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TURBINE INSTABILITIES - CASE HISTORIES

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Several possible causes of turbine rotor instability are discussed and the related design features of a wide range of turbomachinery types and sizes are considered. The instrumentation options available for detecting rotor instability and assessing its severity are discussed.

INTRODUCTION

Turbines of course are built in many forms, from very large multi-cylinder power generation units, through process/feed pump drives to the smaller specialized designs using process fluid in an integrated expander/compressor configuration. The causes of rotor instability in turbines, whether they are steam or gas driven, relate closely to the causes of instability in compressors and pumps. In recent years, instability problems with high pressure compressors have been perhaps more prominent than with turbines - light rotors, high rotational speeds and high stage pressure ratios inevitably result in a rotor system which is really fundamentally unstable in the presence of very significant fluid derived forces. Only the adoption of bearing designs with strong stabilizing characteristics and seal designs with resistance to the development of destabilizing static and dynamic forces can ensure the stable operation of many high pressure compressor designs; the position is similar with turbines.

Arguably, it was the electricity generation industry which pushed forward the design of large rotating machinery, particularly turbines and high pressure pumps. Paradoxically, in power generation, rotor speeds are effectively limited to 3000/3600 rpm by system frequencies of 50/60 Hz; gear-box transmission of 660 MW is absolutely not economically desirable. In the petrochemical process industries, attention has centered on improved efficiency and reliability (common to both industries), but in turbine design there has been freedom to exploit the use of increased rotor speeds, as well as elevated steam conditions, to achieve improved efficiency.

Rotor instability problems have occurred therefore in a somewhat cyclical manner as technological progress occurred in the thermodynamic sense; in many ways the design of steam turbines in the power generation and petrochemical process industries has converged. In two important respects however, the steam turbines employed in these industrial environments still differ - partial admission of steam is a necessity in most process-drive steam turbines, but is a rarely used technique in power generation turbines, and fortunately not usually necessary (except in feed pump drives) since the size of the machine permits the use of full admission with reasonable nozzle areas. Coupling between multi-cylinder turbines and to the driven machine is universally solid or rigid in the power generation industry, almost universally flexible in the process industries. The effect of coupling flexibility is profound in terms of coupled rotor dynamic behavior; from the point of view of turbine stability, both of these design differences have a large influence on the potential for instability in the turbine and driven machine.

Several years ago, the most common type of rotor instability experienced on turbines was associated with bearing oil film instability resulting in "oil whirl" or "oil whip." Very often the cause was relatively simple - turbine designers tended to be more concerned about ensuring that the bearings used were unlikely to suffer from premature failure due to overloading, either due to dynamic stresses leading to fatigue of the babbit metal, or wipes when oil film thickness was minimal at low rotational speeds.

Solidly coupled power-generation type turbines (and their associated generators) were quite commonly affected by this problem some twenty or so years ago. Ensuring that a bearing load at least approximating to the natural proportion of the rotor weight associated with the bearing location was maintained was obviously a problem, since changes in alignment could easily deload a bearing during normal operation.

Nevertheless, the bearings used in large turbines are rarely more sophisticated than fixed geometry, elliptical bore designs, perhaps with central grooving and sometimes with a tri-lobal geometry. Use of variable geometry bearings, typically of four or five shoe design is very unusual, indeed is positively avoided. The disadvantages of initial cost, high losses in operation and risk of damage during slow turning (provision of jacking oil is difficult) are very significant when very large, heavy rotors are involved. Insufficient system damping at the critical speed(s) may also be a problem.

Even on relatively small process drive turbines, the amplification factor at the first flexural critical speed is often quite high when tilt-shoe bearings are employed. Although this is largely due to the characteristics of the bearing, it may also be due to a relatively poor state of balance. Process-drive turbine rotors are often balanced in low-speed balancing machines or balanced at high speed using a multiplane technique optimized for good running speed vibration levels. Very accurate correction of balance at the critical speed(s) may require the application of a true "modal" technique, widely practiced by the manufacturers of large turbines and generators, simply because the inherent balance defects present in new rotors demand balancing at each critical speed in order to run the rotor up to its normal operating speed. Even with the application of good balancing techniques, the trade between available damping at critical speeds and inherent stability at normal operating speed is a very significant one. Ironically, the achievement of a truly excellent state of balance manifested as near zero levels of synchronous (1X) vibration amplitudes at the rotor bearings can lead to instability of the rotor due to oil whirl or whip; quite understandably, in the absence of any distinct 1X forcing function, it is easier for the rotor system to cross the stability/instability threshold if the margin of stability is small.

1. MEASUREMENT OF TURBINE VIBRATION - DETECTION OF ROTOR INSTABILITY

Unstable rotor operation, generally indicated by the presence of nonsynchronous vibration behavior, is normally interpreted as a malfunction or fault condition. Often, the additive effect of synchronous or 1X vibration due to residual unbalance and nonsynchronous (typically subsynchronous) vibration leads to overall vibration levels in excess of agreed acceptance or tolerance limits whether during initial customer acceptance tests or later in commercial operation. Existing manufacturing standards, etc. generally state a maximum level of rotor relative or casing absolute vibration at synchronous (1X) frequencies and at nonsynchronous frequencies, but little guidance is offered as to the real significance or potential for damage of vibration related to instability of the rotor system. This leads, in the author's experience, to two fundamentally different problems. On one hand, sometimes undue importance is attributed to well controlled vibration resulting from a suppressed level of instability, while on the other hand, potentially or actually serious instability may be underestimated or even ignored.

The vibration measurement technique used on a particular machine may significantly affect the user's perception of the severity of an instability condition. The inherent frequency dependent relationship between vibration displacement, velocity, and acceleration is very well known and understood; clearly, subsynchronous vibration of a rotor will be measured on a one-for-one basis, compared with other frequencies if expressed in displacement terms, but will be measured in velocity terms as an amplitude proportional to frequency.

In acceleration units, the relationship with displacement is proportional to frequency squared; consequently, low frequency vibration due to instability, typified by "oil whip" or re-excitation of the rotor's first critical speed will be measured in acceleration terms as an apparently insignificant vibration level - a potentially dangerous situation when very flexible rotors are being considered. Some vibration monitoring systems fitted to turbines may actually conceal the presence of unstable rotor vibration completely.

There is clearly a need to do two things. One, ensure that the vibration monitoring technique used is capable of measuring all relevant vibration behavior of the rotor. Two, use a sensible interpretation technique which neither overstates nor understates the importance of any measured unstable rotor vibration. In many respects, it is more important to assess how much margin of stability a turbine system possesses, so that in conditions of unusually severe operation it may be expected to remain stable, rather than simply documenting unstable operation of a gross nature. Although the solution to the latter problem may not be so easy, the vibration levels experienced on turbines suffering from severe instability are often so unsatisfactory that an immediate solution to the problem is mandatory.

Quite often, a change in the instrumentation system used for vibration measurement on a particular turbine, for instance, after a retrofit, will reveal an unstable condition, perhaps several years after the machine was commissioned. If the turbine has not been a problem, is it justifiable to be worried? In some cases the answer is quite definitely no; in others, an investigation of the turbine's operating history may reveal that certain problems have occurred which can be attributed retrospectively to the continued or occasional presence of unstable rotor vibration.

2. CASE HISTORIES

Bearing oil film instability, leading to "oil whip" on a 3,000 rpm double-end drive gas turbine-driven generator set, produced unacceptably high rotor vibration at a frequency of 800 c.p.m., corresponding to the first flexural critical speed of the generator rotor. The unit was intended for rapid start-up and loading for peak-lopping and system support during emergencies; near 100% reliability of synchronization was an essential requirement.

Instability, when it occurred, developed during start-up at about 2,000 rpm; the standard vibration monitoring instrumentation was a single velocity seismoprobe at each end of the machine train. Even when rotor vibration amplitudes due to "oil whip" were virtually as large as the bearing clearances, high bearing vibrations were not indicated, due to the poor frequency response of the velocity transducer and the high bearing pedestal stiffness relative to the oil film stiffness. This misleading situation was discovered when eddy current proximity probes were installed for analysis purposes.

The bearing oil film instability and resultant rotor "whip" were not of particularly unique character; however, the solution of the problem through the adoption of improved bearing design was made unusually difficult by the design of the generator rotor, its bearing locations and the use of extremely heavy clutch couplings overhung at the generator rotor ends.

Due to the overhung masses, the first flexural critical mode shape resulted in nodal points very close to the bearing centerlines at both ends of the generator rotor. The bearings were therefore relatively ineffective in terms of controlling the rotor response, either at the first critical speed during start-up or shutdown, or whenever re-excitation of the first critical occùrred. The adoption of tilting-shoe bearings, which would have solved the oil film instability problem at normal operating speed, would have resulted in totally unacceptable rotor vibrations when passing through the first critical speed due to the inevitable reduction in system damping which would have resulted. Very precise rotor balancing was already necessary to obtain an acceptable maximum level of vibration at the first critical speed and small, but significant, thermally induced bow of the rotor was present so that sufficiently good effective balance could not be maintained at all times.

This case history illustrates the importance of ensuring that the bearings are located as far away from nodal points as possible; coupling overhang effects are influential on virtually all turbines, indeed on rotating machinery of all types, and will often influence the position of nodal points sufficiently to make the solution of rotor instability problems particularly difficult.

The second case history refers to a typical modern process-drive turbine of single casing design using relatively high steam inlet pressure. Considering the horsepower developed and the "energy density" within the expansion, the rotor weight was small; tilting-shoe journal bearings of conventional design were employed, indeed were necessary, at the operating speed range of about 5,000-12,000 rpm.

It was found that during normal operation, when synchronous (1X) rotor vibrations were very acceptable, small changes in process conditions could initiate high vibration levels on the inlet and exhaust end bearing journals, leading to frequent trips of the unit. The standard vibration monitoring instrumentation was eddy current proximity probes at each bearing in conventional X-Y configuration.

On-site vibration analysis showed that the problem was one of re-excitation of the first flexural critical speed of the turbine rotor, leading to a "whip" type response, illustrated in Figure 1. Oil film instability was not suspected. Analysis of the rotor position within the bearing clearance showed that, particularly at the inlet end, the rotor was being lifted into the top half of the bearing (see Figure 2). Steam induced lifting/perturbating forces were considered to be responsible, but were difficult to quantify in a field situation.

The subsynchronous vibration frequency corresponded closely to the first flexural critical speed of the rotor, confirmed during start-up/shutdown tests. A temporary modification was installed by the turbine manufacturers, effected by reducing the mean bearing clearance in the inlet and exhaust end bearings by shimming behind the tilting shoes in the top half of each bearing bush. This was very successful in suppressing the subsynchronous vibration so that, even under the worst steam inlet/extraction pressure transients, the rotor re-excitation was below 0.25 mils peak-peak (see Figure 3). At a later date, modified bearings with increased shoe stiffness and preload were installed, leading to a further reduction in subsynchronous vibration amplitudes. Although steam-induced forces within the turbine continued to lift the rotor journals into an abnormally high position in the (reduced) bearing clearance, particularly at the inlet end, the modification was considered to be a completely acceptable method of overcoming the problem.

Low amplitude subsynchronous vibration response can be detected on many turbines of virtually every manufacture. Figure 4 illustrates the point; this turbine drive operated in the long term without any failure. Although complete elimination of subsynchronous vibration is of course a desirable objective, it should be remembered that the cost and inconvenience incurred may not be justifiable in every case.

Case history number three documents an unusual instability found on a relatively old Ljungstrom type radial-flow steam turbine generator which had been in service for many years without suffering the problem; machinery of this type rarely exhibits unacceptable vibration, except of course when rotor balance is poor. It was found that at loads above 4.0 MW (the unit was rated at 6.25 MW), casing vibration levels, particularly at the exciter end (see Figure 5) increased rapidly as the load increased. At about 4.8 MW the measured casing vibrations were totally unacceptable (above 30 mm/sec peak velocity). Although casing vibration measurements clearly indicated that a problem existed, even very detailed measurements could not identify the source of excitation of the machine; it was possible only to determine that, when high vibration developed, the main component of casing excitation was at 3700 cpm (61.67 Hz) at the normal operating frequency of 3000 cpm (50 Hz system). Below 4.0 MW load the super-synchronous frequency was completely absent. Eddy current displacement probes were temporarily installed on the machine at the four main bearing locations and at the exciter bearing; integrated velocity transducer measurements were also taken at the exciter bearing housing to permit, by electronic summation, exciter rotor absolute vibration levels to be determined.

It was found that, when the increasing super-synchronous vibration developed, the rotor response was greater than the bearing housing at the exciter end, although during start-up the structure responded strongly at the same frequency when 2X rotor vibration frequency corresponded to the 3700 cpm (61.67 Hz) frequency.

Clearly a structural response of the turbine generator and a strong rotor response were present at the 3700 cpm (61.67 Hz) frequency, capable of being excited by 2X rotor vibration during start-up and also, when on load, at a super-synchronous frequency on the rotor (see Figures 6 and 7).

However, study of the rotor position, particularly at the exciter bearing showed that when the unit was operating at 3000 rpm, the exciter bearing was totally deloaded, with the journal virtually at the center of the bearing clearance (see Figure 8). In this condition, the exciter rotor was clearly unsupported at the outboard end and was effectively a cantilevered overhung mass which would be expected to considerably change the response characteristics of the rotor system. At a later date the exciter bearing was raised to retain a gravity preload at normal operating speed, allowing for the greater lift of the much larger diameter main generator bearing journal; full load (6.25 MW) was then achieved with acceptable overall vibration levels and negligible super-synchronous vibration components.

The fourth case history deals with two examples of rotor instability found on turbo-expander compressors of the integral double overhung design, although they were of different manufacture. One unit was found to exhibit strong subsynchronous vibration at 0.47X rotor rotational frequency; clearly "half speed" oil whirl of the rotor, which operated well below its first flexural critical speed. Rotor relative vibration was monitored with eddy current probes located at each end of the rotor in "X-Y" configuration. This machine has been in operation for over 5 years without any mechanical failure or process interference through trips. Some reduction in the whirl amplitude was achieved by selecting a less viscous oil, but no opportunity to cure the condition through bearing design change has been permitted; the rotor orbit and run-up cascade spectrum are shown in Figure 9.

Another expander compressor of similar design but smaller size and higher speed was found to suffer from occasional heavy seal rubs. Rotor vibration was measured using a single eddy current probe positioned approximately mid-way between the two bearings. Tests showed that when the oil temperature at bearing inlet was increased from 35°C to 45°C, the normally stable rotor started to go through a cyclical rise and fall in 1X vibration; a second eddy current probe was installed during testing to permit the analysis of the rotor orbit - examples are shown in Figure 10. These measurements also identified that when the vibration at 1X rotational speed increased, the phase angle increased in lag by about 50 degrees. Increasing the oil inlet temperature by 10°C had a significant affect on the viscosity and would therefore change the bearing coefficients. However, the rotor vibration was considered to be occurring in an essentially rigid conical mode; therefore, measurements made at the central region of the rotor were likely to be in the region of a node and potentially much smaller than at the bearings and seals. Changes in bearing oil film parameters due to oil inlet condition changes, especially if the effect was greater on one bearing than the other, were predicted to affect both the frequency of occurrence and node position for such a coaxial mode. Vibration measurements taken on the machine casing were extremely small and increased insignificantly when the rotor vibration levels increased; detailed analysis of the rotor behavior and adequate machine protection would clearly have required installation of shaft-relative probes next to both bearing journals.

CONCLUSIONS

Turbine rotor instability can occur for a variety of reasons associated with fundamental bearing design, interaction with the rotor dynamics and perturbation forces developed within the machine as a direct or indirect result of fluid forces.

Application of high speed lightweight flexible rotors demands bearings with strong stabilizing characteristics, typically of the tilting shoe or multi-lobe profile type. Unless the design of the turbine eliminates rotor lifting and perturbing forces derived from the expanding fluid within the machine case, it is almost inevitable that some evidence of well suppressed instability, seen typically as re-excitation of the rotor first flexural critical speed, will be detectable. Provided that the level of re-excitation is small at all times, such behavior may not be necessarily considered to be a problem or a defect in the machine; clearly good judgement must be used in assessing turbine behavior of this type.

Bearings with high intrinsic stability cannot always be specified since the available damping, particularly at the critical speed during start-up or shutdown, may be insufficient to prevent the occurrence of unacceptably high rotor vibration.

Some types of turbomachinery appear to be able to operate with the rotor vibrating relatively strongly in an unstable regime without failure; most are not of course. Establishing the level of stability available on marginally stable turbines is possibly the most important objective.

A most important requirement, both for machine protection and investigation of rotor behavior, is the selection of an appropriate vibration measurement technique. Use of the wrong technique can conceal potentially serious rotor instability and prevent effective machine protection or meaningful analysis of machine behavior.

Although good rotor dynamics/bearing analysis techniques are available for initial design and post-design assessment of turbine behavior, it is still difficult to use this type of facility in the field.

There is a considerable need for the development of portable hardware and software systems capable of being used in the field to enable more specific analysis of both machinery instability and stability margins to be made. In many situations it is more important to establish how much margin of stability a rotor system has, particularly when it is in as-designed condition. It is often of even greater importance to establish how sensitive the rotor system stability margin is to deterioration in bearing condition, effective bearing load and the development of perturbing forces within the turbine.

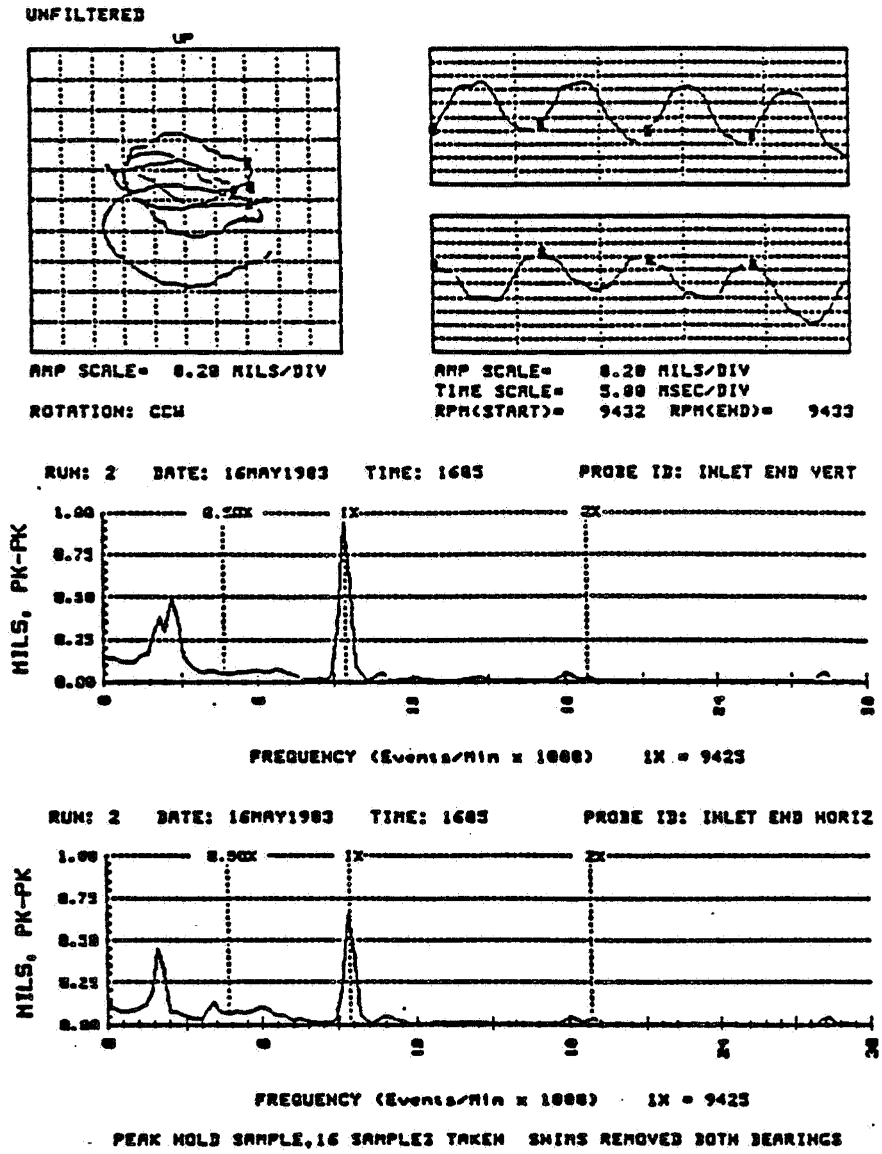


Figure 1. - Process drive turbine inlet end orbit, waveform, and spectrum plots.

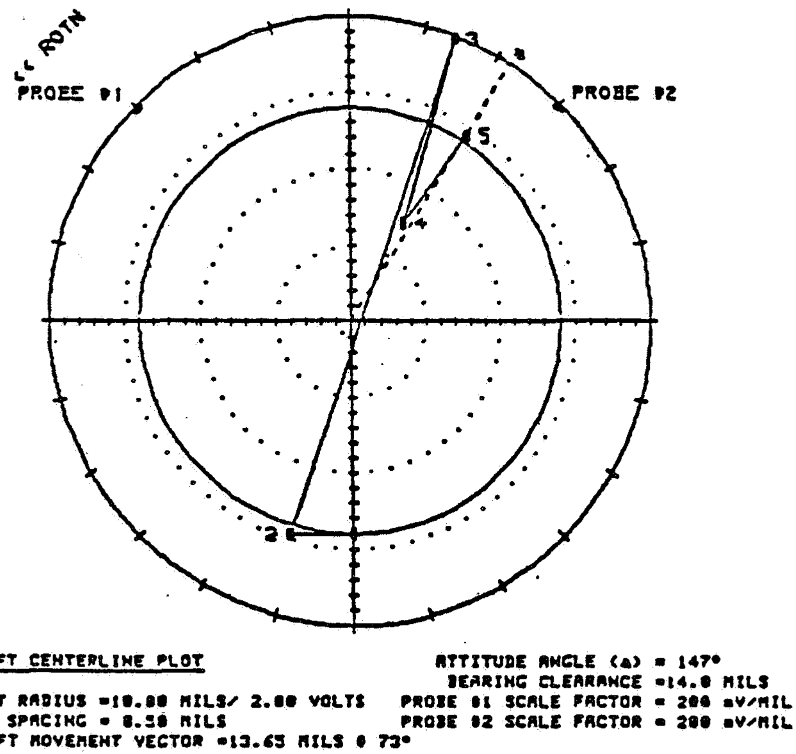


Figure 2. - Process drive turbine inlet end rotor position.

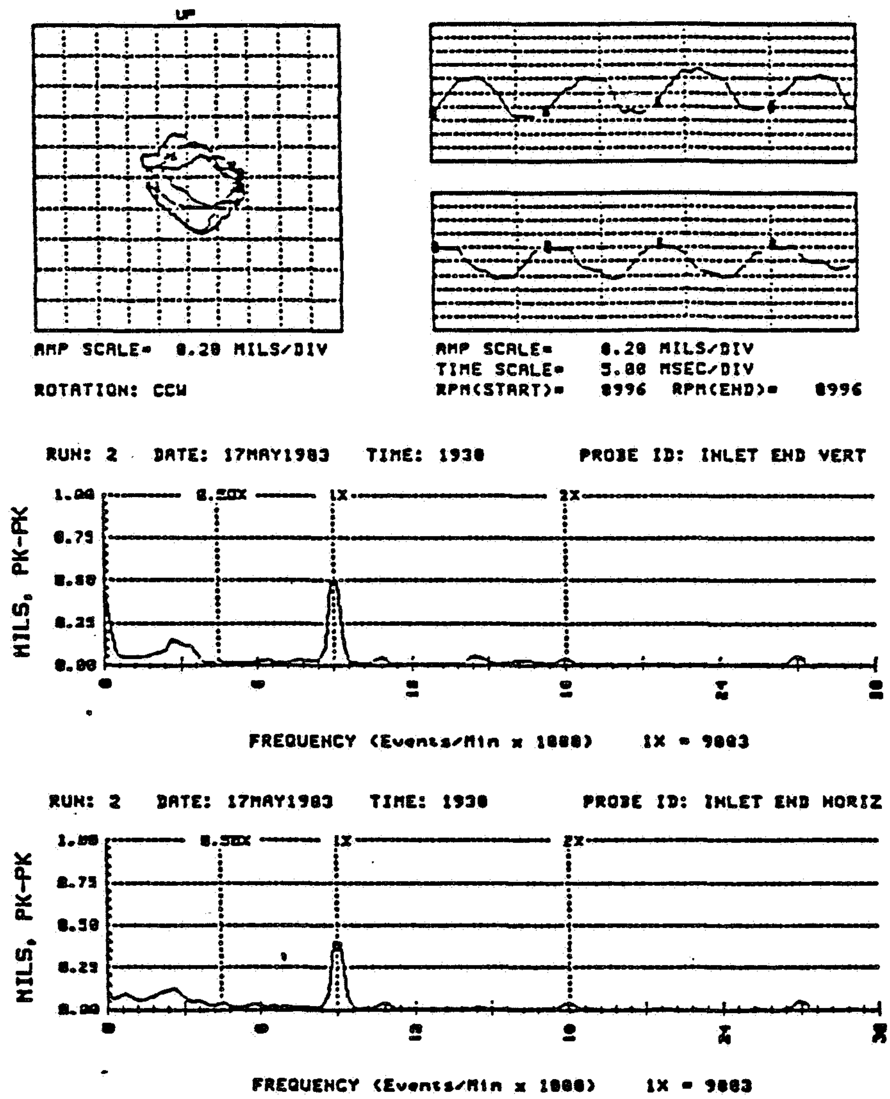


Figure 3. - Process drive turbine modified bearings.

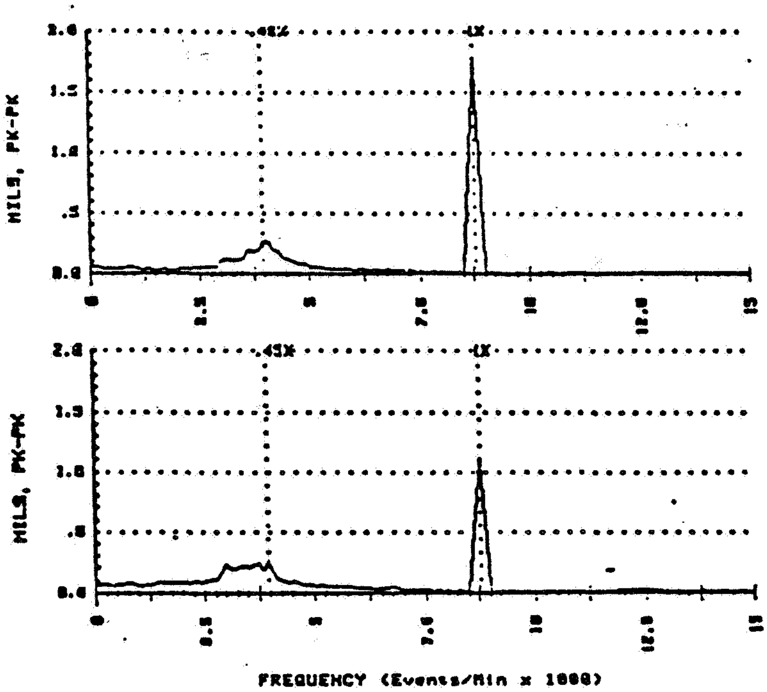


Figure 4. - Example of subsynchronous vibration - process drive turbine B.

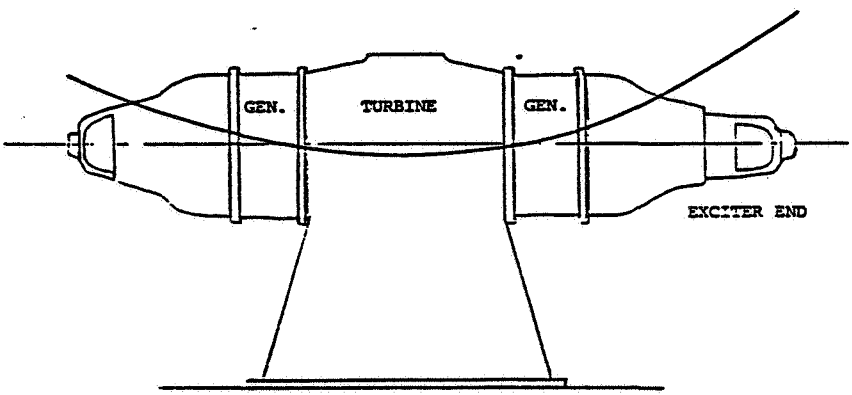


Figure 5. - Radial flow steam turbine generator casing vibration.

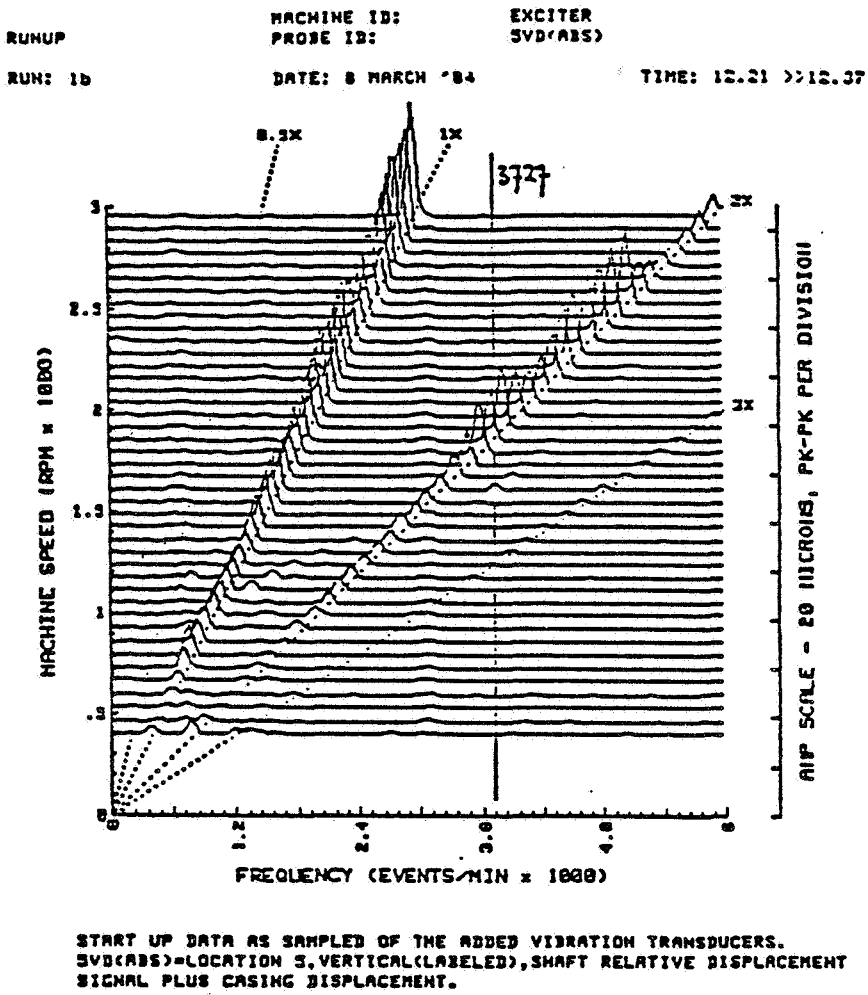


Figure 6. - Cascade spectrum plot for radial flow turbine generator exciter outboard bearing - rotor absolute vibration.

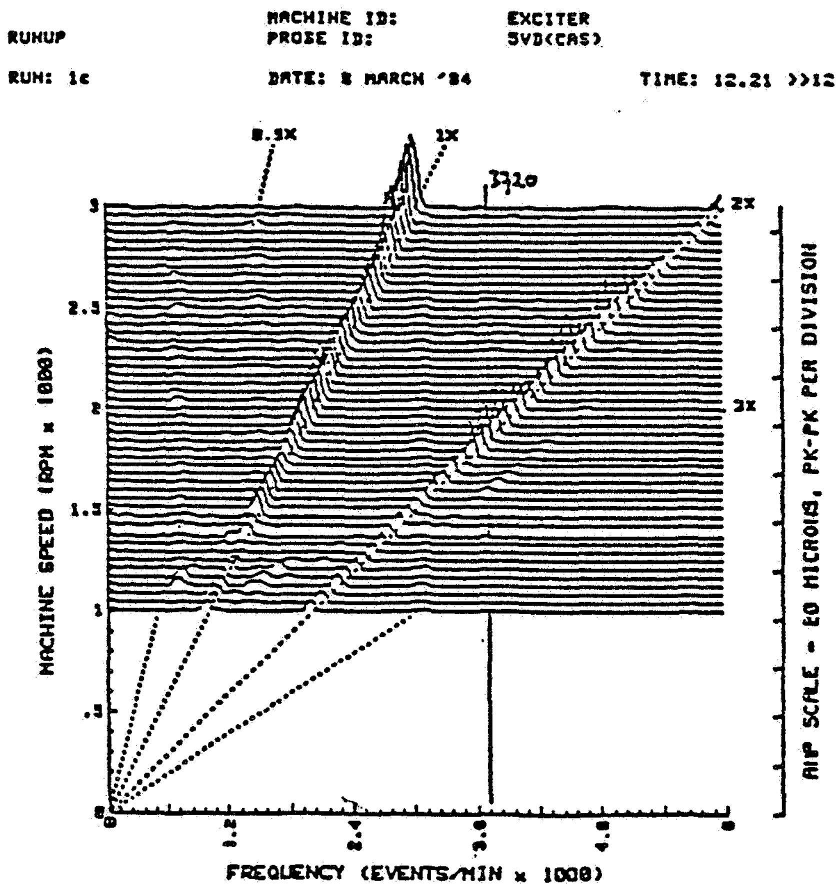
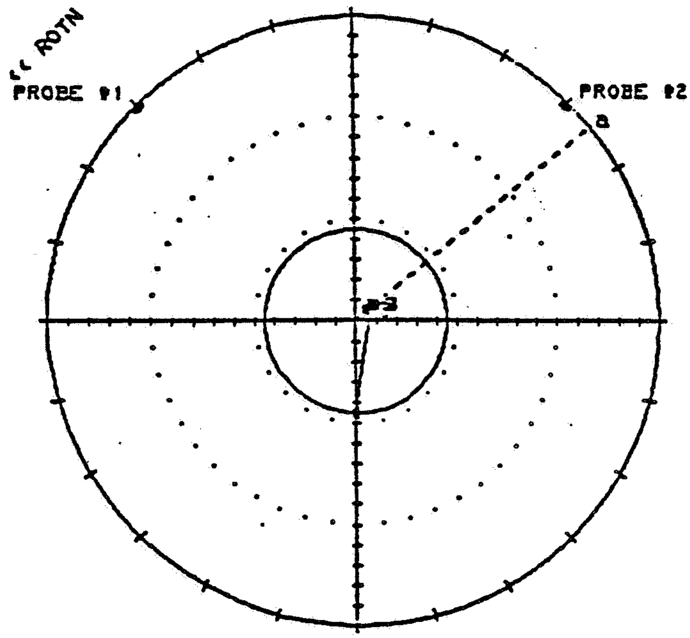


Figure 7. - Cascade spectrum plot for radial flow turbine generator exciter outboard bearing - housing absolute vibration.



SHAFT CENTERLINE PLOT

ATTITUDE ANGLE (a) = 129°
BEARING CLEARANCE = 90 MICRONS

PLOT RADIUS =150 MICRONS/ 1.18 VOLTS
TIC SPACING = 10 MICRONS
SHAFT MOVEMENT VECTOR = 54 MICRONS @ 79°

PROBE #1 SCALE FACTOR = 8 mV/MICRON
PROBE #2 SCALE FACTOR = 8 mV/MICRON

#	RPM	TIME	DC VALUES (Volts) PROBE #1	PROBE #2	REMARKS
1	0	1210	-8.94	-9.83	LOAD 0 Mwatt
2	3000	1235	-8.69	-8.69	LOAD 1 Mwatt
3	3000	1325	-8.69	-8.72	LOAD 1 Mwatt
4	3000	1544	-8.70	-8.68	LOAD 4.6 Mwatt

SHAFT AVERAGE POSITION PLOTTED USING BEARING CLEARANCE OF 90 MICRONS

Figure 8. - Journal position of radial flow turbine generator exciter outboard bearing.

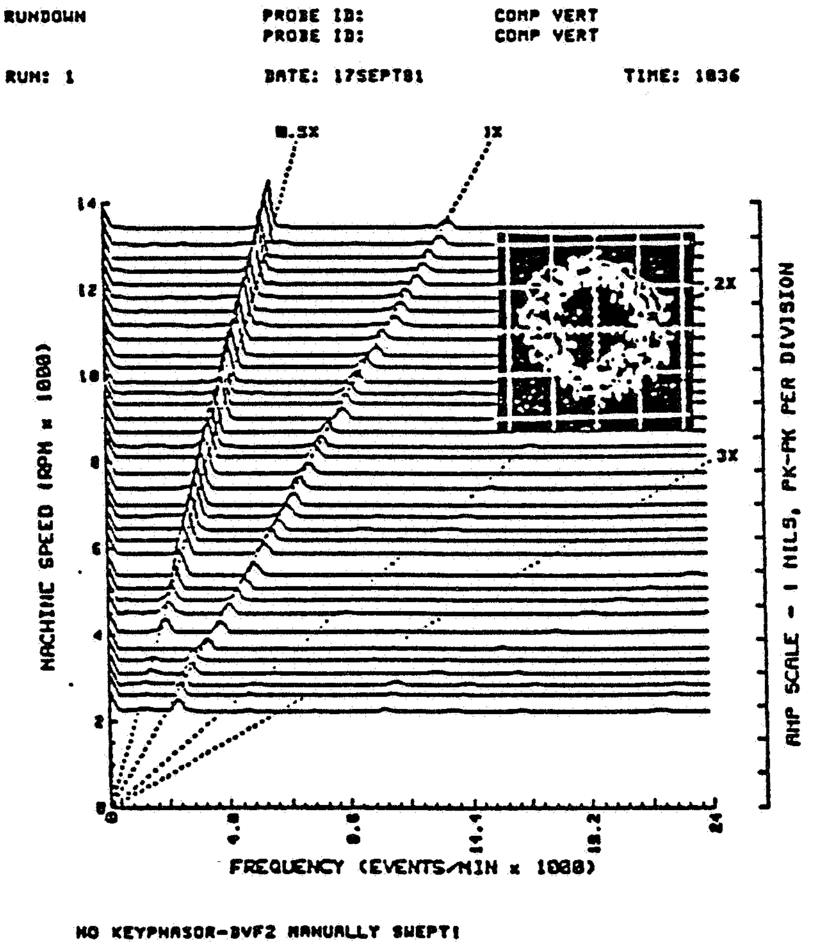


Figure 9. - Cascade spectrum plot and rotor orbit for turboexpander/compressor A rundown.

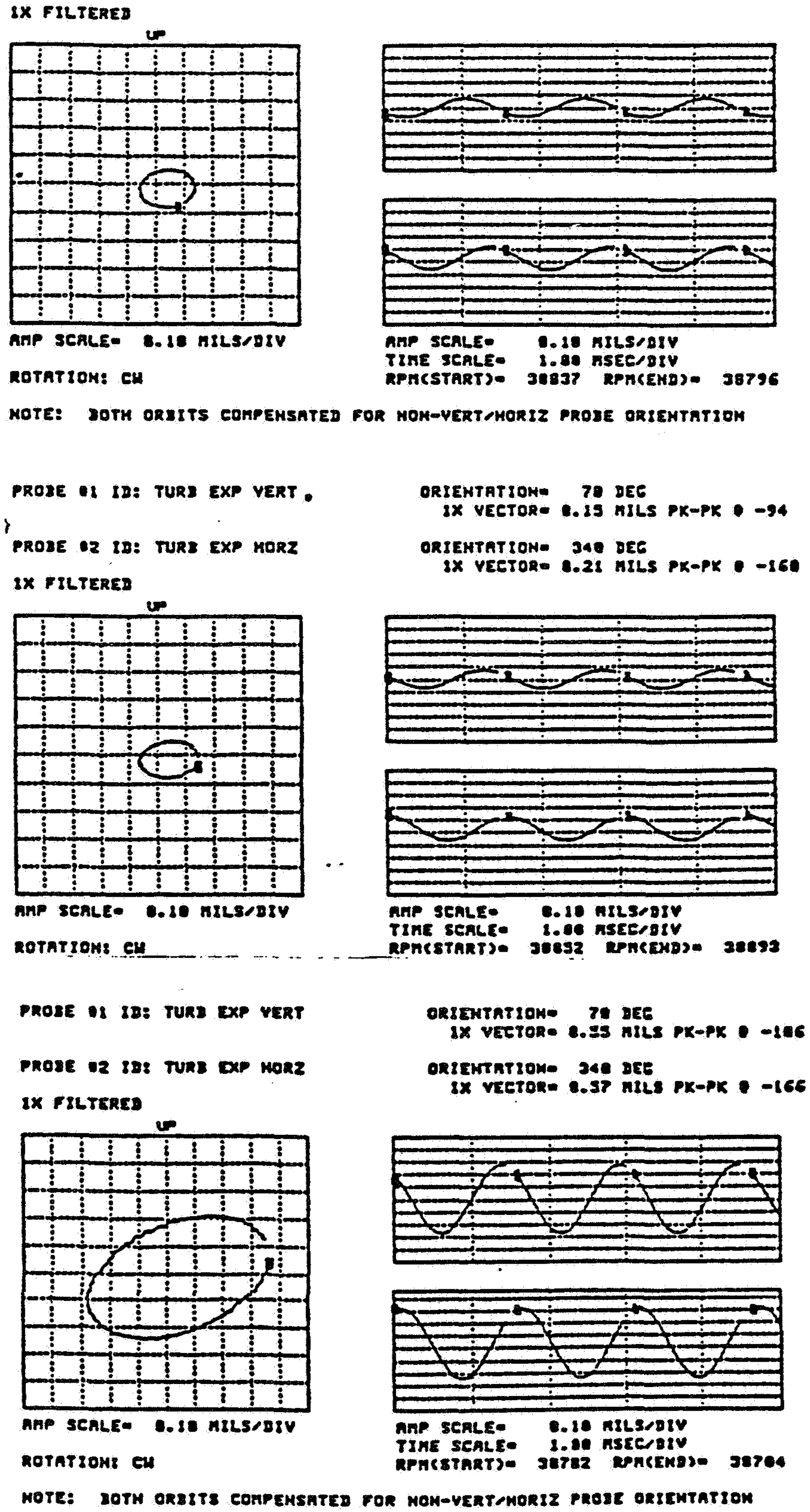


Figure 10. - Rotor orbit changes at high oil temperature for turboexpander/compressor B.

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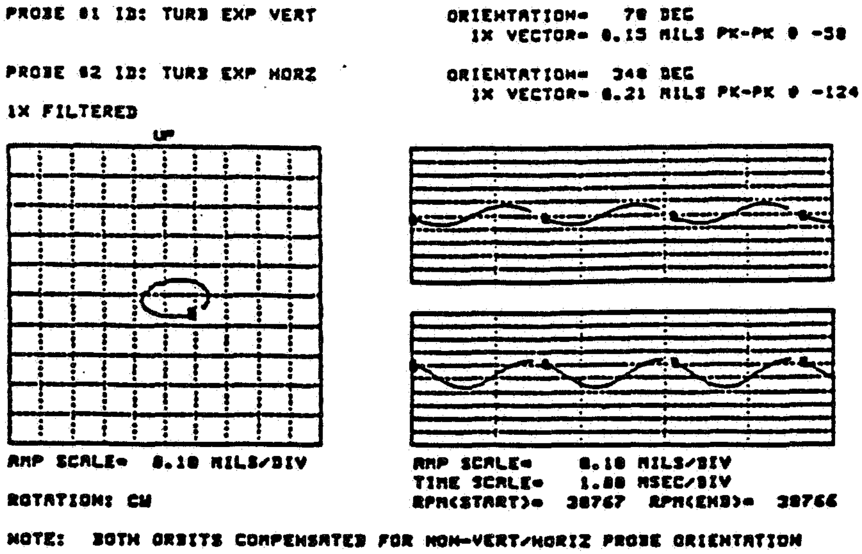

Figure 10. - Concluded.