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 unload 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-topping 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 instal-
led 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 over-
hung 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 when-
ever re-excitation of the first critical occurred. The adoption of tilting-shoe
bearings, which would have solved the oil film instability problem at normal opera-
ting speed, would have resulted in totally unacceptable rotor vibrations when pass-
ing through the first critical speed due to the inevitable reduction in system damp-
ing 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 suf-
ciently 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 sol-
ution 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 horse-
power 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 vibra-
tion 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 re-
sponse, 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 op-
erating speed, allowing for the greater lift of the much larger diameter main gen-
erator bearing journal; full load (6.25 MW) was then achieved with acceptable over-
all 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 vibra-
tion 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 in-
creased from 35°C to 45°C, the normally stable rotor started to go through a cycli-
cal 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 there-
fore 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 funda-
mental 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 in-
evitable that some evidence of well suppressed instability, seen typically as re-
excitation of the rotor first flexural critical speed, will be detectable. Provi-
ed 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.

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

Figure 5. - Radial flow steam turbine generator casing vibration.
START UP DATA AS SAMPLES OF THE ADDED VIBRATION TRANSDUCERS.
SVDCABS=LOCATION S, VERTICAL(LABELED), SHAFT RELATIVE DISPLACEMENT
SIGNAL PLUS CASING DISPLACEMENT.

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.
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.
NOTE: BOTH ORBITS COMPENSATED FOR NON-VERT/HORIZ PROBE ORIENTATION

Figure 10. - Concluded.