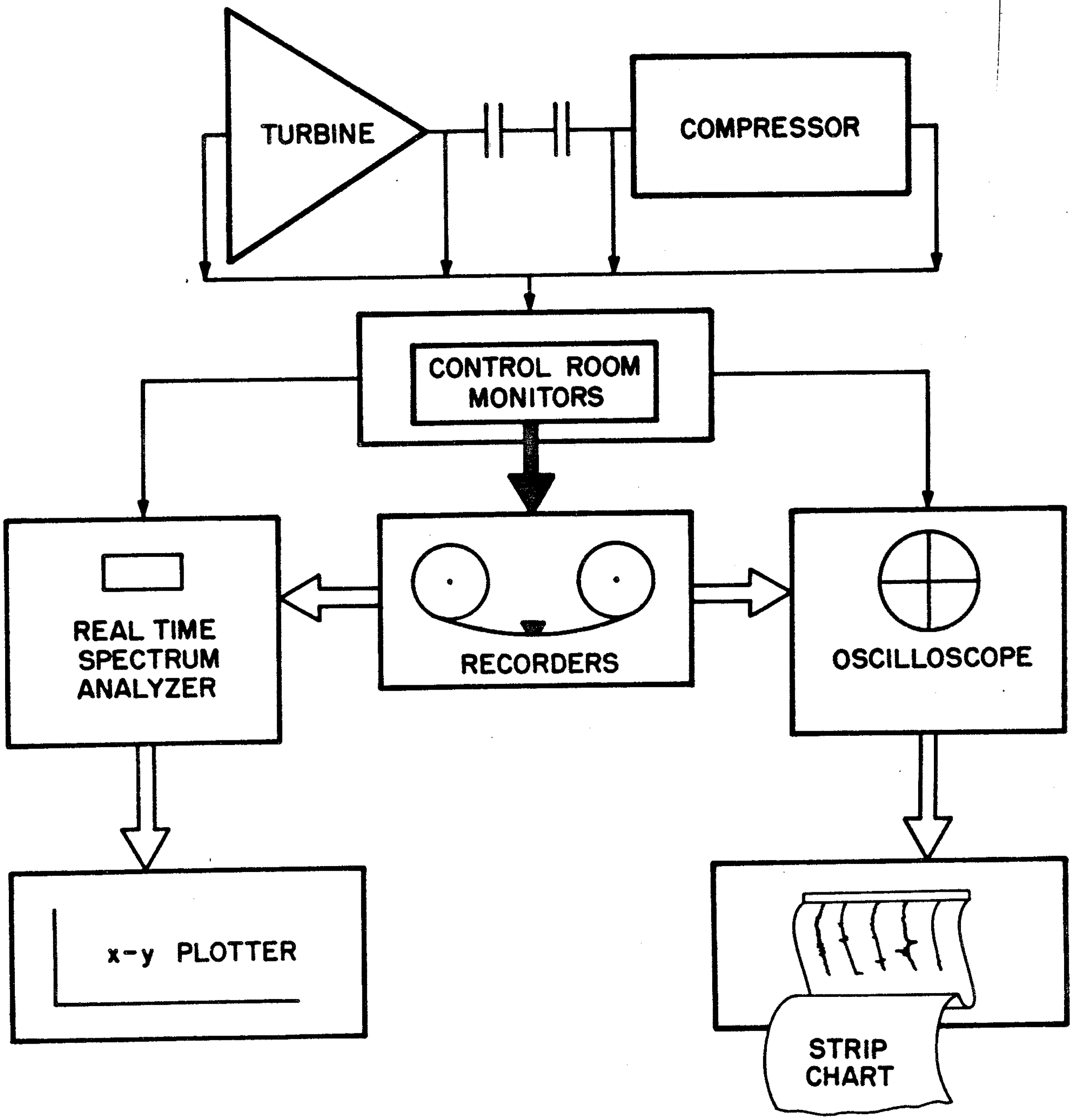


FIG. 2 DATA POINTS AND INSTRUMENTATION SYSTEM

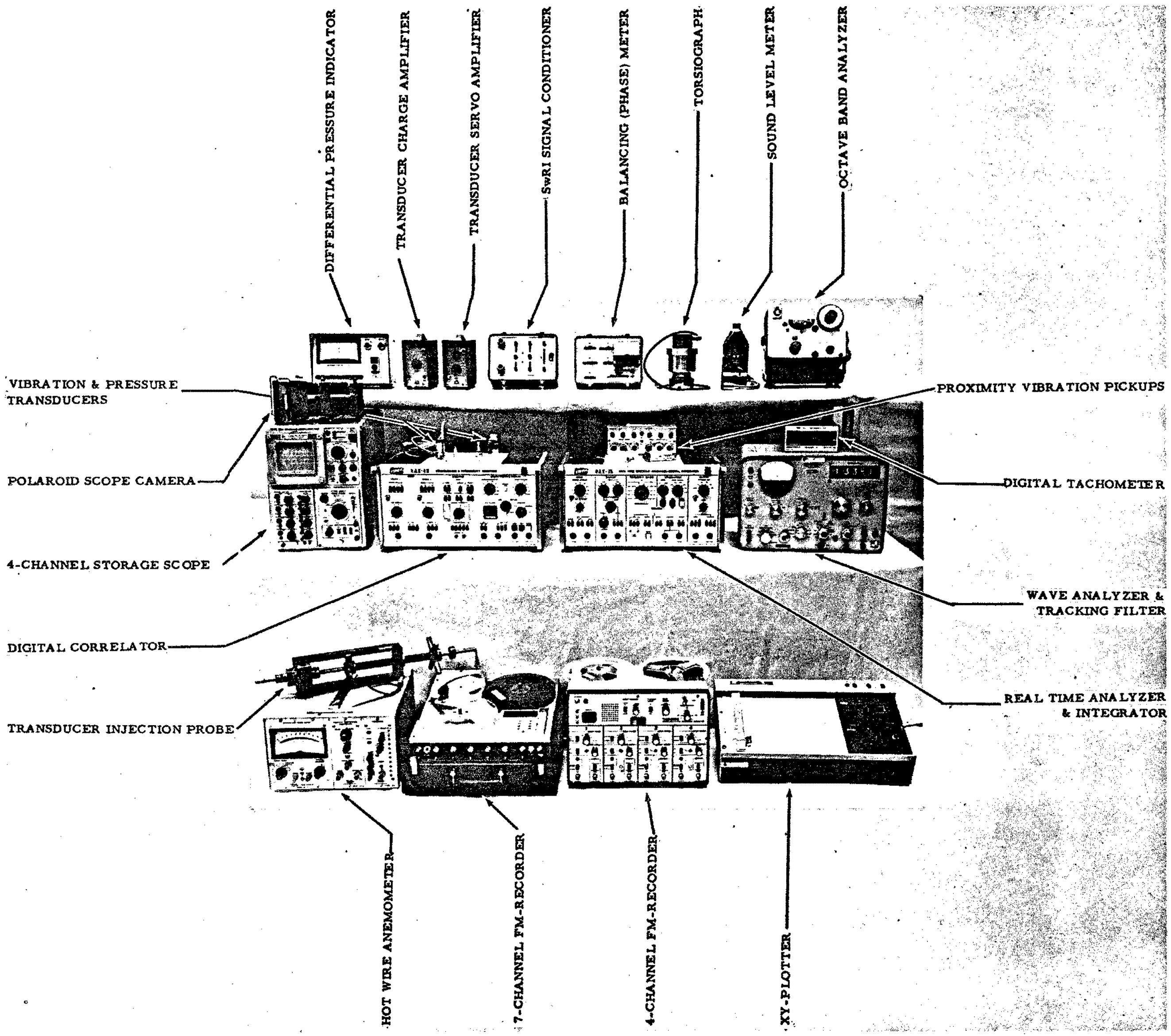


FIGURE 3

INSTRUMENTS USED IN SIGNATURE ANALYSES

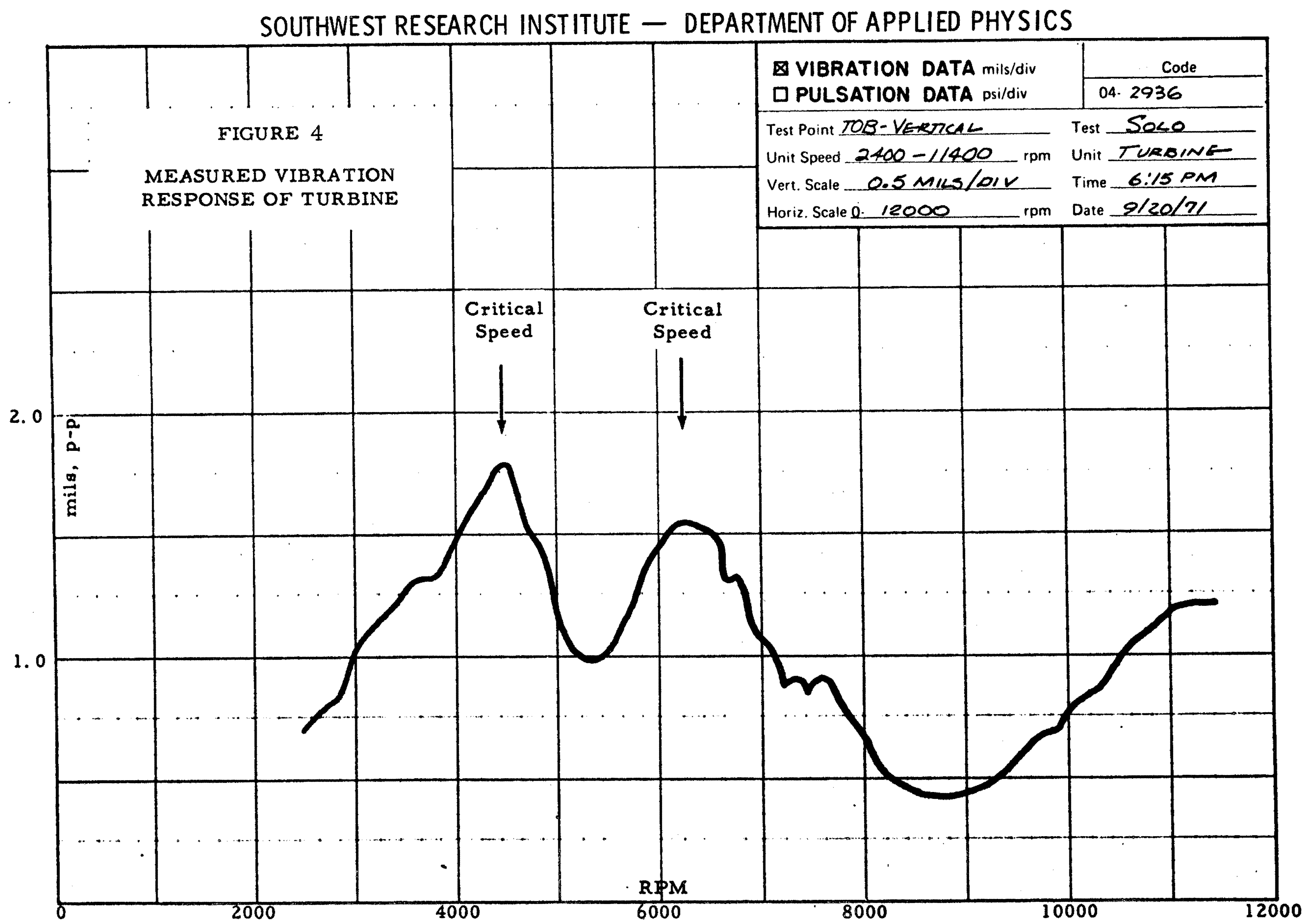


FIGURE 4

MEASURED VIBRATION RESPONSE OF TURBINE

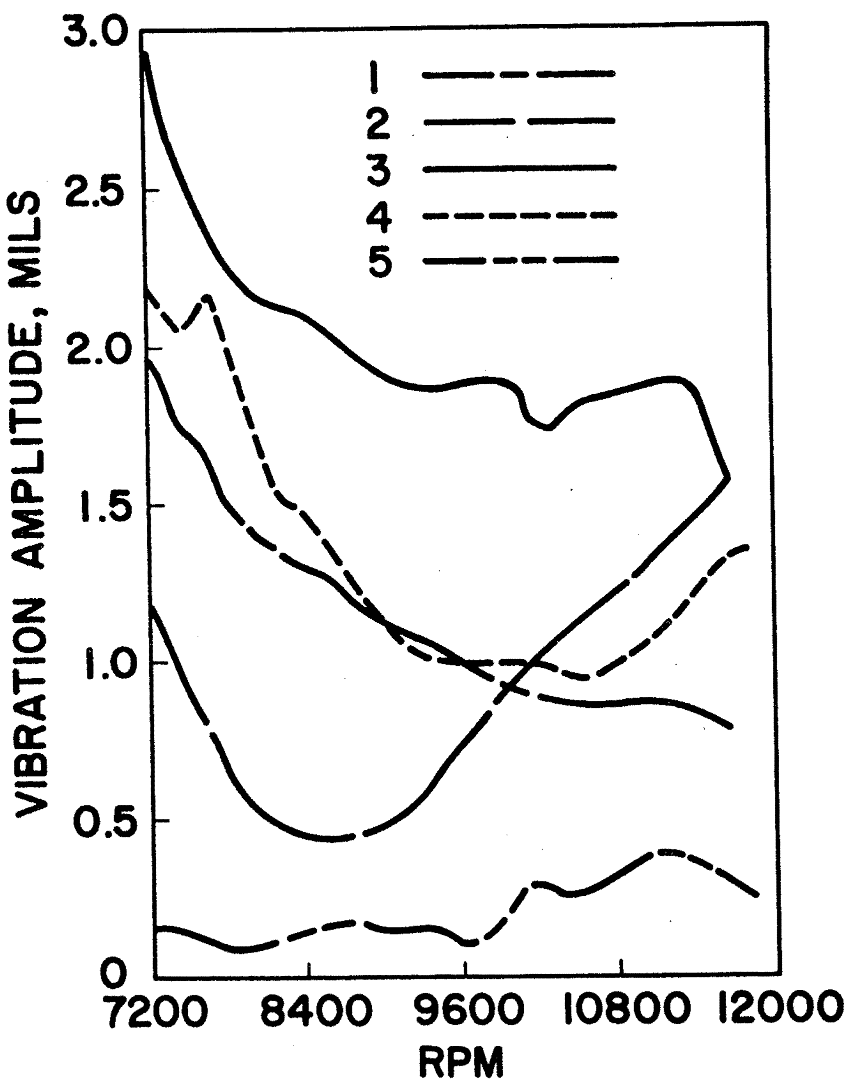


FIG. 5 NON REPEATABLE VIBRATIONS OF TURBINE

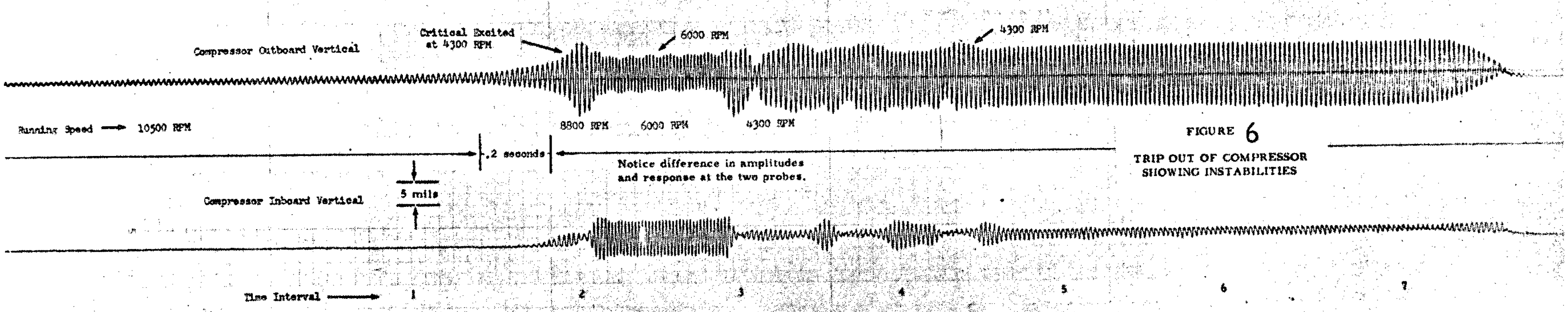


FIGURE 6

TRIP OUT OF COMPRESSOR SHOWING INSTABILITIES

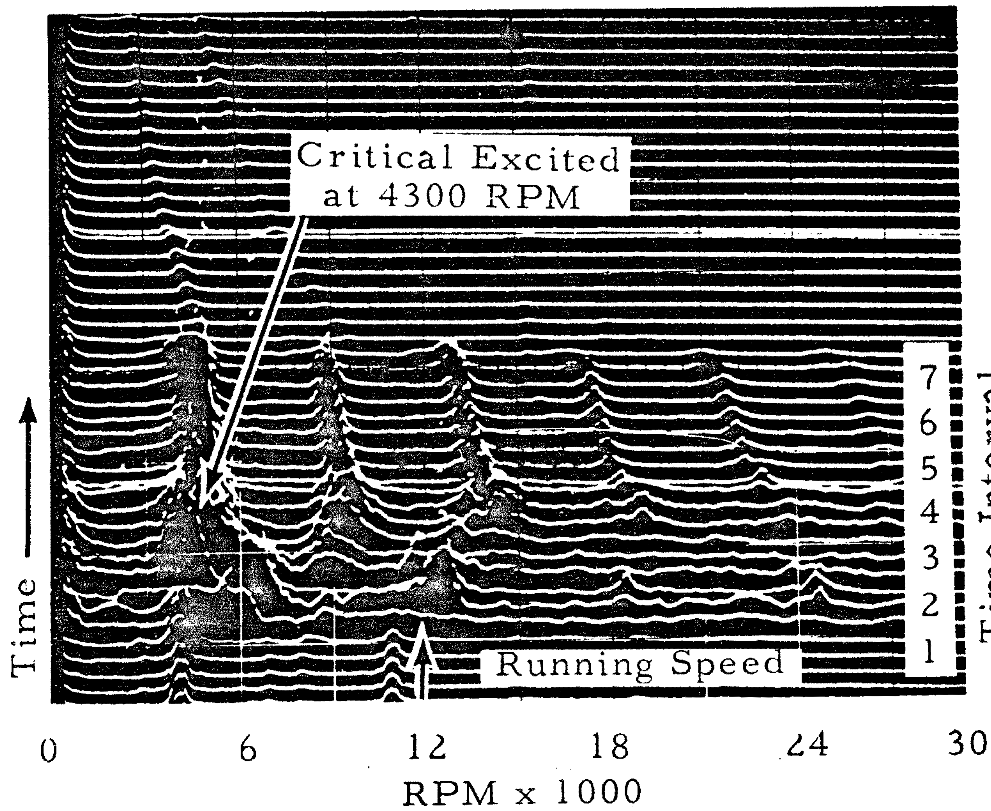


**FIGURE 7**

SPECTRAL TIME HISTORY OF COMPRESSOR
TRIP-OUT SHOWING INSTABILITIES

10 mils/div, 0.16 sec/line

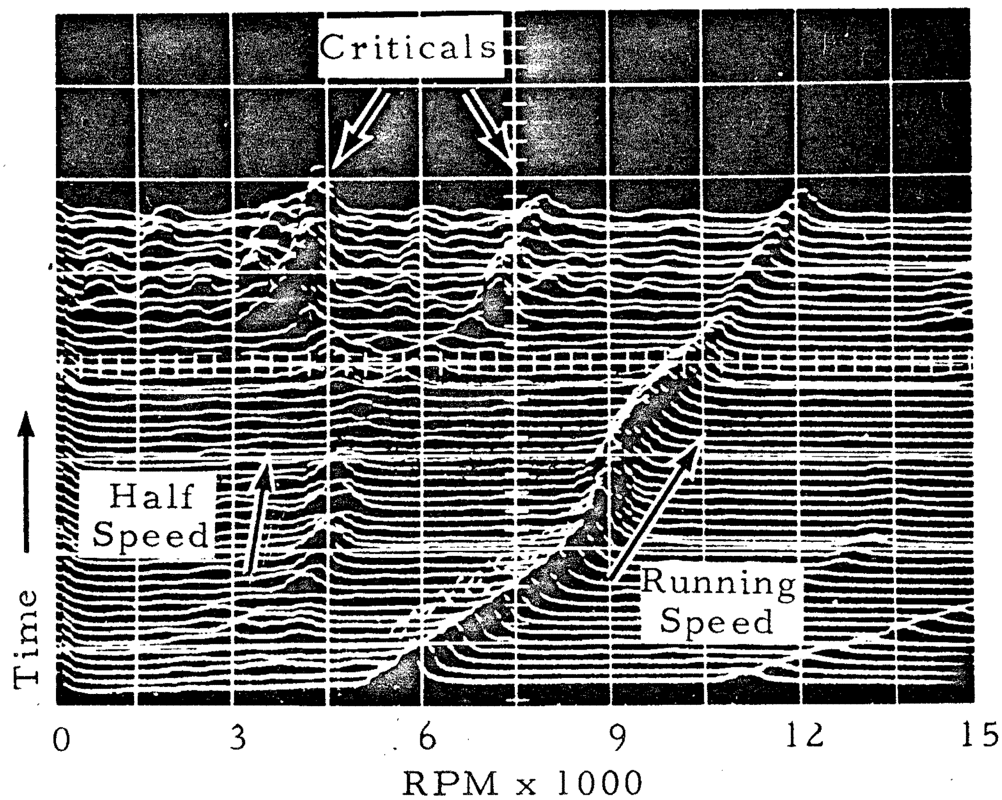


FIGURE 8

SPECTRAL TIME HISTORY OF TURBINE
SHOWING SUBHARMONIC VIBRATIONS
IN NORMAL SPEED RANGE

4 mils/div, 0. 64 sec/line

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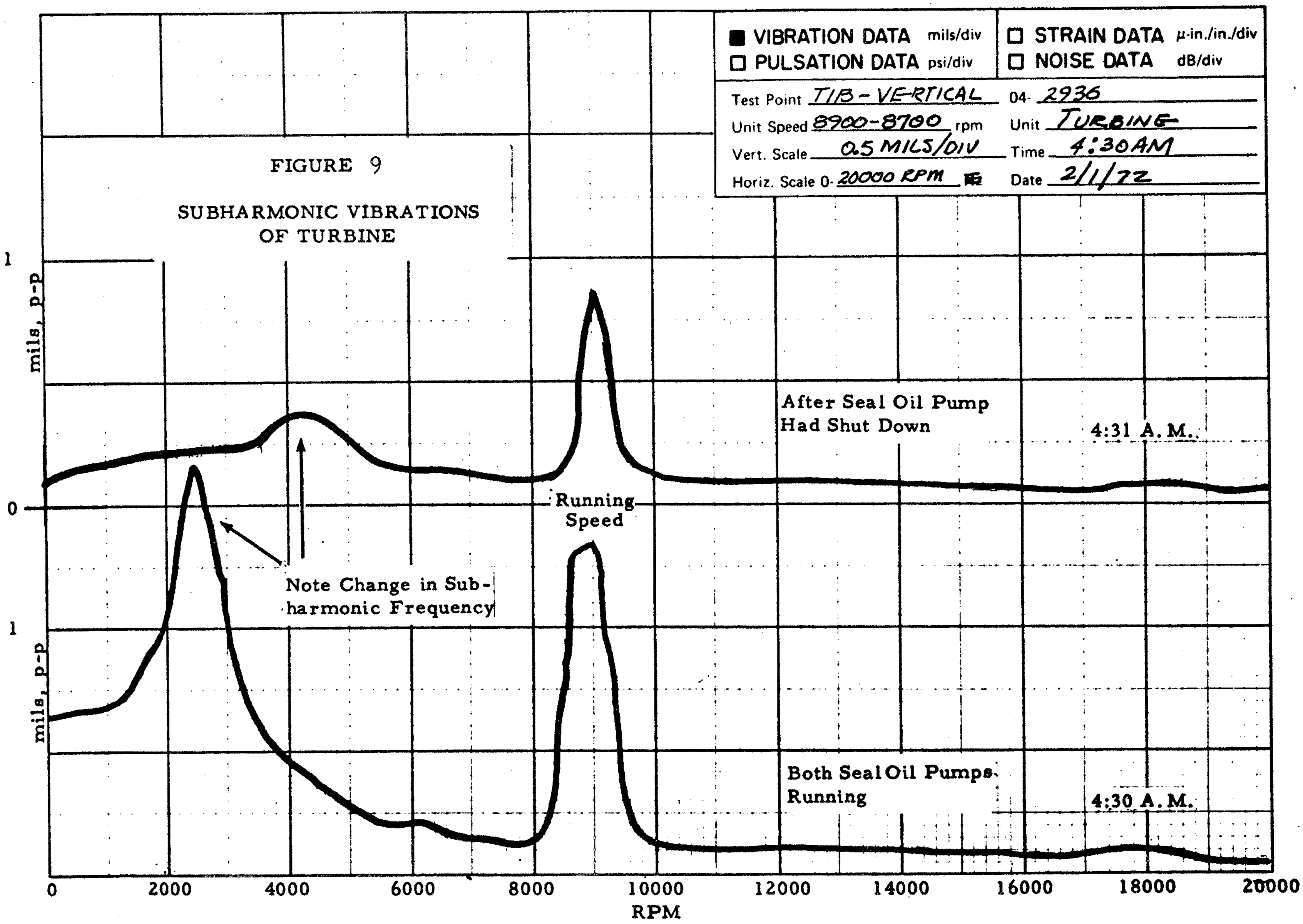


FIGURE 9

SUBHARMONIC VIBRATIONS OF TURBINE

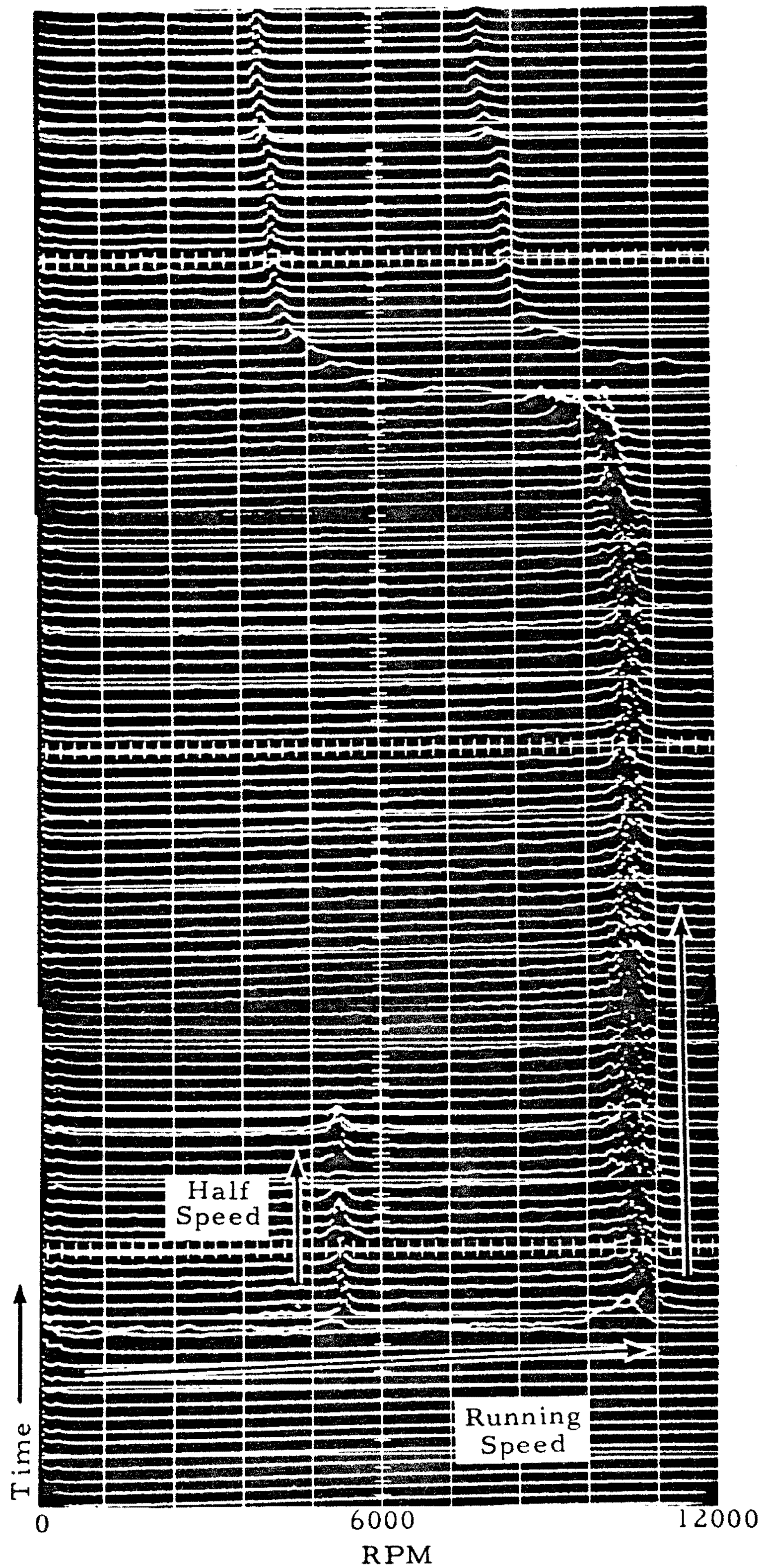


FIGURE 10

FAST RESTART OF COMPRESSOR
4 mils/div, 1.28 sec/line

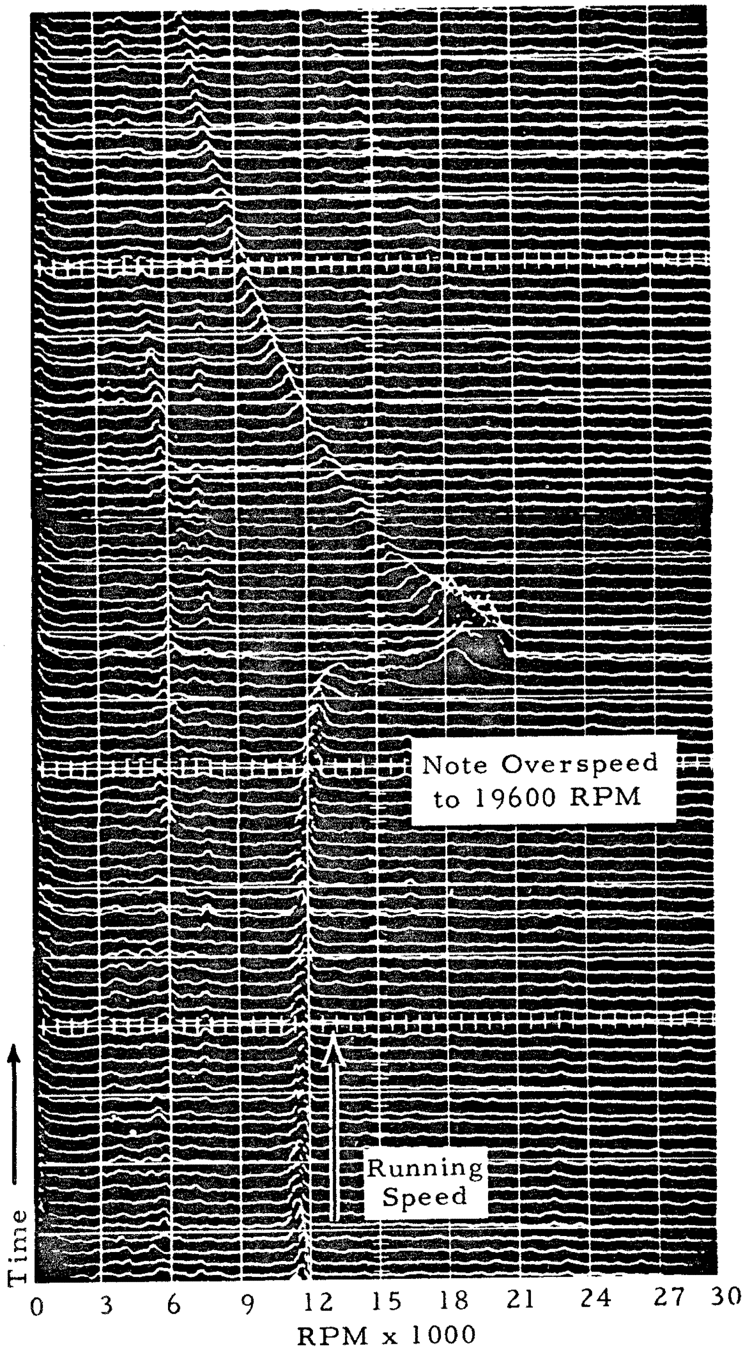


FIGURE 11

SPECTRAL TIME HISTORY OF SPEED
OVERSHOOT ON TRIP OUT OF TURBINE

1 mil/div, 0.16 sec/line

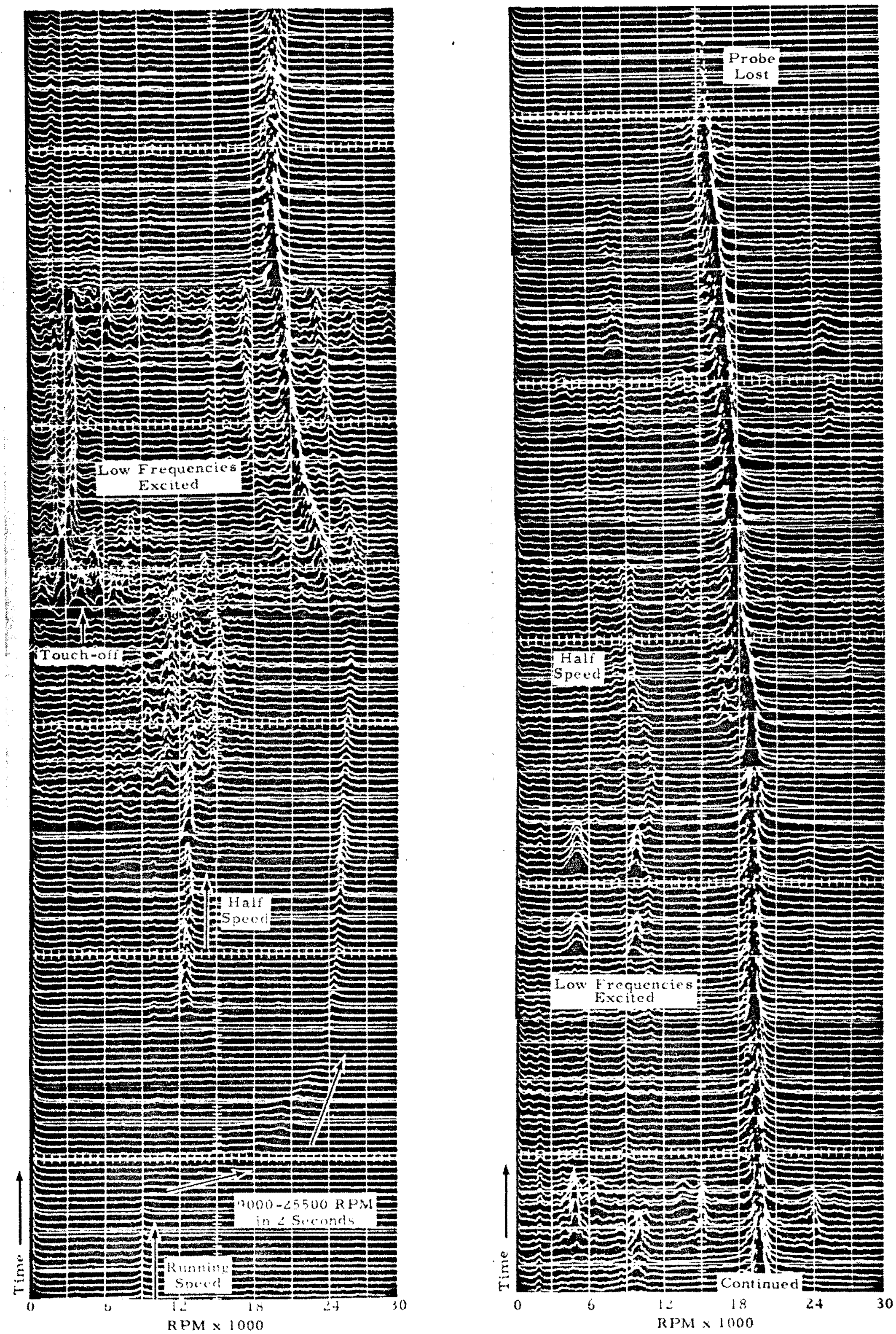


FIGURE 12

SPECTRAL TIME HISTORY OF OVERSPEED FAILURE

4 mils/div, 0.08 sec/line