ELECTRIC POWER RESEARCH INSTITUTE

Guidelines for the Uprate Design of Turbine-Generator Rotor Support Systems

2011 TECHNICAL REPORT

Guidelines for the Uprate Design of Turbine-Generator Rotor Support Systems

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Abstract

The focus of this guidelines document is the rotor support structure for uprated turbine-generator (T/G) systems.

The overall objective of the guidelines is to provide utility personnel with the guidance needed to validate the design of the components prior to a plant uprate/upgrade to ensure that the unit will operate without unplanned and unanticipated issues related to the rotor bearings, bearing support structures, and pedestals. The guidelines are applicable to both nuclear and fossil T/G rotor support structures.

A review of T/G rotor support system design is made that includes the loading and significant technical challenges presented during power uprate redesign. Industry experience with power uprate projects is reviewed, and the most significant issues are discussed—including rotor vibration and loss of last-stage turbine blades. Recommendations are then made for utilities to pursue in both analytical and testing activities prior to and during a power uprate.

Keywords
Bearing support structures
Pedestals
Power uprate/upgrade
Rotor bearings
Turbine-generator (T/G)

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Section 1: Introduction

Many utilities are seeking to uprate their current turbine-generator (T/G) sets to increase capacity rather than seeking to build new units. This has become a particularly attractive option for nuclear generating stations, in which turbine and generator modifications can lead to capacity increases up to 20% in much less time and at far less expense. Cost per MW for power uprate is 15–50% of new construction alternatives and requires half the time (or less) to implement. While these upgrades and uprates increase efficiency and capacity, they also present significant reliability challenges and can lead to negative consequences if not properly and comprehensively assessed during the uprate planning and implementation stages.

Utilities often entrust the entire design, analysis, and installation of the upgraded T/G equipment to the manufacturer(s). However, because the uprate process has proven to be problematic in some cases, utilities now desire to play a more proactive role. The ability to participate in or manage the different steps of the upgrade is compromised by a lack of information regarding the suitability of these components and structures for uprated conditions. Additionally, utilities frequently have inadequate information regarding the processes required to assess the structural integrity of these machines and may be unable to request or specify the appropriate evaluations and reviews. This document presents guidelines intended to fill this void.

The planning and execution of power uprate is the subject of an EPRI program that has produced two previous guidelines [1, 2]. The present document is intended as a companion to these guidelines and focuses on technical issues associated with the T/G rotor train and its support structure.

1.1 Background

There have been several failures associated with power upgrade projects. The most recent and probably the most concerning of these to the nuclear utility industry is the failure at D.C. Cook Unit 1 that resulted in the separation of five 56-in. (14.22-mm) long last-stage turbine blades (LSBs) in September 2008. The root cause of the failure was originally attributed by the utility company to a blade-rotor attachment design that failed to provide adequate stress margin in the L-0 blades [3]. There were thought to be many contributing factors, including torsional modes in the vicinity of 120 Hz and blade rubbing suspected of producing high cycle fatigue cracking. Torsional modes close to 120 Hz were discovered by analysis during installation late in the uprate process. This discovery led to L-0 blade modification with the addition of blade tip weights. The failure resulted in significant damage to the T/G and surrounding equipment and an extended loss of electric generating capabilities.

Although the exact nature of the root cause of the Cook failure remains somewhat controversial, the use of longer blades in the last stages of the LP turbines to extract additional power from the expanding steam is one of the biggest challenges that must be addressed when selecting turbines for power uprate. These challenges are evident in the failure of LSBs in other low-pressure (LP) turbines. Examples include Detroit Edison Fermi and South Texas Project (nuclear); TransAlta Centralia, ESKOM Duvha, and TVA Cumberland (fossil); and the Doswell Cogeneration Station (gas turbine). All but Centralia involved extensive damage to the turbines and auxiliary equipment.

1.1.1 Summary of Power Upgrade Projects

Across the installed base of fossil and nuclear power generating stations, many power uprates have been completed. This uprate activity includes a wide range of increased capacity, from 1 or 2% to increases as high as 20%. For nuclear installations, utilities have implemented power uprates since the 1970s. As of April 2011, the NRC had approved 139 uprates, resulting in a gain of approximately 6,020 MW at existing plants. Collectively, these uprates have added generating capacity at existing plants that is equivalent to about six new reactors. Lists of these NRC approved uprate projects as well as a list of uprate applications under review and anticipated are documented in [4].

The design of every U.S. commercial reactor has excess capacity needed to allow for an uprate, which can fall into one of three categories: measurement uncertainty recapture power uprates, stretch power uprates, and extended power uprates [4].

1. Measurement uncertainty recapture power uprates are power increases less than 2% of the licensed power level and are achieved by implementing enhanced techniques for calculating reactor power. This involves the use of state-of-the-art devices to more precisely measure feedwater flow, which is used to calculate reactor power. More precise measurements reduce the degree of uncertainty in the power level, which is used by analysts to predict the ability of the reactor to be safely shut down under possible accident conditions.
2. Stretch power uprates are typically between 2% and 7%, with the actual increase in power depending on a plant design's specific operating margin. Stretch power uprates usually involve changes to instrumentation settings but do not involve major plant modifications.
3. Extended power uprates (EPU) are greater than stretch power uprates and have been approved for increases as high as 20%. Extended power uprates usually require significant modifications to major pieces of nonnuclear equipment such as high-pressure turbines, condensate pumps and motors, main generators, and/or transformers.

EPU (that is, power uprates greater than 7%) is associated with nuclear power plants much more than with fossil units. One reason for this is the environmental (carbon emissions) regulations that inhibit uprating of fossil plants in the United States. In Canada, where environmental regulations have traditionally been less stringent (although this is now changing), EPU of fossil plants is more common. Examples include uprates at Sundance Units 3, 4, 5, and 6 in Alberta for a total of 165 MW. Another plant, Keephills, is planning a 46-MW uprate in 2012. All of these uprates in Canada are for coal-fired plants, and—if one includes the uprates at the Centralia units—the total uprate capacity associated with these projects is 1340 MW.

In the U.S. nuclear industry, EPU has been much more prevalent for BWRs than PWRs. This is because there was more conservatism in the original design of BWRs. Unlike PWRs that had been developed for smaller applications prior to the onset of commercial nuclear power generation, the "boiling water" reactor concept was new and relatively untested at that time [5]. Today, the greater margin in BWR design allows for increased flow and pressure that can result in EPU as high as 20%. However, the market for EPU of PWRs is expected to expand with the development of advanced fuel pin materials.

Problems with power uprate are not confined to EPU projects. The data reported in [4] show that the power uprate at D.C. Cook was only 1.66%. The potential for failure resulting from any size power uprate project is evident in the problems experienced by the industry over the past several years.

1.1.2 Problems Encountered with the T/G Support Structure

Although some power uprate projects are conducted with original plant equipment, many involve replacing turbine and generator rotors and frequently the stators with new equipment that may be larger, heavier, of a different geometry, and in many cases designed by a different manufacturer than the original T/G OEM. Large rotating apparatus such as turbines and generators are subject to significant operational issues, including vibration that can potentially damage the equipment itself, the structures that support it, and other equipment in and around the machines. Modifications to these machines for any reason must be carefully executed to avoid potential vibration problems and catastrophic failure in the case of blade loss.

T/G rotor systems often operate near critical speeds, both lateral and torsional. Consequently, any change in the rotor dynamics can impact the proximity to critical frequencies, which can lead to dangerous vibration levels. For example, it is well known that torsional vibration of the rotor system can excite harmonics in the turbine blades, leading to blade failure by high cycle fatigue. And it is well recognized that any change in the stiffness or mass of the rotor system or its support structure can have an impact on the (lateral) natural frequencies. Therefore, rotors must be analyzed to ensure that they are operating sufficiently away from natural frequencies to preclude such occurrences. Even comprehensive analysis can at times fall short of providing the needed assurances. Because of the complexity of these machines, even the most rigorous design analysis may have a tolerance of a few Hertz on the resulting frequency calculations. Consequently, dynamic testing is often required to calibrate the analytical treatment and thereby ensure that these computational tolerances do not facilitate operation too close to a natural frequency.

Critical components that must be analyzed during this process include those that make up the support structure for the T/G rotors, inclusive of the bearings, bearing support structures, turbine and generator pedestals, and even possibly the foundation. These components must support the new apparatus weights, short circuit torque, and other fault conditions and may need to be repaired, strengthened, or replaced during the upgrade.

Because all of the support components are directly coupled, they all interact statically and dynamically. Static deflections are usually not of concern because they are routinely measured and compensated for during installation and balancing. However, different masses from turbine uprate redesigns can cause deflection changes that can cause misalignment, rotor crank, increased bending stresses, and increased risk of fatigue. Dynamic interactions can cause significant damage to any of the components, both rotating and stationary. All of these issues represent challenges to the design and must be carefully considered throughout the uprate process.

1.2 Scope

The focus of this guidelines document is the rotor support structure. The overall objective of the report is to provide utility personnel with the guidance needed to validate the design of the components involved in a plant uprate/upgrade to ensure that the unit will operate without unplanned and unanticipated issues related to the rotor bearings, bearing support structures, and pedestals. The guidelines are applicable to both nuclear and fossil T/G rotor support structures.

1.3 Purpose of This Guidelines Document

The objective of this guidelines document is to provide the following with regard to the T/G and its support components:

- An overview of the different elements of the T/G rotor support system, their design, and the common engineering practices employed in both fossil and nuclear plant installations
- Sufficient information to understand the design process that the vendor should implement prior to upgrade
- Sufficient information to enable utility personnel to ensure that the correct assessments are being performed as well as guidance on the appropriate and acceptable results of such analyses
- Guidance on field testing and monitoring that should be conducted pre-and post-power uprate implementation
- A design review checklist for implementation during the design process

Section 2: Historical Perspective

This document is primarily directed toward the larger units for which the incentive for power uprate is added capacity (and/or efficiency) at lower than new construction cost. This effectively includes all of the nuclear fleet and many large fossil units. As a starting point, a look at the evolution of large turbo-machinery likely provides insight into the issue of how to extract more output from a given machine. That is, a fundamental understanding of how machines evolved from 50 MW to 1200 should provide some insight on how to extract an additional 100 MW from an existing 1000-MW unit.

2.1 Generator Development

Looking first at the generator, the history of the evolution of generator capacity provides significant insight into the means of uprating a machine to provide additional output capacity. Over the years, major increases in the T/G set capacity nearly universally have been associated with major changes in the ventilation of the generator. Historically, once the means to increase generator capacity had been defined, the increased power needed to drive the generator (that is, the turbine output) could be achieved simply by adding turbines and steam supply in the absence of any other viable means of increasing power. So, the generator dictated the rate at which unit size increased.

Very early machines were open air cooled, and the means of extracting heat was by conduction of the heat through the stator and rotor winding insulations, conduction through the stator iron and rotor steel, and then convection from the rotor and stator surfaces to the air ventilation medium. This conventional cooling of the generator (that is, direct cooling by convection only at exposed surfaces of the rotor and stator and no direct cooling of the coils themselves) continued for many years—yet step improvements in capacity were still possible. One major innovation that led to the stepwise capacity improvements in T/G unit output capability was the move away from air as the ventilation medium. Hydrogen gas was found to be an excellent cooling medium, providing both lower density for easier circulation and higher heat capacity. The use of hydrogen provided several additional output capacity improvements as the gas pressure went through the stepwise increases, initially from 15 psi to 30 and 45 psi (103 kPa to 207 and 310 kPa), and so on.

Following this was the use of improved ventilation schemes such as more direct cooling of the rotor windings via sub-slots beneath the windings and finally direct, inner cooling of the windings—both for the stator and the rotor. For the stator, gas inner-cooling involved providing hollow tubes intermixed with the copper winding strands of each coil. For the rotor, inner cooling evolved by providing various means of passing cooling gas directly through the copper coils, some via axial flow, others radial flow, and others a combination of the two. Ultimately, the stator coils moved to fluid inner cooling, which is true “inner” cooling: the copper strands are hollow and directly cooled by water flowing through the strands. Some early inner cooling designs used oil as the cooling medium, but water quickly replaced oil and became the industry standard. In Europe, some manufacturers even moved to direct water cooling of the rotor windings, although this was never pursued extensively in the United States.

Improvements in insulation materials also provided potential for improvements in generator capacity. Materials that provide the requisite electrical insulation yet with improved heat conduction capacity or in thinner layers, or a combination of the two, provide meaningful improvements in two ways. First is the ability to more effectively move heat by conduction to exposed convective heat transfer surfaces. Second, thinner insulation provides the capacity to support additional conductor cross section within the same space. Simply moving to insulation materials that can withstand higher temperatures without material degradation can also be useful; however, the insulation temperatures are not the only consideration in setting maximum allowable temperatures at various locations within the generator. Other considerations may dictate temperature limits.

So, for the generator, step improvements in output have historically been related to step improvements in heat dissipation capacity. Consequently, for any meaningful uprate of a given unit, the likely modifications will similarly include one or more improvements in the ability to remove heat from the generator and to maintain all components at or below their temperature limits. This may well involve the use of higher rated insulation materials for the stator and/or rotor windings. It may also involve redesign of the stator coils to provide an improved cooling scheme, for example, from gas inner cooled to water direct-cooled coils, increased cross section of the copper components, and/or higher temperature insulation. The uprate may include modification of the rotor windings from conventional cooling to inner cooling and/or the use of improved insulation. The uprate can even involve modifications to support increased gas pressure.

2.2 Advances in Steam Turbine Technology

For the turbines, historically the solution for additional capacity could well have been to merely add turbines, so the turbine was never the limitation in single-generator units. That is, a simple solution for adding the power necessary to drive increasing generating capacity is to add more turbines in tandem—a method that was used successfully in new plant construction. However, because of the constraints of existing plant configurations, adding more turbines is not applicable for power uprates. What is required is increased steam mass flow and expansion within the envelope of the plant and major components, involving choices of new turbine designs or modifications to increase thermodynamic efficiency and, in some cases, increase steam pressure and temperature.

The considerations for turbine evolution to larger capacity units are, for the most part, quite different from those for generators. Whereas for the generator the main limitation was heat dissipation, for the turbine a primary factor limiting size was related to material technology. To a much greater degree than for the generator, turbine capacity was dictated by material properties and fabrication processes. For example, stresses are typically related to blade diameter, which in general dictates the rotational stresses, and to temperature gradients and the transient nature of thermal excursions. For the generator, because temperatures are generally much lower, only the mechanical component of stress is usually considered. The turbines are therefore generally subjected to higher stresses, which in turn further limit their tolerance to flaws. Additionally, the temperatures to which fossil turbine components are subjected are sufficient to drive certain material degradation mechanisms; of these, high-temperature creep and temper embrittlement are the most noteworthy. Turbine components are also exposed directly to steam (which can contain contaminants that can lead to environmentally assisted mechanisms, such as corrosion, erosion, and stress corrosion cracking [SCC]), and so certain aspects of turbine life expectancy are related to the steam generator and the ability to deliver quality steam—or the corollary, that is, in the ability of the turbine to withstand certain likely chemical events and operating conditions. Because of these factors (component chemistry and the processes used to form the various components, such as ingot pouring and forging), the resultant properties are more critical in general for turbine components than for the generator components.

Obviously, another means of increasing turbine output involves increasing steam temperature and pressure, which is directly related to boiler/steam generator design and output. However, the resulting potential output increases must be considered along with these material considerations.

Increased turbine output involves in large part the efficiency with which the blades extract useful work from the steam, which is directly related to blade design. The designs of the turbine inlet and exhaust hood also have a significant effect on the steam mass flow. The growing availability of improved design and design analysis tools marked the beginning of an era highlighted by improved analytical tools to more accurately define stresses, temperatures, dynamic interactions, and steam flow and to improve the precision with which other similar design calculations could be made. Prior to the use of such tools, design analysis was largely a manual endeavor, featuring hand-calculated approximations, and with the true design details evolving largely by extrapolation from earlier successful designs. One of the issues that eventually arose because of this design approach involved the duration of the design, manufacture, and installation schedule. Because of this lengthy cycle, any problems with the extrapolated designs were typically not identified for a number of years and therefore until many units and possibly even further extrapolated designs were well through the process. The use of advanced analytical tools, primarily computational fluid dynamics (CFD) and finite element analysis (FEA) for mechanical, thermal, electrical, and mixed assessments, provides the means not only to remove conservatism from the designs by “sharpening the pencil,” so to speak, but also the ability to analytically test the designs for compliance with design limits prior to manufacture.

For the turbines, the increase in output likely will be achieved by redesigning the steam path. This may include modified blade design for higher efficiency, particularly in the latter rows of the low-pressure sections. It may involve new high-pressure rotor(s) with improved blade designs and correspondingly modified stationary blade rings and the conversion from partial arc to full arc admission. The redesign activity for power uprate will certainly take advantage of the latest advancements in analytical techniques to optimize the designs. Similarly, rotor fabrication techniques have progressed so that new rotor forgings will take full advantage of disks that are all fully integral to the rotors (as opposed to shrink-fit assembled disks) and modern forging practices capable of producing super clean rotor forgings. These advances in analytical techniques and fabrication processes allow the use of substantially increased blade lengths in the last rows of the turbine, permitting greater expansion and power extraction as well as eliminating other life-limiting factors such as SCC at the bore and keyway region of shrink-assembled disks and life-limiting flaws in the rotor forging body and central bores, which lowers rotor centerline stresses.

Section 3: Rotor Support System Design

The historical perspective provided in Section 2 describes how advances in engineering technology as applied to T/Gs have resulted in units that are more efficient and generally of a larger size in MW output. During power uprate modifications, all of these advances in technology may be used. Therefore, for the T/G rotor support system, the questions become the following:

- How do these design modifications impact the different components of rotor support under the uprated conditions, normal operating conditions, and a number of defined fault conditions?
- What assessments should be performed to verify that the rotor support system will perform satisfactorily for the intended life of the machines?
- What tests can (should) be performed to validate the integrity of the rotor support system?

3.1 Major Components of the T/G Support Structure

The rotor support system includes everything from the rotor bearings to the foundation mat. The three major components are the bearings, bearing support structure or bearing pedestal, and the pedestal foundation, which is the large stand-alone reinforced concrete structure that supports the entire rotor train.

3.1.1 Bearings

Starting with the T/G components and working toward the foundation mat, bearings can be one of a number of designs that are typically specific to the manufacturer. However, all bearings for large turbines and generators of the size that would typically be the subject of an economical power uprate use hydrodynamic fluid film journal bearings in which the cylindrical rotor journal is surrounded by a cylindrical support having a clearance gap filled with a hydrodynamic fluid upon which the rotor is lifted during rotation. Hydrodynamic force is created by shaft rotation as the rotor moves along the viscous film and creates an oil wedge that lifts the rotor away from metal-to-metal contact between the shaft journal and the bearing. The bearing surface itself is covered with a soft babbitt material, which is a tin-/lead-based material that typically contains about 88% tin, 7.5% antimony, 4% copper, and the remaining 0.5% lead. The softer babbitt material is used so that metal-to-metal contact does not mar or otherwise damage the rotor journals.

Bearings are typically spherically seated such that they are self-aligning with the shaft, which collectively with the other shafts making up the tandem rotor train is aligned to account for gravitational sag and the changes that occur due to thermal effects as the unit heats. Tilting pad bearings are the prevalent choice for some manufacturers for nonnuclear utilities, particularly for generators and certain high-pressure turbine rotors. Tilting pad bearings are made up of a series of segments around the journal that are free to tilt or pivot such that a positive film pressure is created at each pad. Tilting pad bearings are more expensive but provide the advantage that they are stable at much lower bearing loads. Nuclear utilities generally use round or elliptical bearings, which use a ball-seat to mate to the journal.

3.1.2 Bearing Pedestal

It is important to note that rotor bearing support structures (that is, the bearing pedestals) can be either integral to or independent of the turbine casing, and they vary greatly from one manufacturer to the next and from one machine type to the next. A typical integral pedestal is shown in Figure 3-1. For the generator, however, the bearings are always supported within massive bearing brackets that are bolted to the ends of the generator frames. This design feature is dictated by the generator design in general, which features a cylindrical stator core housed in a cylindrical frame. For the turbines, on the other hand, the fully assembled rotor is not a simple cylinder that can be removed axially. The turbine casing is a split cylinder from which the top half must be removed to enable removal of the rotor by lifting it vertically from the bottom half of the casing. This configuration lends itself more readily to the two options, that is, integral or independent bearings. It is also important to note that in most cases, each rotor within the T/G set has two bearings—one at either end of the rotor. However, it is not unusual to also find nonnuclear units in which two turbine rotors are supported by only three bearings, one on either end of the pair, with a single bearing between them.

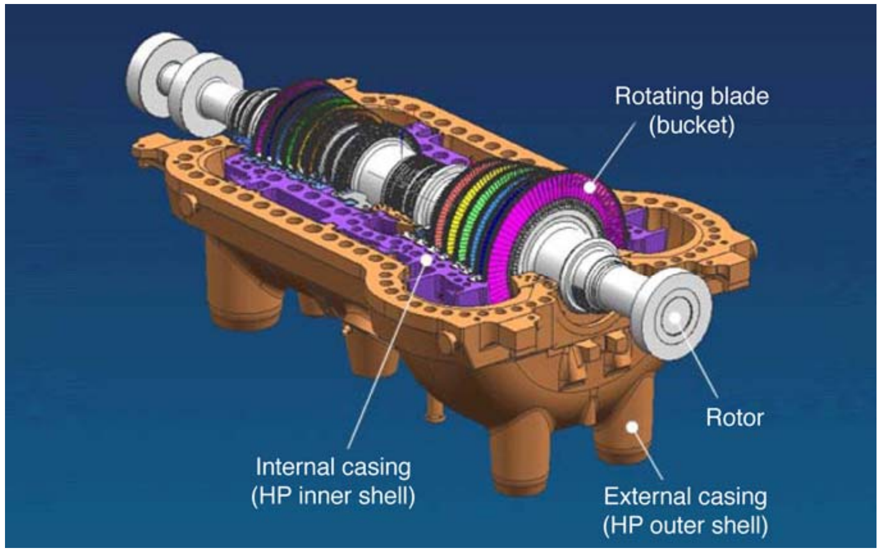

Figure 3-1
Example of Bearing Pedestal Integral with the Turbine Casing [6]

An important part of the bearing pedestal is the bearing housing that supports the rotor vertically (see Figure 3-2). The strength of the housing and bolting that secures the top and bottom sections is most important especially for impact loadings that arise from blade loss events and other accident loadings. As these loads are most often modified by uprate redesign, the bearing housing and supporting structure become key elements that require careful attention. Quite obviously, these various designs significantly impact the transfer of static and dynamic loads to the foundation. Consequently, they become a key part of power uprate considerations.

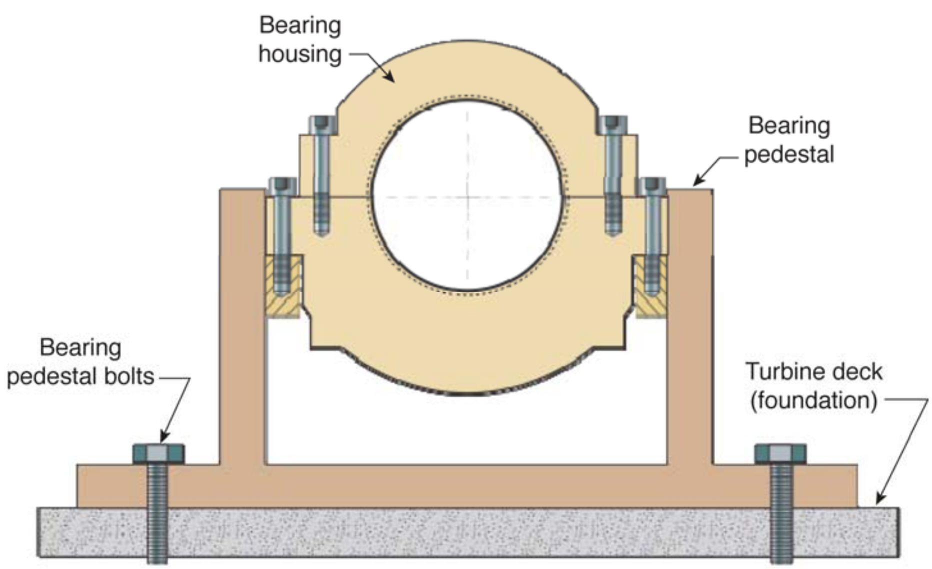

Figure 3-2
Bearing Pedestal and Bearing Housing Design

3.1.3 T/G Island Foundation

The T/G foundation is a stand-alone structure isolated from the remainder of the power station that supports the entire T/G system. There are primarily three types of designs: concrete, steel, and spring mounted. In this overview, the focus is on concrete foundations as this design is almost universally employed in the United States. The steel and, to a lesser extent, the spring-mounted designs are used in Europe and other countries

Reference [7] provides a good treatment on the basic considerations involved in the design of T/G foundations and is highly recommended reading for those desiring a broader understanding of foundation design. The same nomenclature used in the referenced document is shown in Figure 3-3 and is used in this report to enable the reader to easily move from one document to the other without confusion.

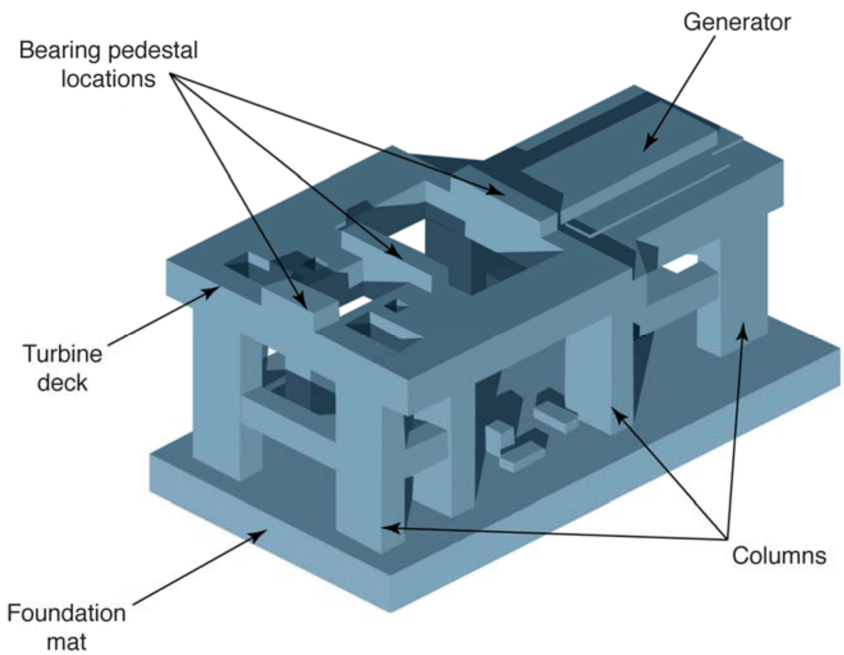

Figure 3-3
Typical Concrete T/G Foundation

T/G foundations were historically "designed" in the United States by simply providing a relatively massive foundation and then later by extrapolating from those that operated successfully. In Europe, the prevalent design going back many years has been the low-tuned foundation, that is, one in which the fundamental vertical natural frequency of the foundation is less than the frequency of the machine by a factor of 3 or more. If this had been the design basis in the United States, it would mean a foundation fundamental vertical natural frequency of 20 Hertz or less, based on 60-Hertz operating frequency in this country. However, the U.S. designs differ from the European designs due to one major difference involving the condenser. In Europe, the condenser is rigidly attached to the turbine such that there is no variation in the applied load during service relative to the load applied when not in service. In the United States, the practice is to support the condenser on the foundation at its base and to then provide an expansion joint at the connection of the condenser to the turbine. While this relieves the dead load, it means that there is a change in the load as the unit comes into service and the vacuum is created in the condenser. This arrangement causes the deck to carry this vacuum load. Because this load is applied after the rotor is aligned (that is, because this load represents a change when going into service), the deck and deck substructure that make up the T/G foundation must be much more massive and rigid to support this additional load change without compromising rotor alignment. This type of foundation, which has been labeled a *conventional foundation*, is much more rigid than the low-tuned foundations used elsewhere. Even so, the U.S. conventional foundations are still low-tuned—albeit not as low-tuned as the European counterparts. The conventional foundations found in the United States typically have vertical natural frequencies that are still less than the operating frequency but more typically by a factor of between 1.4 and 3. For the 60-Hertz machines, this means foundation vertical natural frequencies between 20 and 40 Hertz—still much lower than the operating frequency of the machinery. The classic definition of a high-tuned foundation requires that the foundation fundamental natural frequency be greater than the machinery frequency by a factor of approximately 1.4, which for the 60-Hertz machine would mean it would be above approximately 85 Hertz. The extremely massive nature of a foundation designed to provide this rigidity makes these foundations extremely uneconomical. The criteria for foundation design in the United States have largely evolved based on past successful practice and have remained empirical. Tests have been used to determine natural frequencies and validate designs, but few attempts have been made to perform detailed machine/foundation dynamic analyses with which to analytically define frequencies and serve as the bases for foundation design.

3.1.4 Foundation and Bearing Foundation Interface

The items defined above and making up the support structure, from the bearings to the turbine deck, are integral to the scope of supply of the T/G manufacturer, while the foundation design is provided as part of the architect/engineer (A/E) firm's scope of supply. The interface between the T/G manufacturer and the A/E responsible for the foundation typically occurs at the junction of the T/G sole plate, which is provided by the T/G manufacturer and the concrete structure to which the sole plates are mounted. The sole plates are grouted and bolted to the foundation. Beyond this structural interface, the T/G manufacturer defines dimensions, applied loads, and other similar pertinent information for the turbine(s), generator, exciter, and governor and for all of the piping, connections, and ancillary equipment requirements for the A/E to enable the A/E to properly design the support structure.

The interface between the T/G set and the foundation additionally incorporates features that allow for the relative motion between the T/G set and the foundation resulting from thermal expansion of the various T/G components, while at the same time maintaining alignment of the tandem arrangement of the shafts. This typically involves lateral keys that are anchored to transverse beams at various locations along the length of the T/G assembly to prevent transverse motion at the centerline of the rotor train but allow motion in either direction away from the centerline. These transverse keys permit axial motion but limit lateral motion along the centerline of the rotors to prevent misalignment due to thermal expansion of the machines relative to the foundation. Keys are typically at each bearing support (that is, between the foundation and each end of the generator frame). Similarly, they are at each end of the turbine casings having bearing pedestals integral to the turbine casing and otherwise at each pedestal that is independent of the casing.

There are also keys to guide axial motion. These are typically limited in number to a single key for each of the larger elements in the tandem arrangement. Because the elements are linked only via the rotors, the casings are free to move—one relative to the next—so each may be keyed, but typically only the larger elements are. A typical axial key arrangement might involve an axial key pair, one on each side for each LP turbine, positioned at or near the axial center of the element, plus a pair on the generator. At some point within the axis of the machine is a single thrust bearing that fixes the axial position of the rotor to the stationary components at that point along the length of the entire assembly. This is designed in conjunction with the axial keys that anchor the equipment to the foundation such that the motion of the stationary components in each direction—relative to the foundation from the keys—corresponds to the motion of the rotors relative to the stationary components to prevent interferences from occurring during normal operation or potential fault conditions. So, in the design of the unit, these are very important considerations—axial clearances, dimensional tolerances, axial key locations, the location of the thrust bearing, axial expansion of the rotor train (which, of course is linked for the full length), and axial expansion of the stationary components relative to the axial keys—such that binding and/or rubs do not occur.

3.2 Loadings

Loads applied to the foundation are derived from a number of sources. Loads originating from the rotor itself are most important as they must be reacted to at the bearings and bearing support structure. Vibration, rotor imbalance, and loss of blade loads are particularly important to quantify and consider in the rotor support design. This section reviews all the loadings associated with the rotor support system with an emphasis on those loads that deserve extra consideration during power uprate redesign activity.

3.2.1 Bearing Loads

Loads that are used in the design of the bearings are gravity, temperature, pressure, misalignment, and vacuum loads. The primary design consideration is the impact of these loads and load changes on vibration. Temperature and pressure increases arise from the increased weight of the rotor and are used in the sizing of the hydrodynamic fluid film bearings. Misalignment of the rotor shaft during installation and operation can influence the pressure loads on the bearing, and limiting misalignment values are used in the design calculations. Vibration loads are based on assumed out-of-balance levels for rotors normally associated with trip level vibration settings. They are computed using finite element (FE) and continuum models for determining critical speeds. Vibration loads affect more than the bearings as they are transmitted into the bearing supports and eventually the foundation. A more detailed discussion of vibration loads is presented in Section 5.

3.2.2 Ordinary Structure

Dead loads. Dead loads include the weight of the machinery supported on the foundation; the weight of the foundation itself; and the weight of other auxiliary components and equipment, such as valves, boiler feed pumps, and apportioned weight of piping not otherwise supported, that is supported by the foundation. Condenser dead load can vary, depending on the condenser support arrangement, of which two different types prevail. In the first condenser support design, the condenser is rigidly supported on the mat and an expansion joint, in place between the condenser and the turbine exhaust nozzle, isolates the dead weight from the turbine. In this design, the entire weight of the condenser is borne by the mat. In the second condenser support design, the attachment of the condenser to the turbine exhaust nozzle is rigid, with spring supports between the mat and the bottom of the condenser. The springs are adjustable such that they can be set to transmit more or less of the dead weight to the turbine and/or nonuniformly to compensate for eccentric loads, for example, water pressure loads. In this arrangement, all of the condenser dead load is still carried by the foundation; however, some is transmitted to the deck and some directly to the mat, depending on the spring adjustments.

Piping and valve reactions. Loads can be transferred through the T/G equipment to the foundation via any number of possible piping and valve connections. These loads result from cold set in the piping, fluid dynamic loads, seismic events, and thermal expansion of the connected components. When various pipes that are attached to the turbines are heated, they expand thermally and impart loads on the turbines. Piping loads are generated by all piping connected to the turbines inclusive of main steam piping, hot reheat steam piping, cold reheat steam piping, extraction steam piping, and crossover piping.

Thermal expansion and friction forces. Temperature changes in the turbine and generator cause expansion and contraction of various components within the elements relative to one another. The most significant of these is obviously the relative thermal motions associated with startup and shutdown of the unit. As the machine heats up, the entire rotor train expands axially. However, because the rotor train is fixed axially at only a single point—the thrust bearing—and because there are no other axial loads or constraints, it is free to expand and contract unrestrained in oiled bearings. This free expansion does not apply to any load to the stationary components or, therefore, to the foundation other than to the net axial thrust load carried by the thrust bearing.

The stationary components themselves also undergo significant transverse and axial expansion as they heat during unit startup. The various elements in the T/G set are all keyed along the axial centerline such that the lateral expansion occurs symmetrically about the axial centerline and maintains alignment of the tandem arrangement of shafts.

Temperature factors. These include the stresses and deflections within the foundation caused by the thermal gradients from heating portions of the foundation near the T/G equipment plus weather-related environmental effects. Environmental effects include direct solar exposure of outdoor units, significantly varying time-dependent thermal exposure such as that experienced at night relative to that experienced during the day (again, mainly for outdoor units), and similar indoor units that experience significant thermal exposure as a function of the time of day and/or at different regions of the foundation.

Shrinkage and creep. This is included for the sake of completeness. Loads due to shrinkage and creep result in reinforced concrete foundations as the concrete cures. However, for T/G foundations, the installation and alignment of the T/G elements typically occurs two to three years after the concrete foundation is poured, providing sufficient time for all significant shrinkage and creep to have occurred prior to initial alignment. Additionally, even where absolute deflections may be significant, the relative deflections between bearings are inconsequential. As a result, for T/G foundations, this is not included in any analyses performed.

3.2.3 Normal Operation

Normal torque load. Torque loads are produced in the turbines and the generator. In the turbine, the normal torque load derives from the steam force—essentially a drag factor that consequently opposes rotation direction. In the generator, the normal torque load derives from the electromagnetic pull of the rotor acting on the stator and consequently is in the same direction as the rotation. The magnitudes of the normal torque loads in the turbines and in the generator depend on the rotational speed and the power output of the generator and the turbine section. Because power output of the turbine section is minimal until the generator comes on line—which can occur only when at full rotational speed—the speed part of the equation is a constant, and the magnitudes for a given unit become dependent solely upon the operating output of the unit. These loads are transferred through the sole plates of the deck to the remainder of the foundation.

Condenser vacuum load. For the first condenser support design case presented above (that is, the design in which the condenser is rigidly attached to and supported by the mat), the vacuum inside the condenser causes a net force on the turbine of a magnitude several times the weight of the condenser, depending, of course, on the rigidity of the expansion joint. For an extremely flexible expansion joint, the vacuum is primarily balanced by strain in the condenser housing. For an extremely rigid connection to the turbine, some of the vacuum load produces strain in the condenser housing and transfers load to the turbine. In any such case, the vacuum load that is transferred to the turbine then passes to the foundation in the form of a deck live load.

For the second design case in which the condenser is rigidly mounted to the turbine, the magnitude of the load that is transferred to the turbine depends on the spring mounts under the condenser. Still, the magnitude that is transferred to the turbine is then transferred to the foundation in the form of a deck live load.

These assessments of load sharing, and therefore the dependence of the magnitude of the resulting load on the springs or expansion joint rigidity, are a bit different from the assessment provided in the referenced document but are believed to be more accurate.

Normal machine unbalance. Unbalance in rotating components can be derived from a number of sources:

- Center of mass not consistent with center of rotation
- Nonuniform thermal expansion
- Liberated mass
- Nonuniform moment of inertia about various sectional axes
- Cracked or failed components that provide or influence rotor stiffness

Turbo-machinery rotors typically receive a high-speed balance at the factory following completion of manufacturing. While all efforts are made to balance the rotors as much as possible, some unbalance always remains—even with new rotors. Additionally, as the machines age and wear, changes can occur within the rotors to cause the balance to change with time. In addition, some rotors become thermally unstable with time and exhibit different cold and hot condition vibration levels. Typically, supervisory vibration pickups are installed at all bearings, with typical and maximum allowable vibration at each. A representative vibration level might be 0.002 in. (0.051 mm) at running speed and 0.004 in. (0.012 mm) through rotor critical speeds during startups.

3.2.4 Emergency

Turbine loss of blade. This emergency unbalance condition is based on loss of mass resulting in an instantaneous out-of-balance condition. The assessment of this load is directed toward the loss of a single blade (one of the largest blades in the turbines): the last row (L-0) blade of the LP element(s). These blades can be quite massive and have a center of mass that is well displaced from the rotor centerline. Consequently, the loss of a single blade will have an immediate and significant impact on unit vibration, which is transferred to the foundation through the rotor, the bearings, and the bearing pedestals. Because most large, modern LP rotors are symmetrical double-flow rotors (that is, having blades arranged in ascending order symmetrically from the rotor center), L-0 rows occur well separated toward both ends of the rotors. Analyses should consider the loss of a blade from the L-0 row in a single LP rotor; in cases in which the T/G set includes multiple LP rotors, all L-0 rows should be included in the analysis. As with other vibration-related analyses, the magnitudes and locations of the forces are provided by the turbine manufacturer as forcing functions for dynamic analyses or equivalent static loads for the simplified static analyses.

The loss or separation of one or more blades represents the greatest risk to the rotor support structure. The importance of considering different T/G operational events that can lead to blade loss cannot be understated. Appendix A presents a summary discussion of the different kinds of turbine blade loss events.

Generator short circuit. A line-to-line short circuit at the generator terminals represents the most severe short-circuit fault that puts severe loading on the generator stator, rotor, and foundation. The calculation of the air-gap torque is normally performed using no electrical damping to define the most severe torque and therefore the most severe forces that can be developed under different fault conditions. The line-to-line fault produces forces that are about 25% greater than the forces developed by a single terminal-to-ground fault and about 30% greater than those developed by a symmetrical three-phase fault at the terminals. Therefore, the line-to-line fault represents the worst case.

Out-of-phase synchronization. The process of starting up turbo-machinery involves first increasing the speed of the machine from stand-still or turning gear speed to full synchronous speed, which is 1800 revolutions per minute (rpm) for a unit in which the generator rotor has four electromagnetic poles and 3600 rpm for units in which the generator rotor has two electromagnetic poles. The poles must pass each coil in the stator winding at a frequency appropriate to produce the required AC current frequency, which in the United States is 60 cycles per second. Therefore, with only two poles—a positive and a negative—rotation must be 60 revolutions per second, or 3600 rpm; with four poles, this is cut in half to 1800 rpm. Once rotating at full speed, the speed must be adjusted to bring the polarity of the generator into alignment with the polarity of the line (that is, synchronized). Then the breakers are closed to connect the generator to the system.

Severe torque levels are imposed when the synchronization is not in phase with the grid when the generator is connected. The worst case is when the synchronization is out of phase by 120 degrees because of the three-phase nature of the generator output. This worst case out-of-phase synchronization produces torques much more severe than the line-to-line short circuit at the terminals, although the exact value is a function of the grid conditions, generator, and step-up transformer parameters. In certain instances, this torque can be higher than the short-circuit torque by as much as 35%.

This notwithstanding, the 120° out-of-phase synchronization is an extremely unlikely event. Consequently, this is not used as a design criterion. Rather, the more realistic practice is to consider the short circuit to be the bounding condition, and the worst-case out-of-phase synchronization is covered by these design margins.

Seismic. Seismic forces are calculated in accordance with the Uniform Building Code (UBC) [8] and provide minimum standards for the design criteria to make structures resistant to earthquakes. The UBC provides guidance on the applicable seismic forces for application in a static analysis and/or the ground response spectra for cases in which a dynamic analysis is to be performed.

Bowed rotor. Bowed rotors can be caused by a number of events and conditions, including the following:

- Turbine packing or generator labyrinth seal rubs
- Bound steam or hydrogen gland seal(s)
- Water induction into the turbine
- Bow caused by lack of sufficient turning gear operation, particularly when the rotors are hot
- Binding of components that are intended to move freely

Regardless of the source of a rotor bow, it creates rotor imbalance and causes increased vibration levels. Because the first critical speed of a rotor is the banana mode (that is, consistent with a bow), the vibration is most highly impacted as the rotor goes up or down through the first critical speed—that is, during startup or shutdown. The magnitude of the vibration is established by the manufacturer according to their standard practices, which vary among manufacturers. The loading may be provided as a forcing function for dynamic analysis or in the form of equivalent static loads for the simplified analysis. Once specified, these are treated in the same manner as the normal vibration loads with the exception of the duration. Because this is an abnormal condition requiring shutdown of the unit, it typically occurs only for a short duration, during which the operator is reacting and possibly running diagnostic testing.

3.3 Industry Standards

There are many industry standards available for the design, operation, and maintenance of T/G systems. For the foundation, the previously referenced Uniform Building Code [8] is used. Also for the foundation, the American Concrete Institute has the general purpose building code requirements for reinforced concrete [9] as well as one for foundations that support dynamic equipment [10].

As vibration-related problems are often the most common and difficult to address, there are many kinds of standards available by the International Standards Organization (ISO). ISO 1940-1 [11] provides requirements for proper balancing of rotors. Other ISO standards [12, 13] are two commonly used standards for addressing vibration for T/G systems. There are several others [14–18] available that address a more general classification of "rotating equipment" that would apply to different types of turbine designs and power outputs. For the petroleum, chemical, and gas industry, the API code has its own standard for steam turbines [19]. And, specifically for generators, the IEEE standard [20] is available.

For the operation and maintenance of T/G systems, the Nuclear Electric Insurance Limited (NEIL) has standards [21] that it applies for T/G systems at nuclear installations. These standards have been in use for several years and have established the industry-accepted practices for inspection intervals and required practices for all parts of the T/G system, including all elements of the rotor support system.

3.4 Design Considerations

Analyses must satisfy two design criteria: one involving deflection and the second involving strength. The deflection criteria address the need to maintain rotor alignment, while the strength criteria address avoidance of mechanical failures. Loads include both static and dynamic loads. Static loads result from sources such as dead weight and condenser vacuum. Dynamic loads are associated with normal rotor unbalance and a number of specific fault conditions, including line-to-line short circuit, out-of-phase synchronization, bowed rotor (which is a more severe case of rotor unbalance), seismic events, and asymmetrical loss of mass, which also is an extreme case of rotor unbalance. Some of these loads are transferred from the T/G directly to the foundation via the sole plates, and others are transferred from the rotors and through the bearings, bearing housings, and bearing pedestals to the foundation. For generators and certain turbines having bearing support structures integral to the turbine casing, loads are transferred from the rotors in a combined path that involves the bearing, bearing support structure, and integral pedestals plus the interface at the sole plates.

For certain applied loads—specifically, those that are applied statically—a static analysis is obviously all that is required. This includes loads resulting from dead weight, condenser vacuum, normal torque, thermal expansion of the T/G set, and piping and valve reactions. Reference [7] recommends full dynamic analysis only for normal machine unbalance load on low-tuned foundations and for seismic events where the response spectrum method is elected, regardless of whether the foundation is low tuned or conventional. Dynamic analysis is also appropriate but not necessarily required for low-tuned foundations to assess other fault conditions, including line-to-line short circuit, out-of-phase synchronization, bowed rotor, and asymmetrical loss of mass. Pseudo-dynamic analysis is recommended for normal machine unbalanced loads for machines mounted on conventional foundations and for seismic events based on UBC or ANSI requirements. Pseudo-dynamic analyses are also suitable for assessing other fault conditions, including line-to-line short circuit, out-of-phase synchronization, bowed rotor, and asymmetrical loss of mass—particularly for conventional and even for low-tuned foundations (although dynamic analyses are preferred for the latter). Therefore, in the United States, where most foundations are of the conventional type, it appears that static and pseudo-dynamic analyses are suitable—at least for the foundation.

For the assessment of T/G uprates, it is the emergency loading of Section 3.2.4 that becomes of critical importance. Emergency loads can increase significantly with increased levels of uncertainty about their maximum values. In the next section, the performance of several uprated turbines is reviewed with particular attention to the problems that they have encountered. An increased awareness of these problems is helpful to understanding the loadings that need the most attention when analyzing the T/G uprate. The specific loadings identified for increased attention during T/G uprate are then discussed in detail in Section 6.

Section 4: Uprated Turbine/Generator Designs

In this section, the most common T/G designs employed by utilities for upgrading large steam plants are reviewed. The scope of recent, current, and contracted upgrade programs has been summarized by EPRI as of August 2010 [1]. This document provides results of an industry survey covering the specific components chosen for replacement as well as the division of responsibilities for major aspects of the program. In most instances, the details of the designs and their integration into the T/G island are proprietary to the OEM or utility; however, some specifics have been made available through EPRI's members and are reported here by way of example.

It is obvious that virtually all upgrades must be highly customized, even when standard building block components are selected, reflecting the unique infrastructure of the plant and its operation and requiring the collaboration of the utility, OEM, and A/E firm [2].

4.1 T/G Component Replacements/Modifications

Configurations of the prime movers selected for upgrade run the gamut from complete replacement by a single OEM to a mix-and-match of repaired and new elements from different sources. Often the choice is made to replace or modify the HP and LP turbines, generator, and exciter during different outages, depending on economic and operating factors. The planning process is extensively reviewed in [2]. The "match" of different components requires extensive engineering analysis and has been the source of some significant outages, as discussed in detail in Section 4.4. Primarily, these problems relate to the dynamic response of the T/G rotor train and the rotor support system to various fault conditions.

4.1.1 High-Pressure Turbine

The primary objective of power uprate is, of course, to increase the output of the unit, which entails either increasing the steam flow or thermodynamic efficiency of the turbine. With improved thermodynamic efficiency, the output can be increased without significant change to the inlet steam temperature and pressure [5]. Increased flow usually calls for improvements in the aerodynamic design of the HP turbine inlet stages along with various mechanical improvements, such as full arc admission, expandable/abradable seals, improved materials, and the addition of blade rows. Replacement of the HP turbine is generally necessary to achieve the uprate potential of a BWR unit or, more generally, to attain the desired level of EPU.

Different OEMs make use of somewhat similar design choices in the redesign of their respective HP turbines for power uprate. Alstom provides for retrofits of existing dual-flow HP cylinders as well as replacement of the complete HP module [22]. In the retrofits, new blading and diaphragms are configured to permit mounting in the original grooves of the outer casing. In these upgrades, the design of the first stage enables partial arc steam admission to the full arc inlet to accommodate single valve closure for fault protection and testing purposes. Replacement of the entire HP module is made most often to address erosion problems with the outer casing. In these cases, the replacement is a single-flow HP module with impulse blading selected to maximize output and minimize net thrust on the existing thrust bearing.

The GE “Dense Pack” designs maximize power obtained from an HP or IP turbine without changing the dimensions of the outer casing or the steam flow [23]. The “dense” name connotes an increase in the number of stages within the same overall length as well as minimizing the inner diameter of the flow path consistent with dynamic stability of the rotor. The number of blades, or “buckets,” per row is reduced to accommodate the smaller root diameter. The ratio of bucket reaction to impulse (steam expansion in the rotating versus stationary blade row) is increased by 20 to 25%. The resulting lowering of steam impingement velocity is said to drastically reduce the rate of solid particle erosion in fossil units. Integral blade tip covers with advanced tip sealing and bimetallic seal rings are claimed to reduce leakage.

Siemens offers replacement HP turbine designs with “convertibility” features [24]. In these designs, blade spacing and root geometry are designed to support a future change of the admission ring and first drum stages. This design supports future increases in steam pressure, allowing the “converted” HP steam path to make use of the existing rotor and stationary components, and requires only minimal equipment changeout during the upgrade. These designs convert the original nozzle chamber design to full-arc reaction blading that is claimed to be most efficient for large baseloaded applications. Table 4-1 lists utilities that are reported to have replaced, or plan to replace, HP turbines with one of these designs.

Table 4-1
HP Turbine Upgrades

OEM	Plant	Year
Siemens	Susquehanna 1	2008
	Susquehanna 2	2007
	Salem 1	2004
	Salem 2	2004
	Grand Gulf	2012
	Turkey Point 3/ St. Lucie 2	2012
	Turkey Point 4/ St. Lucie 1	2011
	Point Beach 1 17%	2014
	Point Beach 2 17%	2015
	Sequoyah 1	2003
	Sequoyah 1 Watts Bar 2	2005 2009
Alstom	Paradise 1	2003
	Paradise 2	1999
	Paradise 3 Callaway Laguna Verde	2004 2005 2010
	Bull Run	2004
GE Dense Pack	Nine Mile Point 2	
	Monticello	2011

4.1.2 Low-Pressure Turbine

Much of the increase in unit output may be achieved through greater steam expansion through the LP turbine. In addition to any increase in steam mass flow afforded by an upgraded HP turbine, replacement of moisture separator/reheaters in a BWR unit, such as at Fermi 2 [25], can achieve a combination of increased flow and higher pressure supplied to the LP turbines. All the major OEMs have redesigned LP turbines with significantly increased exhaust plane areas and computational fluid dynamics (CFD) optimized blade stages and exhaust hoods. Accordingly, there has been much competitive development of very long last-stage blading (LSB). Table 4-2 lists the current offerings for 50-Hz/1500-rpm and 60-Hz/1800-rpm nuclear units and 3600-rpm fossil units.

Table 4-2
LSB Lengths of Current/Recent LP Turbines

Frequency (Hz)/rpm	Plant	OEM	Number of Blades	Length (in.)	Area (m^2)
60/1800	Quad Cities 1 and 2	Alstom	61	47	
	Dresden	Alstom	61	47	
	Unistar	Alstom "Arabelle"		57	
	13 units	Siemens		46	13.9
	D.C. Cook 1	Siemens 2006 Alstom 2009	50	56	18.0
	Quinshan Phase III	GE/Hitachi			
	Gentilly 2	GE			
50/1500		Hitachi		74	
		Alstom "Arabelle"		69	
		MHI		74	
		Siemens		52	
60/3600	Centralia	Siemens		32	6.9
	Aberthaw	Alstom			
	Sundance	Alstom			
	Keephills				
60/1800	Fermi (ESBWR)	GE Hitachi		52	
	Vogtle (AP1000)	Westinghouse		54	
	Callaway Diablo Canyon Ft. Calhoun	Alstom			

1 in. = 25.4 mm

4.1.3 Generator

Issues affecting the choice of repair or replacement of the generator are discussed in EPRI reports [1] and [2]. The key issue is the adequacy of insulation and cooling to supply the additional electrical output, which may be increased by as much as 20%. In general, the stators are considered to require rewinding (replacement of coils, end turns, and wedges), while cooling fans and hydrogen heat exchangers may need greater capacity and the electrical torque associated with a line-to-line short circuit may require review of anchor security. Rotors are replaced or repaired depending on results of a condition survey (bearings, retaining rings, windings, laminations, temperature distribution, cracking, and corrosion). Following reconditioning, the assembly should be checked for critical lateral and torsional vibration modes. A list of utilities that have opted to replace generators or rotors only is provided in Table 4-3.

Table 4-3
Generator Upgrades

OEM	Plant	Upgrade	Year
Siemens	Grand Gulf	New	2011
	St. Lucie 1 and 2	Rewound stator and rotor	
	Turkey Point 3 and 4	New rotor rewound stator	
	Point Beach 1 and 2	New rotors	
	Watts Bar 2	Rewound stator	2009
Alstom	San Onofre 2	New rotor	
	San Onofre 3	Rewound rotor	
	Laguna Verde	New	2010
GE	Monticello	Field only	
	Gentilly	Rotor windings	
	Susquehanna	Refurbished	
Mitsubishi	Ginna	New	

4.2 Changes in Control Systems and Operating Procedures

In the case of nuclear plants, the T/G upgrade is usually conducted in conjunction with license extension. The process may anticipate changes in the distribution system such as increasing the geographic range or adding specific types of industrial demand. These service changes can affect the probability of trips, with attendant thermal and mechanical loads on the T/G rotor system. In addition, the system may be subjected to phase changes that could excite torsional oscillations of the system. Changes may be made to extractions, or, in the case of turbine replacement, there may be mechanical changes in the extraction piping connections to the casings.

The control systems of nuclear plants in the United States have been exclusively analog based. The major reservation against digital control of nuclear plants has been the risk of external hacking. A committee of the National Research Council studied the reliability and vulnerability of digital control in 1997 [26]. The first introduction was into a Korean plant in 2003 [27], and since that time digital systems have become widespread in Japan, France, the United Kingdom, and Sweden. However, introduction into a U.S. nuclear plant, Oconee Nuclear Station 1, was delayed until 2011 [28]. Other plants have reportedly committed to digital control: Seabrook and D.C. Cook as well as the Generation III+ units for Vogtle (Westinghouse AP1000, approved by the NRC [29]) and Detroit Edison (GE Hitachi ESBWR, approval pending).

The significance of the conversion to digital instrumentation and control lies in greatly improved reliability through triple-redundant systems, the reduction of overspeed and the number of trip events, and the capability of real-time diagnostics. The system at Oconee was custom tailored to avoid all external communications, eliminating the possibility of external hacking, avoiding lengthy recalibration of devices, and eliminating what Duke considered to be one of the top three causes of trips [30]. Other advantages claimed for the digital system for the AP1000 are the automatic control of load (generator current) and power (cold reheat pressure) based on the calculated level of thermal stress in turbine components [29]. All these improvements would be expected to reduce the probability of severe loads on the rotor system and its supports. But perhaps the greatest potential benefit to the T/G could be the opportunity for real-time diagnostics of displacement, vibration, temperature, and pressure data to warn of impending problems. The subject of monitoring and diagnostics is further discussed in Section 5.

4.3 Loading Changes

The foundation of the T/G island, consisting of beams, columns, and footings, is primarily designed to withstand seismic, dead weight, and accidental impact loading. The question of seismic loading has recently been brought into focus by the 5.8-magnitude quake in the vicinity of the North Anna nuclear plant [31]. It is not clear whether the investigation of seismic adequacy will require modifications to foundations of other plants or affect plans for future uprate projects. As a minimum, it is anticipated that inspections of the foundations for some plants will be required.

Any increase in supported weight must be factored into the evaluation of the foundation. However, this increase may be negligible or even negative. Recent discussions with utility personnel have shown increases in overall weight of around 3% or less associated with an uprate. The 2006 upgrade of D.C. Cook to the Siemens 18-m^2 LP turbines is believed to be an exception, involving a 26% increase in weight, from 155 to 195 tons [32]. New plants, such as the AP1000, will have spring-mounted reinforced concrete decks and condensers, which effectively decouple the foundation from dynamic loading from the T/G [29].

The limiting loads on the foundation are a direct result of fault conditions. These faults include various electrical faults, the most severe being a line-to-line short circuit, sudden loss of condenser vacuum, pipe break, and loss of one or more LSBs. Fire damage to foundations from broken lubrication oil or hydrogen cooling lines is also a risk. Various other scenarios can be imagined, such as a rotor, disc, or retaining ring burst events, that might destroy the support of bearing pedestals, turbine casings, or generator stator. Fortunately, these events assume a low probability along with terrorist attacks, disc penetration of turbine casing, and impact of a large aircraft. The insurance industry has placed certain requirements for demonstration of the absence of resonant rotor vibration near multiples of the running speed, either by analysis or testing. For power uprates, the utility or its suppliers will have reviewed the analysis of the foundation for the limiting short-circuit event, so that for the case of quasi-static loading, the stresses need only be ratioed by the percentage uprate. The dynamic response must also be considered, requiring the time history of generator current.

The other relatively high probability faults affect either the adequacy of casing supports (pipe break and loss of condenser vacuum during a trip) or the bearing and bearing pedestal.

The latter probability is an indirect consequence of rotor dynamic response via fatigue cracking and separation of one or more LSBs. The incidence of LSB separation has not increased because of uprates, but the consequences have been magnified by the incorporation of very long blades. LSB failure has been attributed to normal stochastic buffeting combined with effects of moisture, that is, corrosion and erosion, water ingestion, stall flutter/high backpressure, and certain features of early turbine designs that led to integral order excitation of vibration modes. Modern designs have been extensively analyzed and tested, and it is rare to encounter a design with a blade response near running speed or any multiple of running speed. However, the very large L-0 and L-1 blades of modern designs are dynamically coupled with torsional shaft modes and require a more complex combination of analysis and testing for acceptance. For example, it was adequate in the past to spin-test a bladed disc stage, but now the entire rotor assembly is tested in a vacuum spin facility. Similarly, the analysis of torsional modes was once adequately conducted with axisymmetric finite elements; however, this analysis now requires the coupling of three-dimensional (3-D) blade models to the axisymmetric rotor model (see Section 6).

The consequences of a single LSB separation are extensive in any design, but for modern large blades, separation is potentially catastrophic. For nuclear plants, NEIL requires analysis or testing of rotor torsional modes to ensure frequency separation of at least 2.0 Hz from twice the operating speed [21], and the OEMs further increase that separation requirement.[1] The OEM also evaluates the centrifugal effect of the loss of a single blade on the integrity of the bearing cap retention and supporting bolting. In reality, of course, the blade separation is a dynamic impulse, and its effect on the response of the rotor should be analyzed accordingly, as discussed in Section 6.

The probability of LSB separation may be suggested by the following list of documented or partially documented failures, which is by no means complete. The list includes fossil and nuclear units and excludes numerous gas turbine LSB failures. The effect on the integrity of the foundations is not known, and what little information is available on the extent, if any, on the bearings or their supports is summarized in the following section.

4.4 Effects on the Rotor Support System

4.4.1 Last-Stage Blade Separations

TVA Cumberland Unit 1 (1992)

This separation of two 30-in. (762-mm) L-0 blades in a row of a double-flow Alstom fossil turbine resulted in fracture of the front standard. It was reported that the blade separation occurred at the end of the LP rotor closest to the HP turbine. The standard was reported to be a steel casting. A break in a lube oil pipe required shutting off the supply of oil, which resulted in bearing failure, rotor interference with the casing, and a four-month outage [33]. The fatigue failure of one blade was attributed to a large corrosion pit [34] and caused the release of the neighboring blade.

Perhaps germane to the probability of such an event is a similar failure of a Siemens V84 gas turbine in the Doswell, Virginia, combined-cycle plant in February 1995. Failure of an LSB was followed by impact separation of the adjacent blade and fractions of neighboring airfoils in the stage. The failure resulted in the collapse of the compressor end bearing support and the fragmentation of sections of the casing. The support structure was primarily made of cast steel. Analysis of the impact of the sequence of blade loss on the rotating shaft showed a magnification of transverse load on the compressor end bearing. These events may indicate three areas to consider: the fracture toughness of cast bearing supports may present a risk of consequential catastrophic damage, the location of the bearing that receives the greatest impact load during LSB separation may not be the one closest to the separation, and the vulnerability of oil lines to LSB separation may be significant and catastrophic.

[1] For example, Siemens adds a further separation of 3 Hz near 120 Hz, for a total margin of 5 Hz. Near 60 Hz, the separation is 2.5 Hz.

D.C. Cook Unit 1 (2008)

The separation of five LSBs occurred in two of three 18-m^2 Siemens LP turbines installed during a replacement uprate in 2006 [3]. Two blades, separated by one intact blade, failed from fatigue in the center turbine, followed by two blades broken by impact. The generator-end rotor had one blade broken by fatigue, with no additional impact failures. The loss was estimated by AEP to total $332 million [35]. The replacement rotors were installed with the existing GE HP turbine, generator, and bearing support system [36]. There were concerns that the pedestals were inadequate for the increased weight of the rotors and that there was insufficient separation between the torsional coupled blade/disc mode and the 120-Hz excitation of the generator. Testing was conducted to verify the adequacy of the torsional design (the blades were modified by the addition of tip masses to lower the frequency). However, the rotor train was found to have a critical speed closer to running speed than anticipated. This difference has been attributed to the reduced stiffness of one of the bearing supports. The possibility of a preexisting crack in the bearing strongback and stripped bearing retaining bolts has been suggested [36].

For the purposes of these Guidelines, it is not known whether the reduced support stiffness was the cause or the effect of the blade fatigue cracks. The critical issue appears to be whether the bearing pedestal stiffness led to excessive transverse vibration of the rotor at critical speed, perhaps resulting in blade tip rub (as is reportedly maintained by Siemens [37]) or whether the fatigue cracking progressed because of operation in the critical speed range, with the vibration causing the loosening of the bearing retaining bolts. This example illustrates some of the risks involved in the mix-and-match approach to uprating and underscores the critical importance of inspecting and modal testing of all the supporting elements of the rotor train in addition to the rotating components themselves.[2]

TransCanada Pipelines Centralia Unit 2 (2006)

A fleet of upgraded Siemens/Westinghouse fossil LP turbines has reportedly experienced three L-0 blade failures. The design, a replacement for the Westinghouse BB72, has exhaust areas of 6.9 m^2 and blade length of 32 in. (813 mm) in the 60-Hz version and 10.0 m^2 in the 50-Hz version. Of possible relevance to the foundation guidelines (although no damage has been reported in any of these incidents) is the fact that in each case the root fatigue separation led to impact fracture of the adjacent blade. The two 6.9-m^2 incidents were attributed to moisture erosion and/or high backpressure operation [39].

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[2] The Cook incident raises the question of blade fatigue at a critical speed for transverse shaft vibration as opposed to torsional response. It is not known if the OEM calculates blade stresses for cases of transverse vibration. Assuming that the critical transverse frequency for the Cook rotor gradually dropped near the running speed either because of a propagating crack in the bearing strongback or, more likely, loosening of the bolts holding the strongback, there might have been a possibility that the blades were subjected to high bending stress at the root over a significant period of time. There is reported to be evidence of two distinct overloads on the fracture surfaces and around 100 "beach marks" [38].

In the Centralia incident, there were reports of significant rotor displacement but no failure of the support; blade tips were “rolled” (folded backwards) with evidence of blade-to-blade contact. In response to failures of longer LSBs, Siemens reportedly issued a product bulletin in August, 2010, to install an extra retention ring over the bearings and increase bolt strength in the 6.9-m^2 fleet [40].

South Texas Project 2 (2002)

An unanticipated change in the generator rotor stiffness led to torsional resonance of a Westinghouse BB380 turbine rotor and cracking of approximately half of the 40-in. (1016-mm) long L-0 blades of the middle rotor [41]. One of the blades separated, leaving a hole in the exhaust guide. The resonant frequency of the rotor was essentially twice the running speed but well removed from any blade frequency; thus, the fatigue cracking of the blade row resulted from forced vibration.

In addition to STP 2, three older torsional failures of LP turbine blades have been reported by EPRI [42], all attributed to electrical grid characteristics reflected through the generator rotor: Prairie Island 1 lost L-1 and L-2 blades in 1974; Maanshan failed eight 43-in. L-0 blades in 1985, causing a fire under the generator; and Susquehanna 1 lost two L-1 blades in 1993.

Detroit Edison Fermi 2 (1993)

Fermi 2 was reported to have had persistent vibration problems prior to the incident [43]. One blade was released, with one or more pieces entering the condenser, severing tubes and causing release of radioactive water. A broken generator hydrogen line resulted in a fire. The forced outage lasted slightly over a year. The extent of any fire damage to the foundation is not known.

ESKOM Duvha 2 (2003)

This 50-Hz fossil unit experienced a catastrophic LSB separation in a GEC dual-flow turbine involving release of 15 adjacent blades upon reaching 3000 rpm [44]. All components of the T/G were essentially destroyed by the resulting imbalance. In the process, a severed oil line caused a fire that damaged the foundation. A sister unit, Duvha 4, lost an L-0 blade during overspeed on February 9, 2011, with similar catastrophic results—including an oil fire. The extent of damage to the foundation is not known.

Duvha 2 is the only reported case of fire damage to a foundation but by no means the only case of oil fire. This raises the question of the integrity of oil piping in instances of severe bearing vibration.

Section 5: Vibration Testing and Monitoring

Limits on vibration levels are essential to ensure safe, reliable, long-term operation. As previously reported in the case of rotors, industry-accepted limits have been established by IEEE, ANSI, and, for nuclear plants, NEIL. For the support structure, the limits depend on the accuracy of the T/G dynamic analysis, particularly the reactions of the bearings to the new dynamic loads. Ultimately, this analysis must be verified and fine-tuned by testing.

Almost inevitably, changes occur either from wear or by accident during the operation of the unit that result in shifts in the vibration frequencies and amplitudes of rotor components and supporting structure. The role of monitoring is to identify these trends to avoid unnecessary trips and, ideally, diagnose their causes and implement appropriate remedial actions.

This section provides the background for the analysis and testing of rotor and structural vibration, leading to a set of guidelines outlined in Section 6.

5.1 Vibration Data Acquisition

Vibration signals that are most commonly measured are acceleration, velocity, and displacement. Both accelerometers and velocity transducers measure vibration of non-rotating parts of a machine and have limited application to T/G systems. Displacement transducers, on the other hand, have the ability to measure small changes in the annular radial clearance gaps between the rotor and the bearing housing, making them most useful for T/G testing and monitoring. The importance of precisely knowing the rotor motion relative to the stationary bearing casing on a continuous basis led to the development of the proximity probe, a non-contacting sensor with a large amplitude range and high accuracy and resolution.

Each of the vibration sensors has specific uses in understanding the vibration of the T/G. Intelligent use of the sensors in combination with high-speed, digital data acquisition systems can provide a high degree of protection for the T/G and, in many cases, early warning of potential damaging vibration.

The sensors are mounted at various locations of the T/G with their electrical cables fed to a nearby instrument cabinet. The instrument cabinet will contain the signal conditioning (power to the sensor, anti-aliasing filters, and amplifiers), analog-to-digital conversion cards, and a computer for data processing, data storage, display, and transmission of critical parameters to the plant computer and control room.

5.1.1 Proximity Probes

The proximity probe displacement sensor employs an inductance-type (eddy current) measurement principal for converting displacement to a proportional voltage. The proximity probe system includes the probe, a specific length cable, the oscillator/demodulator that supplies power to the probe, and a voltage output proportional to the displacement.

The probe is non-contacting; that is, there is an air gap between the probe tip and the rotor surface. The probe is mounted a specific distance from the target point on the rotor. The rotor surface at the target must be smooth and free of scratches and irregularities to avoid unwanted signals.

In order to achieve an accurate voltage-to-displacement calibration factor, the cable must be properly matched to the probe and the oscillator/demodulator and calibrated to the specific rotor material properties. The probe can measure both static and dynamic changes in the rotor's radial motion.

For large T/G sets, proximity probes are typically installed in pairs at each bearing with their measurement axes oriented at 90 degrees to one another. Figure 5-1 illustrates a typical installation of proximity probes. Note that the proximity probes are at the 45- and 135-degree azimuth.

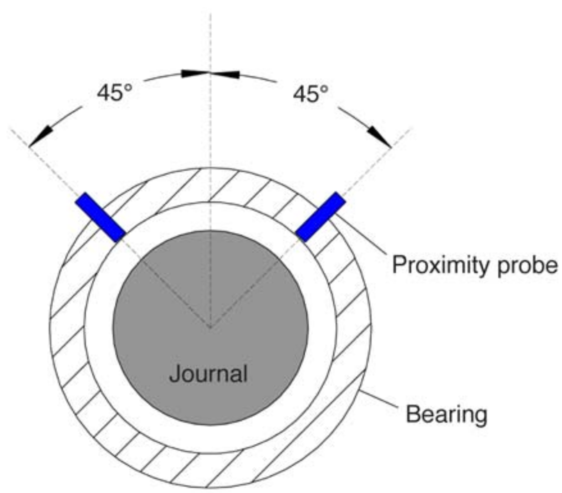


Figure 5-1
T/G Bearing Installation

The measured rotor displacements at the bearings are the primary measurements continually monitored and compared to warning and alarm criteria. Though this has sufficed to protect the majority of T/Gs, more sensitive or sophisticated methods are needed to provide an earlier warning of potential failures due to vibration.

5.1.2 Accelerometer

Another sensor used in vibration monitoring is the accelerometer. The most commonly used accelerometer is the piezoelectric accelerometer that produces a charge (picocoulomb) proportional to the acceleration experienced by the sensor. The charge is converted either internally or externally through a converter to a voltage signal. The accelerometer, through a process of integrations, can provide velocity and displacement. The integrated accelerometer displacement is currently used in T/G monitoring to determine the motion of the bearing housing and added to the proximity probe to produce a total rotor displacement rather than a rotor displacement relative to the bearing housing.

The accelerometer may be used for other purposes in the monitoring of T/Gs. An accelerometer placed strategically on a T/G component will be sensitive not only to the motion of the component, but also to other vibrations (noises) generated on or inside the component—similar to the way a stethoscope placed on the outside of the body measures noises generated inside the body. For example, an accelerometer attached to the bearing housing would be able to detect vibrations caused by rub, whirl, or loosening of the housing or bearing pads as well as the motion of the structure supporting the bearing housing. Accelerometers attached to the turbine housing would measure the motion of the housing and vibration generated by the blade acoustics, attachment looseness, or blade rub, to name a few.

5.1.3 Data Acquisition and Monitoring

Data acquisition and monitoring of the T/G have changed significantly over the years, from the use of strip charts, meters, oscilloscopes, and tape recorders looking at each sensor sequentially over long periods of time to the current use of high-speed, digital data acquisition systems with displays of the data on computer monitors in several locations at one time. The current systems can acquire all the monitoring channels simultaneously and continuously, providing the ability to correlate two or more channels in time and frequency without the loss of phase and amplitude relationships. Real-time analysis in the time and frequency domains can be calculated in parallel. The results are continually updated: amplitude data (peak, peak-to-peak, and rms), frequency spectra, and trending are instantly displayed. Any transients, sudden changes, and intermittent behavior can be detected, and the time history signals for all monitored channels can be captured starting prior to the incident and for a short period after the incident to be later analyzed for cause.

The data acquisition systems have usually been rack-mounted and located relatively close to the T/G. The cabinets consist of signal conditioning modules for the proximity probes, accelerometers, and other monitoring equipment (for example, temperature and pressure) and a high-speed, industrial quality PC with backup disk storage and monitor. The most modern systems look at each channel continuously while simultaneously storing data based on an exceeded criterion or a daily capture for the historical record. The data can be stored and analyzed and pertinent information transmitted to the plant computer for warning, alarm, and display. Typically, rotor displacement at each bearing and displacement trends are available on displays in various parts of the plant. With the advent of the Ethernet and other localized data circuits, the full range of monitoring information is available for observation or analysis on any computer.

5.2 Data Analysis and Rotor Vibration Source Identification

Significant advances have been made in the analysis of T/G vibration data and the diagnosis of vibration problems. However, identification of vibration causes and how best to rectify them has largely been a subjective exercise that relied heavily on industry experience. This section describes some of the more common causes of vibration problems and how they can be detected from the vibration data.

5.2.1 Rotor Orbit Trajectories

Plotting the two orthogonal displacement time histories from a bearing's proximity probes on a vertical and horizontal axis of an X-Y graph creates an orbit graph (see Figure 5-2). The graph portrays the vibration displacement trajectories for a specific period of time. Before the advent of digital acquisition systems, these graphs were observed on oscilloscopes—but today can be computed and observed in real time on computer monitors. The orbit requires that the time histories are acquired simultaneously from the orthogonal probes so that the amplitude, frequency, and phase relationships between the probes are maintained. This orbit plot provides a diagnostic tool that is highly sensitive to rotor vibration conditions.

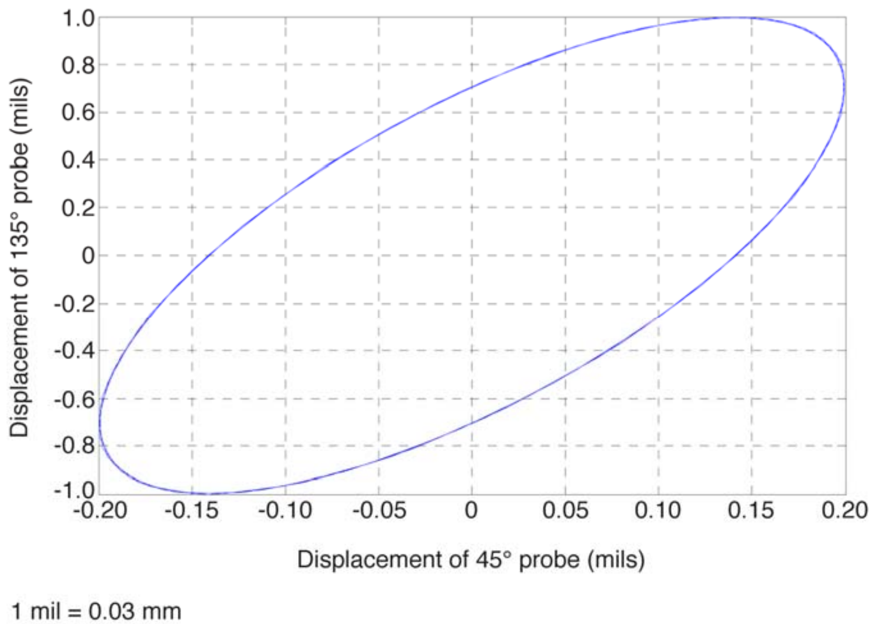


Figure 5-2
Orbit Graph of Two Orthogonal Proximity Probes at a T/G Bearing

The rotor vibration orbits computed at several bearings can be viewed in real time and compared with computationally simulated rotor vibration models to aid in the diagnosis of vibration problems. Real signals from operating T/G sets can be very noisy and may require filtering. Low-pass and band-pass filtering of raw vibration signals can be used to remove noise and to observe selected frequencies that may pertain to a particular vibration issue. Filtering is conducted selectively and intelligently so as not to remove parts of the signal necessary for accurate representation of the problem.

During roll-up, coast-downs, and rotor balancing, the proximity probes can use a tracking filter that creates a narrow, band-pass filter that tracks rotor rpm. This provides the ability to create a sharp time history of the rotor motion and a "synchronous" orbit picture at the rotor rpm that accurately detects critical speeds.

Figure 5-3 illustrates what is meant by sub-synchronous and synchronous frequencies. Figure 5-4 illustrates an orbit with a period of two revolutions containing synchronous and half-synchronous components. Sub-synchronous rotor vibrations are often associated with instability or self-excited rotor vibrations. For the case in Figure 5-4, the sub-synchronous frequency is exactly half the spin speed and is easy to illustrate. In general, sub-synchronous rotor vibrations are more often not an integer fraction of rpm, and the motion is not periodic.

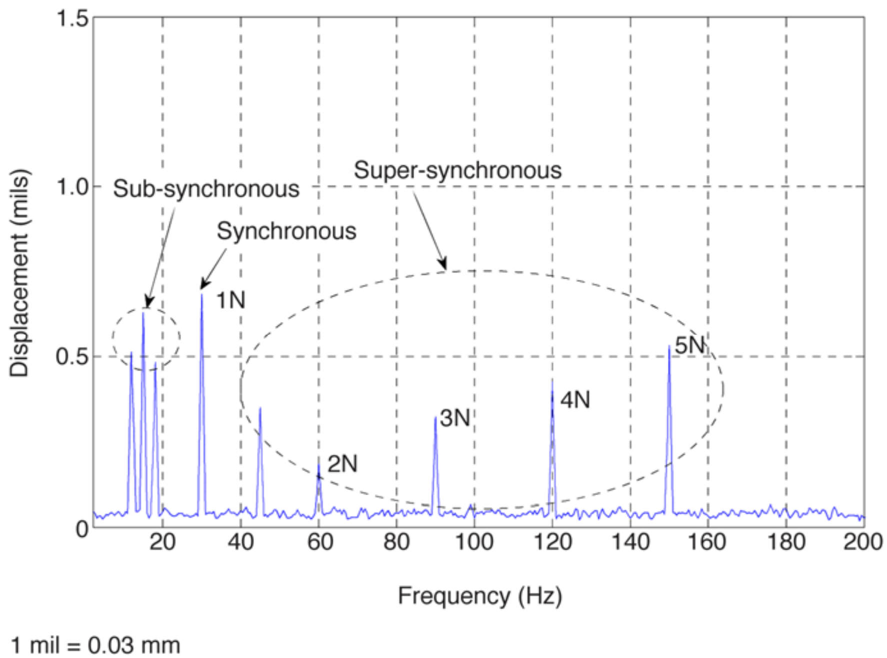


Figure 5-3
Synchronous, Sub-Synchronous, and Super-Synchronous Frequency Components of a Vibration Signal

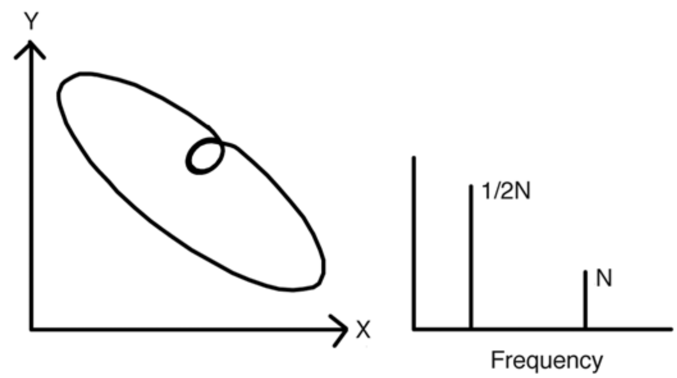


Figure 5-4
Orbit with Half-Synchronous and Synchronous Components

Figure 5-5 is an example of rotor vibration in a deteriorating condition due to increasing loads on the bearing or misalignment of the bearings and rotor with increasingly nonlinear behavior. More specifically, normal unbalanced forced vibration of the shaft orbital motion shown in (a) begins to intensify as the radial load (or misalignment) increases, causing dynamic nonlinearity in the journal bearing. As the rotor orbit plot changes from a normal elliptical to a distorted ellipse and then to the figure 8, a progressive increase in the 2N harmonic is observed. Similar types of distortion to the normal elliptical orbits can result from other higher harmonic contributions (3N, 4N, and so on) of rpm. Higher bearing loads produce higher eccentricities, resulting in increased bearing film dynamic nonlinearity. Therefore, the presence of high harmonics in the rotor orbit trajectories can be indications of excessive bearing loads and/or misalignment.

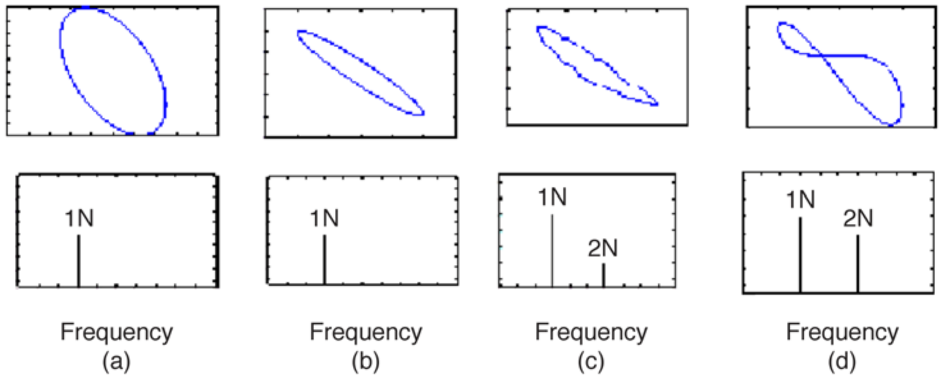


Figure 5-5
Orbit and Spectra Representation for Increasing Load or Misalignment: (a) Normal Load (1N), (b) Slight Load Increase (1N), (c) Significant Load Increase, (d) High Load Increase

This subsection on rotor orbit trajectories is merely a brief introduction to the usefulness of this monitoring tool for the diagnosis of rotor vibration issues. A more complete treatment of this subject can be found in [45] from which much of this material was taken.

The high sensitivity of the orbit to rotor dynamic change as shown in Figure 5-5 reveals the usefulness of the orbit as an early indication of a change in the rotor dynamics from an established baseline. To emphasize just how sensitive the orbit is to change, Figure 5-6 provides the orbits for two subtly different simulated rotor conditions. The initial condition (normal operation) is a synchronous 1N rotation with the two displacement amplitudes of 1 and 0.2 mils (0.0254 and 0.00508 mm) with a 45-degree phase difference. By adding a 2N component with an amplitude equal to 10% of the 1N amplitude, the change in the orbit is significant and detectable. Though this increase would not necessarily be observed in the bar chart or the trend, it is immediately observed in the orbit comparison.

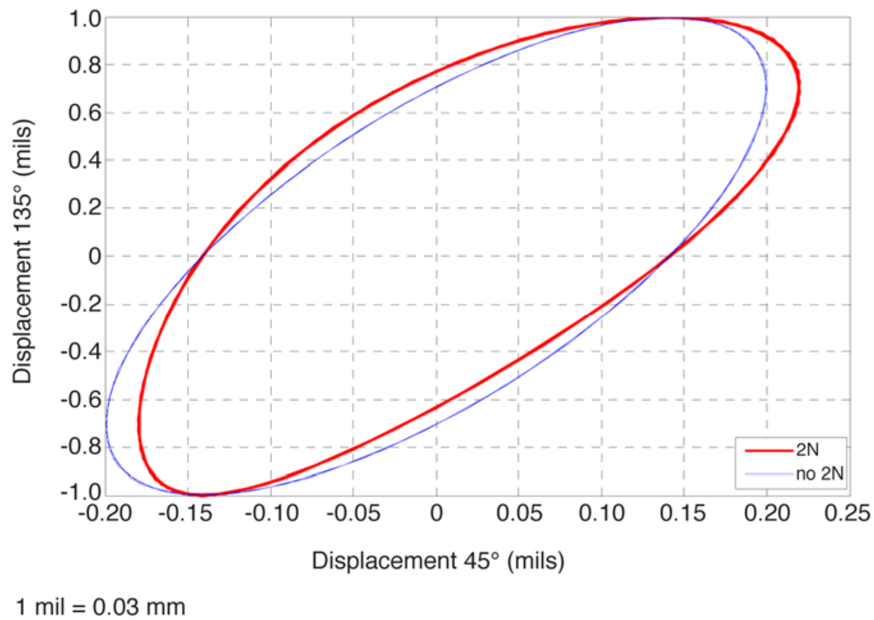


Figure 5-6
Orbit Comparison for a Small Change in the 2N Amplitude

By establishing a set of normal operation orbits at each bearing that include various combinations of low-pass, high-pass, and band-pass filters where there is an additional statistically computed band for normal operation variation, small changes from the normal may be detected and used to provide an early warning capability.

5.2.2 Spectral Analysis, Dual-Channel Analysis, and Cascade (Waterfall) Plots

Three of the more useful techniques for diagnosing T/G vibration problems are spectral analysis, dual-channel analysis, and cascade (waterfall) plots. Spectral analysis transforms time series data into the frequency domain, identifying the frequency content of any signal. With the advent of the fast Fourier transformation (FFT), real-time spectral analysis in parallel with complex monitoring tasks can be performed and used for automated, real-time diagnostics. Modern T/G monitoring systems typically have this capability.

Figure 5-7 is an example of a frequency spectrum calculated for the displacement channels used in the orbit plots in Figure 5-6. The addition of the relatively small 2N component is readily observable above the background noise; in fact, a signal possibly a factor of 10 smaller may be detectable.

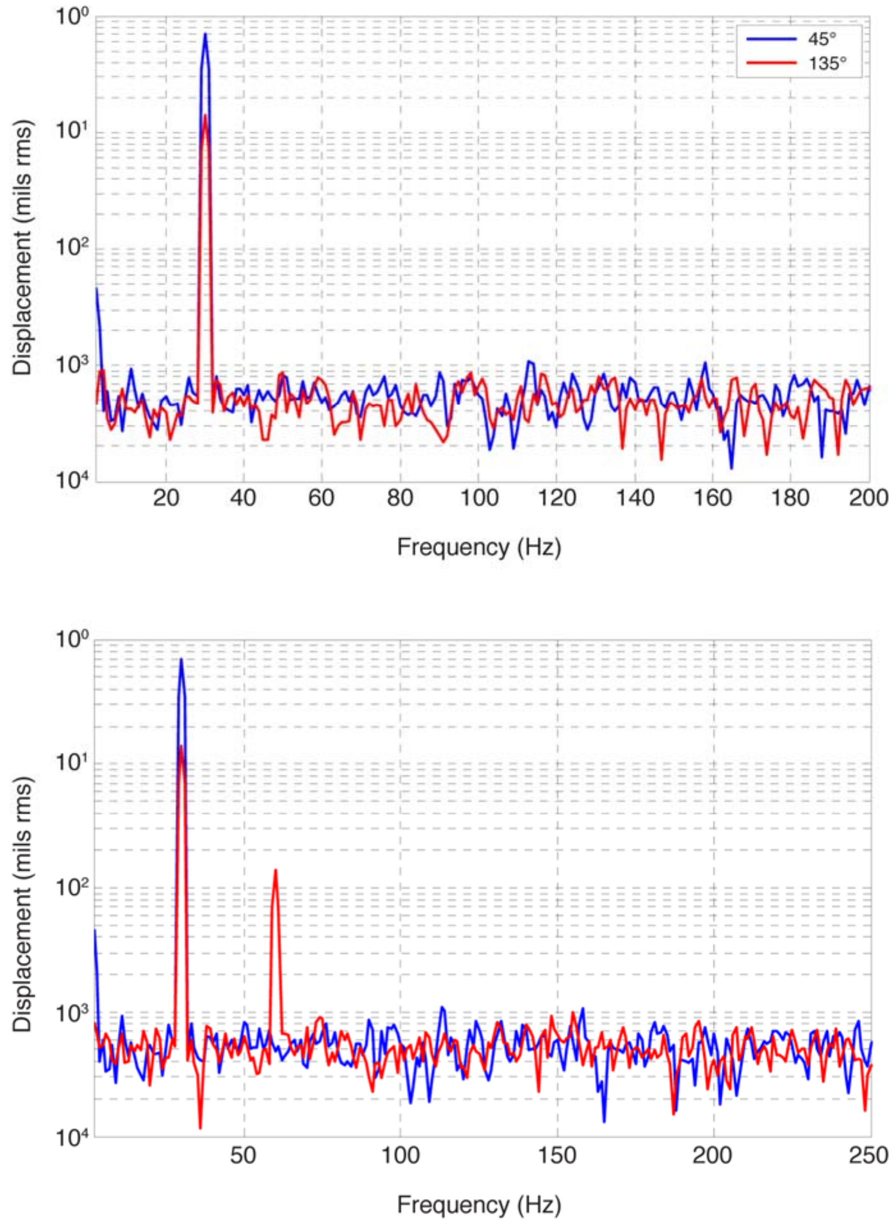


Figure 5-7
Frequency Spectra

Dual-channel spectrum analysis (transfer function, cross spectra, and coherence) provides the ability of relating two channels of data by transforming the time histories of two data channels to the frequency domain while retaining the amplitude and phase relationship. One form of displaying the magnitude and phase is the Bode plot (see Figure 5-8). This figure displays the cross spectra of the two orbits with and without the 2N component. The phase relationship at 1N can be seen to be 45 degrees. Though the 2N is small, it is still detectable in this type of analysis. The power of this type of analysis is its ability to observe the amplitude and phase relationship as a function of frequency; small changes are readily detectable.

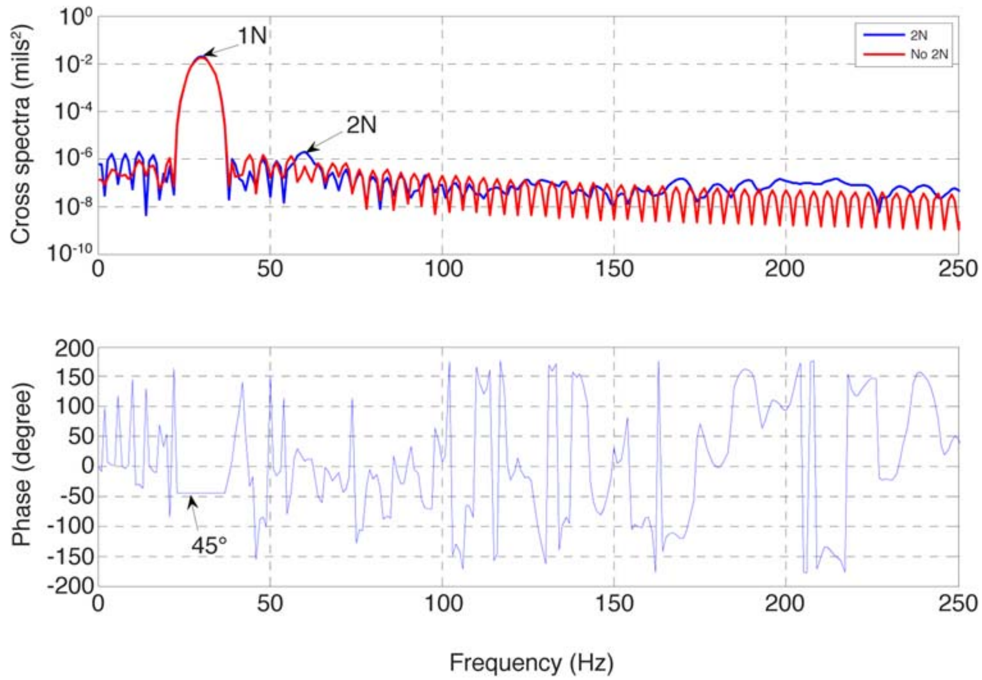
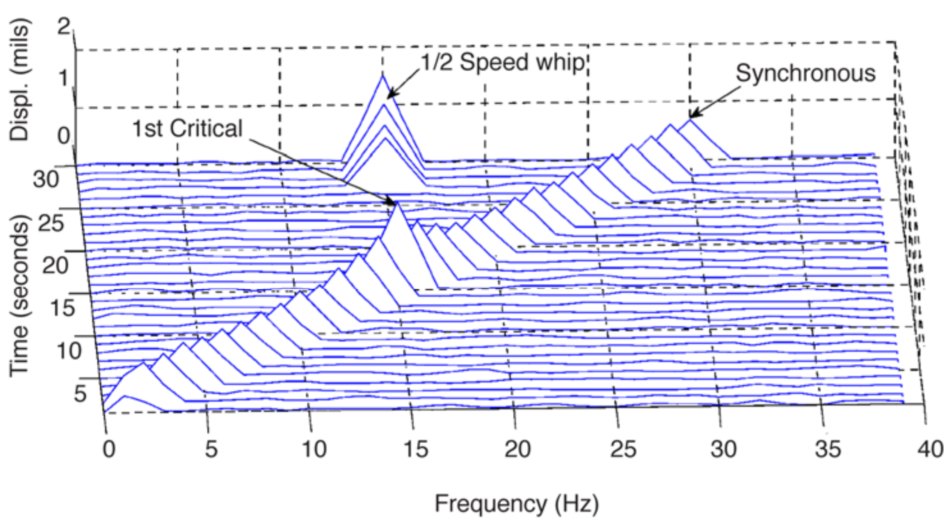


Figure 5-8
Cross Spectra of Proximity Probes with and Without a 2N Component

Cascade plots, also referred to as *waterfall plots*, are contour-map presentations of vibration amplitude data across a 3-D space defined by amplitude, frequency, and running speed. The cascade plot shows a rotor's vibration characteristics (amplitude and frequency) over an entire speed range. Figure 5-9 shows a typical cascade plot that is used to identify the oil whip threshold speed. Oil whip behavior is discussed in Section 5.2.6.



1 mil = 0.03 mm

Figure 5-9
Waterfall Graph of Roll-Up with Half-Speed Whip

5.2.3 Rotor Displacement

Rotor displacement is the primary measurement for protection of the T/G used in vibration monitoring. The rotor displacement signal is composed of signals representing many types of rotor motions, including the rotor's centerline position relative to the bearing casing, vibration at the operating speed and its multiples, and vibration due to whirl/whip phenomenon (self-excited), rotor broadband motion, and at times external loads due to foundation motion from earthquakes or other operating equipment.

The displacement is displayed as bar graphs (see Figure 5-10) and trends (Figure 5-11). The bar graph provides the current displacement of each bearing proximity probe. The trend graph shows the history of the bearing displacements. It is updated periodically and is useful in observing changes from the normal operation values. Under normal operating conditions, the amplitude is low in comparison to the alarm and trip setpoints.

Figure 5-11 includes indications for the alarm and trip setpoints. Two trend cases are shown in this graph as examples of how vibration problems progress in power plants. In both trend cases, there is a long period of normal operation followed by either a gradual or sudden increase in amplitude. At the juncture between normal operation and increasing displacements ("early warning"), either sudden or gradual, subtle changes in the vibration characteristics begin to occur. These changes, though detectable using the more sophisticated diagnostics techniques described above, are not observable in the commonly used trend graphs. It is at the "early warning" point that the use of on-line vibration diagnostics could provide an alert and initiate a process to detect an incipient failure.

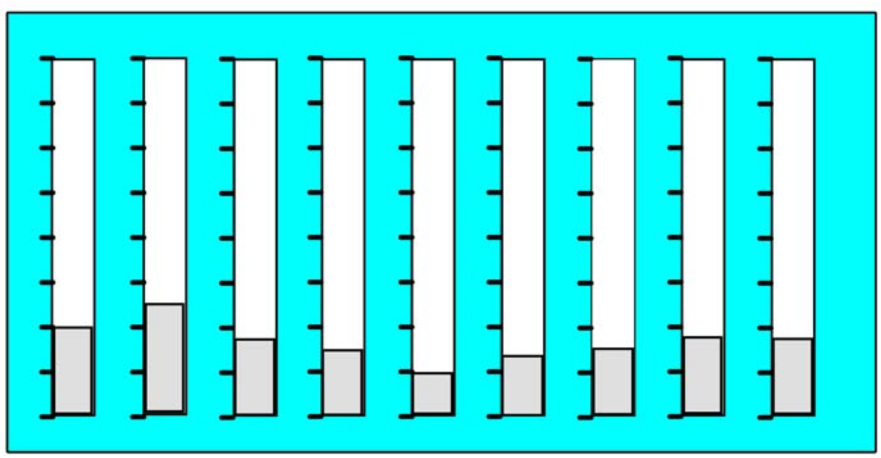


Figure 5-10
Monitor of Turbine-Generator Bearing Proximity Probes

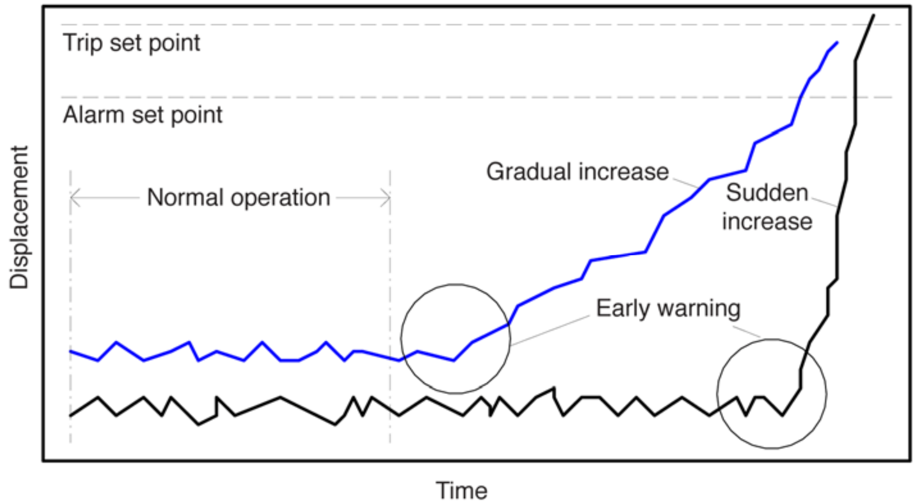


Figure 5-11
Trend Graph of Rotor Displacements

Collecting vibration data during normal operation and creating statistically bounded parameters (frequency, amplitude, and phase) that capture the normal swings in these parameters such that accurate baselines can be developed would provide early warning criteria. If these criteria are exceeded, alerts are issued, and data review can be implemented.

5.2.4 Rotor Mass Unbalance

Mass unbalance of the rotor is the most common excessive vibration problem for T/G systems, as particularly observed in the once-per-rev, synchronous vibration at running speed. As the vibration increases from the normal baseline, the amplitude soon becomes nonlinear due to the effects, for example, of journal bearing oil film stiffness, damping changes, and rotor-to-stator rubs. The nonlinearities are exhibited by increases in the vibration amplitude at the integer multiples of the operating speed as well as changes in the once-per-rev amplitude. Additional changes may occur in the phase relationships and the amplitude balance between the proximity probe pairs at each bearing. This is not to say that other vibration issues may cause increased amplitudes at operating speed harmonics. The diagnosis of each problem needs to consider all possibilities and requires looking at the vibration information in different ways.

Vibration due to mass unbalance has the potential to cause bearing damage, including wear, excessive heating of the oil (changing the stiffness and damping), and component breakage. This damage will accumulate over time unless the problems are identified and resolved early on.

Another concern with mass unbalance vibration is the amplification roll-up or coast-down. As the operating speed moves through a critical speed, the vibration amplitude may exceed the bearing or blade-to-casing clearance, potentially causing severe damage to the bearing, blade, or seals.

5.2.5 Critical Speed

The critical speed frequencies are designed to be sufficiently away from the operating speed and its multiples so as not to influence the rotor vibration during operation. The rotor is initially adequately balanced to avoid excessive vibration at critical speed during roll-up and coast-down. Yet, the critical speeds are sensitive to changes in the T/G components that may occur during operation due to deterioration of parts, design or installation error, loosening, wear, changes over time to the support structure or foundation, or oil degradation. A change in system mass or stiffness that shifts the critical speed toward the operating speed or its harmonics can cause significant increase in the rotor vibration during operation. A decrease in damping due to oil property changes or bearing pad loading can also cause a significant increase in the vibration amplitude at critical speed during roll-up or coast-down.

As with mass unbalance, the parameters that cause critical speed change probably occur gradually over a long period of time and could be caught and mitigated early in the plant operation. One method is to design the plant monitoring system to capture the T/G vibration sensors either by manual or automatic trigger during the periodic roll-ups (planned) and coast-downs (planned and unplanned). A record of the vibration response characteristics (frequency, response amplitude, and damping as calculated from the critical speed peak bandwidth) should be maintained and reviewed for change. Any significant change should be flagged and investigated.

5.2.6 Oil Whip: Self-Excited Instability

Oil whip is a self-excited vibration in which the forces causing the vibration are produced by the vibration itself. Specifically, during oil whip, the bearings are acting as negative dampers that cancel and then exceed the dissipative positive damping forces of the rotor system. When this occurs, subsequent increases in running speed lead to increases in vibration amplitude as shown in Figure 5-9. Therefore, vibration due to oil whip cannot be passed through as can be done for critical speeds.

Typically, oil whip occurs at running speeds that are twice the critical speed because there is sufficient energy at this frequency to produce the oil whip phenomenon. The changes in damping forces in the rotor bearing responsible for oil whip are difficult to predict analytically due to small differences in bearing manufacture and rotor operation. As a result, identical turbines do not all exhibit the same degree of oil whip vibration. It can be severe in some rotors and non-existent in others. In addition, modifications made for power uprate can bring on the occurrence of oil whip vibration where previously there was none.

5.2.7 Misalignment

Misalignment of the rotor bearings can cause a vibration with significant 2N component and increased axial vibration. Misalignment can occur due to three possible configurations: an improperly seated bearing, offset of the rotor centerline in the radial direction, and angular offset of the rotor centerline.

Rotor misalignment and rotor unbalance can both produce higher harmonics, but misalignment has increased axial vibration and unique phase characteristics. In diagnosing increased rotor vibration, this distinction may be important.

5.2.8 Loose Connections

Looseness in the components that are not rotating may have a significant effect on the rotor vibration. Components such as the bearing casing and mounts provide loads that maintain the rotor centerline. The looseness creates nonlinearity in the vibration characteristics of the operating speed and its harmonics as well as changes in the phase relationships between bearing proximity probes. Loosening of rotating components such as the impeller ring, thrust collar, slinger disc, and spacer collar may produce increased synchronous or non-synchronous vibration.

To diagnose these problems, both the displacement time histories and the accelerometer data must be analyzed for changes in their spectrum. Some of these loosening actions have a directionality, possibly requiring the addition of accelerometers or the use of handheld accelerometers to determine the primary characteristics in a 3-D space.

5.2.9 Rubbing

The potential interaction of the rotating components with the stationary components creates conditions of potentially high heating, wear, and breakage. Changes either statically and/or dynamically to the rotor centerline are required for a normally operating T/G to begin to rub.

Rub will manifest itself in the vibration data as an increase in both the displacement of the operating speed harmonics and in the accelerometer overall vibration amplitude. Deeper inspection of the vibration data using orbit plots, time histories, and spectral analysis would provide the following observations:

1. The orbits would show a shift from the normal operating centerline with a potential flattening at the angle corresponding to the contact and an increase in motion around the flattened side, representing the higher excited harmonics as shown in Figure 5-12.

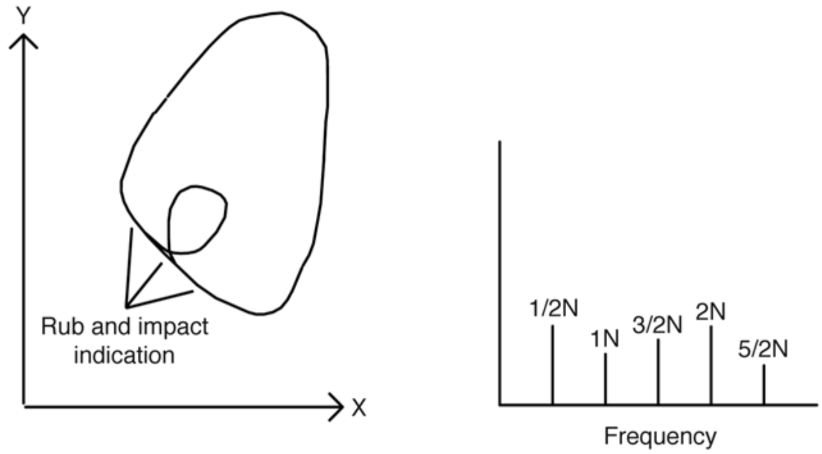


Figure 5-12
Example of an Orbit Plot Indicating Strong Rub-Impacting

2. Review of an accelerometer time history near the rub would show intermittent acceleration bursts as shown in Figure 5-13. These bursts could occur at a period of once-per-rev.

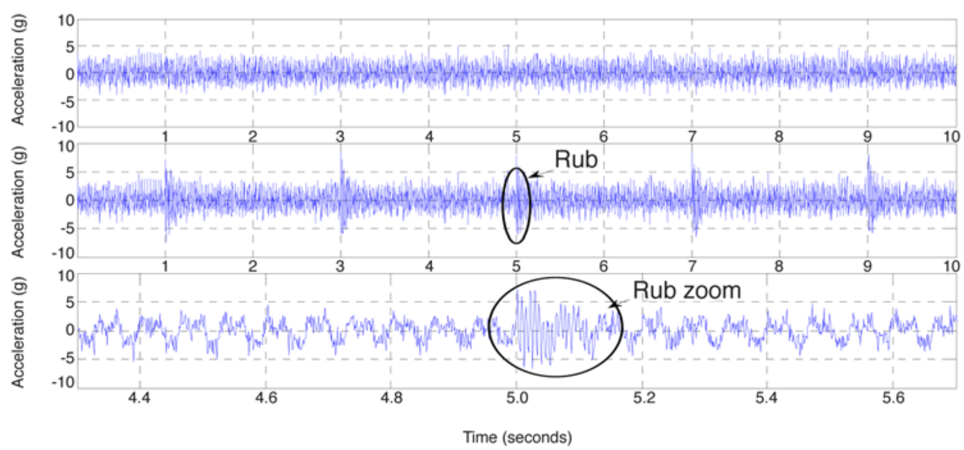


Figure 5-13
Acceleration Time History for No Rub, Rub, and Zoom of Rub

3. A comparison of frequency spectra for the acceleration baseline, normal operation, and the condition where rubbing is occurring would indicate an increase in overall frequency content due to the impact/rub phenomenon. In addition, the stationary component's resonant frequencies may be excited. Figure 5-14 provide two conditions with and without (baseline) rub. In this figure, the operating frequency and its harmonics are shown to increase, and there is an additional frequency related to the stationary component's resonance.

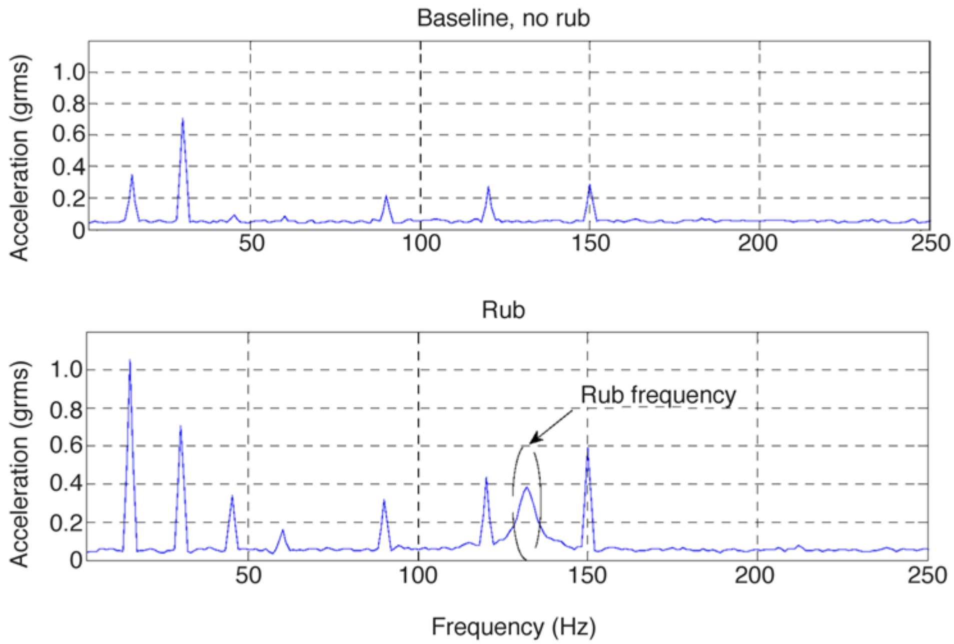


Figure 5-14
Acceleration Spectra for Periodic Rub versus No Rub

5.3 Power Uprate Monitoring Summary

In summary, these vibration monitoring examples emphasize the possibility of "early warning" for deleterious conditions. It was shown that the processed (orbit, spectra, and phase relationships) measured vibration is sensitive to changes in the rotor dynamics. Some vibration characteristics are more sensitive than others for a particular failure mechanism; so the data need to be selectively processed to evaluate a particular failure mechanism. With the ability to trend and create baselines for each vibration characteristic, software can be implemented or may already exist to provide constant surveillance with "early warning" alerts.

The baselines are established from the vibration data gathered during normal operation. These data will be analyzed to provide statistical limits on the baseline that allow for acceptable variation in the vibration characteristics before an alert is issued.

Available today are high-speed, digital data acquisition, monitoring, and protection systems for vibration that work in parallel with real-time condition monitoring and diagnostic platforms that allows the identification and evaluation of potentially harmful conditions at the onset rather than at the alarm or trip setpoints.

It is recommended that this type of baseline monitoring be implemented for power uprates in nuclear power plants to ensure early detection of potential vibration failures.

5.4 Blade Vibration Monitoring

Development of blade vibration monitoring (BVM) systems has received increased attention as the blade lengths of gas turbine compressors and low-pressure steam turbines have increased, carrying with them greater risk and consequence of fatigue failure. Such systems have been employed in research and development for many years but recently have been adapted to continuous monitoring of aircraft engine turbines [46]. Application to large steam turbines has received increasing importance in view of the experience summarized in Section 4, especially the 2008 incident at D.C. Cook, to the extent that similar LP rotors are subjected to quality assurance testing in the OEM's vacuum spin pit facilities with BVM instrumentation.

Two issues should be addressed by the utility contemplating procurement or significant uprating of LP turbines: the retention of OEM or third-party specialists to conduct BVM testing as part of the commissioning process and the installation of one of the commercially available BVM monitoring systems as part of the upgraded turbine instrumentation and control package. The election depends in part on the history of the turbine model, the OEM's ability to demonstrate the robustness of the design, the attitude of the insurance carrier, and the utility's plans for further unit modification. The systems are sophisticated and expensive and require some expertise on the part of the operator in order to maintain them and avoid false calls. On the other hand, installation of a suitable monitoring system could, arguably, have prevented the enormous costs associated with most of the blade losses cited in Section 4. Properly installed and interpreted, a BVM system should warn of the presence of blade cracks in time to take corrective action.

The available BVM systems are based on the timing of the arrival of the tips of freestanding blades or targets on the shroud ring. The position sensors are normally mounted in pairs and may be optical, eddy current, or proximity probes. Optical fibers are mounted at a skew angle of around 90 degrees or as a seven-fiber bundle to minimize the effect of scattering by steam. The dual sensors are positioned in exactly the same axial plane so as to record the time of arrival of the same location on each passing blade tip. Differences in the length of time between arrivals are processed statistically to determine a mode of blade vibration. Obviously, a single probe would be unable to detect a synchronous mode, and multiple probes are employed to detect more complex motion, such as twist. If the motion of a blade differs from that of the remainder of the row—especially if this difference increases with time—the possibility of a crack would exist.

Rotor torsional vibration monitoring is not a substitute for BVM. Torsional vibration has certainly been a leading contributor to blade failures; as a consequence, much attention has been focused on understanding torsional excitation and response, to the extent that the damage to blades can be avoided or, if not, at least estimated. However, as discussed in Appendix A, there are several other mechanisms of blade failure that may not be detectable from measurements of shaft vibration. While there may be lower cost alternatives to BVM for monitoring these mechanisms, they are much less direct and more difficult to interpret than BVM. Examples include erosion/corrosion (conductivity) and stochastic steam buffeting (structure-borne noise).

Section 6: Recommended Analysis and Testing

In Section 3, the major components of the T/G rotor support system are reviewed. The function of each component is described, including the various loadings considered during design. This background is intended to provide a basic understanding of the function of the rotor support system so that the challenges presented by uprates can be understood. In Section 4, different turbine uprate configurations offered by the OEMs are reviewed along with changes to the control systems, operating procedures, and, most importantly, the loading. Section 5 provides the foundation for understanding the requirements for testing and monitoring T/G rotors and their support systems for vibration excitation and response. This section presents a synthesis of this material into a set of recommendations for the evaluation of proposed uprates.

6.1 Rotor Analysis

This section describes the rotor analyses recommended to be performed for T/G power uprates. Accurate dynamic characterization of the entire rotor train is essential. Critical speed assessment—both analytical and experimental—must be performed to minimize risk to the rotor and its supports. In addition, the calculations for critical speed can be used to evaluate bearing loads associated with vibration, misalignment, and imbalance. Finally, proper characterization of torsional modes is necessary to minimize the chances of shaft and blade resonance that can lead to tip rubs and blade loss. The complexity of the required analytical modeling is beyond the capability of most, if not all, utility companies. In this section, the critical issues are presented to assist the utility in discussions with the OEM along with guidance on the information to be requested for independent assessment.

6.1.1 Critical Speed

The transverse vibration of the rotor imposes vibratory forces on the bearings during run-up and coast-down as the rotor passes through critical speeds, both vertical and horizontal. The critical speeds are dependent on the stiffness of the bearing supports and pedestals in addition to the construction of the rotor train. These periods of vibration are brief and infrequent in a baseloaded unit and normally of no consequence from a fatigue standpoint. However, the vibration can cause fretting of bearing retaining bolts if the bolts are insufficiently sized or improperly preloaded. The experience of the D.C. Cook incident suggests a gradual loosening of the retaining bolts, reflected by a change in bearing vibration phase angle over a period of several months, which may have moved a critical speed close to running speed.

The separation between running speed and any critical speed should be sufficiently large that errors in the preloading of bolts or in the assumptions of pedestal dimensions and mounting could not reasonably cause this difference in frequency to be approached. The OEM will likely have constructed a lumped mass or FE model of the rotor train with stiffness values for each support. It is **recommended** that the support stiffness calculations be independently reviewed. The review should include dimensional checks of the supports and their condition along with the structural models. If the support system is not to be changed for the uprate, the accuracy of support modeling should be checked via comparison between the measured and analytically predicted critical speeds. The OEM should adjust the stiffness values to achieve a best fit between calculation and measurement of critical speeds. For a modified support system, the sensitivity of the critical speeds to reasonable errors in the assumed dimensions or assembly should be provided. What is "reasonable" will depend on the accuracy of the structure modeling and the conservatism of bolting, welding, and other means of securing the bearing and pedestal.

It is **recommended** that as a check, impact testing be conducted to obtain modal frequencies for each pedestal. The final check will, of course, be direct measurement of the critical speeds of the modified rotor train.

6.1.2 Torsional Response Model

The OEM will normally provide the results of its analysis of torsional mode shapes and frequencies. The analysis may combine lumped mass rotor elements with structural blade models, in which case the OEM may employ a number of empirical design correlations. Alternatively, the analysis may be based on finite element analysis (FEA) of the rotor coupled with detailed FEA of the complex bending/twisting response of L-0 and L-1 blades. In either case, the response of large latter stage blades significantly affects the amplitude and frequency of the rotor train. The torsional modes do not affect the rotor support structure directly but have been responsible for blade fatigue failures, either as a result of blade/disc vibration modes or forced vibration from a resonant shaft torsional mode. The OEM should also provide an analysis of the torsional impact of a line-to-line short circuit in the generator to ensure that the deflection of a last-stage blade will not result in contact between the blade tips and the casing. This applies primarily to the largest freestanding blades that rely on a large chord for stiffness. The OEM should be questioned about its experience with blade tip rubs, as they may contribute to blade fatigue.

Independent review of the OEM-calculated torsional mode shapes and frequencies is **recommended.**

6.1.3 Loss of Low-Pressure Blades

Industry experience suggests that the greatest risk to the rotor support structure results from the separation of one or more last-stage blades from an LP rotor. Where the loss of blades has resulted in failure of a bearing or standard, the consequences have been catastrophic. This is the main reason for requiring a significant separation between running speed and a torsional rotor mode. Of course, there have been instances of blade failure from many other causes, some of which are discussed in Section 4. Industry surveys repeatedly cite the last-stage blade as the number one contributor to outage time in large steam turbines. While other factors such as corrosion have been found to initiate failure, invariably the fracture shows evidence of fatigue, occurring over periods from a few days to several years. The uprate analysis should strive to minimize the risk of blade failure while mitigating the consequences where possible.

The most direct approach would be to monitor the vibration of the blades, since absence of vibration in a baseload plant eliminates the possibility of fatigue. As discussed in Section 5, the monitoring systems have limited resolution of synchronous vibration and are not simple to install and maintain. It is **recommended** that the OEM be required to disclose the performance of similar LP turbines in the fleet and that the managers of the plants that have performed similar uprates be interviewed to determine their assessment of the risk of blade separation and the reasons for it. The following issues should be discussed with the OEM in the effort to minimize the risk of blade loss:

- Results of spin testing of a bladed disc or full LP rotor to verify blade mode shapes and frequencies; it is **recommended** that the OEM's spin test be witnessed.
- Manufacturing processes such as shot peening to reduce the susceptibility of blade root and disc attachment to SCC and corrosion fatigue.
- Quality control steps that govern material properties as well as blade-to-disc fit-up **and witnessing** by the utility.
- The availability of *in situ* nondestructive evaluation (NDE) methods for crack detection.
- The availability of blade vibration monitoring instrumentation, either continuous or on request.

Mitigation of the risk of consequential damage from blade separation should focus initially on the analysis of bearing reactions to the near-simultaneous loss of a blade and its adjacent neighbor. OEM practice at present is to analyze the loss of centrifugal force applied at the LP disc rim as a quasi-static load for purposes of determining the adequacy of the bearing retention bolts and strongback. It is **recommended** that the OEM apply the impact forces of **two** adjacent blades to the disc in a **dynamic** calculation of the resulting forces at each of the bearings. This analysis should be performed for each of the LP flows in the rotor train. The reasons for this recommendation are twofold: 1) data from blade loss incidents indicate that, in most cases, a single blade release is followed by impact fracture of the adjacent blade, and 2) calculations supported by two failure incidents show that the maximum dynamic force occurs at a bearing remote from the turbine end from which the blades are released.

In addition to analyzing the bearing and bearing support structure for the impact loads resulting from blade loss events, the OEM should model the entire coast-down transient following the loss of blade(s) to ensure that the now "out-of-balance" rotor does not subject any bearing pedestal to excessive vibration as it passes through the critical speeds. As a way to minimize the time spent passing through the critical speed range following a trip on high bearing vibration, consideration should be given to the installation of a programmed vacuum break. A manual vacuum break was employed during the D.C. Cook failure, which interrupted power to the main feedwater system.

Other failure incidents have shown that even a single blade loss can lead to extensive damage to the unit, including severing oil or hydrogen lines and consequential fire or explosion damage. The occurrence of catastrophic damage such as that at Duvha has not been explained, but it is suspected that such extensive damage resulted from the failure of a bearing support. It is **recommended** that the utility conduct a risk assessment of the consequences of high vibration and impact on bearings with a view to the orientation and placement of lubricating oil and hydrogen cooling piping to minimize the forces and moments that might be applied to mounting clamps and joints.

6.1.4 Vibration Analysis

Sustained vibration, unlike critical speeds, poses the risk of fatigue failure of rotor and support components. Excessive vibration amplitude can fail bearings, bearing supports, coupling bolts, and blades (by forced vibration) and is limited by establishing and maintaining appropriate automatic trip levels. Bearing loads should be determined for vibration levels associated with the trip level vibration settings. These estimates can be developed with the aid of the dynamic model constructed by the OEM for critical speed analysis. It is **recommended** that the OEM determine the sensitivity of bearing loads to out-of-balance conditions at each bearing.

6.2 Bearing Analysis

Radial impact from the loss of LP blade(s) causes dynamic reactions on the LP, IP, and HP bearing assemblies. The maximum instantaneous force can occur at any bearing and have any direction in the radial plane. Both the bearing retention ring/strongback and bolting must be capable of sustaining this reaction. Also, for excessive vibration loading, the bolting on the bearing retention ring cover is susceptible to fretting fatigue damage. It is **recommended** that both of these issues be reviewed with the OEM.

The dynamic modeling of the bearing, including its coupled behavior with the oil film and bearing pedestal as discussed in Section 5, is important for accurate calculation of critical speeds. It is **recommended** that the dynamic bearing oil film stiffness incorporated into the OEM's finite element rotor analysis code be reviewed for appropriate design conditions such as those specified in the EPRI COJOUR software program [47].

6.3 Pedestal Analysis

Regardless of the design, pedestal stiffness dominates the dynamic response of the rotor. In addition to critical speed, the response of the rotor train to emergency loading requires that an accurate calculation or test of the dynamic properties of the pedestal be available and incorporated into the rotor dynamic response analysis. In the case of pedestals that are independent of the casings, the horizontal stiffness may be the lowest of that of any of the support components. The loss of a blade could result in contact between the tips of blade rows and the casing, as is reported to have occurred at D.C. Cook and Centralia. It is **recommended** that the OEM provide the details of the finite element model and modal analysis for review. In the case of integral pedestals, the modal analysis will identify any portion of the lower casing that has a resonance near running speed.

6.4 Foundation Analysis

In order to meet code requirements, the foundation of the unmodified unit will have been analyzed by an architectural engineering firm for static and dynamic loading as discussed in Section 3. In general, the changes in static loading associated with an uprate are a negligible fraction of the total loads applied to the foundation. Possible exceptions are the D.C. Cook uprate to the 18-m^2 Siemens LP turbines and any temporary storage of new components on the turbine island. In the case of an EPU, the increased dynamic loading from a generator short circuit combined with other loads could exceed the allowable stress on certain beams. It should not be necessary to address the lowest probability events, such as the simultaneous loss of LSBs and a short circuit, but it is important to understand that there is likely a dynamic magnification of structural loads on the foundation. For this reason, it is **recommended** that the original structural analysis be reviewed. The utility should determine whether a dynamic modal analysis or a simplified quasi-static analysis has been performed on the original configuration. An A/E firm is best equipped to conduct a review for the revised loads and to recommend whether a new analysis is necessary.

In any event, the foundation must be inspected to verify the locations and dimensions of key components. A laser survey of the entire foundation is highly recommended (as was reported for Grand Gulf). At the same time, the foundation should be visually inspected for cracks and spalls, corrosion, tightness of joints and anchor bolts, and any suspect areas targeted for NDE.

6.5 Fire Resistance Considerations

The lubrication and generator cooling piping have been sources of fire/explosion, and the utility should review its plans to minimize the risk of piping failure and the availability of modifications that would mitigate the damage in the event of fire. Piping should be tested with a handheld accelerometer to ensure that any vibration modes are well separated from multiples of the running speed. The resistance of piping to large-amplitude bearing displacement that could result from loss-of-blade accidents should be reviewed.

Continued oil pressure is obviously a potential hazard, as illustrated by the Duvha incidents; yet shutting off the oil supply can result in bearing seizure and failure of the support as in the case of Cumberland. A review of oil supply valves is suggested in order to limit the flow of oil to a damaged bearing or supply line while maintaining pressure to others.

A brief review of generator hydrogen explosions failed to indicate damage to the foundation.

6.6 Recommended Vibration Testing

6.6.1 Bearing Support Structure

Prior to installation of the T/G, while access is available, modal testing of the bearing support structures should be performed to determine the modal characteristics (resonant frequency, mode shape, and damping) of these structures. These tests can be performed with portable equipment that includes impact hammers, accelerometers, and digital recording systems. The results of these tests should be supplied to the OEM for use in the analytical models as a check on the calculated dynamic properties of the bearing supports.

6.6.2 Turbine Blades

When an existing LP turbine rotor is to be reused in the uprate, the latter blade rows should go through a set of tests that can determine the individual and group blade modal characteristics for bending and twist motion. These tests can be performed prior to replacing the top section of the turbine housing using equipment similar to that described in Section 6.6.1. In the event that the diffuser or exhaust hood is replaced on an existing LP turbine, it is **recommended** that the response of last-stage blades be measured over the operating range by a BVM.

6.6.3 Rotor Lateral Resonant Frequency and Critical Speed

During startup of the turbine, the turbine will sweep through lateral resonances, and critical speeds may be measured by the vibration instrumentation installed on the bearings. The critical speeds may vary due to the quality of the bearing installation (alignment, stiffness, and lubrication characteristics) and rotor balance. It is important to note the critical speeds each time the T/G is started up to ensure that some part of the system has not degraded.

6.6.4 Torsion

Based on the risk of a miscalculated torsional resonance, particularly near 120 Hz, it is **recommended** that torsional testing be performed as part of the startup test program for T/Gs that have undergone significant modifications.

There are three types of tests that can provide the torsional resonances: 1) torsional sweep testing with one phase shorted, 2) impulse testing during grid synchronization, and 3) normal grid perturbations during operation. The most accurate method and the one that does not require continuous monitoring is Test 1.

All testing requires special sensors to measure torsion. Two methods are generally used: strain gauges applied to the rotor shaft at one or more locations and probes that measure gear teeth velocity. The probe method is actually readily available on most TGs due to the speed measurements made at the turning gear and the HP end gear. These measurements are adaptable to measuring not only speed but torsional vibration.

Torsional response will also be measured when BVM is employed.

6.6.5 Bearing Housing Vibration

It is assumed that all nuclear plants will be upgraded with the latest in monitoring technology, which would include accelerometers mounted to the bearing housings. These accelerometers are used as a reference for the rotor deflection measurements using proximity probes attached to the bearing housing. Since the bearing housing is directly connected to the support structure, the housing vibration measurements—if compared pre-and post-modification—would indicate any significant vibration changes due to the modifications. It is **recommended** that vibration monitoring instrumentation be reviewed and specified in the vendor’s scope of work.

Section 7: Checklist

The following items are a suggested checklist for use when planning and coordinating the uprate of T/G units. The checklist items are organized by utility, OEM, A/E firm, and third-party activities. The checklist items for the OEM, AE firm, and third-party activities can be included in utility uprate technical specifications for bidding purposes. These checklist items should not be considered strict requirements for technical specifications; rather, they are intended as guidelines to assist utilities when preparing bid specification documents.

Utility Requirements:

- Conduct inspections and dimensional check of rotor supports. Determine whether third-party involvement is needed.
- Review original structural analysis with an A/E firm to determine whether a dynamic modal analysis of foundation was conducted; if not, have A/E firm perform an evaluation of the structural model and loading.
- Review the uprated loads and emergency conditions with the OEM and coordinate with the A/E.
- Witness the OEM's rotor spin test, rotor manufacturing processes (shafts and blades), and quality control inspections.
- Interview managers of plants having the same OEM LP turbine uprate to assess the risk of blade separation and the need for BVM.
- Review the OEM's design analysis models of the rotor train, including the models of the support structure.
- Have assumptions and calculations reviewed by third-party expert, to include rotor and blade vibration, bearing stability and multiple blade loss and its effect on bearing integrity during coast-down.
- Discuss the following with the OEM: vibration instrumentation, its integration with T/G controls, and the need for torsional instrumentation and BVM.
- Review the lube oil and generator cooling piping for vulnerability to vibration and loss of LP blade impact loads.

OEM Requirements:

- Provide the fleet experience with proposed T/G uprated components.
- Provide the comparison between analytically calculated and experimentally measured critical speeds.
- Provide OEM analysis of torsional impact of line-to-line short circuit, including LSB deflection and the potential for blade tip rubs. (Also, provide OEM experience with blade tip rubs).
- Provide rotor dynamic analysis for bearing loads due to the loss of two adjacent blades.
- Determine sensitivity of bearing loads to out-of-balance conditions at each bearing.
- Supply analysis that demonstrates bearing retention ring/strongback integrity and bolting capacity for loss of two blades.
- Provide details of FEA modal analysis of rotor train, pedestals, and bearings; show conformity of models to measured dimensions and properties.
- Participate in foundation inspections for locations of key components (for example, bolt pads and steam line connections).

Architect/Engineer Requirements:

- Review existing calculations for static and dynamic loading of the foundation, and update where necessary.
- Verify locations and dimensions of structural components included in analysis.
- Determine need for laser survey of turbine deck and foundation.
- Determine compliance with codes.

Third-Party Requirements:

- Review support stiffness calculations used for critical speed calculations.
- Review OEM torsional mode shapes and frequency calculations to ensure adequate separation between resonance and running speeds.
- Review original structural analysis of foundation to ensure that no dynamic magnification exists for loss of blade and other combined loading events.
- Perform torsional and BVM tests during startup.
- Inspect foundation for evidence of deterioration that could impact assumptions of the structural analysis (such as laser dimensional mapping and nondestructive testing).
- Conduct a "bump test" on completed support structure to verify bearing support stiffness values.
- Conduct modal testing of the bearing support structures.

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Appendix A: Last-Stage Blade Loss Mechanisms

It is not uncommon for a last-stage blade in a large LP turbine to exert over a million pounds of centrifugal force on the blade attachment arrangement. The sudden loss of a blade imposes an impact load on the rotor that may be dynamically magnified at the bearings. Once a blade has been liberated, a constant radial unbalance force is applied to the shaft close to an end bearing. As a result, the separation of one or more blades represents the greatest risk to the rotor support structure. The unbalanced bearing forces will increase significantly as the unit coasts down through its highest critical speed. The loss-of-blade analysis requested by the owner must consider this amplification at critical speed when evaluating the bearing and pedestal. The leading causes of last-stage blade separation are discussed in this appendix.

Torsional Vibration

The utility industry has experienced a number of blade failures from torsional vibration/excitation, which can be either transient or steady state. Steady-state excitation occurring at twice the line frequency (120 Hz in the United States and Canada) has been responsible for the majority of failures. The excitation is caused by unbalance in the transmission line network. For any transmission line network, there is always some unbalance among the three generator phases. This is also called *negative sequence current*. This unbalance produces a 120-Hz oscillating torque in the generator; its magnitude can be several percent of the generator rating. For more detailed information on the design ratings, consult ANSI C50.13-1977.

Sources of Torsional Transients

The generator short-circuit and out-of-phase synchronization transients discussed in Section 3.2.4 are two of the more common events experienced by T/G systems that produce torsional impulse loading that can contribute to blade fatigue as the oscillations “ring down.” Table A-1 lists other generator transient torque events that should be considered from the standpoint of cumulative fatigue if these events are frequent. Again, it is noted that stress/response amplification will occur if any torsional modes are close to the line and double-line frequency.

Table A-1
Generator Transient Torque Events

Generator Transient Torques			
Type	**Torques**		
	DC	**1X Line**	**2X Line**
Short circuit	X	X	X
Out-of-phase synchronization	X	X	
Transmission line opening	X		
Transmission line closing	X	X	
Unit tripping	X		
Open conductor			X

Steam-Excited Blade Vibration: Synchronous

Nonuniform Flow and Pressures

Excitation caused by nonuniform pressures, velocities, and/or flow angles is experienced by the rotating blade as a time-dependent periodic fluctuating flow and force field. The excitation frequencies must be rotational frequency or multiples thereof. The following are some of the major causes of the nonuniform flow field:

- Inexact matching of stationary blade geometry at the horizontal joint
- Leakage in stationary blade shrouds at the horizontal joint
- Thermal distortion of stationary components that causes ellipticity
- Nonuniform spacing of stationary blades
- Extractions and moisture removal slots
- Exhaust hood recirculation/cooling sprays

It is important to quantify the maximum level of excitation possible in a given blade row. This is normally accomplished by field testing. The resulting vibratory response of the blade to the harmonic excitation will dictate whether one or more of the blades' natural frequencies must be tuned away from harmonics of running speed. *Stimulus* is the term that some manufacturers use to describe the unsteady loading.

Stimulus is the ratio of the blades' unsteady to steady loading at a given harmonic. This relationship implies that the vibratory response is approximately equal to the turbine's end loading. This is important since EPUs are synonymous with higher mass flows/stimulus. Larger last-stage LP blading must be tuned to avoid resonance with the first 8 to 10 multiples of running speed.

Steam-Excited Blade Vibration: Non-Synchronous

There are a number of mechanisms that cause self-excited vibration, including stall flutter, unstalled flutter, rotating stall, and unsteady condensate shock. Only stall and unstalled flutter will be considered in this high-level overview. It should be noted that these two phenomena comprise the majority of self-excited failures in the industry.

Unstalled flutter, sometimes called *aeroelastic vibration*, has been the source of a number of turbine blade–related failures. It is a problem unique to freestanding blades (with or without snubbers). When this problem exists, there is an interaction through the steam between blades that acts like a spring to couple the blades, even though there is no mechanical link (that is, tie wire). It has been shown in units that exhibit unstalled flutter that when the frequency of two adjacent blades is equal, the fluid actually enforces the vibration of the two blades rather than reduces it. In reality, the fluid acts as a negative aerodynamic damper. Blades exhibit this vibration concern at very high mass flow levels. Therefore, when selecting a retrofit, ensure that the blade design has been tested at the flow ranges expected with the EPU. Otherwise, significant risk of last-stage blade loss may exist.

Stall flutter usually occurs in the last stage of an LP turbine under operating conditions of low load and high backpressure. At this condition, the angle of flow entering the upper portion of the blade can reach a large negative angle. Stall flutter can result in significant vibratory stresses and can be the cause of blade failure. For design and analysis purposes, curves relating the lift coefficient versus the angle of attack are generated. These curves are normally developed by cascade test. The instability of stall flutter relates to the rate of change of the lift coefficient with respect to the inlet angle. Telemetry and blade vibration tip displacement measurements can also be used to map the response of the production blade at the backpressure and low flow condition when stall flutter occurs.

Blade Rub

Blade rub is reported to be one of the postulated causes of the LSB fatigue failures at D.C. Cook in 2008. Periodic blade tip or shroud contact with the inner casing is possible when eccentricity in the casing or movement of the shaft is obtained under a number of off-design conditions, including imbalance. Advanced seal designs should allow for non-damaging "soft" rubs under load transients. However, excessive bearing support movement or blade vibration amplitude has resulted in periodic blade-tip drag force.

Disc Dovetail or Pin Slot Failure

Blade release has occurred because of cracking of the grooves or projections on the rotor that anchor the blade roots. Modern designs embody multiple axial or circumferential "hooks" or circumferential flanges with pinholes that provide redundant load paths. Considerable emphasis has been placed on material processing, surface treatment, and assembly tolerances to minimize the risk of cracking in the wet environment of LSBs. Nevertheless, the frequency and cost of NDE of the blade attachments deserve careful review in the selection of an uprated LP rotor.

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