DETECTION OF ROTOR CRACKS

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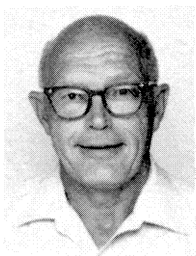

Donald E. Bently is Founder, Owner, and Chief Executive Officer of Bently Nevada Corporation, and President of Bently Rotor Dynamics Research Corporation. He pioneered the successful commercial use of the eddy current proximity transducer to measure vibration in rotating machinery and diagnosing machinery malfunctions. He guided Bently Nevada Corporation to its present position as the world's leading manufacturer of rotating machinery monitoring and information systems, diagnostic equipment, and services.

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ABSTRACT

The method of detection of rotor cracks by vibration monitoring is outlined. Various mechanisms stimulating cracks are discussed. Vibration measuring instrumentation and diagnostic methodology for early detection of rotor cracks are described.

INTRODUCTION

Current trends in rotating machinery design and operation result in severe stress and environmental conditions imposed upon the rotor. High mechanical stress results from the general trend to increase the power of machines which is associated with the rotative motion of shafts. Due to various physial mechanisms, small fractions of the rotative energy are usually transferred into shaft lateral, or more complex lateral/torsional/longitudinal, vibrations, as a side effect of the dynamic process of machine operation.

High torsional and radial loads, together with a complex pattern of rotor motion, create severe mechanical stress conditions which eventually lead to shaft cracking. Design imperfections, often resulting from simple extrapolation of low power smaller machines to larger sizes, introduce unpredicted stress concentrations, which increase the hazard of crack initiation. Most rotor cracks propagate in a high or low cycle fatigue manner. Cyclical stress, related to vibrations, occurs in combination with tensile mean stress. Very often, environmental conditions have a critical effect on crack initiation and propagation. For instance, the presence of sodium chloride in steam or water in turbines can strongly encourage the crack growth due to the concentration of salt in keyways and under disks. Severe thermal fields significantly aggravate the conditions in which most rotors operate.

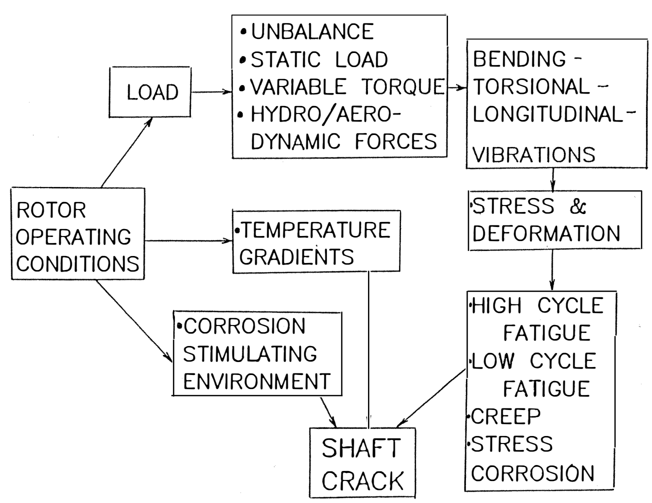

Figure 1. Shaft Crack Stimulating Factors.

Numerous catastrophic failures caused by cracking rotors have resulted in increased interest in detecting shaft cracks in turbomachinery. The early detection of the crack and the prediction of its growth may prevent costly machine failures and subsequent shutdowns of entire plants.

In recent years, rotating machinery protection systems which include vibration measurements, monitoring, diagnostics, and analysis have received widespread acceptance and use. Vibration monitoring provides advanced warning and information of machine malfunctions, which can help to prevent major machinery failures and reduce costly downtime. It requires, however, dependence on instrumentation and correct interpretation of the data to realize the maximum benefit of the protection systems.

More than 30 shaft crack incidents have been reported in North America over the past ten years in the power generation industry alone. On the other hand, in the last few years, at least five major saves of rotating machines have been accomplished, using shaft vibration instrumentation and crack diagnostics methods.

Recent advancements in vibration monitoring instrumentation and computerized data acquisition/processing systems have resulted in the accumulation of well managed experimental/analytical results, as well as properly documented case histories. These results and case histories have added to the theoretical background on shaft cracks, which originated in the early 1930s. Literature on cracked shafts is cited in the REFERENCES [1-30].

Vibration monitoring has gained a widespread application in shaft crack detection in recent years. A shaft crack causes specific dynamic phenomena during machine operation. By observing and monitoring the appearance of these phenomena and, in particular, analyzing their evolution in time, it is possible to successfully detect shaft cracks before complete breakage. If coastdown and startup data is available, detection of rotor cracks becomes even more certain and at earlier stages that can be detected at the operating speed.

The rotor lateral vibration monitoring and diagnostic methods for shaft crack detection at early levels are outlined. In addition, a few crack stimulating machine design features are discussed. Occurrences of shaft cracks could be significantly decreased even further, if certain modifications were undertaken in the rotating machine design.

ROTATING SHAFT CRACK STIMULATING FACTORS

Mechanisms considered to be of primary importance in rotating shaft cracking include high-cycle fatigue, low-cycle fatigue, creep and stress corrosion cracking. The first mechanisms are related to the mechanical conditions in which rotors operate. The last ones include influence of the environment, mostly thermal factors and effects of corrosive chemical substances contained in the rotating machine working fluid.

Modes of Shaft Stresses

There are three principal modes contributing to shaft stresses. In apparent order of importance to cracking, they are lateral, torsional, and longitudinal modes. All of these modes usually have both steady-state and dynamic components. Of course, the shaft stress at a given location is a function of the instantaneous vector sum from all of these sources.

The lateral bending modes are generated by radial rotating and nonrotating forces. These forces include mass and elastic unbalance (bowing) inertia forces, misalignment forces, thermal forces, and gravity bowing. The lateral bending modes also often result from self-excitation factors in rotating machine, leading to shaft lateral vibrations of the subsynchronous "whirl" or "whip" type. These self-excitation factors are usually generated in oil-lubricated bearings and seals, or in the main fluid flow of fluid-handling machines.

The lateral bending modes may be generated by a high number of sources; however, by far, the largest contributor to shaft cracking is the $1\times$ cyclic stress induced by misalignment. This misalignment, both parallel and angular, may be internal, such as a bowed casing as shown in Figure 2, or it may be external as shown in Figure 3. This misalignment may be angular or parallel. External misalignments are more severe with solid coupled rotors such as turbogenerator sets, and least severe on machine trains with elastic disk-type couplings.

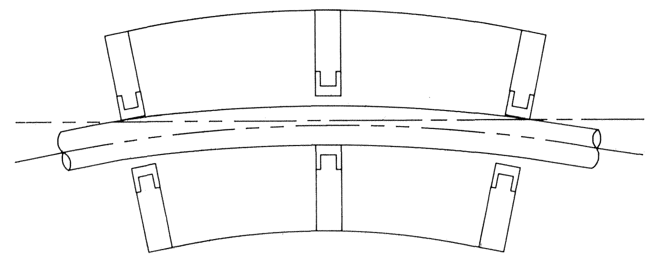

Figure 2. Warped Casing Resulting in the Rotor Internal Misalignment.

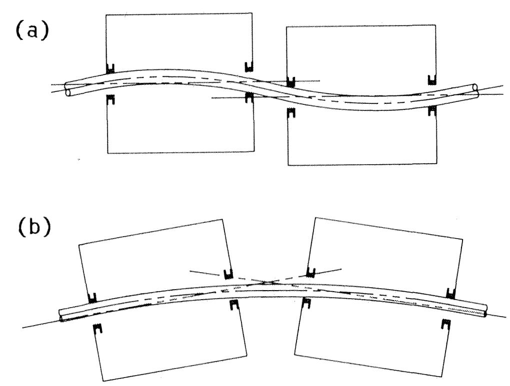

Figure 3. Rotor External Misalignment (a) Parallel, (b) Angular.

There has been a fundamental assembly misalignment problem of turbogenerators (TG) that is only recently sometimes being rectified. It has been a universal procedure to assemble a turbogenerator by setting the middle span (usually an intermediate pressure (IP) or low pressure (LP) steam turbine) level, then adjusting each span toward the corresponding end so that the solid coupling faces are both parallel and concentric (or ready to go). An example is shown in Figure 4. This significantly simplifies the procedure of coupling the shaft together, but clearly causes, the resultant shape of the turbogenerator rotor centerline to resemble a "banana," or catenary form.

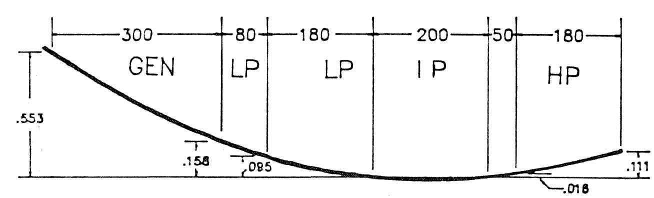

Figure 4. Typical Turbogenerator Cold Alignment Setting.

While this procedure is perfectly good for assembling the TG set, the alignment adjustment of turbogenerator sets should be based on the modern standard rules and methodology of hot alignment, so that couplings, seals, and bearings carry the proper loads. If the large catenary is left in, there will be an additional 1× stress incurred reversal by it.

For better understanding of the role of misalignment contributing to the rotor bending stresses, the synchronous vibrations due to unbalance will be discussed first.

The most common source of rotor vibrations, the unbalance, is least damaging to the shaft, provided that the rotor is appropriately balanced so that the amplitudes are small enough. In the high majority of cases, the unbalance inertia forces cause shaft forward synchronous precession (orbiting), characterized by the frequency equal to the actual rotative speed (1 ×). If at a *constant* operating speed the observed shaft centerline motion represents a *circular* 1× orbit around a neutral axis, it means that the shaft is bent and rotates with its "frozen" bent shape. The shaft fibers are, therefore, constantly either stretched or compressed and there are no variable stresses involved. Acting alone, the unbalance is relatively unimportant for shaft cracking.

The synchronous *elliptical* precession resulting from very common rotor and/or support lateral asymmetry introduces a variability of the steady bending stress. The shaft bending mean steady stress is accompanied by a small component fluctuating with a frequency double the rotative speed (2×) (Figure 5 (a)).

A constant radial force, such as created by misalignment or gravity, results in the shaft centerline being displaced from the neutral axis. This dramatically changes the rotor stress situation. The shaft synchronous vibrations, due to unbalance, result in shaft *reversal* stresses with frequency 1× (Figure 5 (b)).

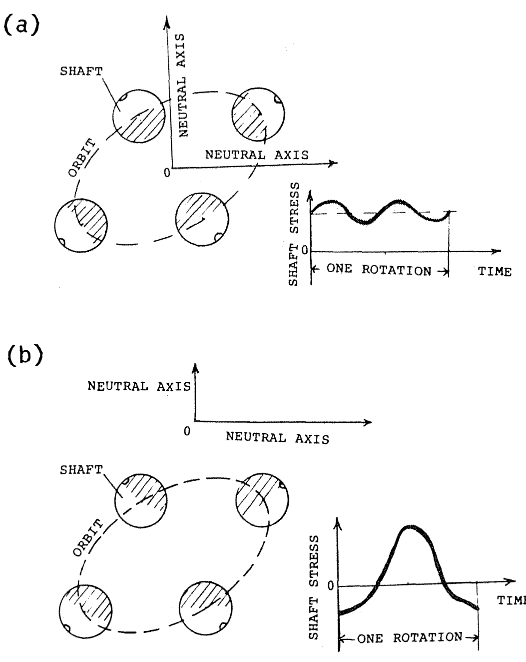

Figure 5. Synchronous (1×) Vibrations of a Shaft at a Constant Rotative Speed Around a Neutral Center, (a) and Displaced Center Due to Constant Radial Force Such as Generated by Misalignment (b).

Operation of rotors at misaligned conditions causes, therefore, the potentially damaging shaft cyclical reversal bending stress with a frequency of once-per-revolution (1×). A rotor usually operating at 10000 cpm during only one hour will acquire over a half of million cycles. Misaligned rotors of 3600 cpm power generating machines are cyclically stressed one million times in four hours and 39 minutes. Even relatively small misalignment, *high-cycle* fatigue results very easily. The situation dramatically worsens if misaligned rotors exhibit higher level of vibration amplitudes and/or vibrations with a rich frequency component content.

There are many other contributors to the lateral bending stresses, including any subsynchronous and supersynchronous activity, such as from the category of rubs and from the category of forward circular instability mechanisms such as steam whip, pumping whip, and oil whirl or whip.

At a constant rotative speed, when one bending vibrational component is present in the shaft response, the net frequency of shaft bending stresses is equal to the difference between the frequencies of vibration and rotation, taking their directions into account. For instance, during the synchronous (1×) backward orbiting the shaft is periodically stressed in reversal manner with frequency 2×. Subsynchronous vibrations such as whirl/whip, as well as vibrations associated with backward modes, may significantly contribute to shaft cracking. These vibrations are often characterized by high amplitudes and may cause *low-cycle* fatigue crack propagation in addition to increasing the high-cycle repetition rate.

During normal operation, the rotor of a rotating machine is under a steady torsional stress resulting from the driving torque and the load torque. This steady stress can, however, often be accompanied by a variable torsional component. The consideration of rotor torsional stress is particularly important during transient processes of startup and shutdown of the machine, when the rotative speed is variable. The torsional stress may also be of concern in machines which operate under a specific variable torsional load. The important source of shaft cracking is the combination of torsional and lateral cyclical reversal stresses. The contributions to the total stress by tension and compression are usually small, but are noted for reference.

Generally speaking, at each rotative speed the rotor responds with lateral and/or torsional/longitudinal vibrations following the axial distribution of applied forces opposed by the *rotordynamic stiffness*. In order to determine the axial locations where the highest stresses occur it is very important to know the shaft modal shape for the given operating speed and the unbalance and other forcing distributions. Rotor lateral mode shape analysis is outlined in the TECHNIQUE TO ASSESS MACHINERY CONDITION USING MODE SHAPE ANALYSIS section.

Stress Concentration

The axial distribution of stresses associated with the bending modes of vibration represents only the first part of the problem. In specific rotor locations the stress may increase dramatically, resulting in a crack, if the rotordynamic stiffness is sharply weakened by "stress concentration" factors.

Classically, the first important stress concentration areas from which cracks can propagate are due to rotor material defects, such as inclusions of slag or other nonmetalic elements. The same effect results, for instance, from hydrogen embrittlements in steels, which weaken intergranular bonds and encourage formations at voids, acting eventually as stress raisers.

Parallel to material defects there are several specific rotor design features which lead to stress concentration in the particular area of the rotor. In such areas, cracks are often initiated. Stress concentration occurs in the rotor elements weakened by holes (such as pressure balance holes in turbine disks), notches, grooves, slots, threads, rotor internal cavities, stampings, scribed markings, and/or keyways, especially when poorly machined and finished with sharp inner angles. Proper radiusing of the weakened area decreases the stress concentration effect. Adequate radiusing is also very important to relieve the stress between two sections of step diameter rotors.

All elements which should be tightly attached to the rotor provide a potential stress concentration area at the "fitting" surfaces. For example, a gap between the disk and the locating shoulder of the rotor may create a stress concentrating notch effect, due to improper fitting. The fitting errors are usually difficult to avoid. During machine operation, a part of the fitting area may develop looseness, followed by fretting which stimulates crack initiation. Fitting surfaces of taper-fit or shrink-on components of the rotors usually contribute to rotor stress concentration, especially when high thermal gradients are involved. Generator rotor slot wedges carrying windings provide another example of notch-type stress concentration effects. Assisted by fretting due to loosened windings or winding retainers, the slot wedges represent potential crack initiation areas.

Rotor Design Features

There are two very popular rotor designs which tend to be a natural target for cracking. These are (i) double flow turbine rotors (Figure 6) and (ii) overhung rotors (Figure 7).

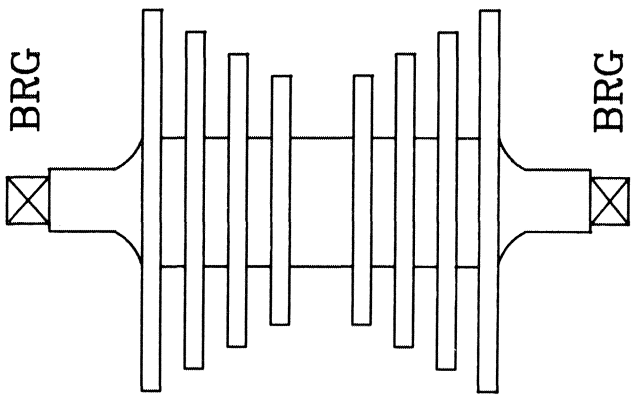

Figure 6. A Sketch of a Double Flow Low Pressure Turbine.

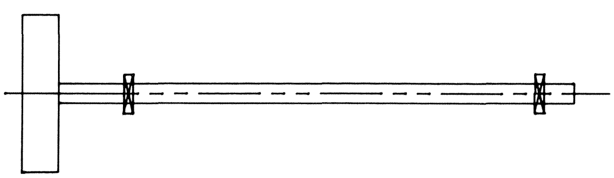

Figure 7. Overhung Rotor.

Due to unfortunate design, the double flow steam turbogenerator operating speed often coincides with, or is close to, half of the second natural frequency of the turbogenerator unit. This means that the double frequency component of vibration has its resonance just at the operating speed. Consider, for example, a typical two cone shape single span double flow turbine rotor for 3600 cpm operation with its first and second natural frequencies equal to 1600 cpm and 4800 cpm, respectively. Since the double flow turbine is virtually always located in the central portion of the multispan turbogenerator set, the coupling with the adjacent units causes the system second natural frequency to increase, often around 1.5 times, yielding the value 7200 cpm, i.e., exactly twice the operating speed. There is also another possible scenario. If the machine second balance resonance is normally higher, at 7400 cpm, the second $2\times$ resonance will be at its maximum in the range of 3700 cpm. If the shaft develops a crack and it propagates, the rotor stiffness is continuously reduced and the resonance frequencies decrease. It may happen, therefore, that in the machine which had its original second balance resonance at 7400 cpm and normally operates at 3600 cpm, half of the second balance resonance may consequently occur exactly at the operating speed. This creates a big problem for the turbogenerator. The $2\times$ resonant vibrations cause high vibrations and high reversal stresses just where the shaft joins the large diameter turbine section—a perfect target for cracks, due to stress concentration. Attempts to provide a very large radius fillet at that point do not fully alleviate the problem. The proper fix is to modify the system natural frequency spectrum and move the resonance further away from twice the operating speed.

The second unfortunate design is that of the single overhung rotor with a relatively large wheel (Figures 7 and 8). This is a very natural design for many machines, and is especially used in vertical configurations for feedwater and coolant water pumps installed in nuclear plants.

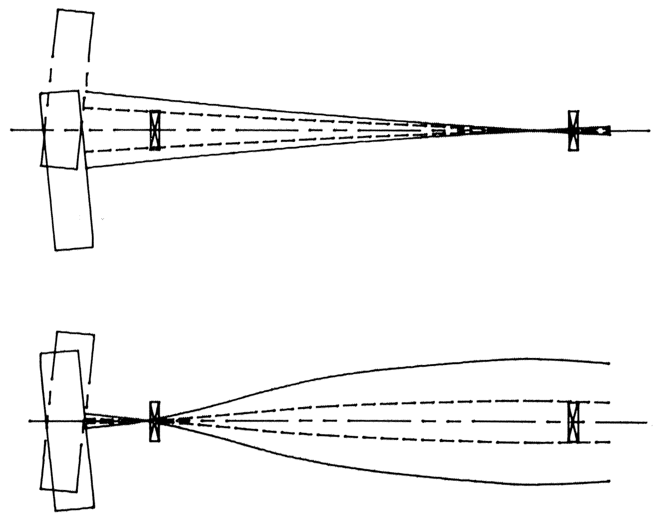

Figure 8. Two Lateral Modes of an Overhung Rotor.

Like the double flow turbine design, the single overhung design may suffer from very high stresses where the shaft attaches to the wheel. Unidirectional high raddial flow-related loads are applied directly to the overhung wheel.

If due to external nonsynchronous forces or self-excitation factors, an overhung rotor ever developed vibrations with frequencies other than the rotative speed, then it is very possible that extremely high cyclical stresses may be generated on the shaft near the connection to the wheel.

Unlike the double flow design, in the case of the overhung rotor, it is much easier to escape the terrible misfortune of operating speed at (or just below) half of any resonance speed. Even so, the authors are of the opinion that there are many machines (including machines in critical service) which carry this design fault. Operation near half of a lateral resonance speed, especially with poor system lateral damping, makes the rotor highly susceptible to crack propagation.

Environmental Factors

Rotor internal cavities, keyweys, notches, clearances in fitting parts, and sharp angles represent not only mechanical stress concentration hazards, they are also especially susceptible to corrosion. Chemical contaminants, such as salts, sulphates, and acids in steam, water, or other working fluids, are particularly aggressive in cavity-like areas, due to possible concentration/crystallization of the chemically active substances. Poor water/steam purity, which occurs for even a short time, creates a crack-stimulating environment. In longer terms, corrosion can lead to material deterioration, which strongly supports the crack formation.

Thermal conditions constitute another very important factor contributing to the initiation and propagation of cracks. Excessive temperatures, for instance, cause acceleration of creep damage, creating intergranular voids in metals followed by micro- and macrocracking. High rates of temperature changes may generate high stress due to large nonuniform thermal expansion. In each case, the thermal stresses worsen the shaft conditions and stimulate crack growth.

The topic of thermal and chemical effects on shaft cracking, well documented in studies by other researchers, will not be further discussed herein.

INSTRUMENTATION FOR SHAFT CRACK DETECTION

The most informative vibrational signals for the early detection of shaft cracks are the rotating machine lateral synchronous response (1×) and the second harmonics (2×). These signals should be monitored at operational speeds and during each startup and coastdown of the machine. The slow roll synchronous signal generated at low speed rotation also provides relevant information. Relatively low frequency characterizes all of the vibration signals discussed previously. The most convenient and reliable transducers for measuring rotor vibrational response with low frequency components are noncontacting displacement eddy current proximity probes that are mounted to the stationary elements of the machine at, or close to bearings. These transducers measure the relative rotor/stator motion, including the rotor static displacement (e.g., journal position inside the bearing or dynamic displacement of the rotor centerline from the neutral axis).

Seismic transducers mounted on the machine housing may be useful for supplementary information, but the obtained signals pertinent to rotor cracking are usually highly attenuated as compared to shaft observing sensor signals.

The generally accepted practice for obtaining the most accurate measurement information on rotating machines is to utilize pairs of XY proximity probes, mounted 90° apart, at a sequence of rotor axial locations. These probes can be installed at vertical/horizontal positions or at 45°/135° orientations.

The American Petroleum Institute (API) has adopted a standard, entitled "Noncontacting Vibration and Axial Position Monitoring System" (Standard Number 670). This standard outlines the system requirements for installing transducers in the 90° configuration on process compressors and their drivers. The transducer installation practices outlined in the standard are appropriate for the monitoring and protection of turbomachinery and other critical machines as well.

One of the most important transducers called for in the API standard is a phase angle reference transducer. This transducer is mandatory to monitor 1× (synchronous) and 2× response phase angle, and to facilitate the machine diagnostic process by indicating a reference of each vibration signal frequency to the frequency of rotation.

A proximity transducer is used in to observing a once-per-revolution event, usually a notch (keyway) or projection on the shaft. The transducer provides a once-per-revolution voltage pulse, which can be input into vector filters, tachometers, and other diagnostic instruments. It is also used to trigger a Z-intensity axis of an oscilloscope, providing bright spots on the time base vibrational signal and shaft orbital motion display.

XY proximity transducers can also be applied to help determine the lateral mode shape of the rotor at resonances and at operational speeds. These "mode identification" transducers are located at various axial positions along the rotor. The use of the proximity probes and a Keyphasor reference should be especially recommended to continuously monitor machines that are susceptible to shaft cracking. The mode identification probes provide adequate machine protection by overcoming the potential danger, when only a single set of XY proximity probes is used, of locating the probes at a nodal section along the rotor. They also provide significant information on crack-related shaft mode changes.

The transducer technology, as well as advances in electronic and computer technologies, now enables machinery users to utilize cost-effective, permanently installed rotating machinery information systems for monitoring and diagnostics. The transducer signals are input into three basic types of rotating machinery information systems:

- Protection systems for continuously monitoring overall vibration levels and rotor position. These systems have become standard requirements on critical machinery in many industries.
- Computerized diagonostic systems for analyzing the transient vibration measured by transducers. These systems have become an important tool for machinery engineers and maintenance personnel to analyze vibration data taken at startup and shutdown. They can consist of filters (a tracking filter of 1× and 2× vibrational components is recommended), oscilloscopes (to continuously observe shaft orbital and time base motion), fast Fourier transform (FFT) spectrum analyzers, and a computer with appropriate software for data storage, processing, and numerical/graphical output. The monitoring system may acquire data continuously or periodically. More sophisticated software may replace the necessity of having separate filters and spectrum analyzers.
- On-line computerized diagnostic systems for capturing and analyzing steady-state dynamic data taken at the operational speed for on-line machine condition analysis.

Computerized diagnostic systems usually acquire machine vibration data, as well as process variable data, such as flow, pressure, external load, etc. The correlation of these two types of data aids in determining the cause of the machine malfunction.

CRACK SHAFT DIAGNOSTICS

Several other malfunctions can cause a machine to exhibit similar vibrational symptoms as those experienced under a shaft crack condition; consequently, it is important that adequate methods are used to diagnose a shaft crack. There are two fundamental shaft crack symptoms, which can be observed by vibration monitoring:

- Unexplained changes in the synchronous (1×) shaft relative lateral vibration amplitude and/or phase at the operating speed and also changes of the slow roll vector on startup and/or shutdown.
- The occurrence of twice the rotative speed (2×) vibration component—occasionally at the operating speed, but especially on startup and shutdown.

The most vital and primary symptoms of a shaft crack are changes in the synchronous (1×) response vector (amplitude and phase) and slow roll vector. The changes in the synchronous (1×) amplitude and phase are caused by the shaft bowing, due to an asymmetric transverse crack ("elastic unbalance" effect). The acquired additional unbalance due to permanent crack-related shaft bow interferes with the original mass unbalance. In this situation, the synchronous (1×) phase and amplitude change—either higher or lower. In all observed cases where the crack has grown larger than 50 percent through the shaft, the synchronous amplitude has grown larger in the final stages (Figure 9). The phase changes may be significant, because it is unlikely that the crack occurs at the same angular location as the previous mass unbalance.

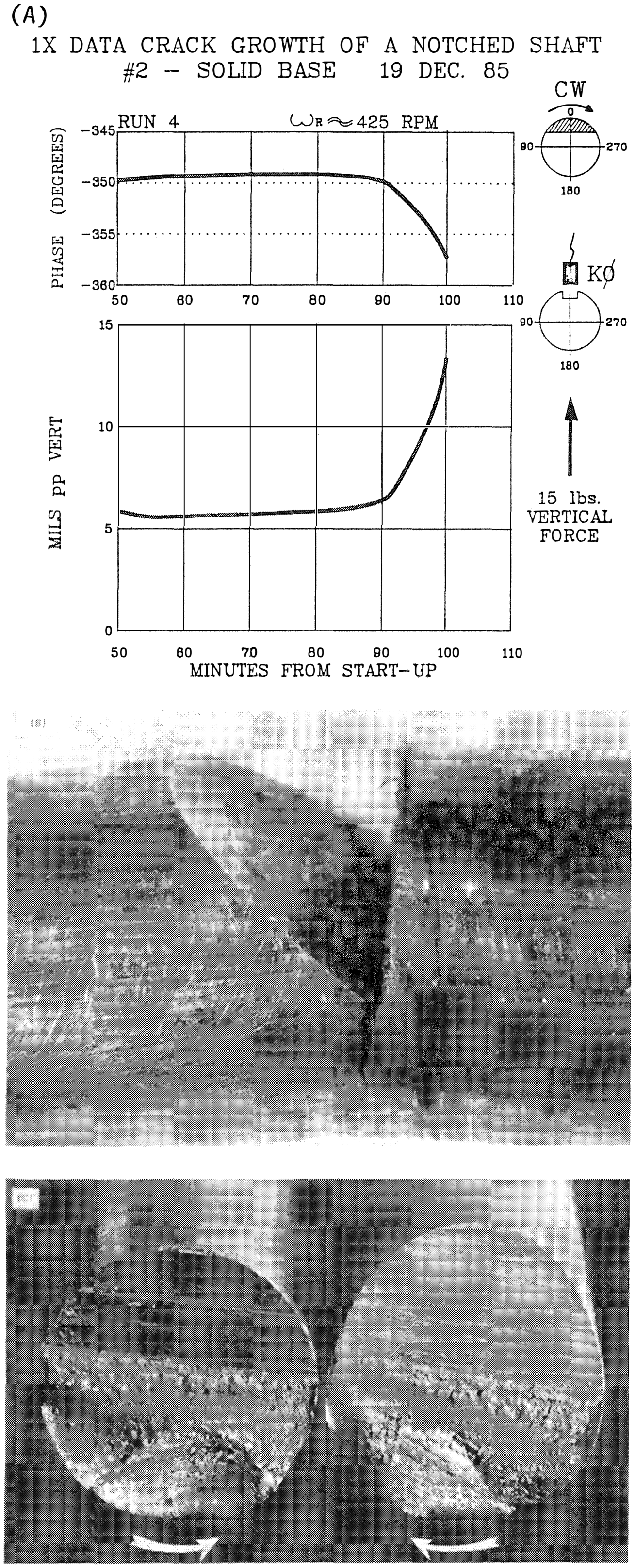

Figure 9. Experimental Data: (a) Phase and Amplitude of Reponse of Radially Loaded 3/8 in Diameter and 2 in Long Shaft with Simulated Crack, Operated until the Total Break of the Shaft, (b) Broken Shaft, and (c) Fracture Surfaces.

The secondary, and classical, symptom of a shaft crack—the occurrence of the 2× frequency lateral vibration component—is associated with the asymmetry of the shaft. The 2× component is due to a combination of a transverse crack (which causes shaft asymmetry) and a constant radial force. On horizontal rotors, the major radial force is caused by gravity. On vertical machines, the radial load can be generated by misalignment or fluid flow. On all machines, the 2× component is especially dominant (has its resonance) when the rotative speed is in the region of half of *any* rotor system natural frequency. The 2× component may or may not be present at the operating speed if the latter does not coincide with one-half of the second natural frequency, and if the system damping is high enough, but almost surely will appear at least once on startup and shutdown in the half balance resonant range of speeds.

On-Line Detection of Shaft Cracks

The changes in synchronous (1×) amplitude and phase, measured by proximity probes, can be monitored under normal operating conditions to provide alarming and early warning of a shaft crack.

The polar plot represents an excellent format for documenting the shifts in synchronous (1×) amplitude and phase (vibration vector). The shifts as functions of time should be monitored at the operating speed and under the normal range of operating conditions. A normal operating range of the 1× vibration vector is determined within the polar plot to form what is called an "acceptance region." The actual 1× vibration vector is then plotted (Figure 10). Deviation of the 1× vibration vector from the acceptance region may be a vital warning of a shaft crack, even though many other rotor disturbances can also cause some deviation from the acceptance region. The area and shape of the "acceptance region" may vary for specific machines and their load conditions.

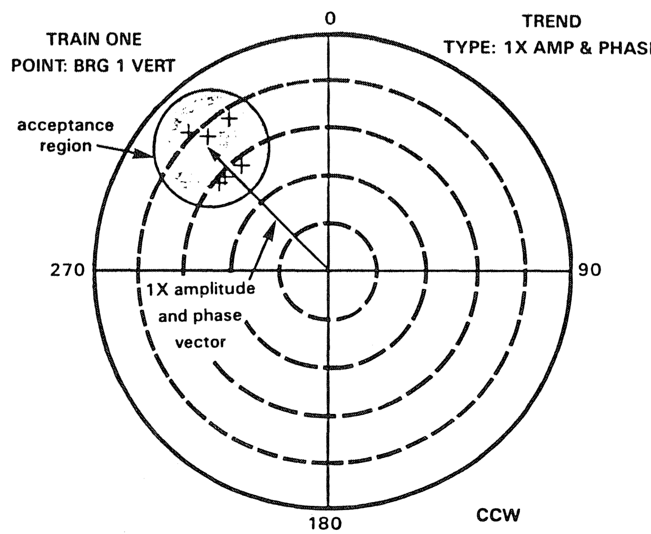

Figure 10. 1× (or 2×) Response Vector in the Polar Plot Format. The response vectors are monitored to ensure that they remain within an acceptance region. Deviation from the acceptance region indicates a probability of a shaft crack.

Similar technique using "acceptance regions" on the polar plots can be applied to monitor changes in the 2× response vector at the operating speed and the slow roll (1×) vector.

Changes in the synchronous (1×) amplitude and phase must be analyzed in conjunction with other information—including the twice rotative speed ($2\times$) machine vibrational response—to determine whether the shifts were caused by an asymmetric transverse crack or other factors. These other factors can include load, field current, steam conditions, thermal expansion, misalignment, etc.

For each machine there should be a set $2\times$ component amplitude level at the operational rotative speed as a mandatory shutdown level for the rotor suspected of a crack. This level may vary, depending upon:

- possible crack axial locations relative to the shaft mode shape,
- rotor system damping (although the damping is reasonably constant for many fluid-handling machines),
- amount of constant radial load (if there is zero load, the $2\times$ component would also be zero),
- location of the operational rotative speed in the natural frequency spectrum, and
- the probe locations along the shaft.

It is normal for a generator (especially a two-pole generator) to have some asymmetry and, therefore, to exhibit residual $2\times$ responses. The existing normal level of $2\times$ vibrations has to be taken into consideration in setting up the shutdown limit.

While it is normal for the horizontal $2\times$ vibration of the shaft to be slightly higher than the vertical $2\times$ vibration when the radial load is gravity, it is suggested that both the vertical and horizontal $2\times$ motions are observed. In at least one documented save on a turbine-generator, only vertical probes had been installed on the machine [28]. By using horizontal as well as vertical probes, the shaft orbital motion could have been observed, and the crack might have been detected earlier.

Even though the $2\times$ component does not necessarily appear at the operating speed, another recent shaft crack save exhibited this classical phenomenon [5]. In this case, the following information on a vertical pump used in nuclear power plant service led to an excellent save:

- increasing overall vibration levels.
- large $1\times$ and $2\times$ frequency vibration components.
- $2\times$ frequency vibration that remained at the same level after the machine was trim balanced. The $1\times$ frequency vibration component was significantly decreased by trim balancing.

The vibration information was used to determine whether the machine response was caused by other malfunctions, such as unbalance, misalignment, thermal bow, rub, etc. After eliminating the other possible causes, a high probability of a shaft crack remained. The machine was taken out of service and inspected. Inspection confirmed the diagnosis.

Shaft Crack Detection Using Transient Data

On startup and shutdown, two types of data formats for the synchronous ($1\times$) and twice rotative speed ($2\times$) components of the shaft lateral vibration can be plotted for both lateral (vertical and horizontal) shaft-observing probe vibrational signals:

- $1\times$ and $2\times$ polar plots (with increasing or decreasing rotative speed).
- Spectrum cascade plot that documents $1\times$, $2\times$, and other vibration components from slow roll to maximum available speed.

It is important to document machine vibrational response at every possible startup and shutdown. (For some electric motor-driven machines, unfortunately, only shutdown data is available.) Comparison of the sequence of polar/cascade plots reduced from the acquired data of previous runs provides very important information on the evolution of the crack growth.

Differences which may occur in the corresponding startup and shutdown data can represent strong indicators of the crack shaft (Figure 11). During startup and shutdown, the torque is different, which may cause a crack to close or open and, consequently, introduce specific asymmetry conditions. It should also be taken into account that alignment, along with thermal conditions may be quite different at startup and shutdown.

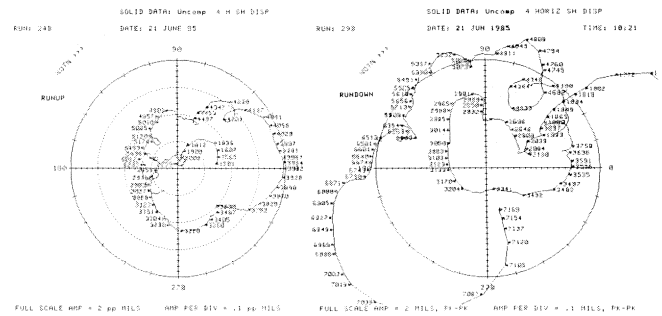

Figure 11. Polar Plots of the Second Harmonics ($2\times$) of the Vibrational Response of Cracked Rotor During (a) StartUp and (b) Shutdown, Indicates Significant Changes [9].

The $1\times$ and $2\times$ startup/shutdown polar plots provide significant shaft crack-related information. The thermal conditions and rotor misalignment, both affecting the shaft elastic unbalance (bow) should, however, be investigated as the first step.

In order to obtain information on shaft crack propagation from the time sequence of polar plots with documented rotative speeds, there should be observed the evolution:

- Slow roll vector changes of amplitude and phase, especially amplitude growth.
- $1\times$ (and $2\times$) resonant response vectors changes of amplitude and phase.
- A shift of the frequencies of the first (and second, third, . .) balance resonance to lower frequencies (shaft cracks cause a decrease of the system rigidity and, consequently, a slight reduction of resonant frequencies).
- Split of balance resonance peaks (crack-related shaft asymmetry causes differences in vertical/horizontal lateral modes; a new peak amplitude may occur at a slightly lower frequency than the original peak of the corresponding mode). Note that resonance peaks are also often split because of laterally nonsymmetric support stiffnesses.
- Changes in the shaft modal shapes.
- Unusual "leading" phase angle, which may be correlated to an opening/closing crack or thermal bowing of the cracked section. This has been observed so far on only one cracked shaft save [9], but is a vital indicator.

$2\times$ polar plots also represent a very important diagnostic tool. When there is shaft asymmetry and radial load, the $2\times$ polar loop will occur at the frequency corresponding to *each* balance resonance of the machine. The rotor speed is then at about one-half of the corresponding balance resonance speed.

Spectrum cascade plot representation of the vibrational signals from all shaft-observing probes is very informative in shaft crack detection, especially when taken during each startup and shutdown of the machine for further data comparison. Spectrum cascade plots clearly illustrate the correlation of all vibration signal frequency components with the rotative speed (Figures 12, 13, and 14). They are very useful for spotting the $2\times$ (and $3\times$, $4\times$, . . .) resonances, and their increasing amplitudes as the crack propagates.

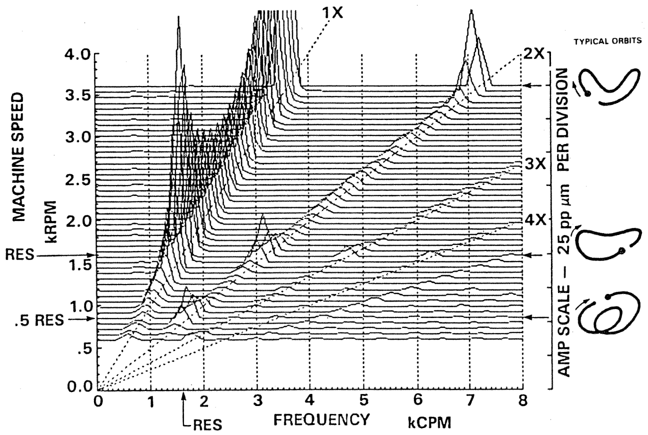

Figure 12. Spectrum Cascade of Lateral Vibrations During StartUp of a Machine with a Shaft Crack: When the Rotative Speed (800 cpm) is at half of the resonance speed (1600 cpm), the 2× component has its resonance. The orbit shows the typical inside loop generated by small 1× and high 2× components with little phase differences. When the speed approaches the first and second resonances, the synchronous (1×) components are dominant. Nonlinear effects creating the spectrum of higher harmonics have smaller amplitudes than these of 1× and the phase differences are higher.

The signal from one probe may not be sufficient for diagnostic purposes. Since spectrum cascade plots do not provide phase reference, more comprehensive information is obtained when the orbital motion at corresponding rotative speeds is presented together with the cascade plot. Observation of orbits, generated by XY probe measurements, is useful for revealing a shaft crack. The typical orbit patterns for a shaft with large 1× and 2× components are shown in Figure 12. A sequence of orbits generated during a startup of the turbogenerator with a broken shaft shown in Figure 15 is illustrated in Figure 13(a).

The orbit patterns can be correlated to the rotor displacement toward the direction of the load. (Displacements of the rotor centerline should be also separately monitored; a DC coupled signal provides the information.) When the rotor is displaced toward the direction of the external force, the shaft is held in a bent position, and its stiffness in the direction of the force increases. This results in shaft asymmetry: stiffness varies twice during each shaft rotation, causing the appearance of a 2× component. Changes in the orbit patterns when the radial load increases depend on the phase angle relationship between the 1× and 2× components. The latter are directly load related.

Difficulty in trim balancing often is a warning sign of a cracked shaft. Further analysis of the other symptoms is required to determine the root cause of the problem.

The following procedures are recommended to avoid missing the evidences of a shaft crack when trim balancing:

- Tape record vibration data during the startup and shutdown of each balance run for further comparison.
- Allow no more than two or three balancing attempts—crack propagation may be significantly accelerated at each unsuccessful balancing run. Further, if the rotor does not respond to proper balancing procedures, this is an indicator of a crack.
- Allow the balancing weights to generate a centrifugal force of no more than 50 percent of the rotor weight.
- Use extreme caution when calculations indicate that the trim weight should be introduced opposite of the location of the previous weight. This is another strong indicator of a shaft crack.
- Immediately reduce the startup and shutdown data in the form of spectrum cascade plots and 1× and 2× polar plots. Study these plots for indications of a shaft crack before the next run. Compare the data with results from previous runs.

TECHNIQUE TO ASSESS MACHINERY CONDITION USING MODE SHAPE ANALYSIS

A technique for the investigation of rotor cracks in turbine generators which have several solidly coupled rotors and up to 15 bearings has been developed. This technique is based on transfer matrix computations and it involves three steps:

Step 1—Background Data Base

Essentially, two forms of data are used to conduct this analysis:

- Observed 1× and 2× lateral mode shapes from before and after a shaft crack is suspected.
- Rotor design, bearing stiffness data, and other dimensional and material information for the machine.

Step 2—Mode Shape Generation

Using a transfer matrix computer program and the lateral mode shapes from Step 1 as a guide, two sets of undamped synchronous mode shapes are plotted for the machine. The mode shapes represent the computer predictions for normal and abnormal machine conditions. The theoretical, computer-predicted mode shapes and frequencies will not match the observed 1× and 2× bow mode shapes, because the deflection is normalized for each shaft section, and unbalance distribution along the shaft is not taken into account. Computer curve smoothing is entirely avoided so that mode shape changes are readily apparent.

Step 3—Analysis

The two sets of computer-predicted mode shapes (before malfunction and after crack-related malfunction) and the two sets of observed bow mode shapes are compared to each other. Changes or localized discontinuities in the mode shapes, between the normal and abnormal conditions, are analyzed. They may provide indications for further investigations.

In addition to examining the effects of shaft cracks on the rotor mode shapes, the results of this technique may be extended to identify rotor balance resonance frequencies, the sequence of rotor mode shapes as the rotating speed changes, the most effective locations for balance weights, possible probe locations, effects of excessive bearing clearances on the mode shapes, and identify multispan shaft mode shapes. This technique is equally applicable to smaller, two bearing/single shaft machines and it has to be recommended for every rotating machine.

An example of commputer-generated results for a turbogenerator set is illustrated in Figure 16. The effect of a crack on each mode is apparent.

CONCLUDING REMARKS

As the number of shaft crack incidents has increased, so, too, has the need for the early detection of shaft cracks. The reason is simple. The consequences of shaft cracks are catastrophic. Companies—from a monetary, safety, and public image point of view—cannot afford to have a disastrous shaft crack incident.

The instrumentation and diagnostic methodology is now available to detect shaft cracks before catastrophic failure of the machine occurs. Recent saves of turbogenerators and pumps prove that this methodology is effective for early detecting shaft cracks.

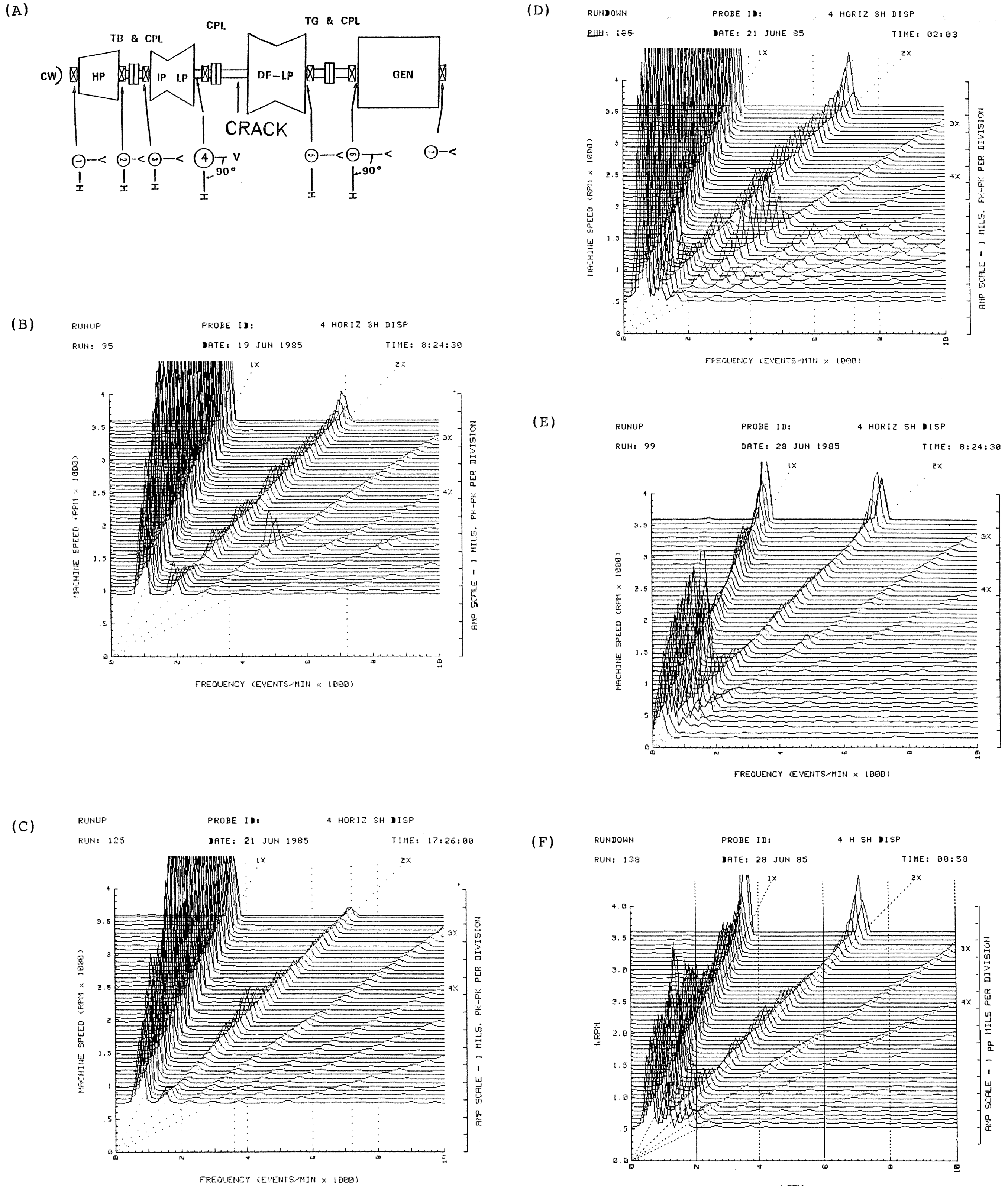

Figure 13. Cascade Spectra of the Rotor Vibrational Response Measured at Bearing #4, Horizontal of the Turbogenerator Unit Presented in Figure 7(a). Data taken on June 19 (run-up, Figure 7(b)), June 21 (both run-up, Figure 7(c) and shutdown, Figure 7(d)), and June 28 (run-up, Figure 7(e) and shutdown, Figure 7(f)). Note differences in the spectrum contents. In particular, note a significant increase of higher harmonic amplitudes during rundowns. Note resonances of the second (2×) and the third (3×) harmonics at low rotative speeds.

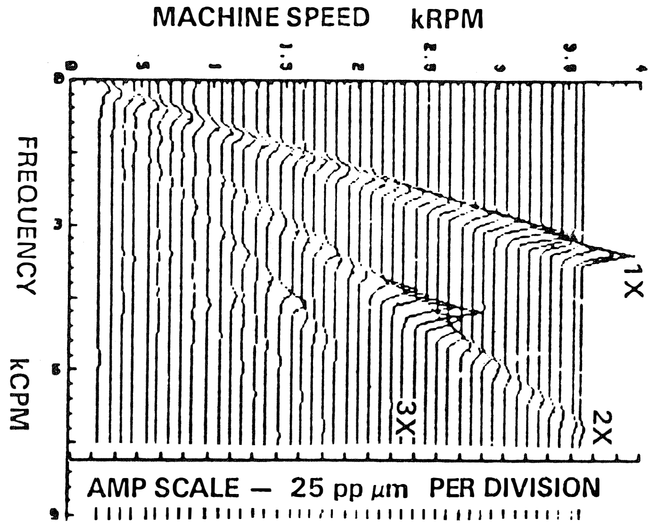

Figure 14. Cascade Spectrum of the Response of a Pump with Cracked Shaft. Note the increase of 1× vibrations, 2× resonance at 2400 cpm and 3× Resonance at ~ 1600 cpm.

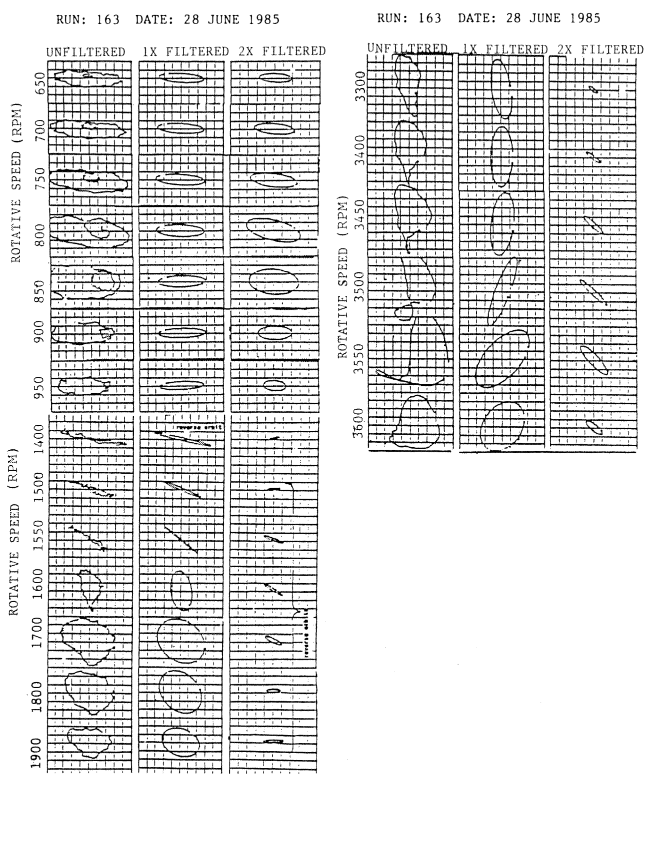

Figure 15. Sequence of Rotor Unfiltered and Filtered (1× and 2×) Rotor Orbital Motion Measured at Location #4 of the Turbogenerator Illustrated in Figure 7(a). Rundown on June 28. Note the amplitude and phase changes including backward precessing orbits.

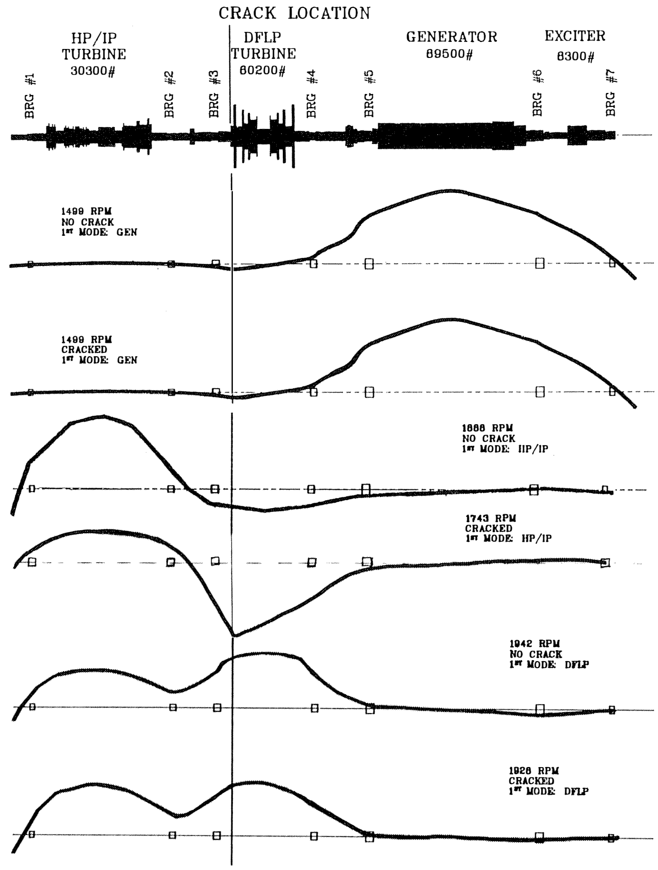

Figure 16. Mode Shape Analysis of a Turbogenerator Set. The circumferential crack of 5 in depth was assumed to occur at 17 in diameter shaft of the dual flow low pressure turbine.

REFERENCES

1. Muszynska, A., "Cracked Shaft Reference Material, 1933-1986," Shaft Crack Detection Seminar Proceedings, Bently Rotor Dynamics Research Corporation, Atlanta, Georgia (1986).
2. Thomas, G. R., "K-301 Syngas Compressor Train Gearbox Pinion Shaft Cracking," Shaft Crack Detection Seminar Proceedings, Bently Rotor Dynamics Research Corporation, Atlanta, Georgia (1986).
3. Laws, W. C., "A Brief History at Cracked Rotor Saves—Turbine Generators in the U.K. Rated 60 MW to 660MW," "Cracked Shaft Detection on a Refinery Boiler Feedwater Pump." "A Summary of Rotor Design Features Leading to Crack Susceptibility," Shaft Crack Detection Seminar Proceedings, Bently Rotor Dynamics Research Corporation, Atlanta, Georgia (1986).
4. Bently, D. E., "Vibration Analysis Techniques for Detecting and Diagnosing Shaft Cracks." Orbit, Bently Nevada, 7(1) (January 1986).
5. Jenkins, S., "Cracked Shaft Detection on a Large Vertical Nuclear Reactor Coolant Pump," Bently Rotor Dynamics Research Corporation Symposium on Instability in Rotating Machinery, Carson City, Nevada (1985).
6. Jones, R. M., "Vertical Reactor Coolant Pump Instabilities," Bently Rotor Dynamics Research Corporation Symposium on Instability in Rotating Machinery, Carson City, Nevada (1985).

7. Nataraj, C., and Nelson, H. D., "The Dynamics of a Rotor System with a Cracked Shaft," ASME Paper No. 85-DET-31, Cincinnati (1985).

8. Rauch, A., "Shaft Cracking Supervision of Heavy Turbine Rotors by FMM Method," Proceedings of the Third International Modal Analysis Conference, Orlando, Florida (1985).

9. Bently, D. E., "Data Reduction of Start-Up and Rundown Tapes of Florida Power & Light Port Everglades #2 Turbogenerator Set," Report, BRDRC (1985).

10. Ammirato, F., "Crack Detection in Large Steam Turbine Shafts," Report, EPRI NDE Center, Operated by J. A. Jones, Applied Research Company, Charlotte, North Carolina (1984).

11. Armor, A. F., Iman, I., Azzaro, S. H., and Scheetz, J., "On-Line Rotor Crack Detection and Monitoring System," EPRI Report (1984).

12. Davies, W. G. R., and Mayes, I. W., "The Vibrational Behavior of a Multi-Shaft, Multi-Bearing System in the Presence of a Propagating Transverse Crack," ASME Paper 83-DET-82 (1983).

13. Dimarogonas, A. D., and Papadopoulos, C. A., "Vibration of Cracked Shafts in Bending." Journal of Sound and Vibration, *91*(4), pp. 583-593 (1983).

14. Iman, I., "Method for On-Line Detection of Incipient Cracks in Turbine-Generator Rotors," U.S. Patent No. 4,408,294 (4 October 1983).

15. Mayes, I. W., and Davies, W. G. R., "Analysis of the Response of a Multi-Rotor-Bearing System Containing a Transverse Crack in a Rotor," ASME Paper 83-DET-84 (1983).

16. Tachegg, E. K., Ritchie, R. O., and McClintock, F. A., "On the Influence of Rubbing Fracture Surfaces on Crack Propagation in Mode III," Int. J. Fatigue 5(1), pp. 29-35, (1983).

17. Baumgartner, R. J. and Ziebarth, H., "Vibration Monitoring Criteria for Early Discernment of Turbine Rotor Cracks," Hartford, Connecticut, pp. 25-27 (1982).

18. Bently, D. E., "Detecting Cracked Shafts at Earlier Levels," Orbit, Bently Nevada, *3*(2) (1982).

19. Grabowski, B., "Shaft Vibrations in Turbomachinery Excited by Cracks," Proceedings of Rotordynamic Instability Problems in High Performance Turbomachinery Workshop, Texas A&M University, NASA CP2250 (1982).

20. Grabowski, B., and Mahrenholtz, O., "Theoretical and Experimental Investigations of Shaft Vibrations in Turbomachinery Excited by Cracks," Proceedings of Rotordynamic Instability Problems in High Performance Turbomachinery Workshop, Texas A&M University, NASA CP2250 (1982).

21. Muszynska, A., "Shaft Crack Detection," Seventh Machinery Dynamics Seminar, Canada (1982).

22. Nelson, H. D., "Analysis of the Dynamics of Cracked Rotor," Technical Report to DTH (1982).

23. Nilsson, L. R. K., "On the Vibration Behaviour of a Cracked Rotor," IFTOMM, pp. 515-524, Rome (1982).

24. Rogers, G. W., and Rau, C. A., "Analysis of a Turbine Rotor Containing a Transverse Crack at Oak Creek Unit 7," Rotordynamic Instability Problems in High Performance Turbomachinery Workshop, Texas A&M University, NASA CP-2250 (1982).

25. Bently, D., "Breakthroughs Made in Observing Cracked Shafts." Orbit (1981).

26. Ignaki, T., Kanki, H., and Shiraki, K., "Transverse Vibrations of a General Cracked Rotor Bearing System," ASME 81-DET-45, Design Engineering Technical Conference, Hartford, Connecticut (1981).

27. Lindinger, R. J., Curran, R. M., "Experience with Stress Corrosion Cracking in Large Steam Turbines," Paper No. 7, Corrosion 81, NACE (1981).

28. "Diagnostic Evaluation of the Excessive Vibration Levels at the Unit No. 7 Turbine Generator." Burlington Generating Station, PSEG Research Corporation, Research and Testing Laboratory, Report No. 68401 (1985).

29. Consolidated Edison, New York ("Big Allis"), Private Communication.

30. Ohio Edison, Ohio (Sammis Plant No. 3, Miami), Private Communication.