Turbine and Compressor Vibrations

Additional vibration testing and monitoring procedures for shop tests and plant startups are needed because the state-of-the-art of analytical techniques are not sufficient to accurately predict subharmonic equipment vibrations.

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The use of high speed turbine and compressor strings in chemical process plants has 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 article will present vibration data measured on a steam turbine and syn gas compressor rated at 10,600 rev./min. No problems arose during the shop tests of these machines; 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) starup 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.

Description of Equipment

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

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

The response of the turbine and compressor rotors to unbalance was calculated to determine their sensitivity to unbalance magnitude and location. As pointed out elsewhere *(1)*, the only ways the measured 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,

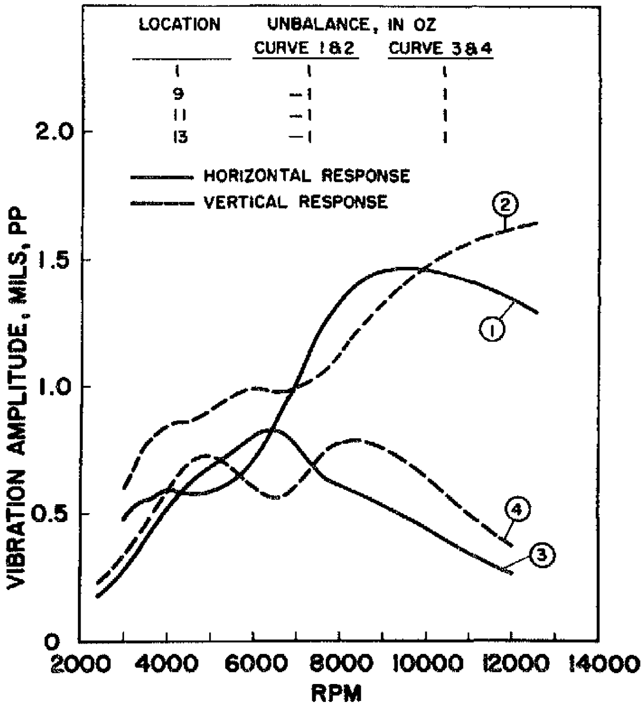

Figure 1. Predicted vibration response of turbine rotor.

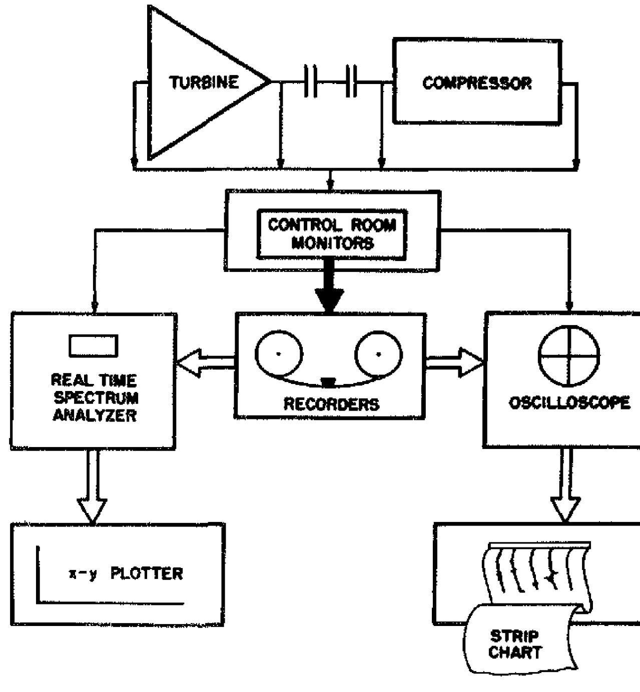

Figure 2. Data points and instrumentation schematic.

6,900 rev./min.

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 requires at least two probes 90° apart 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 to require installation of the spare rotor. New bearings were installed at this time. 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 rev./min. This curve gives a complete response plot which defines the critical speeds. One was at 4,320 rev./min. and another at 6,200 rev./min. This is similar to the response predicted in Figure 1, curve 2.

When the turbine speed reached 11,580 rev./min., 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

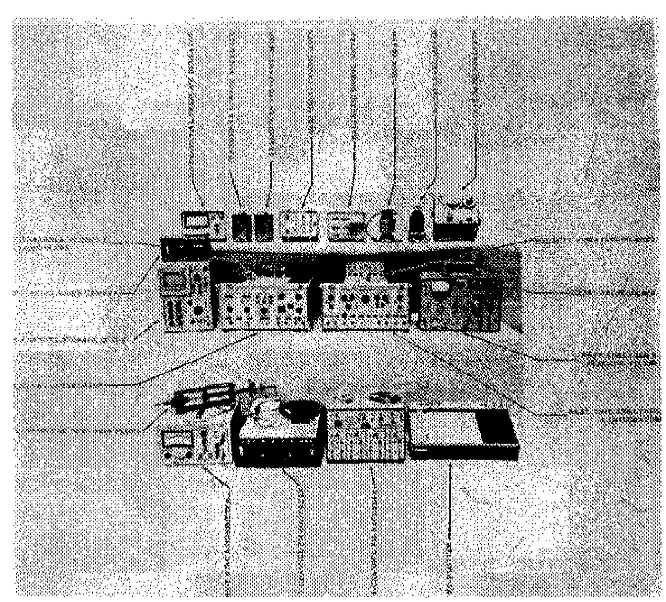

Figure 3. Instruments used in signature analyses.

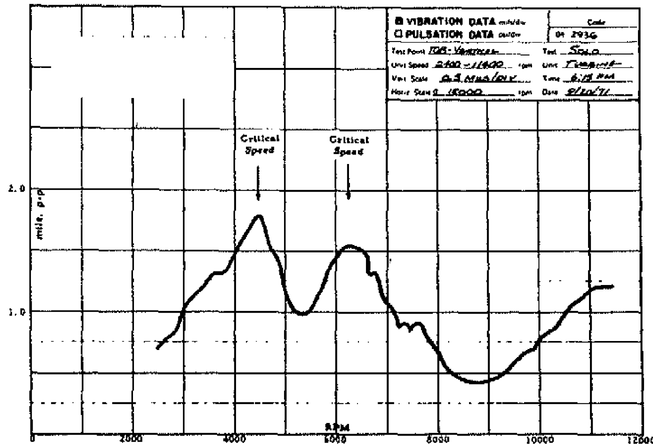

Figure 4. Measured vibration response of turbine.

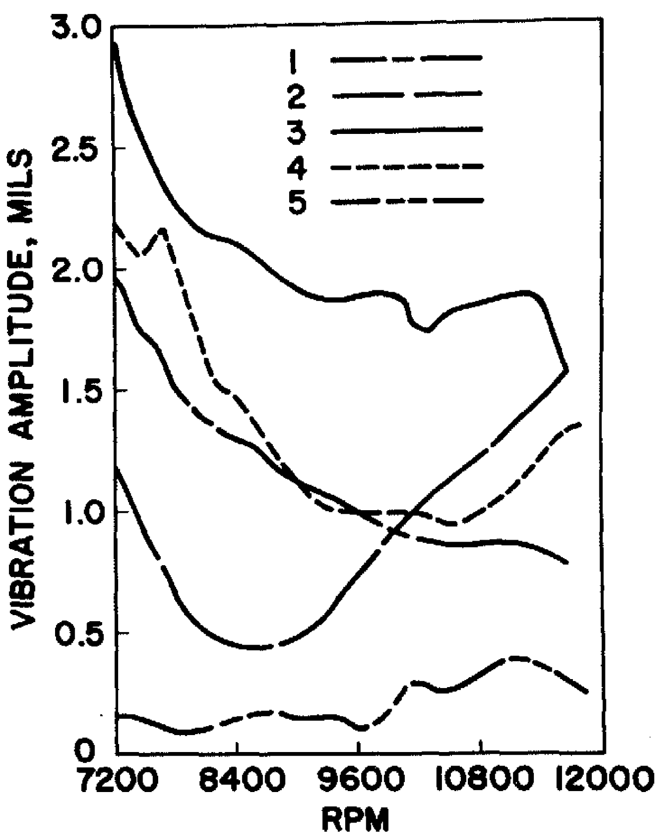

Figure 5. Nonrepeatable vibrations of turbine.

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 3,000- to 11,000 rev./min. 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 9,500 rev./min., the vibration amplitudes were less than 1 mil. due to the shifting of the sleeves 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 9,600 rev./min. 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 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 4,500 rev./min., 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

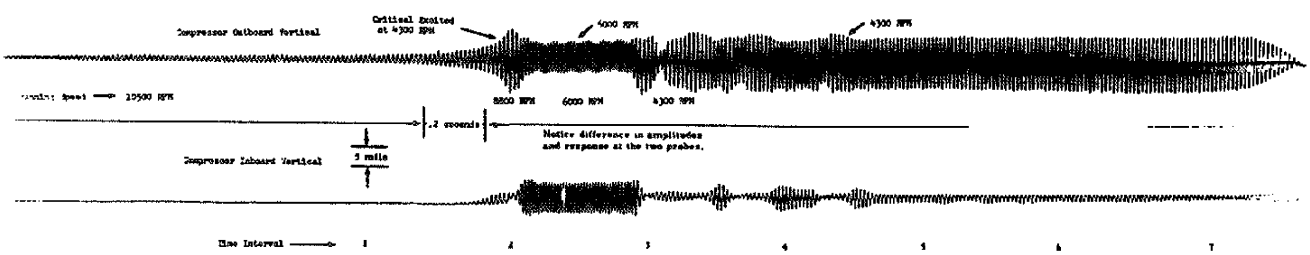

Figure 6. Trip-out of compressor showing instabilities.

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 4,000 rev./min. increased in amplitude (approximately 16 mils) until the bearing clearances were exceeded. The rotor then vibrated primarily at 6,200 cycles/min. for approximately 0.5 sec. with an amplitude of 8 mils. The speed then locked onto the first critical at 4,500 rev./min. 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 9,000 rev./min., apparently could have been 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.

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; the clearances in the labyrinths and seals were increased; and the five shoe tilted pad bearings were 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 article.

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 discussed earlier. During

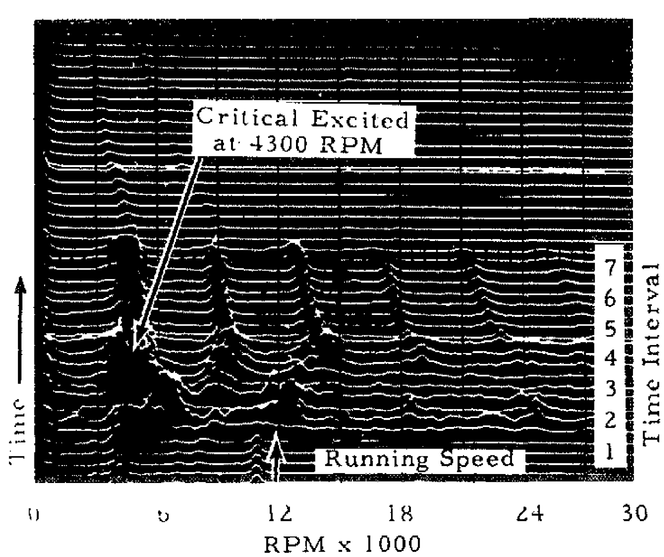

Figure 7. Spectral time history of compressor trip-out showing instabilities.

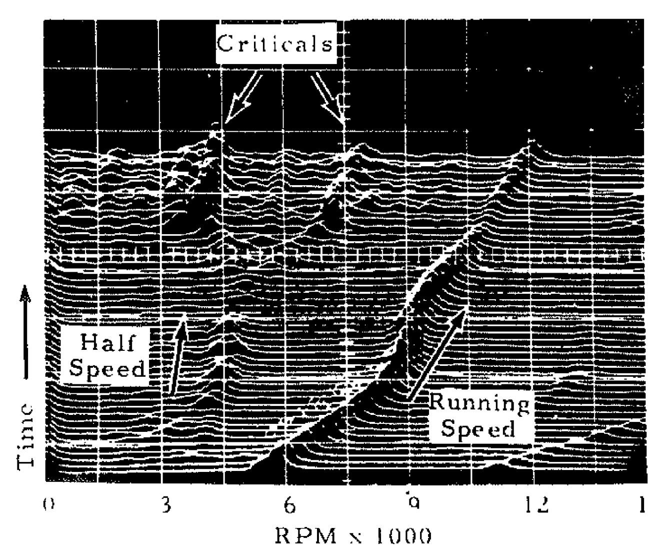

Figure 8. Spectral time history of turbine showing subharmonic vibrations in normal speed range.

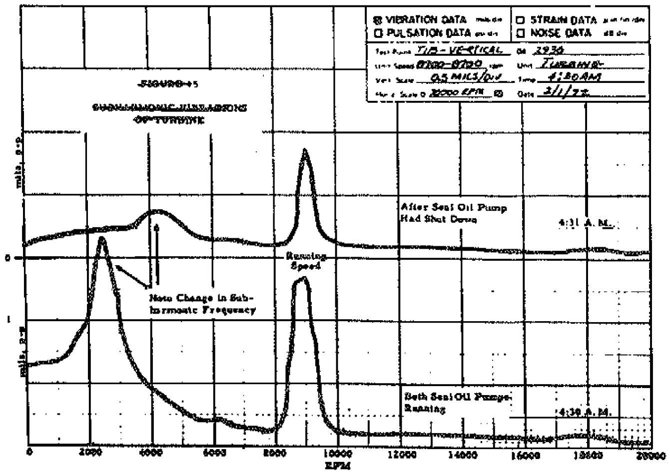

Figure 9. Subharmonic vibrations of turbine.

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 4,800- and 7,000 cycles/min. and a component at approximately one-half speed were excited.

Figure 9 shows that a larger amplitude component at approximately 0.3 times the running speed occurred at 2,500 cycles/min. when the speed was 9,000 rev./min. When the speed suddenly dropped 200 rev./min. the instantaneous frequency analysis, Figure 9, shows that this moved the subharmonic component from 2,500- to 4,400 cycles/min.

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.

During a solo warm-up, the turbine speed unexpectedly increased from approximately 3,000- to 15,500 rev./min. in less than a second. Some adjustments were made and the turbine was again soloed and successfully checked out 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 rev./min. 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 rev./min. in 2 sec., 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 rev./min. 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 rev./min. Upon trip-out, the speed jumped to 19,600 rev./min. instantaneously and then dropped, Figure 11. Due to the response time of the digital tachometer in the control room it did not indicate any overspeed.

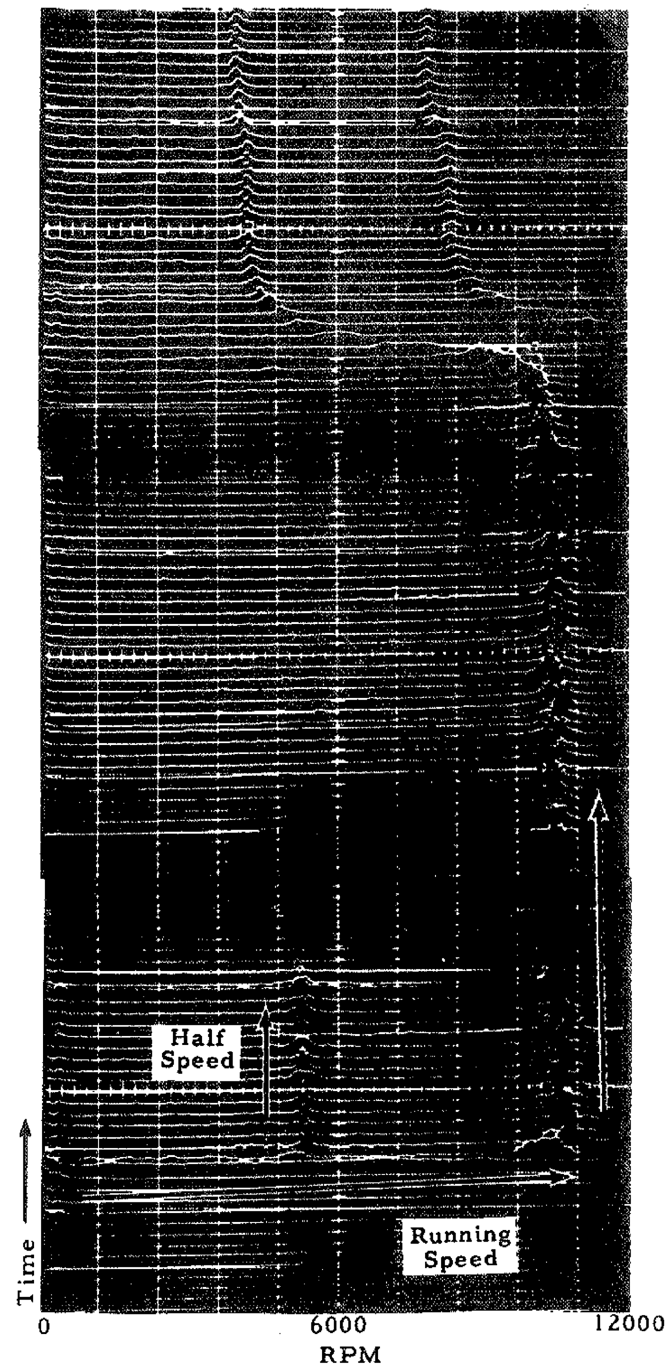

Figure 10. Fast restart of compressor.

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 rev./min. in

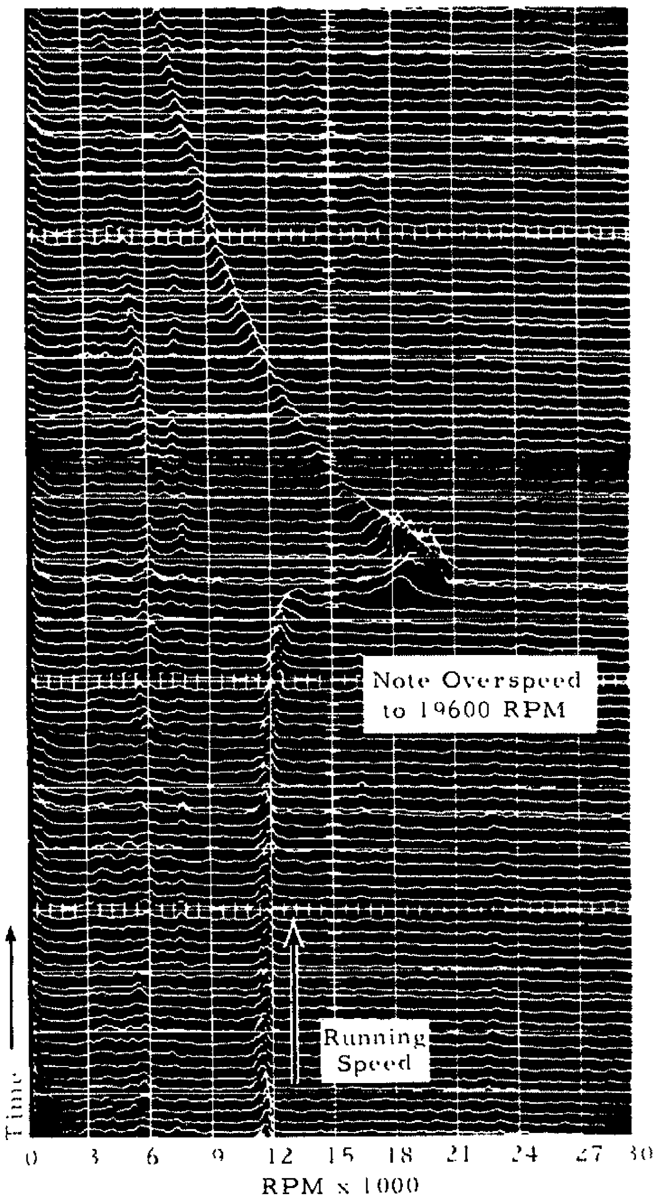

Figure 11. Spectral time history of speed overshoot on trip-out of turbine.

approximately 0.4 sec. It was then shut down.

After rechecking all the governor speed controls, the turbine was brought up to 9,000 rev./min. and vibrations were less than 0.5 mil overall. Suddenly, without warning, the speed jumped from 9,000 to 25,500 rev./min. in less than 2 sec. The overspeed trip did not work and efforts to shut down the machine in the control room were unsuccessful. The thrust collar was finally 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 amplitude at 25,500 rev./min. were less than 2 mils for about 3 sec.; however, at approximately one-half speed, 12,750 cycles/min., 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 rev./min. and the amplitudes were greater than 12 mils. The speed decreased to 16,000 rev./min. where the amplitude reached 14 mils, at which time the probe was lost. All these events took place in approximately 30 sec.

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 article 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 subharmonic vibrations.

The signature analysis techniques presented here can be used to identify these and other types of vibration problems. The use of these techniques during plant startups is desirable because:

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

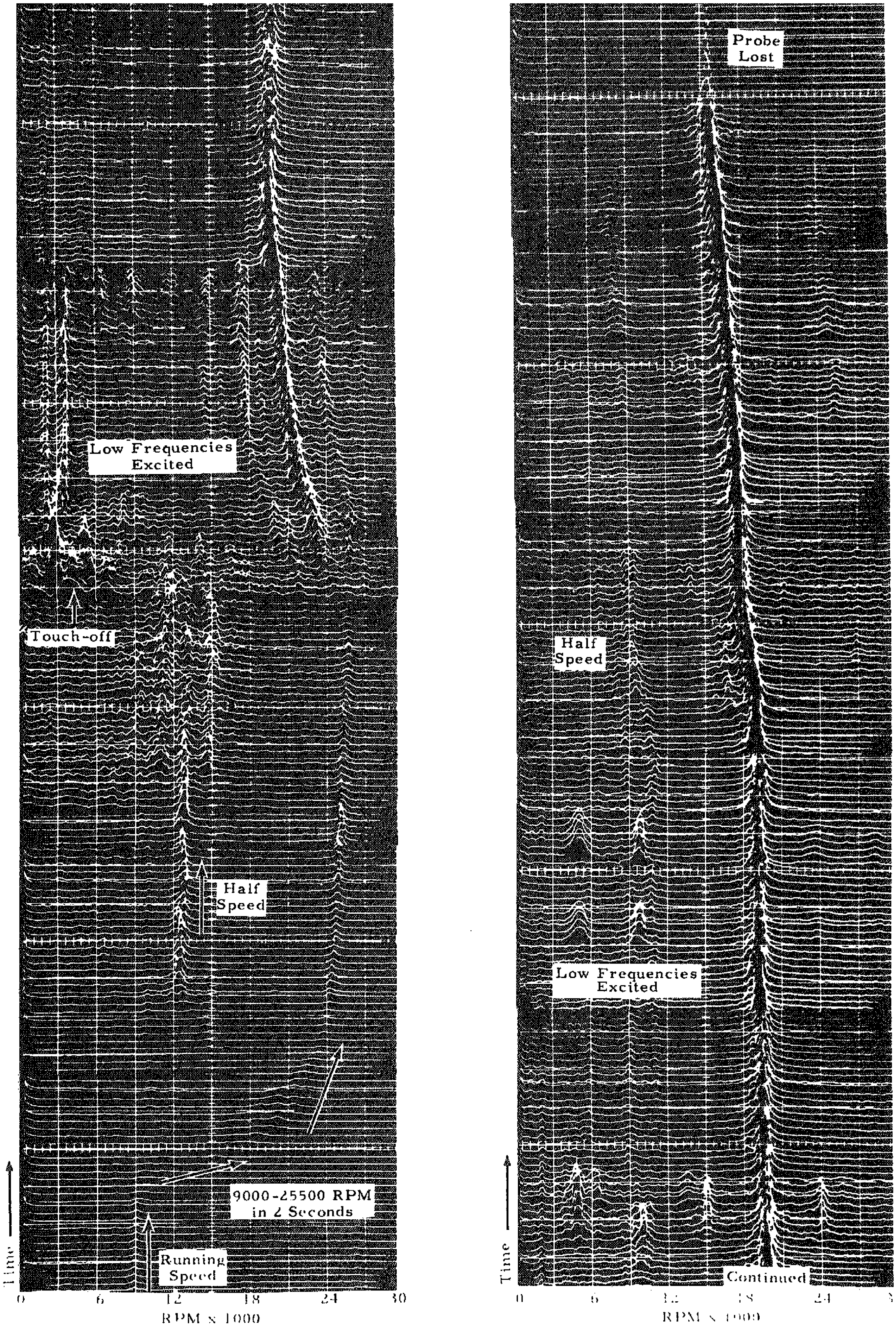

Figure 12. Spectral time history of turbine overspeed failure.

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 neeeded.

Literature Cited

1. Nimitz, Walter, and J.C. Wachel, "Vibrations in Centrifugal Compressors and Turbines," ASME Paper No. 70-PET-25.
2. Gunter, E.J., Jr., "Dynamic Stability of Rotor-Bearing Systems," NASA SP 113 (1966).
3. Gunter, E.J., Jr., and P.R. Trumpler, "The Influence of Internal Friction on the Stability of High Speed Rotors with Anisotropic Supports," ASME Paper No. 69-Vib-2 (1969).
4. Tondl, Ales, "Some Problems of Rotor Dynamics," Chapman and Hall, London (1965).
5. Ehrich, F.F., "Identification and Avoidance of Instabilities and Self Excited Vibrations in Rotating Machinery," ASME Paper No. 72-De-21 (1972).

DISCUSSION

QUESTION: Could you give us your comments on your success in detecting cracked blades and also your comments on using vibration proximity probes for hot and cold alignment checks.

WACHEL: We have had some experience with installing proximity probes in the stator near the blade tips in a power turbine, and actually displaying on a scope the distance to every one of the blades. With the proper correlation techniques you could actually pick out each individual blade and tell the height of it. By installing several probes one should be able to depict blade vibrations. Accelerometers mounted on the bearing have been used successfully to determine excessive blade vibration.

One has to use a practical approach as to the effect of alignment on vibrations. It seems to me that it would be better to slightly misalign a machine and study the effect on vibration rather than say that it's lined up properly; therefore, alignment can not be the problem. You can use proximity probes, taking the static DC level, to determine how the shaft moves or you can measure how the cases move and determine the misalignment from these measurements. Optical measurement and laser techniques are good; however, the problem is that, in many installations, you can't find a quiet stop to set them up on. So this remains a problem.

ANON: I was just curious. You talk about amplitude and frequency measurements. At our plants we use velocity. Wouldn't that be a little easier to use?

WACHEL: Well, it depends. If you have a system which only has vibrations at the running speed, then it doesn't matter whether you use acceleration, velocity, or displacement. Now velocity has a lot to say for it, because as you know, the overall acceptable level is constant as a function of frequency. A value of 0.5 inches per second is a typical quoted standard. In high speed machines where you have small clearances, we have found that in troubleshooting you're not really interested in the total velocity indication, but what the amplitude and frequencies are and how they vary as system parameters are changed. If you have a machine that you have already analyzed and you know it doesn't have any subharmonics or multiples of running speed, then you can set up a realistic velocity measurement criterion. Measurements can be made on the bearing housing and this will be adequate because if it should change, you still have to find out why it changed.

As an example, on a recent problem the vibration specification stated that if the vibration velocity was above 0.5 in/sec, the unit had to be shut down and repaired. This unit had to be repaired four times in one month, and nothing could be found wrong. The velocity criterion was exceeded because of a low frequency component that was transmitting through the foundation and piping. These vibrations were not detrimental to the machine, but it was there in the measurements and it had changed since they had originally installed the machine. This is one of the advantages of establishing baseline signatures or baseline data so that you can determine the changes in the system if vibrations increase.

DICK DAZE, M.W. Kellogg: You've emphasized taking signature traces of the vibration patterns in the rotating train and locating the horizontal and vertical probes so you can take you lateral displacement measurements. Have you done anything taking tortional traces—tortional signature traces. Have you found any relation between tortional and shaft lateral vibrations?

WACHEL: Yes, we have done a lot of measurement of torsional vibrations in the field. What we find is that the analytical techniques for calculating torsional natural frequencies and mode shapes are nearly 99% reliable. If you have the right information to feed into your computer program, you can calculate the natural frequencies within the accuracy needed for engineering work. In some cases where you're tying in reciprocating compressors with centrifugals, you can get pulsations that can couple in and cause failures in the centrifugal shafts. For that reason, we do a lot of measurements for acoustical resonances in centrifugal systems. We have several torsiographs that can be installed on stub shafts to accurately measure the torsional natural frequencies.

D.E. CLAPPER, American Cyanamid Co: I believe all of us here are familiar with the vibration guideline charts, which show whether your machine's running smooth, fair, rough, very rough. My question in regard to this, since people like to be told whether the machine is running rough, smooth, or fair; if you have a complete vibration analysis at various frequencies sometimes significant vibration may be measured at double or triple the running frequency, which one of these tests would you use in referring to guideline charts to say whether the machine is rough or smooth and there is quite a deviation in amplitudes at these various frequencies?

WACHEL: The curves you refer to such as given by Rathbone were useful in establishing general guidelines for the conditions of the lower speed units but I feel that one should use such criteria with caution. By this, I mean that one should realize their basis. These curves are generally based on bearing housing vibration and assume that the shaft vibrations are two times that amplitude. This ratio can be as high as 20:1 for high speed compressors and turbines. Also, the curves are based on the assumption that the only component of vibration is at one times running speed caused by unbalance only. In most problem cases, these assumptions are not valid; therefore, one should not use these curves. If baseline information is obtained, reasonable criteria can be established for each particular machine. #