TROUBLESHOOTING TURBINE STEAM PATH DAMAGE MECHANISMS

by

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ABSTRACT

Steam path damage, particularly of rotating and stationary blading, has long been recognized as a leading cause of steam turbine unavailability for large fossil fuel plants worldwide. Turbine problems cost the utility industry as much as one billion dollars per year. Failures of blades, discs, and rotors in both fossil and nuclear steam turbines represent a serious economic loss of availability and reliability for electric power generation suppliers and other energy supplies worldwide. Turbine problems such as deposition and erosion of blades can result in severe efficiency losses, resulting in significant economic penalties. The primary objective of this tutorial is to provide a methodology to identify the underlying damage or failure mechanisms, determine the root cause, and choose immediate and long-term actions to lessen or prevent recurrence of the problem.

INTRODUCTION AND BACKGROUND

Failures of blades, discs, and rotors in both fossil and nuclear steam turbines represent a serious loss of availability and reliability—with significant economic consequences—for steam turbine operators worldwide. Preventing these failures from occurring requires strict adherence to three philosophical beliefs (McCloskey, et al., 1999):

- Understanding the mechanism and root cause of each incident is of paramount importance to permanent alleviation of the problem.
- By understanding what causes a problem to occur, it should be possible to anticipate its development, monitor evolving "precursors," and take early action to avoid a significant condition from occurring.
- A formalized companywide program for correction, prevention, and control can minimize steam path problems in the turbine. Events can emanate from inadequate initial design, poor operation and maintenance, cycle chemistry environments, or lack of proper management support.

Figures 1 and 2 depict some very early turbine steam path failures showing rotating blade erosion and axial disc fatigue (Stodola, 1905; Campbell, 1924).

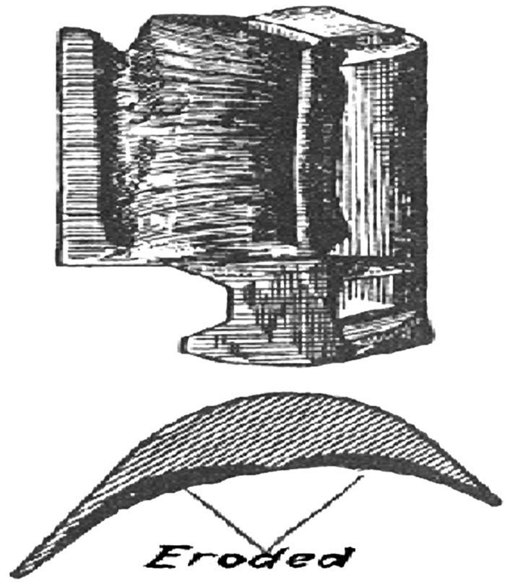

Figure 1. Double Sided Rotating Blade Erosion from Exfoliated Boiler Oxides. (Courtesy Stodola, 1905)

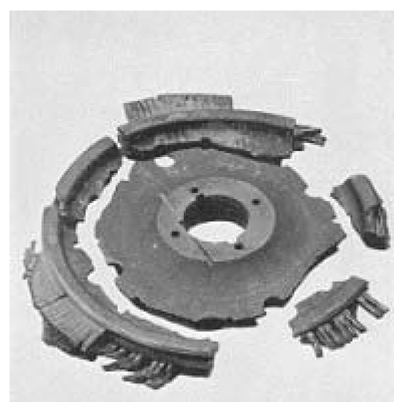

Figure 2. High Cycle Fatigue in Turbine Discs from Axial Blade/Disc Resonances. (Courtesy Campbell, 1924)

FORMALIZING A COMPANYWIDE PROGRAM FOR CORRECTION, PREVENTION, AND CONTROL OF STEAM PATH DAMAGE

This tutorial focuses on technical guidance to understand, prevent, and correct turbine steam path damage. However, it is clear from previous experience that more than just access to proper technical guidance will be necessary to reduce the costs associated with turbine damage. Organizations with formalized, companywide programs and a commitment to reducing turbine steam path damage will be the ones that garner the most significant benefits from the technical experience base.

Aspects of successful programs include:

- Emphasis on the importance of a corporate directive reflecting continued management support for steam path damage reduction activities (as part of ensuring the continued performance of the turbines).
- Emphasis on training and commitment of personnel. It is not possible to overemphasize the importance of operator and maintenance personnel training, experience, and commitment to the health of the unit.
- An emphasis on the importance of a multifunctional team, i.e., steam path failures are not just a maintenance problem, but should involve operators, chemists, and other functional groups.
- A recognition that the long-term view of damage prevention is not only cost effective, but a requirement to enable extended turbine outage intervals.
- Training of key personnel (operators, chemists, maintenance personnel, and management) is central to the success of the program and is a continual process with the addition of new program personnel and/or management. The maximum time between training sessions should be limited to two years.
- The necessary technical understanding and the solutions needed to mitigate outbreaks of steam path damage are known and are available; it is important to make sure that the information is systematically applied to outbreaks of turbine damage and to prevention of damage.
- A commitment to determining the correct underlying cause of damage. Much of the time damage is wrongly characterized, making it impossible to prescribe the appropriate action and avoid a repeat of the same damage in the future. In many cases, the final failure is remote enough from the causing event or series of causing events that the true cause is obscured.
- Remaining life assessment for damage components is a critical part of the successful turbine program. Such assessments, combined with risk analysis, are particularly critical for units moving to longer outage intervals.
- A detailed nondestructive evaluation (NDE) inspection of a turbine by trained staff is essential to detect developing or emerging problems such as corrosion fatigue, stress corrosion cracking, creep, high cycle fatigue, and low cycle fatigue in known susceptible areas such as rotors and blades.
- Established shutdown procedures such as to provide a dehumidified atmosphere to the steam-touched components.

Three key parts of a formal program are:

- A formal corporate directive or a “philosophy statement” to provide action oriented directives and procedures.
- Forming a multidisciplinary team. The turbine condition assessment team (T-CAT) to take responsibility for all actions required to ensure the continued reliable and safe operation of the turbine including preventing steam path damage.
- The comprehensive reporting and trending of steam path condition. There clearly needs to be a responsible and accountable person in each power plant whose specific task is to coordinate all turbine steam path condition investigations.

Figure 3 details these three key aspects of the corporate program along with the activities of the T-CAT.

Other success factors for the formalized program that are addressed throughout this tutorial include:

- Attention to indicators that damage is accumulating,
- Evaluation of unit “precursors” to turbine damage,
- Optimizing inspection and outage intervals,
- Identifying the appropriate root cause of damage,
- Determining the residual life of damaged turbine components,
- Applying permanent engineering solutions to problems identified,
- Maintaining established procedures and careful control over startup, shutdown, and layup conditions in the turbine.

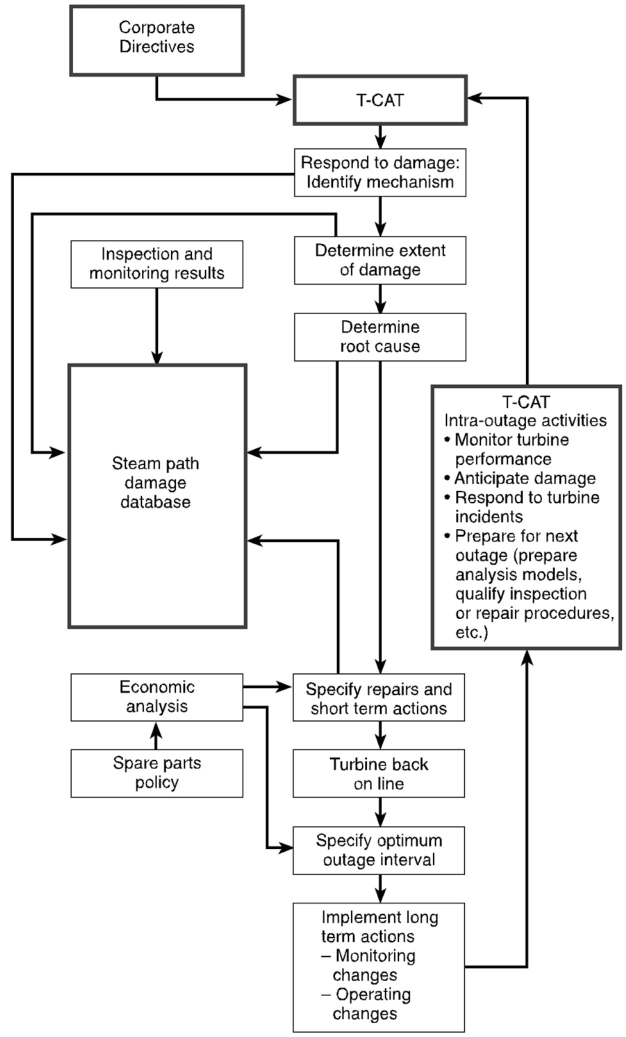

Figure 3. Steam Turbine Root Failure Cause Analysis Flowchart.

Once established, the T-CAT and the formal program will have representatives from, and continued interfaces with, plant operating, maintenance, chemistry, and engineering personnel. For example, cycle chemistry can have a significant impact on steam path damage. Therefore, there is a need to allow operating personnel to direct unit activities so as to set and achieve cycle chemistry goals and thus protect the turbine. Actions might include: application of permanent engineering solutions, development of “controllable” procedures, and the use of instrumentation to monitor critical control parameters. Interaction between the T-CAT and other such plant decisions will be critical.

A useful classification for influences on steam path damage is:

- Operation-controllable,
- Maintenance-controllable,
- Chemistry-controllable,
- Management-controllable,
- Design-related,
- Manufacture-related, and
- Installation-related.

The last three of these are termed “-related” as the owner/operator will have little control over them once the installation has been completed, the exception being when replacements or upgrades are contemplated. In contrast, the first four factors are controllable by the organization. The key recognition provided in such a schema is that specific activities, choices, and controls within the job function of a variety of personnel will affect the occurrence of turbine steam path damage. Training and clear directive is critical for each group to have an appropriate role in preventing turbine damage. Without specific directives, it can be very difficult for operating personnel to convince system control personnel that significant conditions in the unit are "harmful" to the turbine.

CORPORATE DIRECTIVES/PHILOSOPHY STATEMENT AND PROGRAM GOALS

The most important step in implementing an effective turbine steam path damage reduction program is to develop and issue a corporate philosophy statement signed by senior management. The statement provides corporate direction and support for all functional groups within the organization to engage in activities specifically targeted to ensuring the continued safe and reliable operation of the turbine, including reducing turbine damage and associated costs.

The philosophy statement should be built from short- and long-term goals, and must also provide direction for the necessary corrective and preventive actions necessary to reduce damage accumulation.

THE TURBINE CONDITION ASSESSMENT TEAM—MULTIDISCIPLINARY APPROACH AND PERSONNEL TRAINING

A turbine condition assessment team consisting of representatives of all pertinent functional groups (maintenance, engineering, operations, chemistry, and management) should be set up. It is important that all functional groups are represented as this increases the likelihood that each will then understand the implications of their actions on the performance of the turbine and in meeting the overall goals established in the philosophy statement.

The T-CAT will have multiple functions, and between outages the T-CAT should:

- Monitor turbine performance. This may include all existing instrumentation and monitoring systems, the installation of new systems such as through access ports, periodic checks of efficiency, etc.
- Determine whether specific damage indicators are starting to appear.
- Determine whether precursors to turbine damage are occurring in the unit. These precursors might include such conditions as steam chemistry upsets, operational upsets, problems with other equipment, etc.
- If conditions arise that require turbine work, the T-CAT should recommend whether:
  - The unit condition is acceptable to run until the next assessment,
  - The unit requires work to be performed at the next weekend shutdown,
  - The unit requires work at the next unit boiler outage.
- Anticipate and be able to respond quickly to any turbine incidents.
- Establish the optimal interval between turbine outages (refer to discussion below) and to review each step in the planned outage to minimize the length of outage required.
- Anticipate and be able to respond to any damage found during planned turbine outages. This may include such anticipatory steps as:
  - Building preprepared finite element models,
  - Accumulation of databases of materials properties,
  - Comparison of prior inspection records so changes in component condition can be easily made (APPENDIX A Figures A-1, A-2, A-3).
  - Assuming qualified repair procedures are in hand,
  - Advanced analysis to determine which run/repair/replace choice will be used depending on the damage found once the turbine is opened.
- Review spares policy and establish the optimum inventory.
- Determine what ramifications to other parts of the unit may be implied by an incident of damage.

COMPREHENSIVE REPORTING AND TRENDING

Standardized report forms, such as shown in APPENDIX A Figures A-1, A-2, A-3, for blading along with a means to store, evaluate, and disseminate information about turbine condition/damage are required. This will help in diagnosing damage outbreaks, judging the efficacy of imposed solutions, and predicting future problems. This aspect of the formalized program is particularly key in helping to evaluate the risk associated with extending outage intervals. As discussed in detail below, a first order analysis to judge the likelihood of a future problem is typically based on an evaluation of unit or industry-wide history. Without a formal means of accruing the necessary information, such an analysis will have to rely on generic data.

ECONOMIC EVALUATIONS

An economic assessment of technically feasible options is central to the correction of turbine damage. Each organization has, or will develop, a preferred means of performing such analyses when faced with a run/repair/replace decision. Some basic concepts about such analyses are presented here.

It is interesting that there is a long history of economic accountability related to thermodynamics and power production, in fact it was necessary during the earliest evaluations of thermodynamics to have an "effective accountancy for the forms of energy, so that all could be equated with the universal standard—"money" (Bernal, 1970).

The economic analysis will typically consist of some form of cost-benefit analysis or, preferably, a discounted cash flow method such as net present value (to take account of the timing of benefit streams). An integral but difficult part of either type of analysis is determining the expected benefits of a particular run/repair/replacement, and in the case of discounted cash flow methods, in determining the timing of such benefits.

Costs to be considered include:

- Capital cost of repair or replacement. This can typically be calculated from original equipment manufacturer (OEM) or consultant estimates, or, if the work is to be done inhouse, by comparison with prior projects of a similar nature.
- Cost of outage, including replacement power costs.
- Cost of ongoing maintenance for a particular option. This may be obtained from OEM or consultant recommendations, or expectation from inhouse experience.
- Requirement for and cost of periodic inspection or monitoring of the component.

Benefits to be considered include:

- Changes in outage schedule. Run (with existing damage) and monitoring options may require a more frequent scheduled outage than if the component is replaced. Such changes should be factored into the benefit analysis.
- Increases in unit output or improvements in efficiency, if relevant.
- Improvements in operating flexibility and/or unit reliability.
- Benefits of decreased emissions levels, if relevant.

EXTENDING THE INTERVAL BETWEEN TURBINE-GENERATOR OUTAGES

Worldwide there are strong economic pressures to move toward longer intervals between major overhauls. For example in North America, major turbine inspection outages traditionally have been scheduled every five years with a duration of approximately six to eight weeks. Current achievable targets for outage intervals are now 10 to 12 years with a decrease in duration by one to two weeks (McCloskey and Pollard, 1995; Roemer, et al., 1997).

Obviously, with longer periods between major inspections, there is an increased risk of equipment failure. Outage extension needs to be executed with no compromise to reliability of the turbine-generator.

At the most basic level, what is required to make the extended outage decision is an understanding of issues such as:

- Identification of components that will limit turbine-generator safe operating life, e.g., what components are at risk by extending the period between outages?
- Periodic check and accurate assessment of current machinery condition/risk and developing conditions. In many cases, this can be performed using a fiberscope and video probe to examine the condition of the control, reheat, and exhaust stage blading.
- Application of methods for tracking and monitoring the critical components; ideally methods should be utilized that can provide online, real-time reports on accumulated damage or performance degradation.
- Risk and decision analysis based on probability of failure for components being monitored. Such assessments should be able to assess the risks of deferring inspections and take into account a specific machine type and operating and maintenance history. The probability and consequences of outage extension decisions should be calculated.
- Assessment of operation and maintenance practices and the effects on machinery condition/risk.

A proper, quantitative, probabilistic engineering analysis can provide the foundation on which the risk of outage extension issue can be judged, particularly when combined with an economic or financial analysis of the implications of the outage extension decision. A schematic showing the overall problem is shown in Figure 4. Lengthening the interval between inspections increases the equipment failure cost, but decreases the inspection cost. The overall objective is to minimize the total cost.

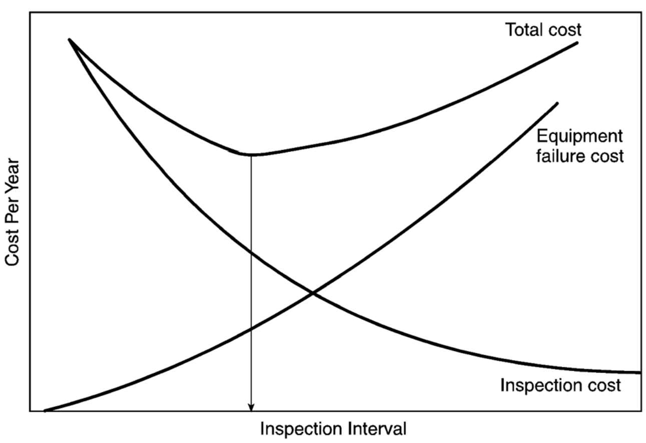

Figure 4. Optimized Turbine Inspection Interval as Function of Failure and Maintenance Costs.

A number of methods have been proposed that can be used to obtain estimates of the probability of failure. (McCloskey and Pollard, 1995; Latcovitch, 1997). Any such methodology should rank components for inspection, focus inspections according to risk, and allocate inspection resources cost effectively. A balance between economics and safety constraints is inherent in such risk assessment.

Three-level approaches will include such steps as:

- *Level 1*—Industry risk data can be combined with unit-specific experience to establish a probability of failure, and from that to establish a net present value versus time for the unit. Such a Level 1 will be formulated to provide guidance about overhaul interval optimization even if no monitoring or engineering analysis has been done on a specific component. It therefore relies on statistical analysis of failure histories. A generic database, such as the North American Electric Reliability Council Generation Availability Data System (NERC GADS) database, can provide knowledge of the high-risk components for a particular unit type, along with history-based probability curves for the highest risk components. Knowledge of unit-specific conditions, as derived from interviews with plant personnel, can also be entered into the calculation of failure probabilities.

The calculation of net present value will be constrained by maintenance budget and forced outage rate limit. Figure 5 indicates schematically the analytical methodology of this level of analysis. Using such an approach, a net present value curve is calculated by analysis; however, choice of outage interval is further constrained by limits on the probability of failure.

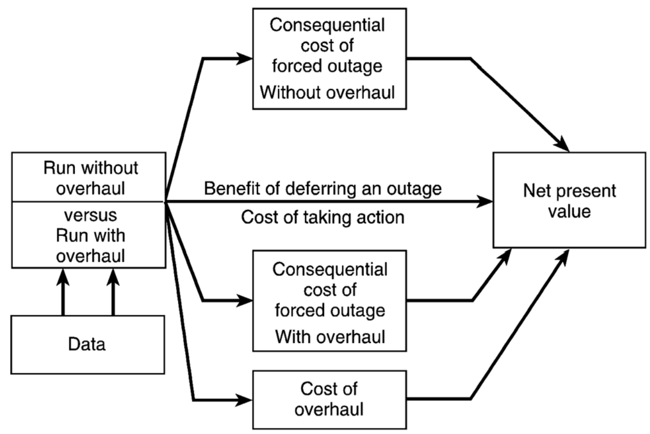

Figure 5. Determination of Net Present Value as Function of Scheduled Versus Forced Outages.

- *Level 2*—Specific component performance and integrity can be evaluated using a rules-based approach to assess key parameters such as stresses. At this level, results from tools such as finite element models, coupled with periodic monitoring to refine the estimate of probability of failure will be utilized.
- *Level 3*—Applies online sensor inputs to assess the consequences of changes in operating conditions and continuously update the probability of failure. Level 3 would apply measured unit data to update component life analyses from a Level 2 analysis. It will track damage accumulated during operating periods.

An overhaul outage inspection interval needs to be optimized by net present value (NPV) and constrained by (McCloskey, et al., 1999):

- Maintenance budget limits.
- Forced outage rate limits.
- Safety limit, expressed as probability of failure or risk limits.

The system will track:

- Cumulative damage or performance degradation,
- NPV incremental and cumulative cost of degradation by turbine section,
- Automate the performance of risk/decision analysis with probabilities of failure and their associated costs.

Shortening Steam Turbine Outage Length

Large economic savings are also inherent in minimizing the length of scheduled outages. Potential timesavings that are being actively pursued include:

- Preparation
  - Listing qualified sources

- "Premodeling" of blade disc stages for damage types and locations that are expected to require analysis or decision about repair choice. Such activities typically include a database of stresses.
- Establishing databases of material properties, chemistry effects, and surface treatments for rapid run-repair decisions
- Support repair specifications for quality assurance of blades, discs, partitions, seals, bearings, spill strips, and packing glands
- Having adequate spares available

- Shutdown and startup
  - Procedures, practices, and techniques are being compiled that can reduce the margin of conservatism in generic loading and starting curves.
- Disassembly
  - Advanced bolting and coupling techniques
  - Cleaning and NDE of key components
  - Automating the recording of clearances during tear down
- Outage activities
- Making alignment changes to rotors and couplings with the minimum number of moves
- Potential for modifications to allow bearings to be isolated and drained on an individual basis, as well as other potential time savings in the oil flushing process
- Assembly
  - Optimization of assembly procedures such as internal and unit realignment and rebalancing

ROOT CAUSE INVESTIGATION OF TURBINE STEAM PATH DAMAGE

This tutorial is focused on problems with the turbine blades, rotors, discs, and seals. For each specific problem, topics include microscopic features of the damage mechanism, common locations, and susceptible components, failure mechanisms and root causes, determining the extent of damage, repairs, and intermediate and long-term actions to be taken. The following is a list of the most prevalent turbine steam path damage mechanisms. (Please note: Because of schedule limitations this tutorial will cover only those damage mechanisms shown in italics.)

- Creep in high pressure (HP) and intermediate pressure (IP) rotors
- Creep and high-temperature damage in HP and IP blades
- Overheating caused by blade windage
- Solid particle erosion
- Deposit effects and removal in HP, IP, and low pressure (LP) turbines
- Copper deposition in the HP turbine
- *Low and high cycle fatigue in LP, IP, and HP blades*
- Fretting wear and fatigue
- Environmentally related damage in blades and blade/disc attachments
- Pitting and crevice corrosion
- *Corrosion fatigue in LP blades*
- *Stress corrosion cracking in LP blades and around the disc rim and blade attachments*
- Liquid droplet erosion of rotating and stationary blading
- Water induction damage, flow-accelerated corrosion, and moisture-related damage

Training and clear directive are critical for each group to have an appropriate role in preventing turbine damage. Without specific directives, it can be very difficult for operating personnel to convince system control personnel that significant conditions in the unit are "harmful" to the turbine. Figure A-4 of APPENDIX A provides a flowchart for the investigation of steam path damage or failures. As shown, there are three avenues open to the investigator(s) depending on the status of the event:

- *"Turbine not opened, damage indicator found"*—The turbine is unopened, but an "indicator" listed in Table 14-1 in McCloskey, et al. (1999) provides an alert that damage is accumulating in the turbine. "Indicators" are discussed in more detail later in this tutorial. Table 14-1 (McCloskey, et al.,1999) can be used to point the investigator toward the correct mechanism, actions to evaluate each indicator in more detail.
- *"Turbine open, damage found by inspection during scheduled outage or as a result of a failure that forced an outage"*—The turbine is open and damage has been found. The damage may have caused a shutdown, such as a failure that released broken blades into the condenser or an extraction line, or damage may have been found during a planned shutdown. The features of damage (macroscopic, microscopic, and locations) can thus be used to identify the possible mechanisms for either case.
- *"Unit precursor observed"*—Table 14-1 (McCloskey, et al.,1999) can also be utilized to determine which mechanism may result from a particular "precursor." Precursors are also discussed below.

As shown in APPENDIX A Figures A-4 and A-5, each of these three paths is an entry point for a series of actions to work through confirmation of mechanism(s), determination and confirmation of root cause, (fatigue, corrosion fatigue, and stress corrosion cracking), and taking short-term and long-term actions to prevent or minimize future damage by this mechanism. A final, but often overlooked, step is to look and consider carefully whether the damage in the turbine steam path is a warning or precursor to damage occurring in the boiler, condenser, feedwater heaters, piping, or other major system components.

Case Studies of Cracking and Damage of Blades in Low Pressure Turbines

Figures 6 and 7 are typical generic examples of the diagnosis of cracking and damage of blades in the LP turbines of fossil and nuclear units. APPENDIX A Figure A-5 provides screening questions to direct an investigator to quickly determine the most probable damage mechanism. Then, in the relevant chapter(s) of McCloskey, et al. (1999), the investigator will find additional means to confirm the initial diagnosis.

Figure 6. Combination of Pitting, Corrosion Fatigue, High Cycle Fatigue in Low Pressure Blade Airfoil Section.

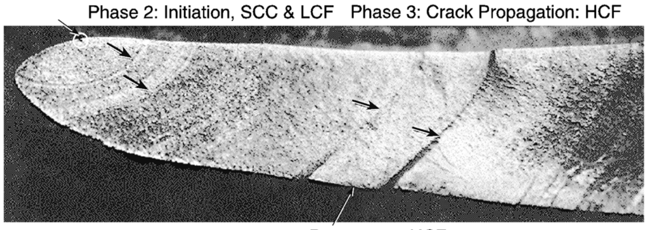

Figure 7. Evolution of Cracking in LP Blade from Corrosion, SCC, LCF, and HCF.

Fatigue, corrosion fatigue (CF), and stress corrosion cracking (SCC) of major steam turbine components, such as blades, discs, and rotors, have been consistently identified among the main causes of turbine unavailability. Particularly susceptible are rotating blades in the phase transition zone (PTZ) of low pressure turbines that have historically been the leading cause of steam turbine unavailability for large fossil fuel plants. SCC of discs and corrosion fatigue of blades and their attachments also affects nuclear units in phase transition regions. The economic impact is even higher than in fossil units.

In McCloskey, et al., (1999), several chapters look at "pure" fatigue of blades, i.e., in the absence of environmental effects; other chapters examine the effects of "pure" environment, leading to damage such as pitting and localized corrosion. Many more chapters examine the synergistic effects of static and dynamic stresses, pitting from the steam environment, corrosion fatigue of blades, SCC of disc rim attachment areas, and SCC of blades (refer to APPENDIX A Figures A-6 and A-7).

Fatigue in Low Pressure Blades

Fatigue in LP turbine blades, with or without environmental effects, is one of the most common underlying causes of steam turbine failures. This tutorial looks at the pure mechanical (i.e., without environmental) aspects of cracking by cyclic loads in LP blades. It cannot be overemphasized that any blade failure by cracking that occurs in the phase transition zone of the LP (typically the last two rows in fossil units and the area beginning anywhere from about the L-3 to L-6 rows in nuclear units) will be influenced by the effect of the local environment.

Unfortunately, although fatigue is a very common damage mechanism, many of the analysis methods for evaluating the root cause and most of the options for preventing fatigue damage will require users to seek outside assistance. Thus, this tutorial covers fatigue in some detail so as to provide an understanding of the underlying issues. In those cases where the influence of environment is extensive, the operator may have significant control over the incidence of failures by the careful control of steam chemistry.

Nature of Features of Fatigue Damage

The fracture surface of a fatigue failure nearly always consists of an initiating defect, a long section of incremental crack growth, and a final failure by overload.

Macroscopically, high cycle fatigue (HCF) typically manifests in a relatively smooth, flat surface with "beach marks" in the propagation region. The beach marks are macroscopically visible ridges representative of transient conditions. Such changes may be in the load level or in environment. Both beach marks and microscopic "striations" (discussed below) expand concentrically from the fatigue origin or origins and thus can be used to help identify the site of initiation. Oxidation or corrosion of the crack tip will lead to discoloration of the metal and help highlight the benchmark. Figure 8 shows a typical high cycle fatigue crack that has occurred at the intersection between the blade airfoil and a tiewire (Sanders and Southall, 1993).

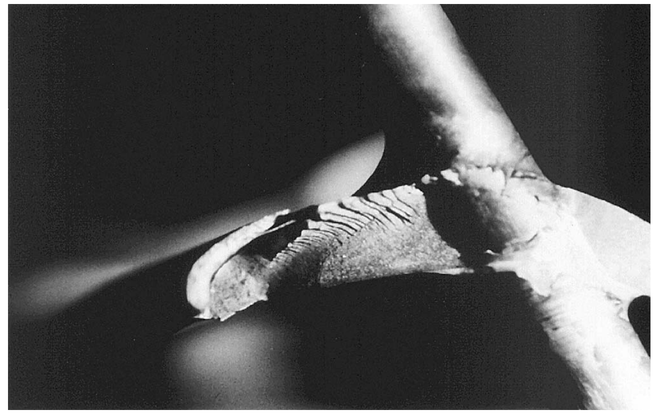

Figure 8. High Cycle Fatigue Failure of Low Pressure Blade at Intersection of Airfoil with Tiewire and Stellite Moisture Impingement Wear Strip.

Microscopically, high cycle fatigue cracking is typically straight, transgranular, and shows fatigue striations. The striations are closely spaced ridges that indicate the crack tip at various points in time.

Low cycle fatigue (LCF) will differ in that there is typically mechanical deformation at the failure surface, i.e., stresses are sufficiently large that they will cause plastic deformation at the leading edge of the crack. Figures 9 and 10 show a low cycle fatigue failure of an L-1 blade. Although low cycle fatigue damage per cycle is significant, the number of large strain cycles that drive such damage is limited, usually less than 1000 cycles for base-loaded machines, and less than 10,000 for topping machines. As a result, the damage caused by crack propagation under low cycle fatigue is localized, i.e., near the initiation site. However, once the crack begins to grow, it is influenced by the much more numerous strain levels that result in high cycle fatigue. Once a blade begins to be influenced by high cycle fatigue, the remaining life of the blade is very limited, only on the order of days or weeks. Thus, in blades, the evidence of low cycle fatigue will typically appear only in the area near the initiation site, while the majority of the surface will show features of high cycle fatigue (Laird and Duquette, 1972).

In blades, the region of final fracture is typically fairly small dimensionally so that common features of fracture such as chevron marks (sharp V-shaped patterns whose tips point back toward the origin) will generally not be obvious.

If there is evidence of staining or deposition on the fatigue surface, it is highly likely that there has been a contribution from the steam and water chemistry environment.

Susceptible Units and Locations

Fatigue is ubiquitous; it can cause damage in the blading of all turbine types and stages. It tends to be more prevalent in the latter stages of LP turbines because the length of the blades leads to higher stress levels and allows for a variety of resonances.

Location on an individual blade. Fatigue damage can occur almost anywhere on a blade, particularly where there is a stress concentration. As a result, on LP blades, the roots, steeple fillet regions, shroud tenons, tiewires, and tiewire holes are the most susceptible. For example, Figure 11 shows the fatigue failure of an LP blade that was caused by breaking of the outer tiewire. The subsequent change in frequency caused the blade to fail by high cycle fatigue, with the damage originating at the inner tiewire hole. Figure 12 illustrates the most common locations of fatigue damage on LP blades.

High cycle fatigue can occur anywhere along the length of the blade. One reason for this is that dynamic (vibratory) stresses can reach a maximum anywhere along the blade length depending on

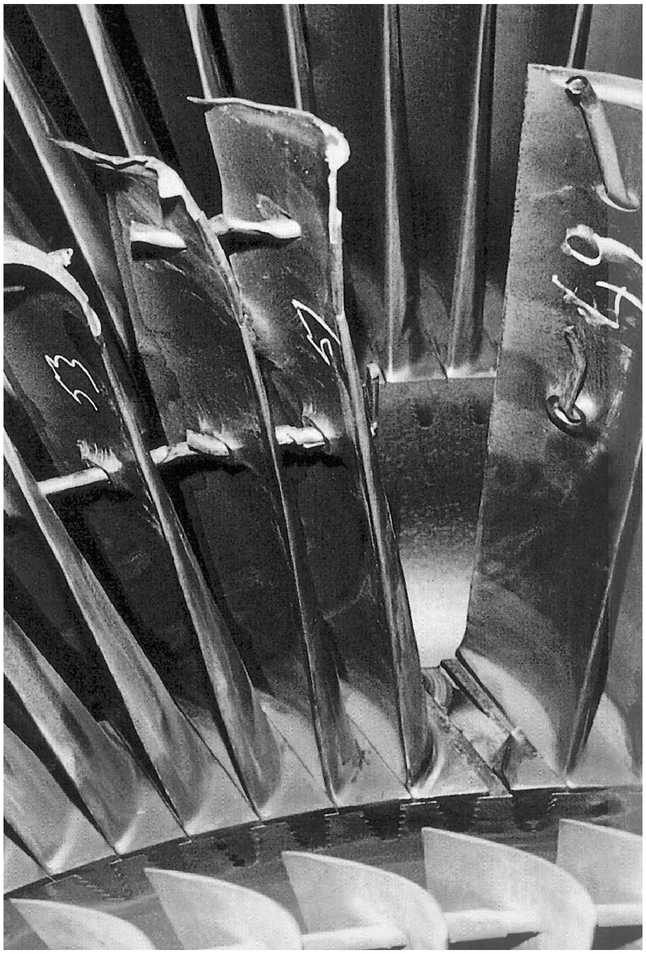

Figure 9. L-1 (Next to Last) Stage Blade Failure at Base of Airfoil Due to Low Cycle Fatigue.

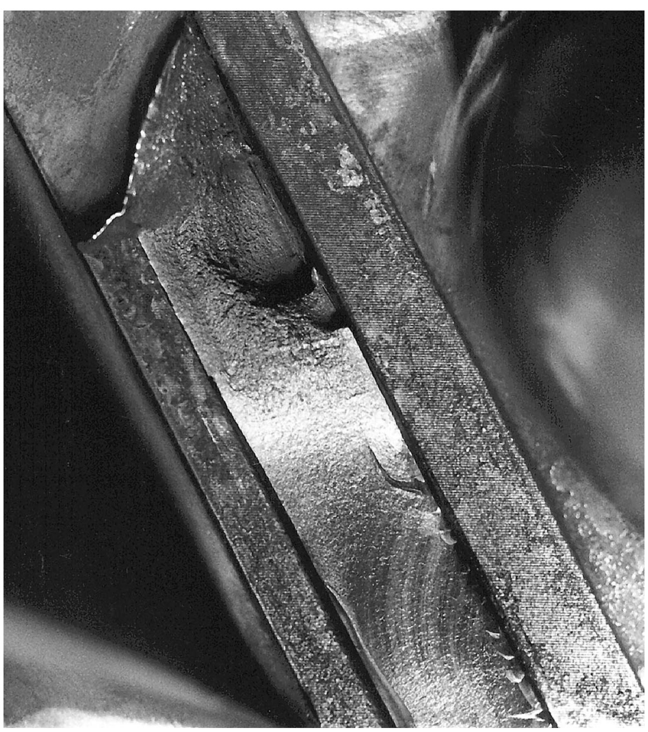

Figure 10. Closeup of Low Cycle Fatigue Failure Shown in Figure 9.

Figure 11. High Cycle Fatigue of Low Pressure Blade at Tiewire, Caused When Crack in Tiewire Changed Natural Frequency of Blade in Group Packet.

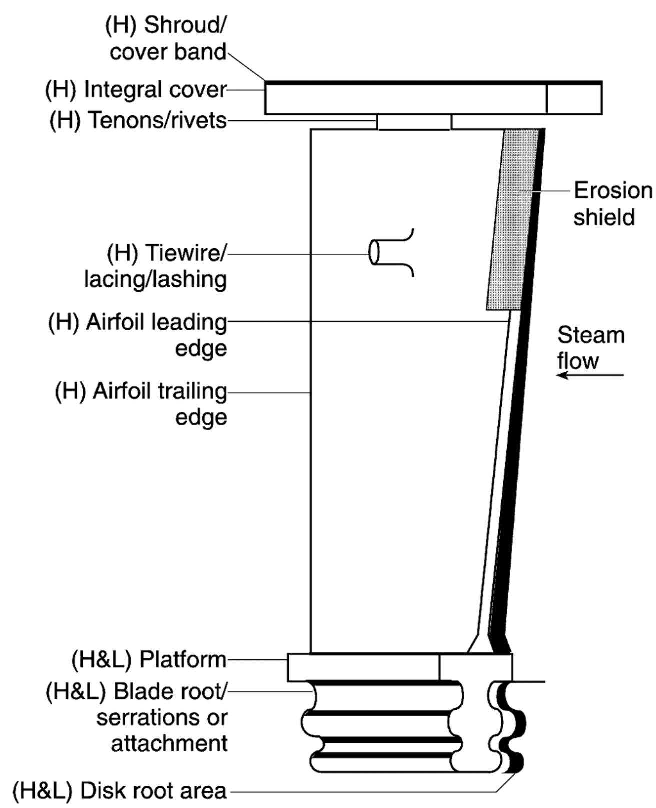

Figure 12. Most Common Locations of Fatigue Damage on Low Pressure Blades. (H indicates typical locations of HCF; L indicates typical locations for LCF.)

the active mode shape. This is illustrated in Figure 13, which shows the level of vibratory stresses for the first five modes of a 3000 rpm blade. The y-axis in Figure 13 is the maximum relative dynamic stress. From Figure 13, for example, a failure by the third mode of vibration might occur at the tiewire as the maximum dynamic stress occurs at approximately 0.8 of the distance to the blade tip (Somm, 1976). Low cycle fatigue typically occurs only in areas near the blade root and is typically caused by speed cycling.

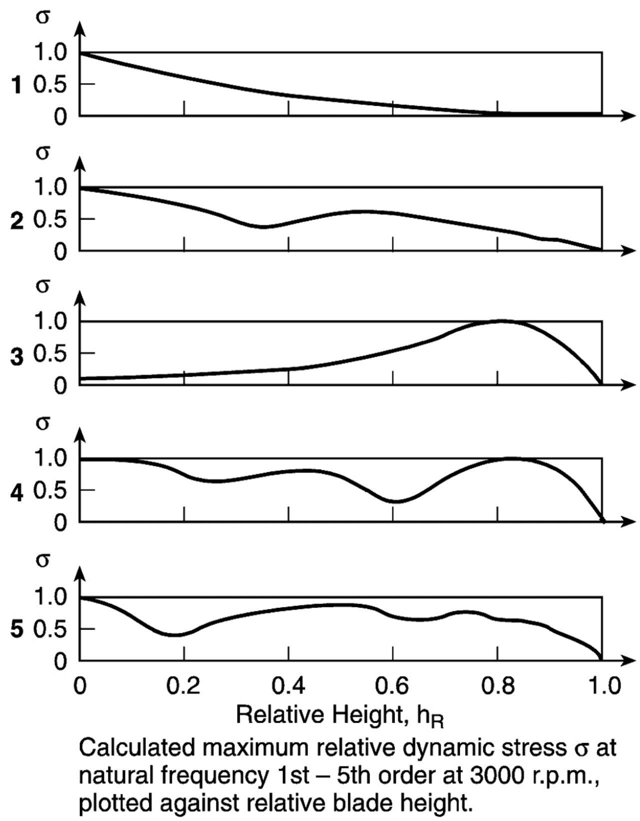

Figure 13. Distribution of First Five Harmonics of Vibratory/ Dynamic Stresses as Function of Blade Length.

Affected rows. Most failures in turbine blading of fossil units occur in low pressure blades, particularly concentrated in the L-0 and L-1 stages. Unfortunately, although in some cases location by row may help determine the applicable loading, there can be a considerable range over which particular causes can lead to blade fatigue failures.

Each turbine blade row of a particular turbine design tends to have distinctive problems. For example, the operating life of L-0 rows of a particular turbine design is generally controlled by low cycle fatigue caused by start-stop cycling, although in extreme off-design conditions stall flutter may also cause fatigue damage by a high cycle fatigue mechanism. In contrast, the L-2 and L-3 rows of the same design are generally controlled by high cycle fatigue typically as a result of resonance with particular blade-disc modes with per-rev harmonics of rotational speed (Dewey, et al., 1983).

Affected blades or groups of blades. Fatigue cracking patterns may indicate the type of loading that lead to a fatigue failure. For example, cracking has been found to originate in microcracks in the tenons, induced during assembly of shrouds that subsequently propagate by steady-state and vibratory stresses. Torsional vibration is more typical in the shroud/cover whereas axial vibration (the higher energy mode) tends to be the dominant mode in the blade root. The dynamic stresses caused by resonances are higher in the end or next-to-end blades for the most troublesome modes, such as tangential and torsional vibrational modes.

High cycle fatigue may tend to selectively affect blades as slight differences in blade manufacturing and installation can result in slight differences in response to excitations. Flutter, a form of high cycle fatigue caused by an aerodynamic instability, may result in clusters of failures. Figure 14 shows the pattern of high cycle fatigue in a row of freestanding, titanium, and L-1 blades (Rust, et al., 1990).

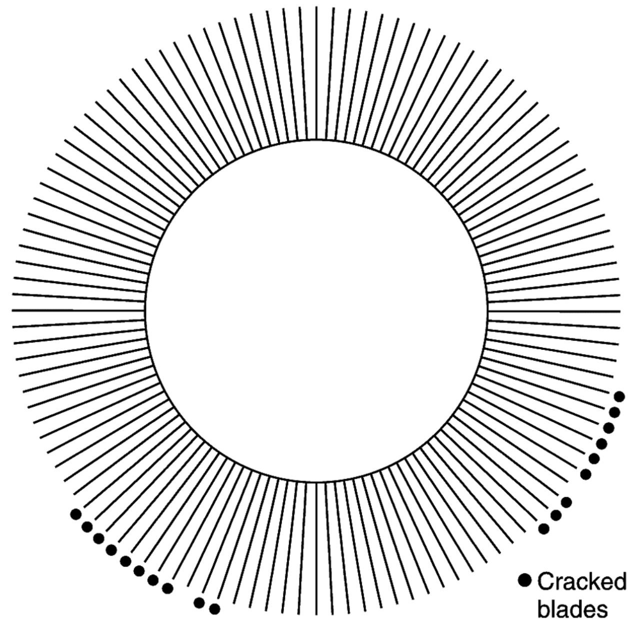

Figure 14. Distinct, Closely Tuned Clusters of Blade Failures in Row of Aerodynamically Unstable Titanium L-1 Freestanding Blades.

In blade groups, the mode of resonance causing a high cycle fatigue failure can usually be determined by examining where in the group the failure occurs, and the origin and direction of crack propagation.

Low cycle fatigue, which is caused by the cycling of centrifugal stresses in the root that accompanies unit speed changes, will typically affect multiple blades since groups or individual free-standing blades in a row experience a fairly consistent amount of centrifugal loading. However, trailing or leading blades within a group may experience somewhat earlier failures as a result of covers or tiewires redistributing the overall centrifugal loads toward the outer members. Indications that this damage mechanism is active are shown in Table 1.

Table 1. Indications Damage Mechanism Is Active.

Appraisal Means	Indicator
Without Opening Turbine	Sudden change in journal bearing vibration and/or change in phase angle caused by loss of a blade or shroud as a result of fatigue damage. Increased vibration of blades may indicate either that fatigue has become active or that loading is higher than anticipated and could initiate fatigue.
With Opened Turbine	Presence of fatigue damage found by inspection. Loss of blade. Unit precursors indicating that this damage mechanism may have become active.

There are few unitwide precursors for this mechanism. One is detecting vibration in the rotor as a result of imbalance, indicating a forcing function that, if unchecked, could lead to high cycle fatigue in blades.

It should be noted that fatigue will occur in steam turbines, particularly in the last stages of the LP and first stage of the HP turbine. A key to the design and fabrication process is to provide acceptably (long) fatigue life, including an allowance for environmental effects.

Mechanism of Fatigue Damage

Fatigue is the phenomenon of damage accumulation caused by cyclic or fluctuating stresses. A simple illustration of the nature of the cyclic loading is shown in Figure 15 where *smax* = the maximum stress, *smin* = the minimum stress, s*a* = the stress amplitude or alternating stress, *sm* = the mean stress (equal to half the sum of *smax* + *smin*), and *sr* = the stress range.

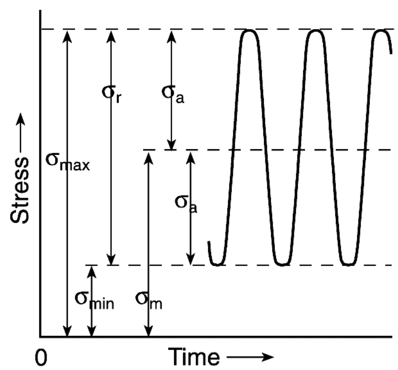

Figure 15. Graphical Definition of Both Dynamic and Static Fatigue Stresses.

In turbine blades, fatigue can be either low cycle (high stress) or high cycle (low stress); there is no fundamental difference in the mechanism between the two types of fatigue. High cycle fatigue is characterized by loads that cause only local plastic deformation (small-scale yielding), while the rest of the component is loaded well within the elastic range; low cycle fatigue is typically caused by loads that produce considerably more plastic deformation. The precise distinction between low cycle fatigue and high cycle fatigue is somewhat arbitrary. Generally failure that occurs in less than 104 to 105 cycles is considered low cycle fatigue; failure in a greater number of cycles is high cycle fatigue.

Low cycle fatigue is incremental crack growth caused by large stress cycles such as centrifugal stresses that occur during unit start-stop, overspeed, or thermal cycles. Low cycle fatigue can also occur if the original design of the blade was inadequate to accommodate the operating loads, or if the blade contains manufacturing defects or shortcomings.

In contrast, high cycle fatigue damage accumulates over a much larger number of cycles at lower stress (strain) levels per cycle. Many blade failures occur as a result of high cycle fatigue, caused by a variety of loading conditions. The most typical failure occurs when the frequency of the excitation is near a resonance frequency or harmonic of the blade. The rate of high cycle fatigue crack growth can be rapid as a result of the superposition of dynamic (vibratory) stresses in combination with the high mean stress induced by centrifugal loading. The dynamic stresses that develop depend on the frequency and shape of the exciting force, the response of the blade, both natural frequency and mode shape, and the energy dissipated through the various damping mechanisms present.

A factor that complicates the diagnosis of LP blade failures is that they typically occur by a combination of mechanisms. For example, initiation may occur as a result of high stresses at a stress concentration (low cycle fatigue) with subsequent damage accumulation (propagation) driven by lower stresses more typical of high cycle fatigue. Further complicating the diagnosis is that in the phase transition zone, the rate at which both these stages, initiation and propagation, occurs is dramatically influenced by the local environment.

Stages of Fatigue Crack Growth

Fatigue is typically manifested as three distinct phases:

- Initiation,
- Stable propagation of a crack, and
- Final failure that ensues when a critical crack size is reached and failure occurs by fracture or overload.

Fatigue is dependent upon the frequency and magnitude of the stress cycles and is generally independent of stress duration.

Origins for fatigue cracks can include:

- Locally aggressive environment that produces corrosion pits or troughs, localized corrosion, or local dissolution.
- Rubbing or fretting damage.
- Stress concentrations such as blade roots, tiewires, shrouds, tenons, and erosion inserts.
- Material processing or fabrication defects such as: forging segregation, exposure of end grains, laps or folds in the material, tears in forming or stamping, casting porosity and cold shuts, and various weld problems such as porosity and cold working.
- Grinding or machining tool marks.
- Inadequate or improper repairs.

Fatigue, although it takes place under globally elastic conditions, can occur only with plastic deformation on the microscopic level, typically only at the crack tip. The sharp crack grows as a result of alternating stresses that cause crack extension and blunting. This extension and blunting causes the characteristic striations. One striation is formed for each cycle, which, in many materials, can allow for careful estimation of the crack growth rate by measuring their spacing using an electron microscope.

POSSIBLE ROOT CAUSES OF FATIGUE

The primary root causes of fatigue failures are cyclic stresses. There are myriad sources of steady-state and dynamic stresses that affect turbine blades. All the principal sources of stresses in LP blading, the nature of the loading (steady-state or dynamic), the source or root cause of the stress, and its contribution to fatigue through increased mean stresses, as high cycle fatigue or low cycle fatigue, all need to be carefully identified and analyzed. The key for the turbine engineer will be to identify the specific cause of the problem so that proper corrective action can be taken; unfortunately, there are few easy ways to do so. Typically, a blade assessment including finite element analysis and fatigue life prediction will be required to fully confirm the underlying root cause of a failure.

DETERMINING THE EXTENT OF FATIGUE DAMAGE

The choice of inspection method will depend on the location. Applicable methods include: visual inspection, liquid penetrant testing, magnetic particle, ultrasonic testing, and eddy current testing. These methods are listed in order of increasing sensitivity for crack detection. Visual and magnetic particle methods are the most widely used for field inspection, although experience has shown that these methods are not always effective in finding tightly closed cracks. If cracks are found, the extent of cracking must be accurately assessed. In many cases, this may require removal of some blades.

BACKGROUND TO REPAIRS AND IMMEDIATE ACTIONS

Most of the strategies to correct, or at least to mitigate, the root cause of fatigue are longer term. Repair, refurbishment, reblading with available replacement, or removal of a damaged row may be used as temporary measures to get a unit back online, or in parallel with longer term actions to deal with the underlying cause. In no case should the simple replacement of blades be considered the end point of the indicated actions; if the underlying cause is not corrected, the failures will typically recur.

When cracks are small, they can usually be removed by local grinding. This has proven effective in corner cracks at top serrations in L-0 and L-1 roots. Alternatively, it may be possible to remove the crack by machining, while still maintaining a sound load-bearing surface between blade and disc. Leading and trailing edge cracks can also be blended out. "Polishing" of accessible areas such as the trailing edges of blades has been found to be beneficial by removing pitting (Ortolano, 1987).

Shotpeening is, on balance, beneficial. It introduces surface compressive stresses that improve fatigue and corrosion fatigue resistance, although it does not change the material's resistance to pitting or generalized corrosion. Further, the long-term benefits of shotpeening are not presently known.

Refurbishment may not be capable of returning damaged blades to new condition, but may be sufficient to return them to operable condition. This may result in some loss of efficiency, but the ability to return the unit to service quickly and with minimum downtime may justify such a decision.

There is probably no part of a blade that cannot be repaired; however, as with all repair/replace/refurbish options, an economic analysis, such as described below, should be performed.

BACKGROUND TO LONG-TERM ACTIONS AND PREVENTION OF REPEAT FAILURES

Figure 16 provides a generic list of options for addressing fatigue failures in LP turbine blades. The shaded boxes indicate those steps or options that can typically be executed by a user; the others may require outside assistance. It is important that the underlying loading problem be clearly identified so that the proper choice of preventive action can be taken. An economic evaluation of the various options should be the first step in the process.

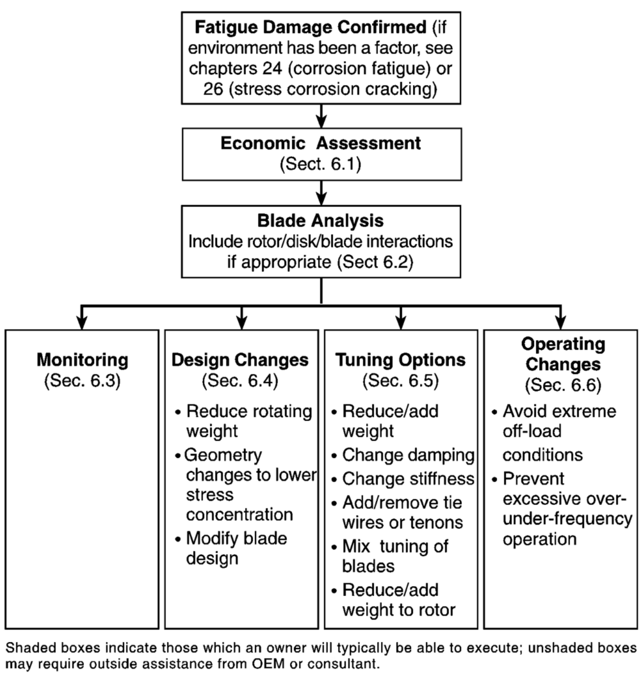

Figure 16. Long-Term Technical Options for Addressing and Correcting Fatigue in LP Turbine Blades.

Economic Evaluation of Blade Damage Due to Fatigue

As with many other damage types, there may be multiple potential solutions. An economic analysis of the options will help guide the user toward the optimal decision. Such an economic evaluation, including the required availability of the unit, the type and cost of repair, the availability of replacement blades, cost of analysis, testing, cost of potential modifications, and expected life of any modification should accompany a set of long-term actions.

Analysis of Steam Blade Path Damage Due to Fatigue

Blade analysis is performed for a variety of reasons:

- Confirming that fatigue is the root cause of the failures observed,
- Comparing alternative fixes to a problem, and
- Estimating future reliability by determining life.

A generic procedure for assessment of blade life when subjected to fatigue is shown in Figure 17 (McCloskey, et al., 1999).

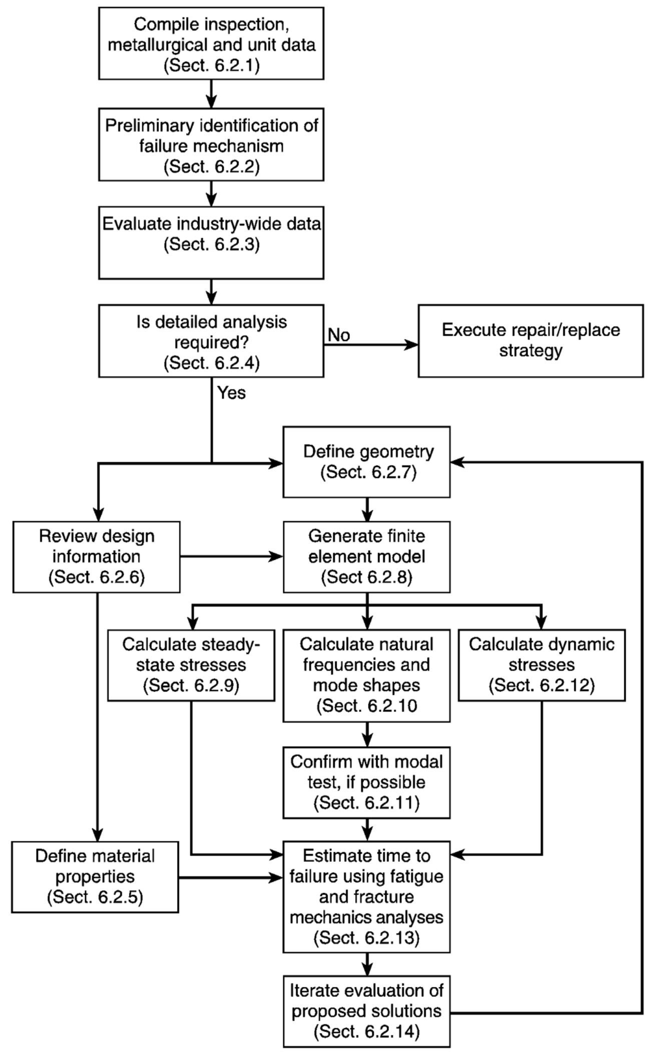

Figure 17. Low Pressure Turbine Blade Remaining Assessment Flowchart. (Sections refer to McCloskey, et al., 1999.)

The following describe the assessment outlined in Figure 17. As it is a generic procedure, the analyst should pick the appropriate steps depending on the nature and extent of the failure.

Compile inspection data, metallurgical results, and unit data. Evaluating where the damage is occurring helps determine which stress sources are causing the problem, and thus narrow the scope of the more time consuming and expensive analytical methods. Inspection results should include answers to such questions as:

- What is the general nature of the damage? Is rubbing, fretting, or deformation seen with the fatigue cracking?

- What can be learned from the location of failures?
- Which stage or stages are affected?
- Are the failures in individual blades or groups?
- Are groups of blades affected? In groups, are all blades equally affected? Which blades in the group are damaged (only leading blades, only trailing blades, both leading and trailing blades, all blades equally, etc.)?
- Where on the damaged blades is the damage occurring?
- What can be learned from the geometry of the blade and attachments near the area of failure? Are there measurable gaps between the blade root and disc rim attachment? Are tiewire, tiewire hole, tenon, or shroud tolerances or geometries implicated in the failure? Are failures associated with a stress concentration?
- Can exciting forces be measured? If such forces can be measured, can they be compared to the speed of rotation?
- Can the blade vibration modes be measured directly? Perhaps the most direct method to confirm the underlying driving force behind a fatigue failure is to directly determine which blades in a row are vibrating, the corresponding modes, and the magnitude of the vibration. The principal means for online vibration measurement is a time-of-arrival method using magnetic sensors such as a blade vibration monitor (BVM) shown in Figure 18 (Rozelle, et al., 1989).

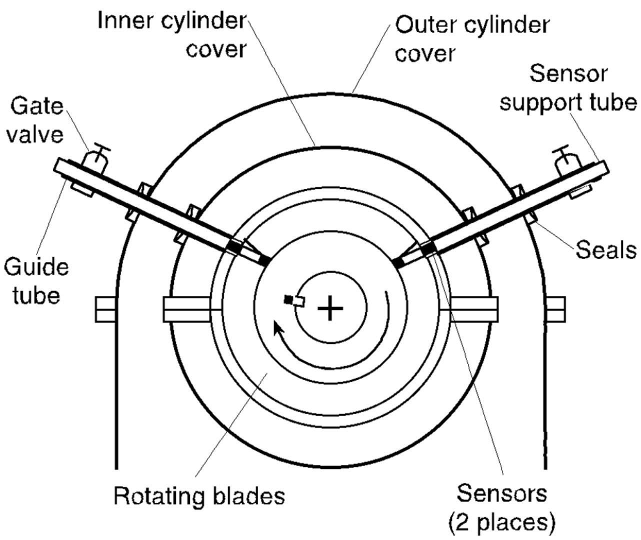

Figure 18. Noncontacting, Online Steam Turbine Blade Vibration Monitoring System.

Information from metallurgical examinations should also be gathered. A summary of key elements of a generic metallurgical failure examination is as follows:

- Identify the source of initiation such as a manufacturing or material defect, fretting, corrosion, erosion, or impact.
- Evaluate the role of the environment. The first step in this process is to plot the locations of failure on the heat balance diagram and on a Mollier diagram. Next is a review of steam purity recommended limits. The third step is a detailed surface analysis of deposits and cracks using techniques to determine the presence of key anions. Is there evidence of significant deposits?
- Measure fatigue crack spacing. Can low cycle fatigue be identified as the primary mode of propagation of the cracking, or is it high cycle fatigue?

Unit information will also be required, including load history and chemical history of steam. The load history of the unit may include such information as:

- Unit load cycling (a key determinant of low cycle fatigue),
- Load changes (which can help account for thermal fatigue),
- Number of overspeed governor trips that have occurred,
- Period of time that blades were at overspeed conditions,
- Conditions that may have lead to subsynchronous or supersynchronous operation.

The period over which the failure has occurred will be an important clue and help identify, for example, whether a source of stress consistent with low cycle fatigue or high cycle fatigue is appropriate.

The following questions should be considered:

- Can failures be related to specific unit operation?
- Are cycles to failure related to unit start-stops, for example?
- Can failures be related to low load operation?
- Can failures be related to operation with high backpressure?

Preliminary identification of the failure mechanism is needed. From the inspection and metallurgical data it should be possible to identify the general failure mechanism.

Evaluate available information on similar failures from unit, sister unit, or industry experience. In some cases, it may be possible to gain considerable insight about the problem and likely fixes from past incidents.

Determine whether a detailed analysis is required. Once fatigue is suspected or has been confirmed, it is likely that a detailed analysis is required. Only very special circumstances would change this decision. Such situations would have all the following characteristics:

- The failure is isolated to a single blade or only a few blades,
- The cause can be clearly identified,
- The fix is straightforward, such as a simple replacement or weld repair,
- It is clear the fix will achieve the owner's goals for blade life.

Such a set of circumstances might occur, for example, where there was an isolated blade manufacturing defect that had been clearly identified as the source of an isolated fatigue failure.

Define the material properties. These may come from historical records or open literature values, preferably with confirmation of critical properties by testing. One of the difficulties in any detailed assessment is dealing with the natural scatter in actual material properties. Judgment will be required to balance the amount of data collected against the cost to do so. Material properties of interest will depend on the damage type being evaluated but will typically include:

- Strength,
- Fatigue crack growth rate,
- Fracture toughness, and
- Hardness.

If the failures have occurred in the phase transition zone, crack growth rates corresponding to corrosion fatigue will be more relevant than those typical of pure fatigue.

Gather and evaluate design information. Is the Campbell diagram available? Are there obvious blade modes that are more likely to have had resonance with per-rev excitation (Campbell, 1924)?

Define the geometry of the blade, attachments, and damage. Dimensions should be specified from plant measurements.

Generate a finite element model of the blade. Models should be defined using actual dimensions of all relevant stress concentrations, nature of blade groups (freestanding or grouped, integral or riveted covers, tiewires, etc.), and detail of attachments. If failure appears to be a disc-blade interaction, the investigator will need to expand the model to include the disc; rotor modeling may also need to be included if rotor torsional modes are suspected to be an underlying root cause of the failure.

Calculate steady-state stresses. This should include an analysis of such stresses as centrifugal stress, steam bending, and any steady-state thermal stresses. This analysis should identify locations of maximum stress and their distribution throughout the blade. The effect of stress concentrations such as at the root attachment and in tiewires or shrouding should be included. Three-dimensional analysis is typically required to identify potential high stress areas; a two-dimensional analysis may be sufficient to focus on key locations for more detailed evaluation.

Calculate natural frequencies and mode shapes. Plot a Campbell diagram or interference diagram to illustrate the relationship of mode frequencies to rotor speed. Suspect a resonance problem if blade frequencies are close to per-revolution harmonics. Plot the modal shapes. Applications of these techniques for the diagnosis of failures are given in the case studies later in this tutorial. It may be necessary to take into account the changes in blade frequencies caused by corrosion, erosion, or deposit build up in the analysis of underlying root causes. This calculation may also include effects that occur at operating speed but would not show up in a static model of the blade such as stress stiffening, causing blade frequencies to be higher than would be measured at zero rpm or spin softening (Campbell, 1924).

Confirm with a modal test where possible. Blade frequencies and modal shapes can be documented at zero rpm and compared with the finite element results. If they compare well, it is reasonable to assume that the dynamic behavior of the blade is also well represented.

Calculate the dynamic stresses in response to unsteady forces using an appropriate forcing function. These might include such sources as nozzle-wake interactions, flutter, or other unsteady dynamic steam forces. Estimate time to failure using fatigue and fracture mechanics analysis. A typical fatigue analysis will use a local strain approach that allows for the cumulative effect of multiple strain sources such as modal resonance and mean strains from centrifugal loading. Input to the fatigue analysis will include the material, unit history (start/stops), and operating temperature. Steady stress amplitude, dynamic frequency, and stress amplitude determined from prior calculations are also used in the fatigue analysis. Fracture mechanics will be used to determine the lifetime after the formation of a crack.

Iterate the analysis to evaluate alternative solutions. Options need to be compared to demonstrate that superior performance in fatigue can be expected. This step is critical if geometry or material changes are anticipated, such as the introduction of longer (or shorter) shrouding, changing to or from freestanding blades, or changing root attachment geometry. Where operating practices can be modified to reduce the rate of damage accumulation, these effects should also be quantified.

It has been estimated that given the various uncertainties in material properties, excitation, damping, and cumulative damage estimates, current blade life estimates are within a factor of three to four of measured life values (Rieger and McCloskey, 1988). For this reason, use of a finite element analysis (FEA) code and similar calculations should be used primarily as relative indicators of performance, not as absolute predictions of life.

Monitoring of Blade Vibratory Stresses and Mode Shapes

Monitoring of actual blade vibration should be strongly considered. As a minimum, a means to determine whether any modification has actually lowered the driving forces on a blade is recommended. Many operators may also wish to place long-term online monitors on highly susceptible rows. A continuous monitoring system to detect rotor vibrations and the resulting rotor-blade coupling reductions in blade fatigue life can also correct line operations such as high speed reclosing while taking into account fatigue life consumption (Tsundoa, et al., 1989).

BLADE DESIGN MODIFICATIONS

Several design changes to blading can be made. These typically require the participation of outside consulting organizations or original equipment manufacturers. This category might include going to freestanding blades, changing from freestanding to grouped or continuously banded blading, or adding damping devices between blade groups. Other design changes include:

- Reduce rotating weight. This strategy aims at reducing blade centrifugal stresses by using a lighter shroud (titanium, for example), cover, or lighter blades (such as by switching to titanium). Note that tuning options described below may also reduce weight but with a different intent.
- Geometric changes to lower stress concentration. This strategy also aims to reduce blade stresses, particularly in the root attachment where cracks have often initiated, by enlarging the hook radii to reduce stress concentration. Other susceptible locations that might benefit from modification of geometry include: vane-platform fillets, tiewire fillets or hole edges, and cover or attachment discontinuities.

Blade Tuning Options

- *Reduce/add weight to blade*—Stiffening the lower region of the airfoil or by removing material from the blade tip or coverband assembly can increase the blade frequency (Hesler and Marshall, 1993). The frequency can be decreased by removing material from the airfoil base or adding material to the blade tip or coverband assembly. Case Study I later in this tutorial describes in detail how a resonance problem in the L-2 row was corrected by adding mass to the tip of the rotating blade.
- *Add frictional damping or material damping*—Frictional damping, also called mechanical, interface, or Coulomb damping, arises from the relative motion between contacting parts such as shrouds, snubbers, damping pins, and root cover sealing plates. Mechanical damping is the most widely applied form of damping and can be a significant source of total damping. Material damping is typically of much smaller magnitude; it is an inherent property of the blading material.
- *Change stiffness of blade/disc*—An example of this option is continuous blade cover and tiewires (or sleeves) near the blade midsection of last stage blades intended to provide improved response to buffeting or stall flutter. The continuous design provides both increased stiffness and damping in the tiewire sleeves. APPENDIX A Figure A-8 shows a variety of interblade connection designs that can be used to change the vibration response characteristics of blades.

In the U.S., the use of long arc shrouding for tuning purposes was introduced into boiler feedpump drive turbines (variable speed) in 1973. By 1984, when more than 80 rows of auxiliary turbine blades were modified to long arcs, the first implementation of continuous long arc shrouding was made. There are now more than 30 successful modifications of this concept to rows of previously failing blades. The design consists of using the proper long arc of shrouding, depending on the harmonic nearest to, or most likely to be exciting the tangential mode of vibration, and linking the ends of these long arcs with a Z-shaped joint. In addition, tiewire arcs are the same arc length as the covers, but overlapped to ensure a continuous tie. Case Study II shows an example where long arc shrouding covering 42 to 43 blades was used to replace five to six blade groups in order to reduce the problems associated with tangential vibration modes.

- Add/move/change tiewires or tenons.
- Mixed tuning of blades. In the case of unstalled flutter, mixed tuning of blades has been found to provide significant reduction of a problem (Evans, 1993). This utilizes alternating blades with natural frequencies sufficiently far apart to decouple the vibrations within blade groups. This strategy will avoid sustained negative aerodynamic damping and thus avoid the high levels of aerodynamic response. Examples of before and after BVM test results utilizing mixed tuning in steel blades are shown in Figure 19.

The degree of mixed tuning required depends on:

- Blade geometry.
- Material.
- Aerodynamic properties.

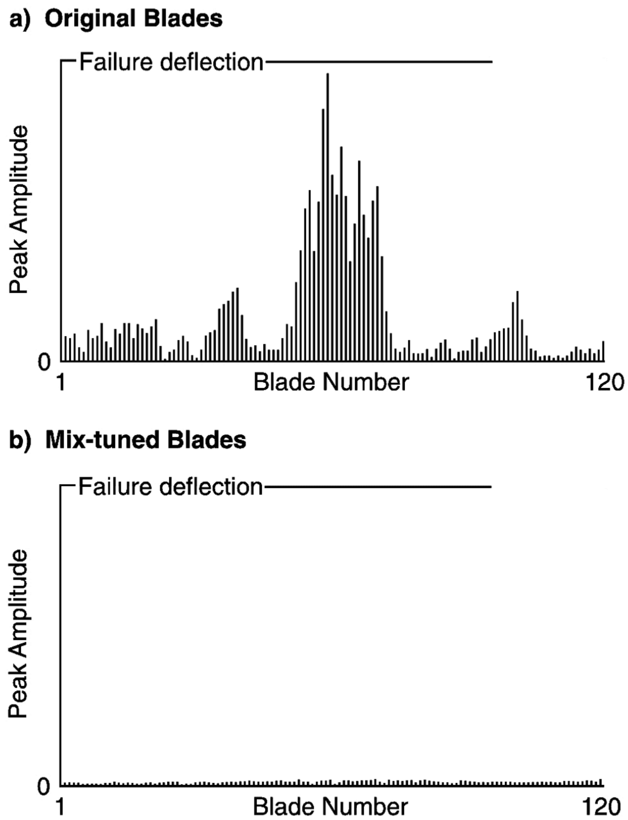

Figure 19. Comparison of Original and Mixed Tuned Rotating Blade Vibration Amplitudes to Reduced HCF Failures from Unstalled Flutter.

Mixed tuning was first applied to freestanding rows of steel blades, and has now been used with titanium with excellent results (Nedeljkovic, et al., 1991).

- Reduce/add weight to rotor. The objective is to ensure that rotors avoid resonance with a frequency equal to two times the electrical grid frequency.

Operating Changes

Operating changes are one of the few areas of improvement that are within the control of the operator to reduce fatigue failures in LP blading. Unfortunately, for the most part, they are not likely to make economic sense. Thus, they are listed mostly for completeness and for those rare conditions where the fatigue problem is so dramatic that a temporary change in operating practice is prudent until the problem can be corrected.

- Avoid extreme off-load conditions. Raising the minimum load and operating at lower backpressure are sometimes used to improve conditions that lead to stall flutter.
- Prevent over or underfrequency operation.

For overfrequency conditions, this can be performed, for example, by immediate reduction of speed to rated levels following generator breaker trip. For underfrequency conditions, system load shedding schemes should be in place to allow system frequency to return to normal (Kundur, 1993).

CASE STUDY I— ANALYSIS AND CORRECTION OF FAILURES IN AN L-2 ROW CAUSED BY RESONANCE

This case study describes one example of a significant and recurring problem in L-2 rows. It includes analysis and confirmatory testing to diagnose the problem and evaluate the potential solution. Tuning of the blades by increasing the weight was found to solve the problem that was caused by resonances (Hesler and Marshall, 1993).

Unit and failure history. The fossil station has two 500 MW turbines, each with four exhaust flows. Repeated fatigue cracking had occurred in the coverband assembly and blade tenons of the L-2 rows following plant commissioning. Over a 10-year period these failures were responsible for an average 4 percent loss in availability.

Blade design. The L-2 blades were 318 mm (~12.5 inches) in length. The blades were arranged in groups of 12 using a coverband attached to the blade tips with two peened tenons per blade. Adjacent groups were joined with a short coverband segment, termed a buttstrap. Each buttstrap spanned a total of four blades in the joined groups. The buttstrap was installed beneath the coverband and held by the peened tenons. Blades under the buttstrap had slightly shorter airfoils to accommodate the 3 mm (0.12 inch) thick material.

Finite element analysis results. A finite element model of the bladed disc was developed. This allowed for detailed analysis of stresses and natural frequencies. The continuous nature of the coverband eliminated most tangential modes resulting in primarily axial modes.

Figure 20 shows the Campbell diagram that resulted. The designation "D" refers to the nodal diameters. The interference diagram at zero and 3000 rpm is shown in Figure 21. A modal shape plot is shown in Figure 22 for a typical axial mode. The modal shape plot is used to help visualize physically the deformation that is occurring and aid in interpreting the results of the Campbell and interference diagrams. The nodes are locations of zero displacement and the highest deformations occur between nodes. Understanding where the locations of highest deformation are located may allow a comparison to the failure locations and confirm that the calculated modes are those relevant to the failure.

Field and laboratory testing results. Two types of testing, modal testing at zero rpm and strain gauging at operating speed, were used to confirm and refine the model and to analyze the proposed solutions. The modal testing confirmed the strong possibility of axial mode resonance involving one or more modes at about five nodal diameters, which can be seen from the interference diagram (Figure 21). The strain gauge data indicated a strong blade response at 300 Hz involving the sixth nodal diameter axial mode (Figure 22). In fact the frequency of the damaging mode was found to be 298.8 Hz or only a margin of 1.2 Hz from resonance.

Life estimation. Using the strain gauge and finite element results, the fatigue initiation life of the coverband assembly was calculated. The life estimate, on the order of a few months, was consistent with the field experience. It was confirmed that the 300 Hz vibration was the cause of initiation.

Modifications. It was clear from the analysis and testing that detuning to get rid of the resonance conditions was the appropriate strategy to pursue. Of the various detuning options available, the chosen strategy was to lower the blade frequency by adding mass to the blade. The finite element analysis and modal testing were reperformed using a specific detuning mass. An additional mass of 38 grams per blade was added (Figure 23). This approach had the following advantages:

- Not changing the airfoil and thus leaving the aerodynamic characteristics intact,
- The modifications could be performed without removing the blades,
- The modification affected mass distribution only, stiffness was not changed. Confirmation by strain gauge testing was performed after the modification was made by the OEM.

Results. The frequency was shifted −11.8 Hz and −11.0 Hz at the six and five nodal diameters. As a result the dynamic strain at 300 Hz was reduced dramatically, by a factor of 3.5 on peak instantaneous strain. The effect was a tenfold decrease in stress at the failure site and a fatigue life estimate that exceeded 100 years.

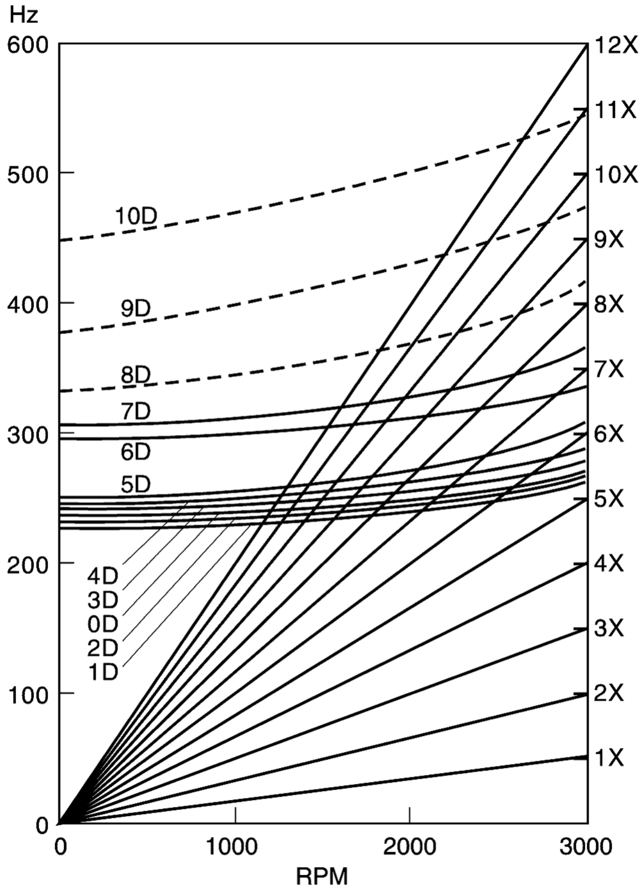

Figure 20. Campbell Diagram of Predicted Natural Frequencies for Case Study I.

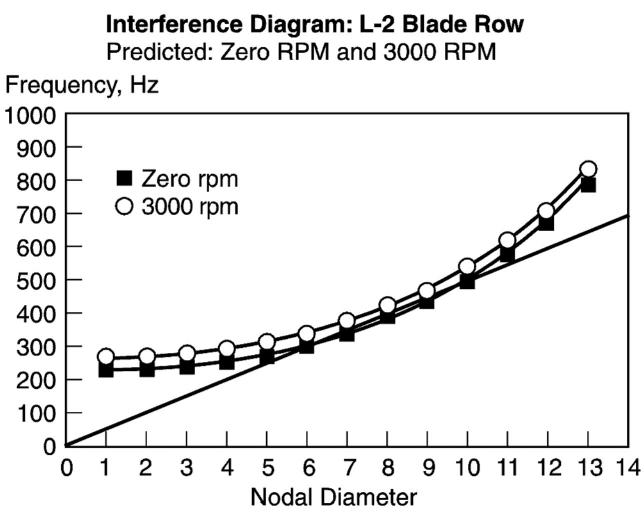

Figure 21. Axial Mode Natural Frequencies and Nodal Diameters in Interferenced Diagram Format.

CASE STUDY II— HIGH CYCLE FATIGUE OF L-1 BLADES IN AN LP TURBINE

Analysis and Field Confirmation of the Cause and Retrofits

This case study was drawn from a research project that had several objectives in addition to just the confirmation of a retrofitted blade design. As a result, it is more comprehensive in scope, for example, running finite element analyses with two

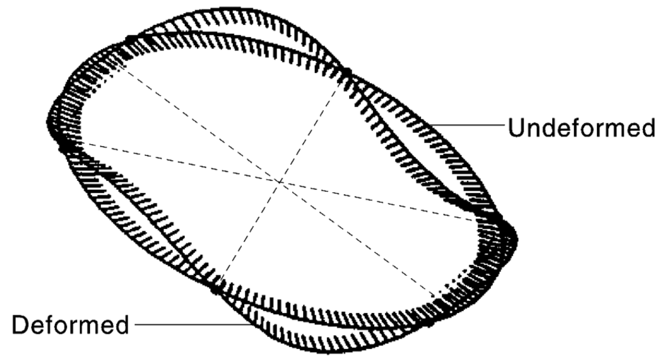

Figure 22. Undeformed and Deformed Shape Plots from Finite Element Model Indicating Three Nodal Diameters.

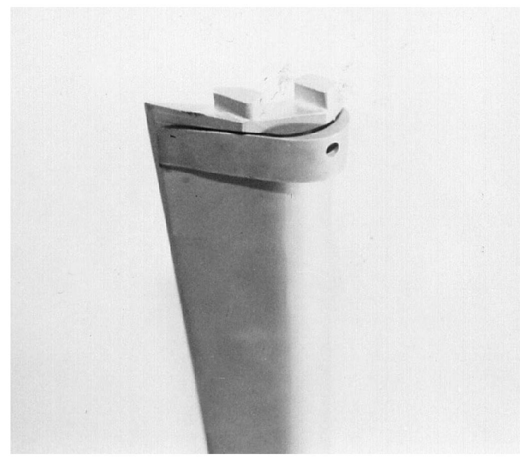

Figure 23. Photo of Blade Modified by Addition of 38 Grams Underneath Shroud of Rotating Blade Shifting Natural Frequency of Blade Down by −11 Hz. (Case Study I)

different programs for comparison. However, the steps taken provide a guide to those that might be considered.

Unit Information. The unit is oil fired and seawater cooled with a cross-compound axial-flow turbine. The turbine-generator was put into service in 1958 with a rated capacity of 215 MW. The L-0 stage blades are 109 cm (43 inches) long and the L-1 blades are 72.4 cm (28.5 inches) long. The L-1 blades are tapered and twisted with a five-finger root attachment. The fingers are staggered and attached to the disc rim with three axial pins per blade. The blades were originally installed in groups of five or six with a 1.3 cm (0.5 inch) diameter solid tiewire and a 3.2 mm (0.125 in) thick shroud.

Blade failures. Cracks had been found in the tiewire, tiewire hole region, tiewire braze, and shroud during every inspection for a 25-year period. In 1983, a section of the L-1 blade broke off 5.1 cm (2 inches) above the blade platform. An examination showed high cycle fatigue was the underlying cause with crack propagation tangentially. Campbell diagram analysis of expected vibration was performed and, as a result, the blades were changed from the short arc groups spanning five to six blades to long arc groups spanning 42 or 43 blades, a quarter of the wheel.

Analysis performed. Finite element models for the six blade group and the 42 blade group retrofit were developed. The objectives were to predict the static and dynamic stress distributions and modal characteristics of the two designs. This information was then used to predict the life to fatigue initiation.

Testing. Modal tests were performed on the original five and six blade groups and on the retrofitted long arc blade groups. Measured natural frequencies and mode shapes were compared to corresponding values calculated by FEA. Dynamic stresses and steady stresses were measured on the redesigned blade during rotating tests using radio telemetry from strain gauges mounted on the rotating blades.

Strain gauge locations were chosen for highest dynamic stresses based on a preliminary finite element analysis. Dynamic gauges were located on the leading edges, beneath the tiewire and 5 cm (2 inches) above the blade platform.

Blade natural frequencies were measured during a series of ascending and descending speed ramps. The turbine was bought up through operating speed to overspeed then allowed to coast down. During coastdown, a stationary steam jet was turned on to provide a once per-rev impulse. During these steam jet tests, the normal steam was turned off. The telemetry testing was run over a 26 hour period. Modal tests were run on a six blade group and on a 42 blade group.

Modal Analysis and Testing Results. An interference diagram for the six blade group is shown in Figure 24. Wherever the 1800 rpm constant speed line crosses a set of nodal diameter modes, a resonant condition is possible. In the six blade group, as shown in Figure 24, this occurs near the fourth nodal diameter of the first tangential mode set, suggesting that this harmonic may have been the cause of the L-1 blade failures. This conclusion is supported by the observation that tangential mode vibration produces high stresses at the location of observed cracking and that the crack propagation was clearly indicative of tangential mode vibration. The Campbell diagram would not have identified the potential resonance since it does not include disc effects or the relation between nodal diameter and per-rev excitation. Although, as shown in Figure 25, the Campbell diagram does indicate the possibility of resonance at 120 and 180 Hz with the second and third modes of vibration, respectively.

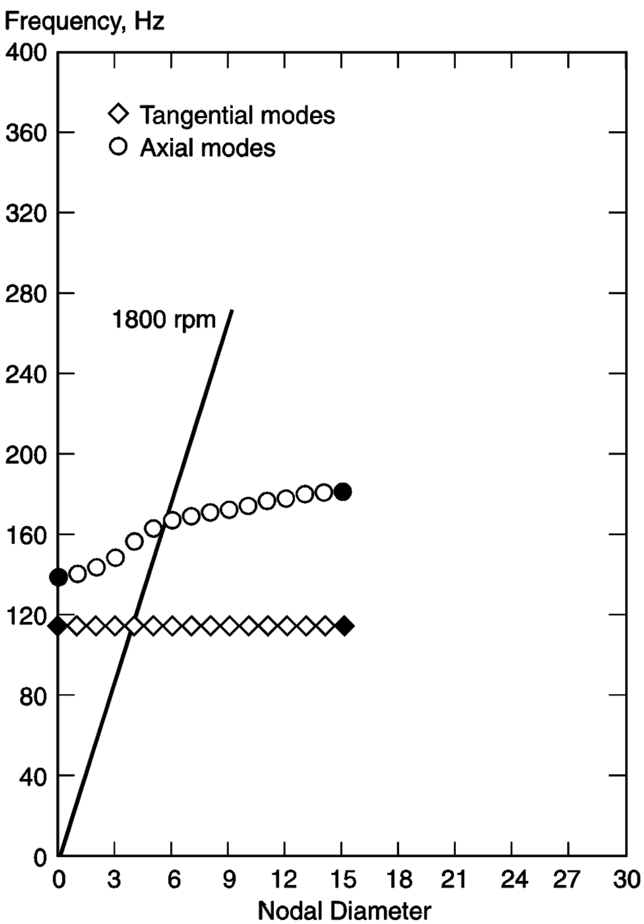

Figure 24. Interference Diagram for Six Blade Group Operating at 1800 RPM (30 Hz) for Case Study II.

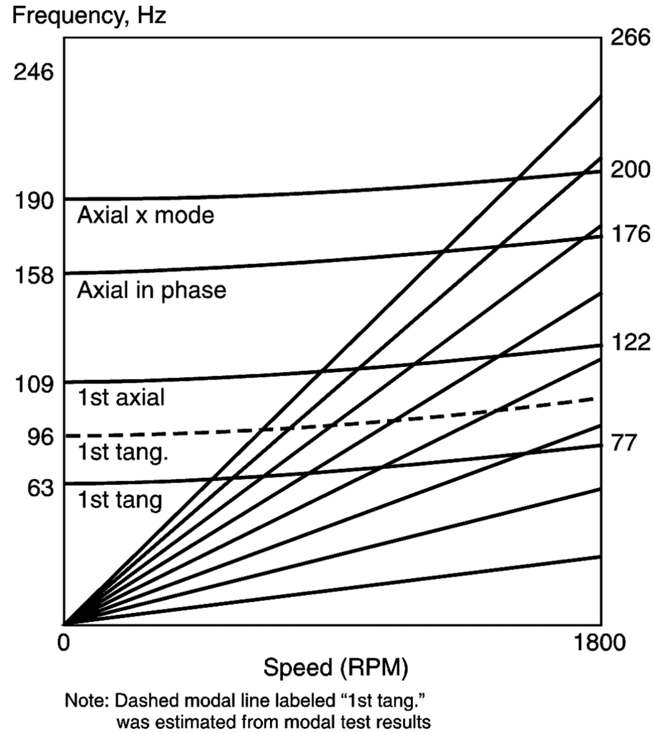

Figure 25. Campbell Diagram of Six Blade Group Indicating Tangential and Axial Modes of Vibration.

A similar analysis for the redesign (42 blade groups) indicated that this tangential mode resonance could be avoided.

Stress and Fatigue Analysis Results. The steady-state stresses were calculated using finite element analysis and compared to strain gauge measurements. Dynamic stresses were calculated with FEA using an estimate of blade excitation based on one-dimensional nozzle flow theory. It was confirmed that the retrofit design would have lower stresses in the tenon and vane region near the tiewire hole, and, although it would exhibit higher stresses at the base of the vane caused by axial mode vibration, it would have much lower stresses for the tangential mode resonance.

Conclusions. Based on the analysis of vibration, stresses, and fatigue life, the 42 and 43 blade groups were retrofit into the affected unit.

CASE STUDY III— PROBABILISTIC METHODS APPLIED TO BLADE ROOT CRACKING

This case illustrates how a probabilistic model of low cycle fatigue crack initiation can be developed.

Problem. The machining tolerances in the root attachment of multiple hook designs can dramatically affect low cycle fatigue life of turbine blading.

Method of Analysis. A probabilistic model was developed that used a Monte Carlo simulation to incorporate the uncertainties of root tolerances and material properties. Three random variables were used:

- Gap sizes,
- The strain-hardening coefficient, and
- The fatigue ductility coefficient.

Figure 26 shows a limited set (51 data points on six bearing lands) of root tolerances obtained from direct measurements, and the assumed exponential distribution that was then used in this model study. In actual field application, considerably more measurements would typically be available. For this data set, the mean gap size was 0.0036 mm (0.14 mil); the median gap was 0.0025 mm (0.1

mil). The distributions for strain hardening coefficient and fatigue ductility coefficient were assumed to be normal distributions, again based on data available.

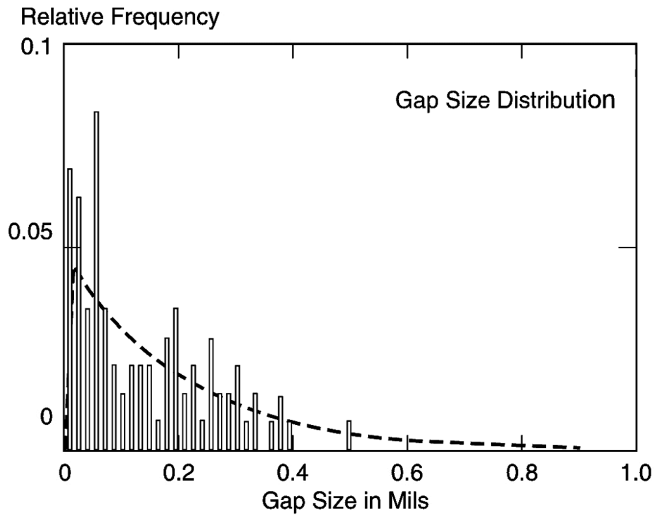

Figure 26. Statistical Distribution of Gap Sizes for Root Attachment in Case Study III.

The simulation procedure started with the calculation of a maximum elastic stress for a random gap combination. Then the plastic strain was calculated based on the strain-hardening coefficient, also randomly chosen. Finally the low cycle fatigue life was evaluated based on the calculated plastic strain and a randomly selected fatigue ductility coefficient.

CASE STUDY IV— LAST STAGE BLADE FAILURE CAUSED BY UNSTALLED FLUTTER

Unit Information. The unit contains a 1300 MW cross-compound turbine with a double-flow high pressure (DFHP) turbine and two double-flow low pressure (DFLP) turbines (LP1 and LP2) on the "A" line, along with a double-flow intermediate pressure (DFIP) and two double-flow low pressure (DFLP) turbines (LP3 and LP4) on the "B" line.

Blade failures. A catastrophic failure of an L-0 blade occurred just above the platform during testing of an overspeed trip. A second blade failed from impact with the first failure. The entire HP line became dynamically unstable because of the large mass imbalance, eventually causing extensive damage to all components in the line, including the high DFHP, two DFLP, generator, exciter, bearings, and turbine auxiliaries.

Prior inspection results. The L-0 blades were inspected approximately four years prior to the failure with wet magnetic particle, and no damage was noted on the airfoils of the blades.

Results of metallographic analysis. The failure initiated at the trailing edge of the airfoil from a corrosion pit that was approximately 0.25 × 0.50 mm (0.01 × 0.02 inch) in size. Other corrosion pits in the same area measured about 0.025 (0.001 inch) deep and 0.05 mm (0.002 inch) wide. No corrosive species were identified in the blade deposits. No microstructural or fabrication anomalies were identified in the origin area. The failure surface was found to be basically transgranular, typically of that resulting from high cycle fatigue. There were 54 "beach marks" on the fracture surface indicating 54 transient events during a four year period during which the bulk of the crack propagation occurred.

Results of material testing. The ultimate strength, yield strength, elongation, and reduction in area were found to be within the OEM specification; thus, material deficiency was determined not to be a root cause.

Results of analysis of resonance. An analysis was performed to determine whether blade resonance was an underlying root cause. The analysis produced Campbell and interference diagrams and determined that the L-0 blades were well detuned away from natural frequencies. Verification was completed using a modal test. The analysis was extended to determine the effect of a crack in the blade on its resonance frequency. It was found that for a crack length of 31.75 mm (1.25 inch), the blade was tuned in resonance with its axial mode; however, as the crack grew beyond that point it became detuned. As the final failure was at a crack length of 88.9 mm (3.5 inches), it was determined that blade resonance did not cause crack propagation.

Was stalled flutter a potential root cause? The unit was base loaded and did not operate below 50 percent load with high condenser backpressure. For this reason stalled flutter was also eliminated as a potential root cause.

Was unstalled flutter a potential cause? An analysis of the potential for unstalled flutter was conducted. It was found that the available aerodynamic damping to resist nonsynchronous vibration response to the fourth axial mode was low. Aerodynamic damping decreases with increased flow rates and increases with increased backpressure. The analysis found that although the L-0 blades were predicted to be stable under design conditions, at off-design conditions of flow > 100 percent and backpressure < 6.68 kPa (2 inch Hg), the blades were predicted to respond strongly (become unstable) if stimulated by nonsynchronous excitation.

Conclusions. The blade failure was a result of a progressive failure. Corrosion pits formed, and, as the pits grew, the blade airfoil became susceptible to significant operating stresses. A microcrack initiated from the bottom of the pit as a result of a combination of low cycle fatigue and stress corrosion cracking. Once the crack reached an advanced length, it propagated by high cycle fatigue driven by unstalled flutter. The flutter occurred as a result of off-design operation (high flows and low backpressures).

Solutions. New (spare) rotors were installed in the line "A." The four L-0 rows of line "B" were refurbished. An online blade vibration monitoring system was installed. Off-design tests to detect zones of distress from unstalled flutter were conducted. Over the longer term, a plan was set up to refurbish all L-0 and L-1 blades on a 10-year reinspection cycle.

An economic evaluation should be executed in parallel with the decision about which option to pursue. In most cases, a detail blade assessment, such as outlined in Figure 17 (McCloskey, et al., 1999), will be required to identify and confirm the underlying stress source (Table 2), compare alternative fixes, and evaluate the expected life of the chosen fix. Three generic approaches to reducing fatigue damage are design changes, tuning, and operating changes. The main text contains a more detailed list of generic choices in each of these categories. The optimum choice will depend on the source of the problem and the expected economics of the tradeoffs. The operator may also wish to implement online monitoring utilizing a blade vibration monitor as a means of anticipating future fatigue conditions.

CORROSION FATIGUE OF LP BLADES

Corrosion fatigue is one of the leading causes of damage in the rotating blades of steam turbines. It occurs as a result of the combination of cyclic stresses and environmental effects. Damage that accumulates by corrosion fatigue typically originates from pitting or other localized corrosion. Subsequently, the developing flaw is affected by cyclic loads leading to corrosion fatigue or, in a few cases, by steady-state loads causing stress corrosion cracking.

Fatigue, corrosion fatigue, and stress corrosion cracking are closely related. In rotating blades, corrosion fatigue is far more common, while in the disc rim attachment area, SCC is predominant. SCC and CF differ in the range of affected materials and environments. CF affects nearly every alloy, as the crack growth rate in any environment is typically more rapid than that in

Table 2. Detail Blade Assessment.

Stress Type/Load	Specific mitigation options
Centrifugal tensile stresses	Reduce stresses in blade roots by enlarging the hook radii or other geometry change to reduce stress concentration.* Improve hook-to-hook contact between the roots and disc; decrease assembly tolerances.* Reduce weight such as by use of a lighter shroud, cover, or blade (titanium, for example).* Adopt freestanding design (eliminate shroud) to reduce weight and thus hook stresses.
Geometric untwisting*	None
Centrifugal bending stresses	Not usually practical
Steam bending loads	Reduce stresses in blade roots by redesign such as by enlarging the hook radii to reduce stress concentration.
Synchronous resonance of blades with a harmonic of theunit running speed*	Identify by analysis those blades that are within Approximately 10 Hz of the nearest harmonic. Redesign to reduce resonant stresses such as by optimizing the blade profile, airfoil, and width. Tuning strategies, such as: Add/reduce weight to alter frequencies and resonances Use Campbell diagram Add structural material or Coulomb damping to alter resonance and/or reduce resonance stresses. Add/move tiewires or tenons to change stiffness of blade or disc, thus altering resonances and stresses.
Nonuniform flows	For synchronous vibrations:* Identify by analysis those blades which are within 10 Hz of the nearest harmonic.* Redesign to reduce resonant stresses Tuning strategies, such as: Add/reduce weight to alter frequencies and resonances Use Campbell and or interference diagram Add material or Coulomb damping to reduce resonance stresses. Add/move tiewires or tenons to change stiffness of blade or disc.* Redesign of steam admission and discharge areas to reduce intensity of excitation. For nonsynchronous vibrations: Will depend on the source
Blade torsional vibration induced from rotor or disc	Change frequency of rotor by adding/subtracting rotor or blade weight. Change frequency of blades by adding/subtracting weight or changing the stiffness of the blades Provide appropriate controls to prohibit operation outside recommended frequencies.
Self-excitation	For stall flutter: Avoid operation of unit at extreme off-design load conditions, specifically increase minimum load and/or increase backpressure. Redesign to use continuous tie strategies to provide restraint at cover tips. Redesign blade sections to install various damping devices such as "Z" cuts in shrouds and loose tiewires.* For unstalled flutter (not as well understood as stalled flutter): Establish aeroelastic properties of blades to identify marginally stable modes. Use mixed-tuned blades (alternating blades with natural frequencies sufficiently different from one another) to decouple aerodynamic forces. Apply damping devices between blade groups. Consider derating unit to within load conditions that do not cause excessive stresses in the blades of the latter stages.
Start-stop transients and overspeeds.*	As for centrifugal tensile stresses above
Manufacture and assembly stresses*	Depends on source, may require redesign.* In the case of localized residual stresses from weld repair, heat treatment may be useful.

air; corrosion fatigue also occurs over a range of electrochemical potentials, even active dissolution, whereas SCC occurs only in a narrow range of potentials.

NATURE AND FEATURES OF CORROSION FATIGUE DAMAGE

Corrosion fatigue cracks are surface initiated and are found in conjunction with a stress concentration. In many cases in turbine blading, corrosion fatigue will initiate from a corrosion pit. However, CF cracks can also initiate from other types of local damage such as:

- Fretting,
- Manufacturing defects,
- Inclusions,
- Microscopic imperfections in the material such as preferential dissolution of persistent slip bands or by mechanical rupture of the passive film at slip steps, or
- Specific adsorption of species that locally reduce surface energy (Macdonald, et al., 1985; Laird and Duquette, 1972).

Corrosion fatigue cracks in turbine blading materials can display a variety of morphologies depending on the environment and stress levels, including both transgranular and purely intergranular. However, initiation will typically be transgranular; subsequent propagation in turbine blading materials is also typically transgranular, except in a narrow stress intensity range where it is possible for propagation to be intergranular.

The surface of the corrosion fatigue crack will typically manifest a brittle appearance, i.e., little macroscopic evidence of deformation, and will not be nearly so rough looking as stress corrosion cracking. There may be some evidence of corrosion products on the fracture, although often these may not be present. Typical deposits include white or gray chloride, sulfate, and/or carbonate deposits that form a local coating over the blade surfaces. There is generally a loss of sharpness or definition of the fracture morphology. Other features typical of corrosion fatigue include: visible striations, crack arrest lines, and little or no branching.

Figure 27 shows developing corrosion fatigue cracks originating from the roots of two adjacent L-1 blades. Figure 28 shows another location in the same row in which a final overload and completed fracture have occurred just below the platform after initiation from the first hook. Corrosion fatigue and SCC occur under similar circumstances and may coexist on the same fracture.

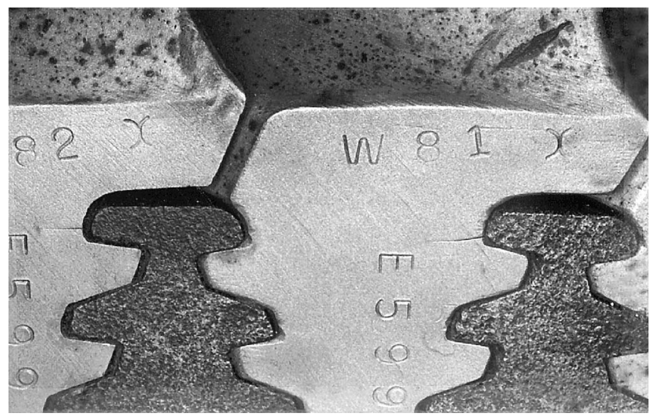

Figure 27. Corrosion Fatigue in L-1 Blade. (Crack initiation and propagation can be seen in first hook (serration) of two blade roots shown.)

Susceptible Units and Locations for Corrosion Fatigue

Damage by corrosion fatigue occurs in the latter rows of LP turbines in both fossil and nuclear units starting in the phase transition zone and extending to the exhaust. The PTZ moves according to load changes, but is typically near the L-1 row in most LP fossil turbines and near the L-3 and/or L-4 region of nuclear LP turbines.

Units going to increasing cycling duty may be subject to worsened corrosion fatigue problems as:

- The blades pass through resonances more frequently since the unit is ramped up and shutdown,
- The phase transition zone shifts, potentially affecting more stages during transients,
- The general steam purity levels are significantly worse during transients than during steady-state operation,
- If the unit is shutdown as part of the cycling operations, significant degradation of the local environment can occur.

Figure 29 shows the locations most commonly affected by corrosion fatigue. Although shrouds are typically constructed of the same material and subjected to the same bulk environment, corrosion fatigue tends to be less of a problem because of the lower vibratory stresses. Indications that CF damage mechanism is active are shown in Table 3.

Figure 28. Another Location in Same Row as Figure 27 Showing Where Fracture and Subsequent Blade Loss Has Occurred Just below Root of Blade.

Unit Precursors Indicating That this Damage May Become Active

Excessive steam impurity levels (particularly of chloride, sodium, and sulfate) and the resulting deposition are underlying causes of corrosion fatigue. If feedwater and steam impurity levels exceed the recommended limits, such as the result of a major condenser leak, then corrosion fatigue damage is a possibility. Poor shutdown procedures are a primary contributor to the aggressive environment that can lead to corrosion fatigue. High steam cation conductivity levels many indicate conditions that can lead to rapid accumulation of deposits and liquid films on blade surfaces; this may be caused by the steam bypassing the drum separation equipment.

Effect of Corrosion Fatigue on Turbine Blades

Corrosion fatigue is one of the most damaging mechanisms affecting the last stages of the LP turbines in both fossil and nuclear units. Extensive damage including broken blades is typical. Distress occurring in any of the rotating parts can cause immediate shutdown of a unit.

MECHANISM OF CF DAMAGE

Damage starts in a localized region as a result of a process such as fretting, the presence of manufacturing defects, inclusions, and pitting or other localized corrosion. Once an initial defect has formed, a rapid decrease in the local fatigue resistance occurs, leading to damage acceleration through the synergistic effects of environment and stresses. Where the stresses imposed are cyclic, either purely cyclic stresses or moderate cyclic stresses imposed on high steady-state stresses, the resultant mechanism is termed corrosion fatigue. High mean stresses and moderate cyclic loads in the presence of an environment form a particularly damaging combination.

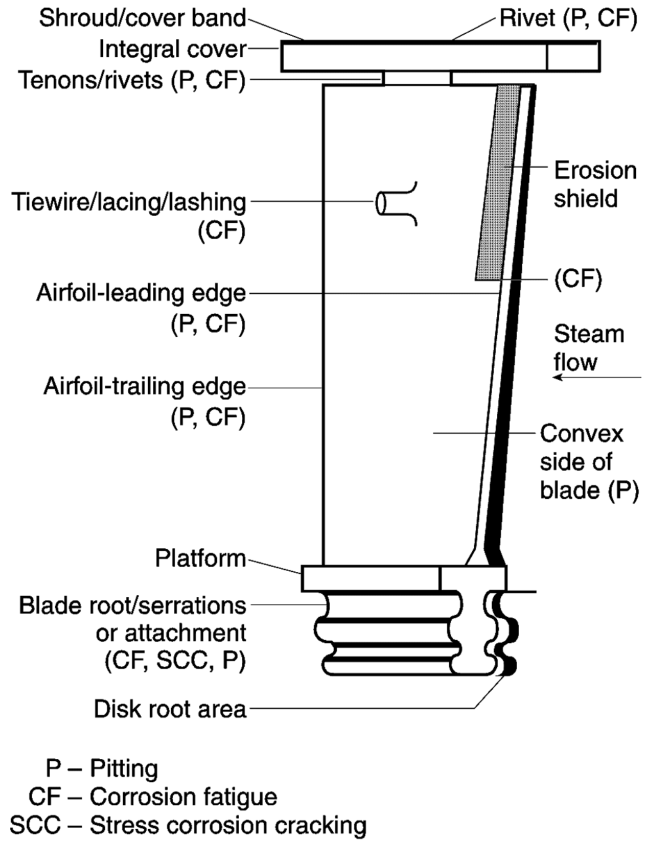

Figure 29. Typical Locations of Rotating Blades of Low Pressure Turbine Affected by Localized Corrosion and Corrosion Fatigue.

Table 3. Indications CF Damage Mechanism Is Active.

Appraisal Means	Indicator
Without Opening Turbine	If the unit is equipped with a blade vibration monitor, it should be able to detect a change in single blade vibration as a result of a growing crack. In the absence of blade vibration monitoring, indicators will be limited to the detection of severe corrosion fatigue damage such as the loss of a blade or shroud. This typically is detected by a change in the amount and phase angle of journal bearing vibration.
With Opened Turbine	NDE survey indicating the presence of corrosion fatigue cracks

Effect of the Local Steam Chemistry Environment on CF

Corrosion fatigue requires an aggressive local environment. The local environment consists of two separate aspects:

- Dynamic environment produced during operation. These are local conditions produced by the natural condensation processes that accompany steam expansion as it moves through the turbine. Such processes as:
  - Precipitation of chemical compounds form the superheated steam,

• Formation of concentrated liquid films,

• Evaporation, deposition, and drying of wet steam on hot surfaces lead to the formation of potentially corrosive surface liquid films and deposits.

• Environment produced during shutdown. During shutdown, moist, liquid, oxygenated films form on blade and disc surfaces as a result of dew point effects. These films result directly from inadequate shutdown practices.

There is an interaction between the two aspects of the environment that is central to understanding the corrosion fatigue process. Figure 30 illustrates schematically how these environmental influences develop and interact, along with cyclic loads, to produce corrosion fatigue.

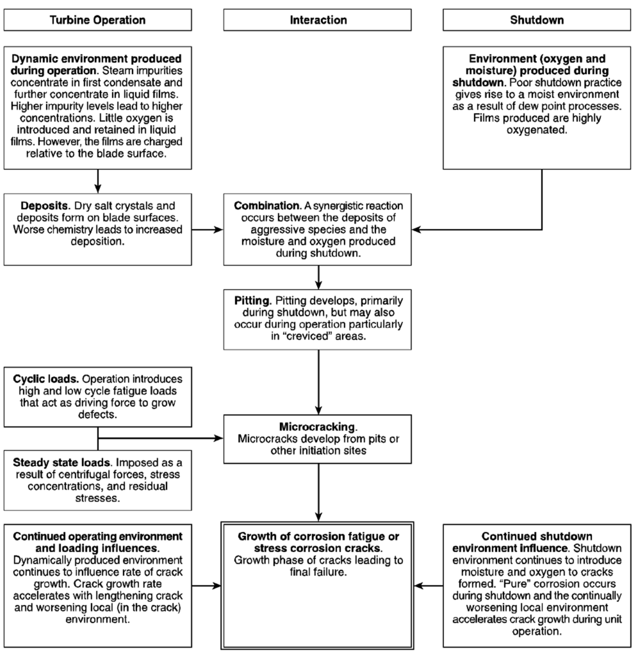

Figure 30. Steam and Water Chemistry Environmental and MW Load Influences on Corrosion Fatigue Flowchart.

The dynamically produced environment concentrates steam impurities and allows the formation of deposits on blades and disc surfaces. Particularly important are the formations of liquid films, which are responsible for supplying the environment for both corrosion fatigue and stress corrosion cracking to occur. Liquid films concentrate anions such as chloride and sulfate and reduce the pH of the local environment, factors that have long been shown in laboratory tests to contribute to an increase in corrosion fatigue damage. It should be remembered that these liquid films are also charged or have a potential that can drive the electrochemical corrosion fatigue process. Such charges, which change with inlet steam purity, may be a key driver in the propagation of corrosion fatigue cracks.

The role of oxygen in the corrosion fatigue process has been subject to extensive investigation. In some laboratory testing, increasing levels of oxygen in the test environment have been found to lead to reduced levels of corrosion fatigue strength in 12 percent cromium (Cr) steels. For example, one investigation found that the fatigue strength at 5 × 107 cycles in 3 percent sodium chloride (NaCl) (deaerated) solutions at 80°C (176°F) was about 30 percent less than the limit in air; but about one-fourth of the air limit for aerated 3 percent NaCl. In contrast, other researchers have found no significant effect of an oxygenated environment on growth of corrosion fatigue cracks in blading materials (Ebara, et al, 1983; Holdsworth, et al., 1997). Given the apparently conflicting results, key issues are:

• Why are there differences in the results of various researchers, but, more importantly,

• How do laboratory results relate to the field environment?

• What, if anything, does the operator need to do to control oxygen levels as a partial means of controlling corrosion fatigue?

One of the most important observations from recent work, which provides guidance for action, is that oxygen does not concentrate in the early condensate and liquid films under dynamic (operating) conditions. Levels in the early condensate are < 1 ppb for turbine steam inlet levels of oxygen in the range of 30 to 250 ppb. However, oxygen does accumulate in moist, oxygenated films during unit shutdown if proper recommended practices are not used.

Recent converging-diverging nozzle tests have shown that impurities deposited from steam during onload periods (the dynamic environment) were able to initiate corrosion pitting, a corrosion fatigue precursor, under the unprotected wet aerated conditions encountered while a unit is offline (Jonas, Inc., 1998).

These results confirm that both aspects of the environment play a key role in producing corrosion fatigue. The dynamic environment allows for the formation of deposits and the accumulation of potentially corrosive anions in charged liquid films. This environment occurs when the dynamic stresses that drive corrosion fatigue cracks are present. During improper shutdown, the environment provides a source of excessive levels of oxygen along with moisture, and thus can produce damage leading to eventual pitting and crevice corrosion.

It has been observed that the presence of small amounts of oxygen may even be beneficial in slowing corrosion fatigue crack growth rates (Macdonald and Cragnolino, 1989). This is a result of two effects:

• The passivation of the alloys should be more effective.

• The generation of hydrogen through the concurrent cathodic reaction should be less.

Both effects should reduce the rate of cracking. This position is strengthened by the observation that there is a lack of environmental effect on crack growth rates for turbine steels at passive potentials, even in highly aggressive solutions (Rungta and Begley, 1980; Macdonald, et al., 1985). There is a fine balance between excessive levels of oxygen that can promote pitting (during shutdown) and accelerate growing corrosion fatigue cracks (in laboratory tests), and too little oxygen that can speed damage accumulation by preventing passivation of the surface and through the generation of hydrogen.

Clearly, there is need for the further exploration of the roles of charged liquid films and oxygen in the bulk and condensing environments and their specific roles in the initiation and propagation of corrosion fatigue cracks. Our present understanding however indicates that the most troubling aspects of oxygen occur as a result of the formation of moist, oxygenated deposits during unit shutdown; the optimum approach is one that controls oxygen during shutdown and, through steam purity control, deposition and charged liquid films during operation. This dual strategy is discussed throughout this tutorial.

Finally, in regard to the overall effect of environment on corrosion fatigue, it is interesting to compare how strong the environmental effect is as compared to the increase of stress that results from a stress concentration. It has been found in laboratory tests that mechanical notches (with a stress concentration factor, Kt = 2.5) in pure water reduce the environmental fatigue strength of titanium (Ti-6Al-4V) and 17-4 PH by about the same factor as an aggressive test environment of either 22 percent NaCl or 6 percent wt ferric chloride ($FeCl_3$) (Viswanathan, et al., 1983; Bates, et al., 1984). In contrast, for Type 403, the environmental effect in these aggressive environments is significantly greater than the notch effect.

Materials' Response—Corrosion Fatigue Crack Growth Rates

Figure 31 shows how environment affects the fatigue behavior of a 13 Cr steel (AISI Type 403) on smooth specimens. Figure 31 is typical of a wide range of materials. There is an obvious decrease in the cyclic stress amplitude needed to cause failure (termed the corrosion fatigue strength) for a given number of cycles, *N*, with worsening environment.

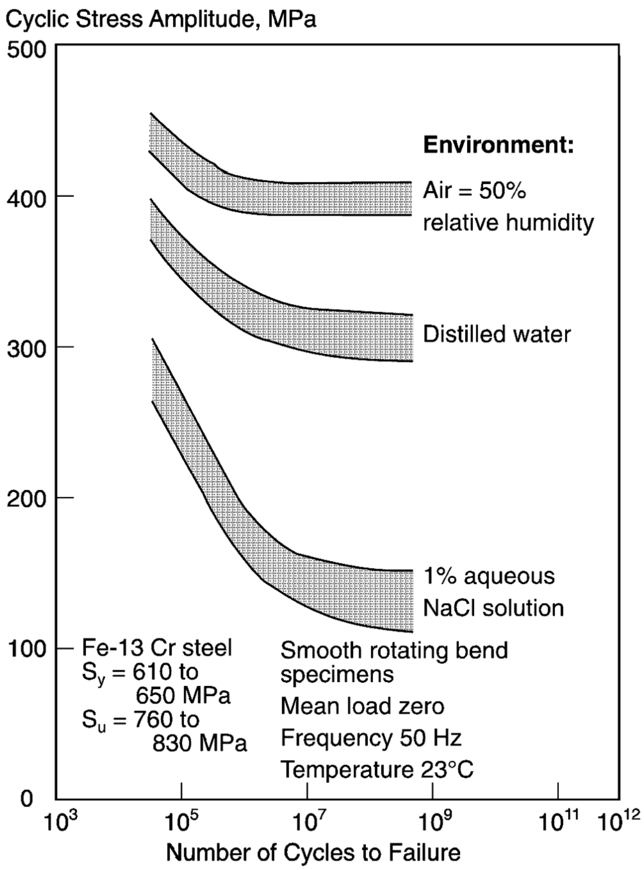

Figure 31. Corrosion Fatigue Curves of Smooth Specimens of 13 Percent Chromium SS, Exposed to Different Steam and Water Chemistry Environments.

In laboratory testing, (Bates, et al., 1984), a concentrated NaCl solution with pH of 10 and less than 20 ppm oxygen causes a twofold decrease in the fatigue strength in Type 403 from its "pure" water value. Decreasing the pH to four or adding large amounts of oxygen (air-saturated) each results in an additional factor of two decrease, leading to an eightfold decrease for 22 percent NaCl, pH of four in an air-saturated solution from its "pure" water value. Synergistic effects with sodium hydroxide (NaOH) and oxides of iron, which might be typical of the operating environment in the LP turbine, further decrease the corrosion fatigue strength of Type 403. The decrease in fatigue strength can be as high as 80 to 90 percent from air to an aggressive environment.

Note that for many materials, true endurance limits can be found only in an inert environment or vacuum, so that even the moist air fatigue limit is considered by some researchers as reflecting corrosion fatigue.

The same behavior is evident on notched specimens of the same material (Figure 32). Again the allowable cyclic stress amplitude is markedly decreased as the aggressiveness of the environment increases. Even for air tests, the presence of a notch greatly decreases the corrosion fatigue life for a given cyclic stress. These two figures indicate why pitting, which forms a stress concentrator, is such a damaging precursor to corrosion fatigue. In many iron-chromium-nickel (Fe-Cr-Ni) alloys, resistance to corrosion fatigue is directly related to the resistance to pitting corrosion (Speidel, 1977).

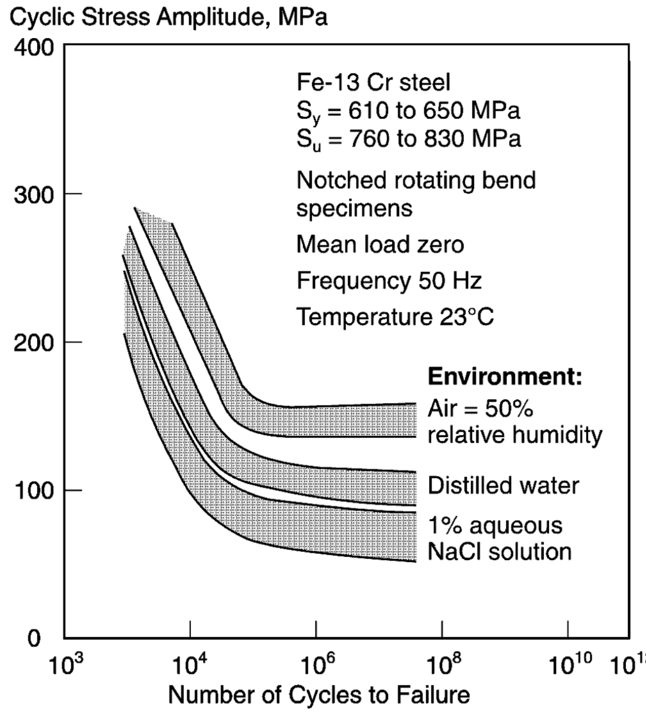

Figure 32. Corrosion Fatigue Curves of Notched Specimens of 13 Percent Chromium SS, Exposed to Different Steam and Water Chemistry Environments.

There are a variety of mechanical, metallurgical, and environmental variables that affect the rate of corrosion fatigue crack growth in metals (Wei and Speidel, 1972). Some are:

- Cyclic stress range, sr = smax − smin. This same effect can be represented by the cyclic stress intensity, DK = f (sr and crack size, a).
- Maximum stress, smax.
- Mean stress, sm = (smin + smax)/2. This effect is also expressed by the stress ratio, R = smin/smax. Increasing the mean stress (or the stress ratio) will decrease the number of cycles to failure.
- Cyclic wave form.
- Loading frequency. Decreasing frequency will reduce the corrosion fatigue strength, likely allowing greater opportunity for the coupled corrosion process to occur.
- Environment.
- Metallurgical variables: alloy composition, presence of impurities, microstructure, and crystal structure.
- Mechanical properties: strength, fracture toughness.

One means of representing the complexity of the data is the modified Goodman diagram or Smith diagram, an example of which is shown in Figure 33 for a 12 percent Cr turbine blade material. Experimental data are plotted for the level of mean stress on the abscissa and for total stress (mean stress plus or minus alternating stress) on the ordinate. The 45 degree line represents the mean stress line or no alternating stress. The farther the data fall away from the mean stress line, the higher the alternating stresses that can be withstood for a given mean stress and stay below the fatigue strength of the material. This diagram is particularly useful for illustrating the effect of mean stress on the corrosion fatigue strength. In Figure 33, the open circles represent tests run in air. As shown, there is little reduction for the alternating stress range to failure for this material in deaerated pure water (dotted line). This behavior is consistent with the observation that 12 percent Cr material serves very well when used in reasonably clean environments. However, there is a marked reduction in alternating stress levels to failure in a 22 percent NaCl (aerated) solution (filled circles).

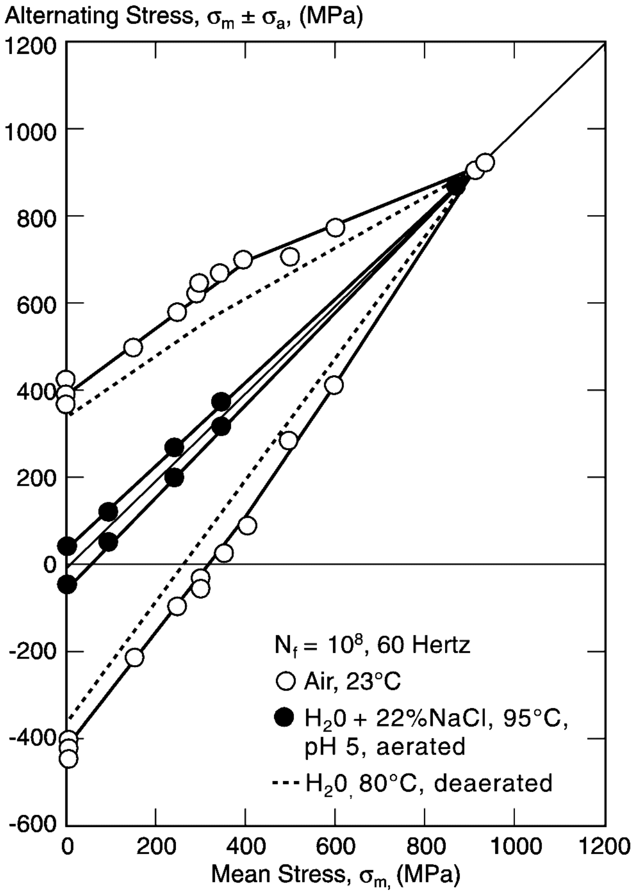

Figure 33. Modified Goodman (Smith) Diagrams for Fatigue and Corrosion Fatigue of 12 Percent Chromium SS for Various Steam and Water Chemistry Environments.

Figure 34 shows that aggressive environments plus high mean stresses (high stress ratios) lead to marked acceleration of corrosion fatigue cracks in 12 percent Cr steel. In Figure 34, the crack growth rate in a vacuum follows the lower curve; in more aggressive environments, the growth rate can be several orders of magnitude higher. Growth is also significantly higher at larger stress ratios (higher mean stresses). Note that in both Figures 33 and 34, the tests were conducted at 60 Hz, frequencies that reflect those similar to high cycle fatigue loads that have one, two, or three times resonances with rotor speed.

The precipitation hardened (PH) stainless steels have somewhat better fatigue and corrosion resistance than 12 percent Cr in reasonably clean environments, and recent extensive testing of 13-8 PH and 15-5 PH materials indicate they have substantially improved corrosion resistance in aggressive environments as compared with 12 percent Cr. They can be developed to have sufficiently high strength levels for use as last stage blades. However, there has been some concern over the susceptibility of PH steels to SCC in aggressive chloride solutions (Denk, 1994).

Duplex (ferritic-austenitic) stainless steel has excellent corrosion fatigue strength. One problem with the duplex stainless steels is that commercially available grades have yield strengths that are less than desired by blade designers (Atrens, et al., 1984). It is possible however, given their superior corrosion performance that they may be suitable for LP turbine blades shorter than those required for the last stage.

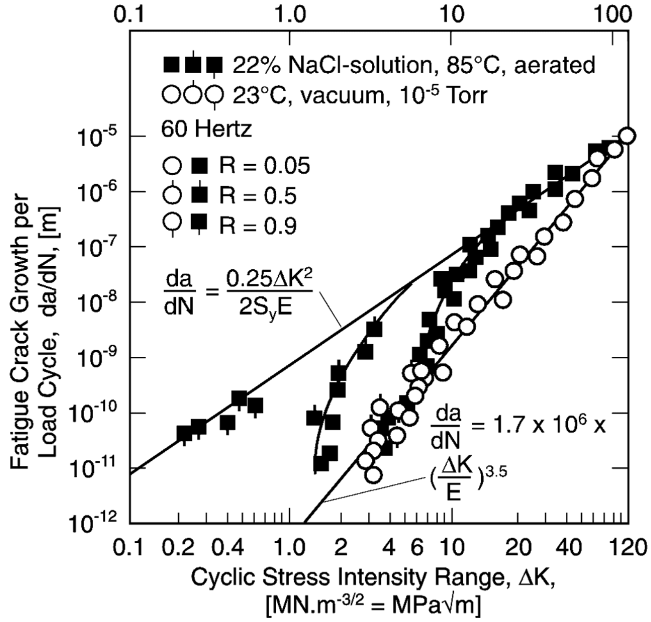

Figure 34. Growth Rate of Fatigue Cracks in 12 Percent Chromium SS as Function of Steam and Water Chemistry, and Cyclic Stress Intensity Range.

The corrosion fatigue behavior for titanium is completely different from 12 percent Cr, 17-4 pH, or the duplex stainless steels. Titanium is fairly insensitive to the environments studied, i.e., there is little change in fatigue strength between air and an aggressive environment consisting of 22 percent NaCl (aerated). However, the resistance decreases markedly for high mean stresses in either environment and is actually less than for 12 percent Cr steel in some conditions. This is shown in Figure 35, compiled from several experimental programs (Speidel, 1983; Atrens, et al., 1983; Rust and Swaminathan, 1983).

How do results from tests conducted in common laboratory test environments for corrosion fatigue such as 22 percent NaCl, aerated environments, or 6 percent wt $FeCl_3$, relate to actual conditions in the LP turbine? Analyses of deposits from LP turbines have indicated an average concentration of chloride of 5 percent (for once-through boilers), with maximum concentrations up to 45 percent (Bates and Cunningham, 1980). Localized conditions of high chloride and low pH can exist in liquid films in the PTZ, and, therefore, these test environments might appear superficially to be representative of field conditions. The two major differences are that there is not any appreciable oxygen in the dynamic turbine liquid film environment, and these liquid films have a charge/potential relative to the blade material.

Figure 36 illustrates that there is a threshold stress intensity, DKth, below which defects do not extend by fatigue. Beyond that stress intensity, crack growth can be rapid, and, in blades, once a crack begins to propagate, the remaining life is short—measured in days or weeks.

Mechanistic Models of Corrosion Fatigue

There have been a number of mechanistic models of corrosion fatigue proposed. Four are briefly described here: film rupture/stabilization, mechanical/chemical dissolution, hydrogen embrittlement, and/or strain induced corrosion cracking.

- *Film rupture/stabilization*—There are several variations of this model, which ascribes accelerated crack growth to the rupturing of protective films and subsequent reoxidation or corrosion when the bare metal is exposed to the environment (Magnin, 1983; Ford and Combrade, 1985). A variation of this model explains the onset of corrosion fatigue or stress corrosion cracking as being controlled by crack tip effects that can be explained by the superposition of an environmental effect and a strain effect (Parkins, 1972).

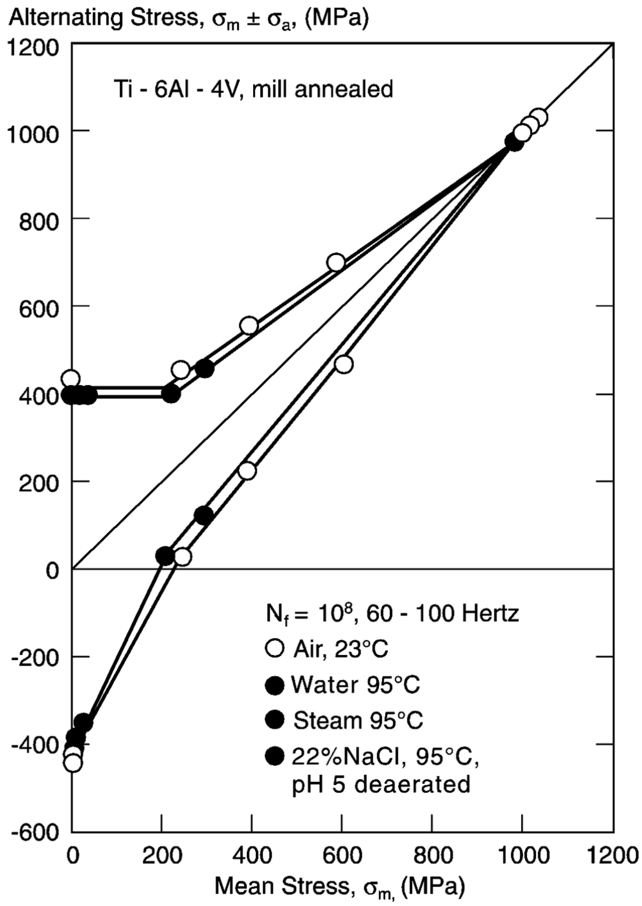

Figure 35. Modified Goodman (Smith) Diagrams for Ti-6Al-4V Alloy in Various Steam and Water Chemistry Environments.

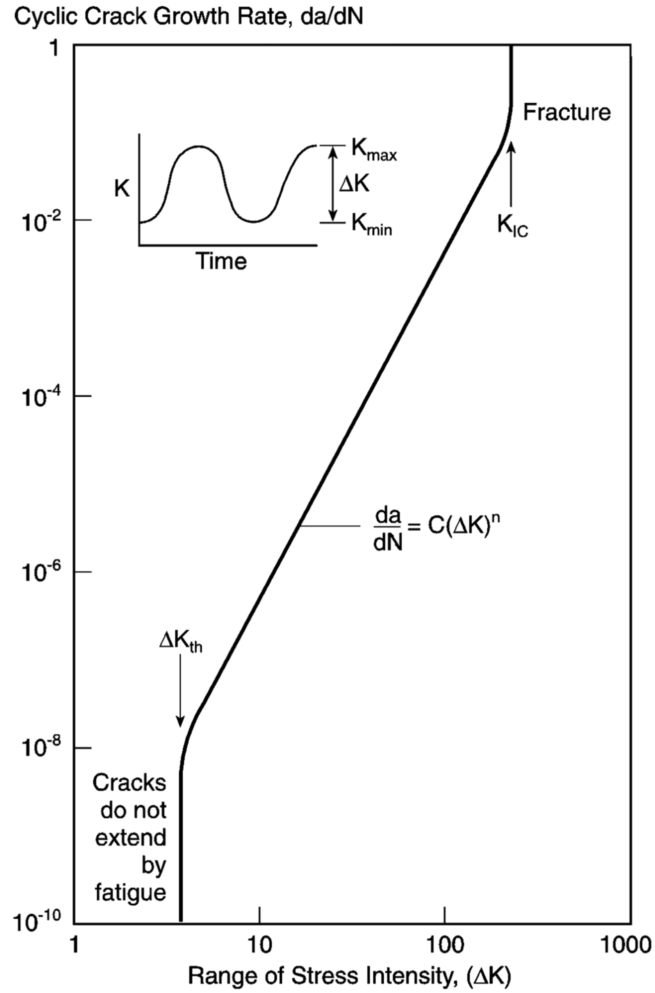

Figure 36. Cyclic Crack Growth Rate of SS as Function of Range of Stress Intensity Factors.

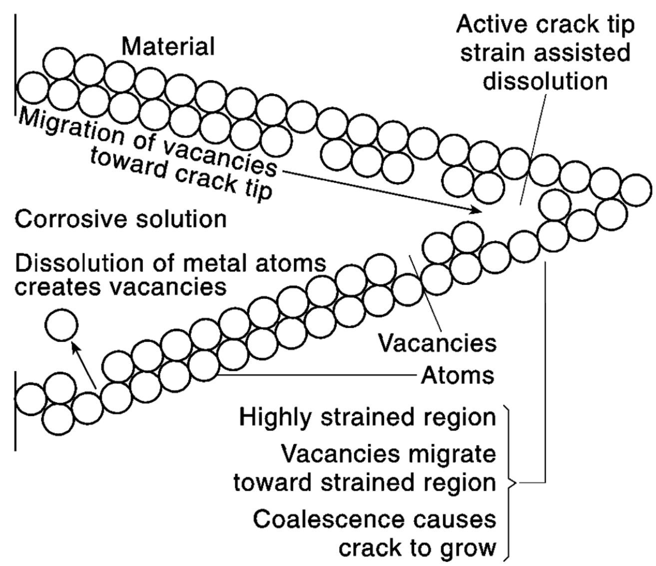

Figure 37. Schematic of Mechanical/Chemical Dissolution Model with Corrosion Generated Surface Vacancies, Migrating to Crack Tip.

- *Mechanically assisted chemical dissolution*—Figure 37 illustrates the basics of this model. Vacancies, caused by dissolution of the metal surface in a corrosive environment are driven by a stress field and accumulate at the crack tip; such coalescence results in incremental crack growth (Galvele, 1987).
- *Hydrogen assisted (or embrittlement) cracking*—Hydrogen is produced by the reaction of steel with water. Absorption of free hydrogen into the metal at the crack tip has been suggested by a number of researchers as being at the root of corrosion fatigue and stress corrosion cracking mechanisms (Hickling, 1990; Newman and Procter, 1990). A schematic of the process is shown in Figure 38.
- *Strain induced corrosion cracking*—Similar to the film rupture model, this concept involves the local disruption of protective oxide (Kussmaul and Iskluth, 1990). Destabilization of the oxide can occur by the environment (dissolved oxygen content, conductivity, and temperature of the water), mechanical means (strain rate and strain level), or by material characteristics (such as sulfur content).

ROOT CAUSES AND ACTIONS TO CONFIRM CF

Influence of Environment

The influence of the environment is pervasive. As indicated in Figure 30, the environmental influence has two main aspects: the dynamic environment, which occurs during operation, and the shutdown environment.

Influence of Static and Dynamic Stresses on CF

Corrosion fatigue cracks are driven by cyclic stresses; mean stresses produced by steady-state loads also have a significant influence on growth rates. One of the most damaging stress combinations in turbine blades is high mean stress from centrifugal stresses and moderate cyclic loads, such as from vibration modes.

Damaging cyclic stresses can range in magnitude. Large magnitude stresses such as result from the zero to full centrifugal stress cycle that occurs during unit starts or overspeeds are clearly

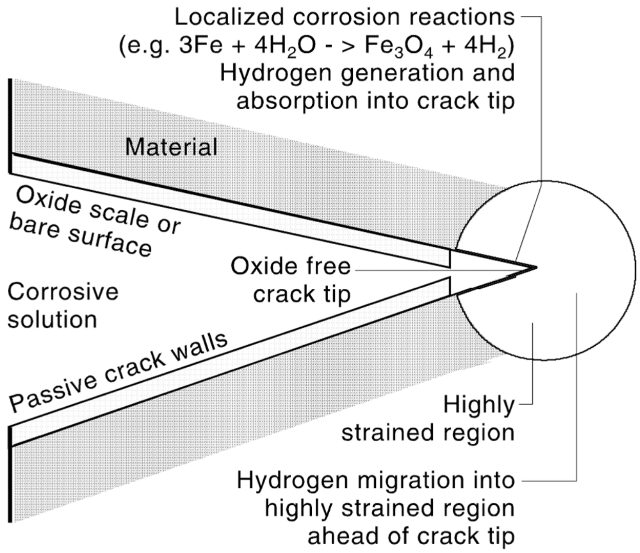

Figure 38. Schematic of Hydrogen Embrittlement Model Showing How Hydrogen Is Generated at Active Crack Tip and Then Absorbed into Material.

damaging, but infrequent (low cycle). Lower magnitude, more frequent (high cycle) stresses include those caused by:

- Synchronous resonance of the blades with a harmonic of the unit running speed,
- Nonuniform flows,
- Blade vibration induced from the rotor or disc,
- Self-excitation such as flutter.

Determining the Extent of Damage Caused by CF

The extent of damage for corrosion fatigue will be determined exactly as outlined for high cycle fatigue.

Background to Repairs and Immediate Actions

Most of the strategies to correct, or at least to mitigate, the root causes of corrosion fatigue are longer term. Repair and refurbishment options will be the same as those described for fatigue.

Background to Long-Term Actions and Prevention of Repeat Damage

Figure 39 outlines most of the available strategies for confronting a persistent problem with corrosion fatigue. Long-term actions for dealing with both corrosion fatigue and SCC begin with an economic and remaining life assessment of the problem. Thereafter, strategies will generally fall into three categories:

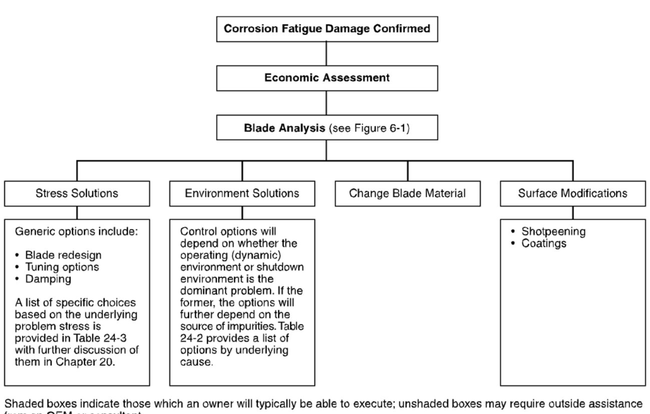

Figure 39. Long-Term Options for Addressing Corrosion Fatigue in Steam Turbine Blades Flowchart.

- Redesigning the blade or attachment to reduce stress levels
- Improving steam purity
- Changing the bulk material or surface of the blade

Redesign to lower stresses, in conjunction with changes to improve steam purity, have typically proven to be successful internationally in reducing LP blade failures by corrosion fatigue in 12 percent Cr material. If such changes are not sufficient, then changing to a more resistant material, such as blade materials with a higher chromium content or titanium, is generally recommended as the next most successful option. The proper choice will depend on the specific turbine design, the results of the economic evaluation, and identifying which of the contributing root causes is most severe.

Economic Analysis of CF Mitigation and Prevention

Proper selection of a long-range strategy for correction of corrosion fatigue will use both an economic analysis and a remaining life analysis. Suggested changes will be compared to identify the one, which can provide the most economic long-term solution.

LIFE ASSESSMENT OF BLADE CF

Life assessment for corrosion fatigue should provide answers to:

- Do blades need to be repaired/replaced immediately or can the unit operate with the existing damage until a more economically favorable outage?
- If the blades are replaced in-kind, what is the expected life?
- If a material upgrade is contemplated, how much additional life can be expected?
- What are the most troublesome stresses, and if mitigation includes a redesign to lower those stresses, to what degree is the approach likely to be successful?

Assessment of blade life will typically consist of an analysis of blade stresses followed by a fatigue analysis. For blades, a typical assumption is that the time to initiation of a crack is the effective life. APPENDIX A Figure A-7 shows a generic approach to blade assessment; this approach is applicable for corrosion fatigue damage, as well as low and high cycle fatigue.

For stress corrosion cracks, the allowable crack size is limited by fracture toughness and overspeed, as cyclic loads such as vibratory fatigue are not dominant. However, the allowable crack size for corrosion fatigue is limited by the levels of vibratory stress and other cyclic loads, and the threshold stress intensity factor.

If off-design conditions, such as operation at low load leading to stall flutter, are a primary contributor to the cyclic loading, the risk of cracking should be assessed for these conditions and monitored.

Stress Reduction Options for CF

Reducing cyclic or mean stress levels can markedly decrease the rate of corrosion fatigue growth. In the 1980s, a declining incidence of blade failures caused by corrosion fatigue in European and United Kingdom utilities was attributed to design improvements to decrease mechanical stresses in blades, such as by the use of:

- Freestanding blades (Atrens, et al., 1983).
- Continuous banding (Mayer and Besigk, 1983).
- Damping design changes and the use of lacing, shrouding, or similar systems.

A major limitation to this action is that it is normally instituted by the blade designer and is not within the direct control of the operator. There are general trends in turbine design toward higher outputs, better efficiency, and better material utilization, all which tend to increase the blade stress level and thus could result in a reversal of this favorable trend.

There are a number of specific actions that can be taken, if economically justified, particularly if damage is severe and/or blade replacement is required. General approaches include:

- Changing the response of the blade by design modification,
- Changing the response of the blade by tuning,
- Changing the damping of the blade,
- Changing operating procedures, for example, avoiding extreme off-design operation such as low load, high backpressure operation that can help to avoid stall flutter.

Within each category are numerous choices. For example, among the tuning options are: adding or reducing the weight of the blade, changing to/from freestanding/shrouded blades, changing to/from blade groupings, adding or moving tiewires or tenons, or adding material. The choice of design change will depend on an accurate identification of the underlying cyclic stress problem. Analysis of blade life will be central in these considerations. Confirmation testing of the changed design is critical to ensuring that the change has indeed lowered the troublesome stress. Note that although mechanical design improvements have had some effect in decreasing failures, the situation is so complex that there will always be some vibratory stresses active, e.g., untuned resonances at high harmonics.

Improving the Steam Chemistry Environment

The role of the environment is dominant in corrosion fatigue. As indicated on Figure 39, primary alternatives to control this factor include:

- Optimizing or changing chemistry,
- Controlling impurity ingress,
- Changing unit operating procedures, particularly for shutdowns and startups.

It is important that a means to determine the efficacy of changes in steam chemistry be used. Such procedures may include testing with a converging-diverging nozzle and an early condensate monitor so that the deposition process can be determined directly. A final option pertaining to the environment is to improve the surface finish of blading. Deposition and subsequent concentration of impurities are a function of blade surface finish, so that improvements may help slow the accumulation of corrosion fatigue damage.

Change Blading Materials

McCloskey, et al. (1999), list some of the advantages and disadvantages of the potential materials for resistance to pitting, corrosion fatigue, and stress corrosion cracking. Note that switching to titanium, the most desirable upgrade, provides an additional benefit of reducing the level of mean stresses. Mean stresses vary with blade length and design, but are typically limited to about 275 MPa (~40 ksi); in titanium, mean stresses for a similar design should be around 170 MPa (~25 ksi) because of the lower material density (Bates, et al., 1984).

Surface Modification and Coatings for Mitigation and Prevention of CF

Shotpeening. The most common surface modification to improve corrosion fatigue resistance is shotpeening. Shotpeening is, on balance, beneficial. It introduces surface compressive stresses that improve fatigue and corrosion fatigue resistance, although it does not change the material's resistance to pitting or generalized corrosion. It creates only a thin layer of compressive stress and it does not slow the growth of corrosion pits. Where such pits extend through the surface layer, the rate of crack growth by corrosion fatigue is not diminished, as shown in Figure 40. Shotpeening can create surface roughness, which can allow increased deposition to occur. The benefits of shotpeening are material dependent. For example, shotpeening has been found to improve the fatigue strength of Ti-6Al-4V and 17-4 PH in 22 percent NaCl solution, but was only marginally helpful for Type 403 stainless steel.

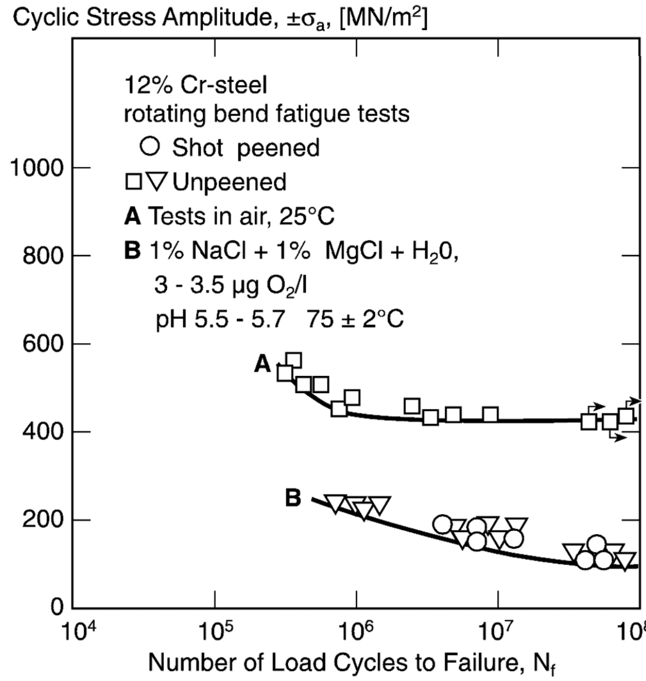

Figure 40. Effects of Shotpeening on Corrosion Fatigue Properties of 12 Percent Chromium SS for Various Steam and Water Chemistry Environments.

Sacrificial Coatings. Numerous coating systems have been evaluated for use in controlling blade corrosion (Kratz, et al., 1987; Ortolano, 1987). Although there have been reports of successful application of such coatings, their use is not extensive, and no comprehensive database on the long-term performance is currently available. Although there are ongoing developments, the most effective coating materials for decreasing pitting and corrosion fatigue have been found to be ion vapor deposited (IVD) aluminum, nickel aluminide (NiAl) diffusion coating, and nickel-cadmium (NiCd) electroplate.

CASE STUDY V—ANALYSIS AND SOLUTION OF CORROSION FATIGUE ON A BLADE AIRFOIL

Unit and Problem Description. Twenty-three of 1120 freestanding L-1 blades on a 600 MW fossil fuel unit were found to be cracked. The unit had experienced about 73,000 operating hours. The majority of the cracks were found to be at the trailing edge, about 15 to 20 cm (6 to 8 inches) from the blade tip. Metallurgical analysis indicated significant corrosion evidence—pits and heavy deposits were evident on the vanes. A review of unit operating records confirmed periodic contamination from condenser leaks. The location of cracking seemed to point to an underlying cause of vibration; when combined with the evidence of corrosion the failure team suspected corrosion fatigue as the underlying cause (Figure 41).

Analyses Performed. The following steps, typical of a blade life assessment, were performed using an FEA code to confirm the underlying cause and choose among proposed solutions:

- The geometry was defined. For this analysis, a freestanding vane profile with axial entry and a fir tree root with four pairs of hooks was chosen. Dimensions were specified by the user from plant measurements.

Figure 41. Campbell Diagram of L-1 Blade for Case Study V. First Three Natural Frequencies Show Good Separation with Harmonics of Running Speed, but Fourth Natural Frequency (Second Bending Mode) Is Coincident with 13th Harmonic of Running Speed.

- The user specified the material (403 stainless steel) and rotational speed (3600 rpm).
- A finite element model of the blade was generated.
- Steady-state stresses were calculated. An analysis of the stresses generated by steam forces applied to the blade surface in both tangential and axial directions was then performed. Maximum stresses were found to be in the root. However, in this particular case, the field failures were in the blade and thought to be caused by the dynamic response of the blade. As a result, the steady stresses at the failure location were recorded for subsequent analyses.
- Natural frequencies and mode shapes were calculated with no forcing applied. Damping was ignored as its effect on natural frequency is small for freestanding blades. The effect of stress stiffening (causing blade frequencies to be higher than would be measured at zero rpm) was included in the calculation.
- A Campbell diagram (Figure 42) was plotted to illustrate the relationship of mode frequencies to rotor speed. The effect of stress stiffening can be seen in Figure 41 as the natural frequencies of the blade (the roughly horizontal lines) increase as the rotor speed increases from zero to 3600 rpm. The first two modes for the blade were removed from natural frequencies of the rotor. However, as shown, the second bending mode (fourth horizontal line up) was very close to the 13 multiple of the rotor speed (“thirteenth per-rev”).
- A plot of the modal shape for the second bending mode was made. A nodal line representing points of zero displacement during resonance of the mode appeared to be about 15 cm (6 inches) from the end of the blade, which corresponded to the failure location. Thus, stresses were confirmed to be high in this location whenever the second bending mode was excited.
- Dynamic stresses were then calculated to determine whether the blade could withstand the resonant stresses (detuning these higher frequencies being generally impractical as discussed above). The second bending mode had maximum stress occurring at the blade edge consistent with the cracked location on the blade. The dynamic stress level was calculated to be approximately 9 to 12 ksi.
- Estimated blade lifetime was calculated from a fatigue analysis. The fatigue program used a local strain approach that allowed for the cumulative effect of multiple strain sources such as modal resonance and mean strains from centrifugal loading. The life was assumed to be the time to initiation of a crack. User input includes the material, unit history (start/stops), and operating temperature. Steady stress amplitude, dynamic frequency, and stress amplitude determined from prior calculations were also used in the fatigue analysis. There was evidence of corrosion in the failure so that the effect of environment was important. The results of the fatigue analysis were:

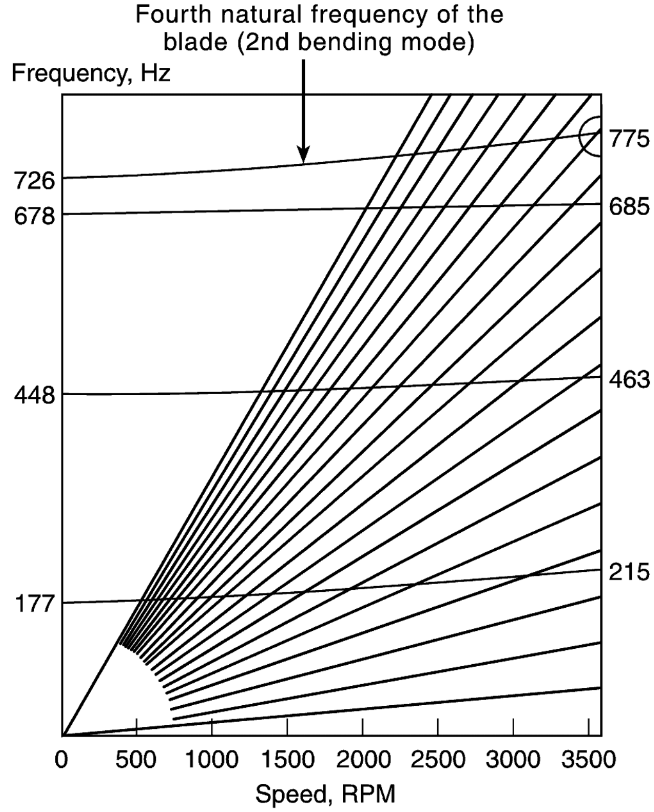

Figure 42. Failures of Low Pressure Steam Turbine Blade in Tiewire Hole Due to Stress Corrosion Cracking.

- Steady stresses at the failure site: 375 MPa (54.5 ksi)
- Dynamic stresses at the second bending mode: 75 MPa (10.9 ksi)
- Life in pure steam environment: 1.5 × 107 hours (infinite)
- Life in aggressive environment (22 percent NaCl): 3.5 × 103 hours (limited)

This indicated that the combination of an aggressive environment and the resonance or near resonance of the second bending mode was sufficient to cause failure in a fairly short period.

Evaluation of Potential Solutions. Four alternative solutions to fix the problem were evaluated:

1. Better control over contaminants in the environment
2. Detuning the blade to reduce dynamic stresses from second bending mode
3. Change to a more corrosion resistance blade material—titanium
4. Modify the blade design to reduce stresses. The modified blade design was considered for both Type 403 stainless steel and for titanium.

Option 1 was examined first. The primary source of contaminants had been condenser leakage. Steps were taken to improve monitoring and prevent recurrence. However, it was decided not to rely on just this option. As a result, the last two options (3 and 4) were considered the most feasible.

The blades were replaced with the modified design using Type 403. There have been no subsequent failures.

CASE STUDY VI— ANALYSIS AND SOLUTION OF CORROSION FATIGUE ON A BLADE AIRFOIL

Unit Background. A 400 MW reheat unit has a once-through boiler, seawater cooling, and mixed bed condensate polishers.

Development of the Problem. During a three month period, cooling water leakage occurred in a discontinuous, but periodic, frequency. Cation conductivity in the condensate and feedwater increased up to 2 mS/cm, for about 30 to 60 minutes per day. At the end of this period, vibration was detected in the LP turbine.

Description of the Damage and Root Cause. The turbine was uncovered and five broken, freestanding L-2 rotating blades were found (Figure 43). According to the turbine design data, the blades were in the phase transition zone. The blades were broken at a position that was visible as the transition between reddish deposits at the blade foot and blank metal in the upper part of the blade, in other words at the phase transition on the blade. A laboratory investigation found chloride and sodium at the crack site and confirmed corrosion fatigue as the underlying mechanism. A calculation of vibration frequencies did not show abnormal conditions.

Figure 43. Five Broken Freestanding L-2 Blades Broken Off at location of Phase Transition Zone.

Actions. The broken blades were replaced. The cooling water leak was repaired and a dampening wire was introduced into the blade design. Thus, both the environmental and stress contributors were improved.

All these mechanisms are closely related, as shown in Figure 44, which shows a schematic representation of the interrelationships and the influence of materials, stresses/strains, and environment. The regimes of load (zero, sustained/rising, and cyclic) and environment (inert or aggressive) and the nature of the accumulation of damage determines that stress corrosion cracking occurs with steady or rising stresses and an aggressive environment; corrosion fatigue with cyclic stresses in an aggressive environment. Pitting is most prominently initiated during shutdown when poor layup practices have been used, thus allowing the formation of a moist, liquid, oxygenated environment.

DAMAGE ACCUMULATION AS A COMBINATION OF SYNERGISTIC MECHANISMS

Most environmentally-assisted damage results from a combination of mechanisms; a factor that complicates its analysis. In blades, for example, damage often consists of three regions with differing fracture morphology (Figure 45):

- An initiation site, associated with a pit or other evidence of corrosion,
- A region of damage accumulation by fatigue, corrosion fatigue, or stress corrosion cracking,
- An overload fracture area.

Early damage accumulates in a highly localized area, most commonly a corrosion pit, but which can also be caused by:

- Fretting,

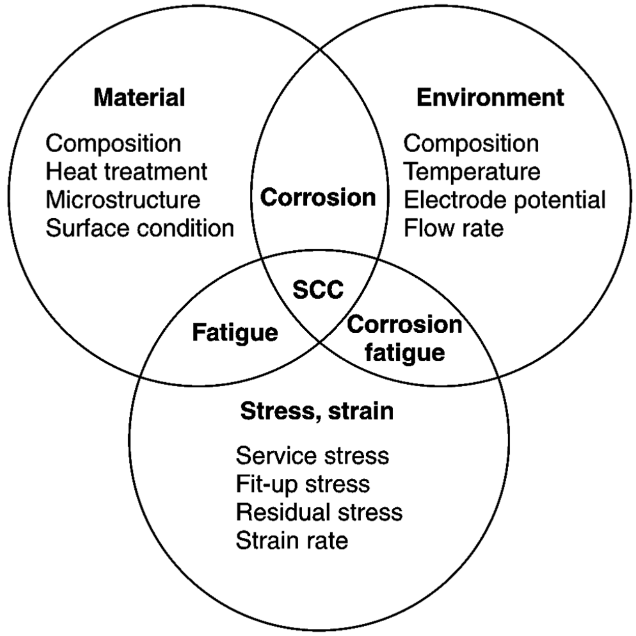

Figure 44. Major Material, Environment, and Stress Influences on Stress Corrosion Cracking, Corrosion Fatigue, and High Cycle Fatigue.

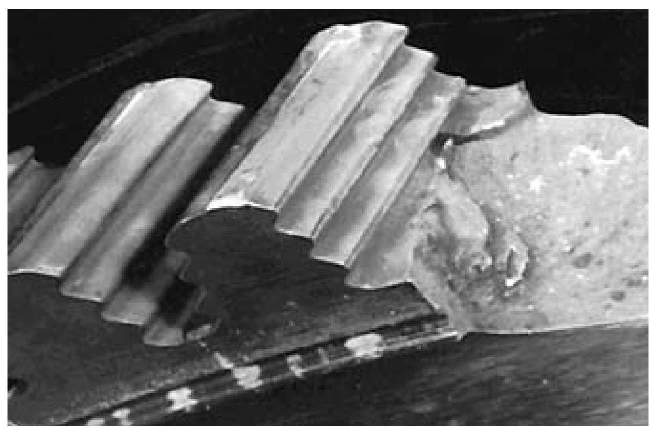

Figure 45. Failures of Axial Entry "Fir Tree" Blade Roots in Low Pressure Turbine Disc Due to Stress Corrosion Cracking.

- The presence of manufacturing defects,
- Microscopic imperfections in the material such as preferential dissolution of persistent slip bands or by mechanical rupture of the passive film at slip steps, or
- Specific adsorption of species that locally reduce surface energy.

Steady stresses and surface effects govern the rate of damage accumulation in this regime. Specific signs of a corrosion related origin will vary by material and local environment. For example, in failed AISI 403 blades, there is usually a region of corrosion related intergranular attack prior to the initiation of crack propagation transgranularly by fatigue. In AISI 630, evidence is typically corrosion along intermartensitic boundaries at the initiation site. Once a corrosion pit has formed, a rapid decrease in the local fatigue resistance occurs, leading to an accelerating synergistic effect of cyclic loads and environment. In this phase, growth occurs as the defect is first affected by the relatively large stress cycles that accompany infrequent loads such as imposed by unit start-stops. Damage accumulates in response to these loads with or without an environmental effect (without = low cycle fatigue; with = stress corrosion cracking).

When the crack has grown to a sufficient size, damage begins to accumulate from the more frequent strain cycling caused by various vibration sources such as: synchronous vibration, nonsynchronous vibration, and rotor torsional modes. Damage accumulates at this stage by high cycle fatigue or, with environmental assistance, by corrosion fatigue. The final phase is overload failure when the section thickness can no longer withstand the applied loads.

Although stress corrosion cracking can occur on blades, its most significant steam turbine manifestation is in rotors and discs. Susceptible locations long recognized as major potential problems include: the keyway, bore, web/disc face, entry slots, steam balance holes, and the disc rim blade attachments (steeples). Stress corrosion cracking is surface initiated. In service situations, pitting or crevice corrosion often precedes SCC, although field failures have been found that did not initiate at pits. Initiation and subsequent propagation can be either intergranular or transgranular with respect to prior austenite grain boundaries. In rotor materials, SCC is almost always intergranular. When the SCC is transgranular, propagation typically produces flat facets that are observable macroscopically. The cracks can appear branched (most typically) or straight: the appearance is a function of applied stress intensity and environment.

SCC fracture surfaces typically contain easily discernible regions of initiation, slow propagation, and final rupture. The region of slow growth will often exhibit corrosion products such as iron oxides or discoloration compared to the region of final rupture. Deposits containing sodium carbonate (Na_2CO_3) and/or NaOH frequently have been found on cracked rotors. Sulfides have also been found in some cases. Multiple, small cracks near the origin of the main fracture are common. Fractures produced by SCC typically appear brittle, that is, there is little or no deformation; fracture surfaces also appear rough in comparison with corrosion fatigue. Striations are typically not associated with SCC cracks (Figure 46).

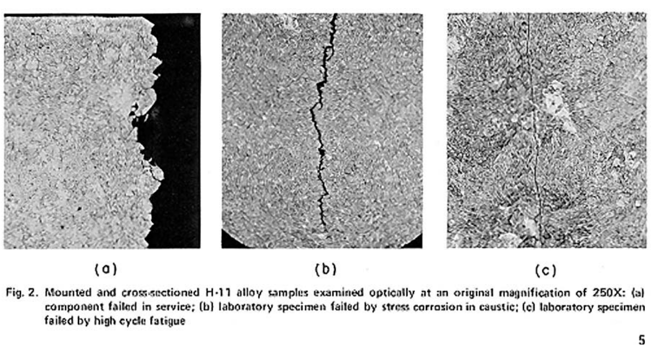

Figure 46. Comparison of Microscopic Photos of Corrosion Fatigue (Transgranular) and Stress Corrosion Cracking (Intergranular).

FIVE STAGES IN THE ACCUMULATION OF STRESS CORROSION CRACKING DAMAGE

Stress Corrosion Cracking Process Stages

Damage starts in a localized, highly stressed region with initiation from locations with fretting, the presence of manufacturing defects, pitting, or another localized corrosion process. Once an initial defect has formed, damage accelerates through the synergistic effects of environment and stresses. Where the stresses imposed are steady-state, the resultant damage is termed stress corrosion cracking. Failure occurs at stress levels that can be much lower than those that result in macroscopic yielding. Flaws propagate by stress corrosion cracking until the loss of section results in fracture by overload. The mode of damage accumulation may also change to high cycle fatigue once the stress corrosion cracks reach a critical size, and failure can occur fairly rapidly, driven by vibratory stresses imposed by the blades.

The process leading to failure by stress corrosion cracking consists of five stages (APPENDIX A Figure A-9 and A-10):

1. An incubation period
2. Initiation
3. Stable crack growth by SCC
4. A period of accelerated crack growth by stress corrosion cracking, high cycle or low cycle fatigue
5. Unstable crack growth or fracture

Stages 1 and 2 are often combined, as are stages 3 and 4, resulting in three regimes of SCC: initiation, propagation and final failure. APPENDIX A Figure A-9 and A-10 illustrates the five steps schematically, indicates the size of defects typical for each stage or the governing growth rate (da/dt), provides a range of times for each stage of defect growth, and shows that there are strategies for dealing with SCC throughout the various stages.

The total time to failure by stress corrosion cracking includes both the initiation and propagation times. Initiation may be the key step in the SCC process. General relationships between material properties, environment, and susceptibility to SCC have been established by numerous laboratory investigations of rotor materials. There is a clear effect of material yield strength on time to initiation: the higher the material yield, the more the susceptibility to SCC initiation. In an analysis of the field experience, materials with a yield strength less than 750 MPa (~110 ksi) have been found to exhibit no SCC cracking. In high purity water (conductivity < 0.2 mS/cm) in laboratory tests, materials show susceptibility only as a function of yield strength and for yield strengths in excess of about 1000 MPa (145 ksi). In this high purity water environment there was no cracking in materials with yield strength below 1000 MPa (145 ksi) even if the applied stresses were greater than yield in notched specimens.

Repairs and Actions to Mitigate Stress Corrosion Cracking

There are some short-term strategies that can be used to deal with SCC in the blade attachment. These strategies can allow the operator some time before implementing the longer term options. These include:

- Run for a limited time with cracks until a major LP turbine overhaul can be scheduled.
- A variety of mechanical repairs.
- Weld repairs. In fossil units, all three options are routine; in nuclear units, the typical actions have been either mechanical repair or replacement with a new disc/rotor with essentially no use of weld repairs, although this is changing as weld repairing in nuclear units becomes accepted.
- Run until the next major overhaul. It may be safe to run until the next major overhaul even with cracking in the disc rim attachment. Depending on the particular circumstances, some significant defects may be tolerated.

A couple of practical points about how stress corrosion cracks grow may make this option quite rational. First, stress redistribution occurs as the compliance of the joint changes with a growing crack. Thus, there may be some unloading around the largest defects, which can temporarily allow that defect to slow or stop growing. Second, as noted previously, the rate at which stress corrosion cracks grow is relatively independent of applied stress intensity, and thus of flaw size. Therefore, a crack growing by SCC will not tend to accelerate with increasing length. However, it is important to remember that once the flaw reaches a sufficient length such that the damage mode shifts to high cycle fatigue, then failure can occur in a very short time, and the rate at which damage occurs is a strong function of crack length.

A remaining life assessment is needed to make the decision about whether to continue to run with a known defect in the disc rim attachment area. Unfortunately such assessments cannot typically be performed within a reasonable period after cracks are first found. As a result, unless existing assessments are on hand, it is more likely that such assessments will need to be considered as central to the long-term and accordingly are discussed with the other long-term options below.

An owner can prepare for the eventual discovery of cracks in the disc rim attachment by performing ahead of time the appropriate stress, fracture mechanics, and probabilistic analyses, based on the best available data. In this manner, acceptable flaw sizes can be determined, and those calculations used during inspections to determine whether immediate action is required. Also, the owner can lay out what short-term or long-term options will be pursued as a function of the severity of cracking that is found.

Mechanical Repairs

Field proven mechanical repairs have included:

- Flaw removal by grinding, light machining, or "skim cutting." This is the most common means to remove shallow indications. Skim cutting typically is not an option on finger root attachment types because of high stresses, particularly with significant crack depths. Skim cutting is typically followed by polishing to provide a smooth surface finish and may be followed by shotpeening to produce surface compressive residual stresses. Shotpeening is also typically used after machining new dovetail/steeple surfaces. If the fit up between the blade and steeples is changed, the contact stresses may increase.
- Machine larger radii in the profile to reduce the stress concentration. This will also remove shallow cracks.
- Blade removal. One or more blades may be removed, with a complementary set removed 180 degrees away to maintain balance, to eliminate loading of cracked sections. This is a useful alternative if damage is localized to a few blades. Deblading of an entire row has been used when cracking is extensive around the circumference. In this case baffles or blocks are installed to simulate the pressure drop across the row. This is an interim measure taken until a more permanent fix can be employed.
- Disc removal. In some discs with severe SCC in the rim attachment and keyway, the affected disc is removed and replaced by a steel or titanium pressure plate (also called "pressure block" or "dummy block") to maintain the pressure drop across the section. This is also an interim measure.
- For side-entry blades, consider "drop notch" modification. Existing blades can be reused.
- Install spacers over the notched portion of the rim to reduce centrifugal stresses at the initiation site.

Two other mechanical repairs fall into the category of activities that can be conducted during current outage if the replacement blades are available. These are:

- For straddle-mount blades, consider the "long shank blades" option. New blades are required. This involves remachining a new attachment region in the disc below the affected area and installing blades with longer shanks. This has also been termed the "steeple drop" approach (Figure 47). It modifies the existing steeple, removing SCC damage, and reblading with "long shank" blades. The results of computational analysis of stresses, flaw growth (through fracture mechanics) by low cycle fatigue and high cycle fatigue and SCC, and remaining life assessment have indicated that the approach can significantly increase the life of the SCC-damaged disc and reduce the need for rotor replacement.
- Reblade with titanium blades to lower centrifugal stresses.

Combinations of these various mechanical options have been utilized. For example, one operator chose to use titanium blades for locations with shallow cracks, titanium blocks (devaned blades) for deeper cracks, and long shank blades for those cracks that were too deep. A stress and fracture mechanics analysis was performed prior to the inspection in order to establish the allowable flaw size ranges for each option.

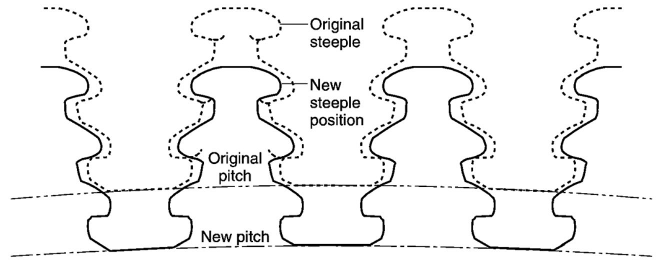

Figure 47. "Long Shank" or "Drop Notch" Blade/Disc Modification to Mitigate Need for Replacement or Repair Welding of Disc/Rotor.

Another operator used skim cutting, polishing, and shotpeening of shallow defects with selective use of titanium blades to reduce centrifugal stresses in some of the dressed out areas as a short-term solution. Over the longer term, they will replace the affected rotors with upgraded designs.

Note that the mechanical methods that rely on replacement-in-kind will not improve the susceptibility of the affected area to further SCC attack. They will likely provide, at best, only the same life as the original part. In order to improve the lifetime of the mechanical repair over the original, an improvement in some condition contributing to the problem will be needed such as to stresses (for example by changing the fit up), surface conditions, or environment.

Weld Repair and Surface Modifications to Mitigate or Prevent SCC

Welding is commonly used to repair rim attachment area damage in fossil units, and has recently been successfully employed in foreign and domestic nuclear units. Weld buildups can provide improved performance to the original rotor, primarily by using a weld material that has superior resistance to SCC. One such material is 12 percent chromium (Figure 48). Damaged shrunk-on discs can be repaired in situ or unstacked and repaired off the rotor or replaced. Damaged rims of integral rotors can be machined off and repaired by adding a shrunk-on disc or by welding. Repair welding of rotors, specifically the rim attachment area, is covered extensively in McCloskey, et al. (1999).

There are a number of surface modifications that logically make sense in responding to SCC. These include:

- Shotpeening to lower surface residual stresses or introduce compressive stresses.
- Use of a corrosion resistant rim or overlay. At least one manufacturer is recommending long-term replacement of selected attachments with the more resistant 12 percent Cr, and weld repair with 12 percent Cr material. This is a relatively recent option (Figure 49).
- Sacrificial coatings seem like a natural option, but are not sufficiently developed for this application at this time. But OEMs and other organizations are working on cost-effective remedial processes.

Longer Term Actions to Prevent SCC

SCC can be prevented in new construction by:

- Specifying low yield strength material. How low depends on other factors such as optimization of the environment along with the level of operating and residual stresses. However, typical specifications might call for rotor materials to have yield strengths less than 900 MPa (~130 ksi).
- Using fabrication processes that provide deep compressive stresses at the rotor surface. Heat treatment, rolling and honing, and shotpeening have been combined, for example, to induce surface compressive stresses of about 300 MPa (~43 ksi), which extend into the material several centimeters.

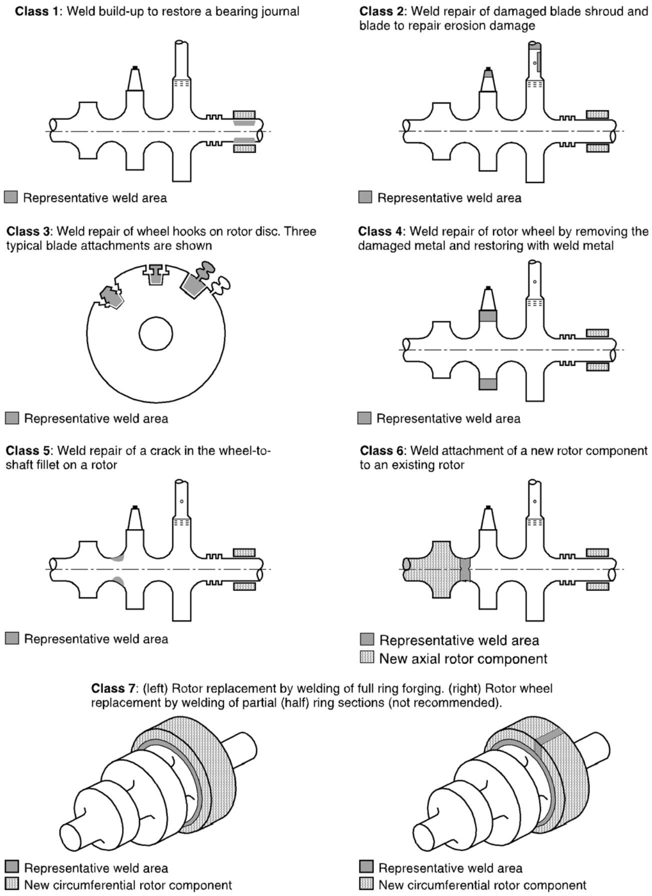

Figure 48. Steam Turbine Rotor Weld Repair Classifications.

• Keeping operating stresses as reasonable fractions of the material yield strength. Again, the limitation on maximum operating stresses needed to prevent SCC initiation and propagation depends on the probable environment, but ranges from 0.6 to 0.9 times the yield strength of the material.

• Protecting the turbine, particularly during shutdown, from damaging environments.

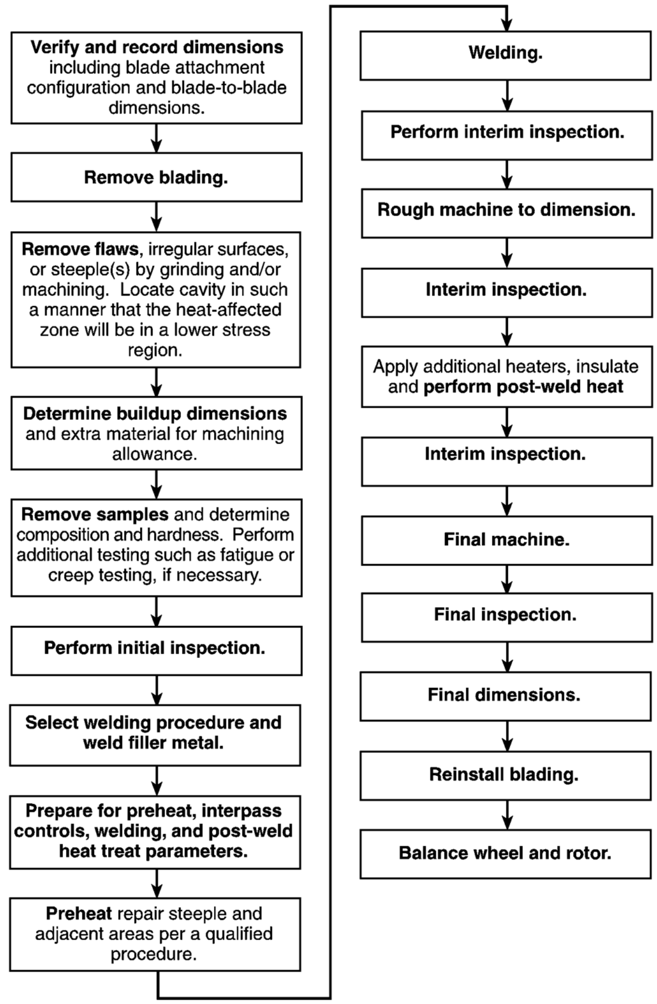

Figure 49. Roadmap of Turbine Disc/Rotor Weld Repair.

APPENDIX A

TURBINE STEAM PATH DAMAGE REPORT

1. Damage ID# ☐☐☐☐☐☐

2. Plant & Unit ID# ☐☐☐☐☐☐ ☐☐
Plant Unit ID#

3. Date Detected: ☐☐☐☐ ☐☐ ☐☐ ☐☐☐☐
Year Month Day Time

4. Unit unavailable due to damage: ☐☐☐☐ ☐☐ ☐☐ ☐☐☐☐
Year Month Day Time

5. Unit available - damage repaired: ☐☐☐☐ ☐☐ ☐☐ ☐☐☐☐
Year Month Day Time

6. ( ) Damage found during scheduled outage
( ) Damage caused forced outage

7. Generation loss: ☐☐☐☐☐☐☐ MWhr

8. Was T-CAT called? Yes / No

9. Operating conditions at failure - fuel, MW, etc. ____________________
____
____

10. Operating symptoms noticed by personnel prior to failure: ____________________
____
____

11. Consequential damage suffered in:

( ) Adjacent turbine rows ( ) Other unit stages ( ) Generator
( ) Bearings ( ) Draft systems ( ) Feed systems
( ) Valves ( ) Other (indicate) ____________________

FAILURE LOCATION

12. Section/Cylinder: ____________________

13. Affected Components:

( ) Rotating blades / buckets ( ) Stationary blades / diaphragms ( ) Casings
( ) Seals ( ) Packing ( ) Rotor - bore
( ) Rotor - keyway ( ) Rotor - disk rim ( ) Valves

FOR BLADE DAMAGE / FAILURES

14. Number of blades that failed or were damaged:

	No. Failed		Total Blades		Row	Rotating/Stationary Blades
a)	________	of	________	in	________	____________________
b)	________	of	________	in	________	____________________
c)	________	of	________	in	________	____________________

15. Nature of Damage:

( ) Surface corrosion ( ) Blade cracks ( ) Complete blade fracture
( ) Rubbing ( ) Nicks / Dents ( ) Blade deformations
( ) Deposits

16. Vendor treatment of blades before failure:

( ) Ground ( ) Machined ( ) Heat treated
( ) Shot peened ( ) Coated, type of coating: ____________________
( ) Other ____________________

Figure A-1. Turbine Steam Path Damage Report, Page 1.

17. Blade material (closest AISI spec): ______________________________

( ) Material determined by specs ( ) Material determined by test

18. Approximate time in service of blades that failed / showed damage: ______________________________

19. Identify locations of failure sites/damage on figure to right

20. Indicate blade root geometry:

FOR OTHER DAMAGED COMPONENTS
Rotor, Casings, Seals, Packing, Valves, Piping

21. Description of damaged location: ______________________________

____

____

____

IDENTIFICATION OF DAMAGE AND ROOT CAUSE

22. How was damage detected? ______________________________

23. Was NDE used during inspection? _________ NDE method and equipment: ______________________________

____

24. Primary damage mechanism:

( ) Creep
( ) Copper deposition
( ) Localized corrosion/pitting
( ) Liquid droplet erosion
( ) Fretting
( ) Creep-fatigue
( ) Other deposition
( ) Corrosion fatigue
( ) Water induction
( ) Overheating by windage
( ) Solid particle erosion
( ) Fatigue
( ) Stress corrosion cracking
( ) Flow-accelerated corrosion

25. Suspected root cause of damage: ______________________________

____

26. Verification of root cause during outage: ______________________________

____

27. Actions taken to correct root cause during outage: ______________________________

____

Figure A-2. Turbine Steam Path Damage Report, Page 2.

DAMAGE REPAIR ACTIVITIES

28. Nature of repair / replacement:

( ) Blades replaced w/ original design
( ) Blade replaced w/ modified design
( ) Blade replaced with changed material
( ) Shielding installed
( ) Blades coated or sprayed: ______

( ) Blade design change
- ( ) Alternative blade grouping
- ( ) Blade root revision: ______
- ( ) Inlet diaphragm modification made: ______
- ( ) Shrouding / coverband modification: ______

( ) Tie wires installed / changed: ______
( ) Disk machined
( ) Weld repair. Specify weld procedure specification number: ______
( ) Other (specify details): ______

Was NDE used after weld? ______
NDE method, equipment, and inspector: ______

29. Repairs made by: ______

POST REPAIR ANALYSIS

30. Sample of damage sent to lab for analysis? ______
Sample ID Numbers: ______
31. Verification of root cause after outage: ______
32. Actions taken to correct root cause after outage: ______
33. Were operating procedures revised following the damage incident? Yes / No
34. Description of changes: ______
35. Recommendations for verification during next scheduled outage: ______
36. Recommendations for corrective activities during next scheduled outage: ______
37. Recommendations for future preventative actions: ______
38. ( ) Attachments, such as photographs, copies of NDE reports, copies of lab analysis, etc.

Form prepared by: ______ Date: ______

Station Manager: ______ Date: ______

Send copies of this form and attachments to T-CAT Leader.

Figure A-3. Turbine Steam Path Damage Report, Page 3.

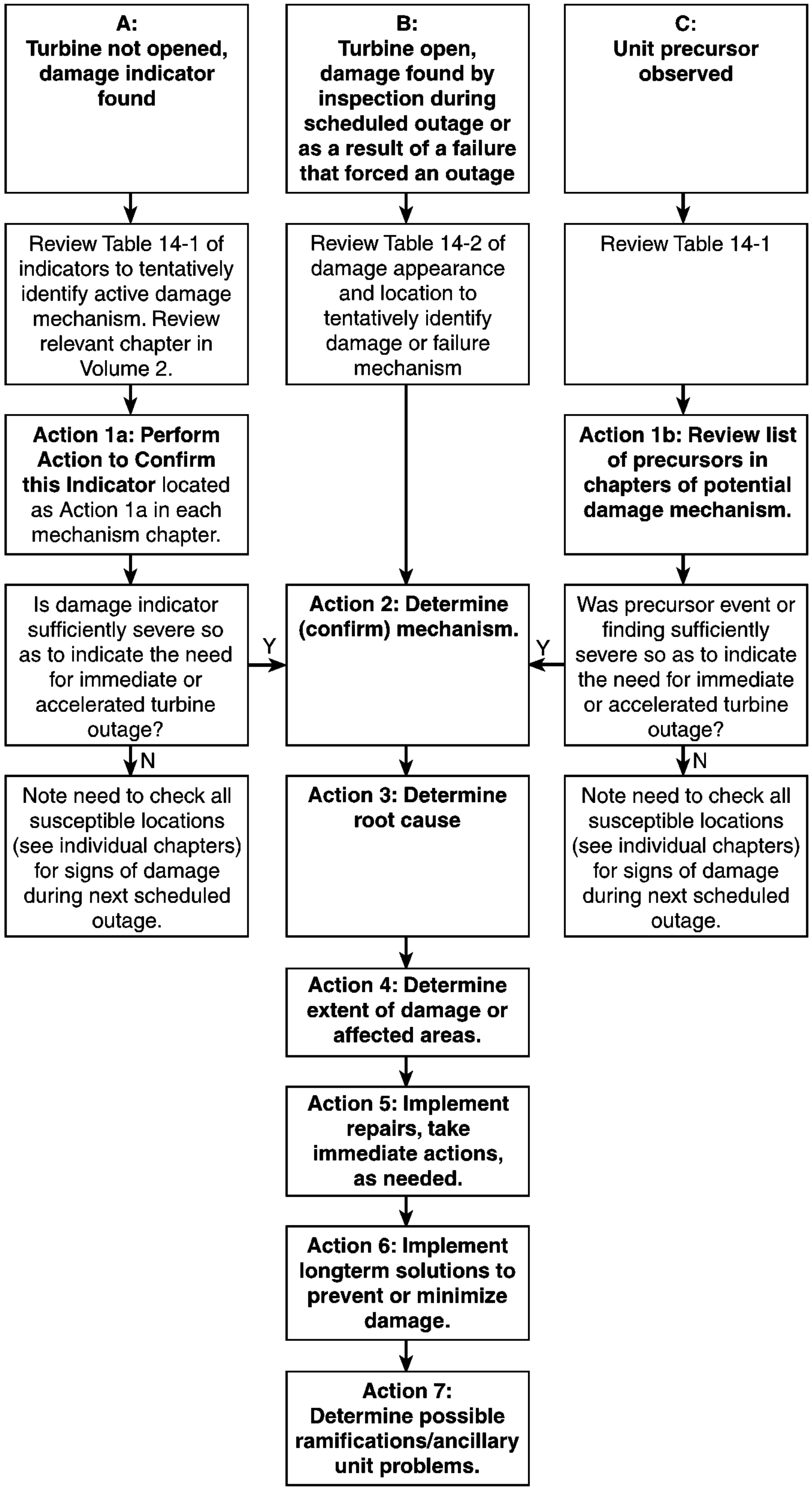

Figure A-4. Turbine Steam Path Damage Investigation Flowchart.

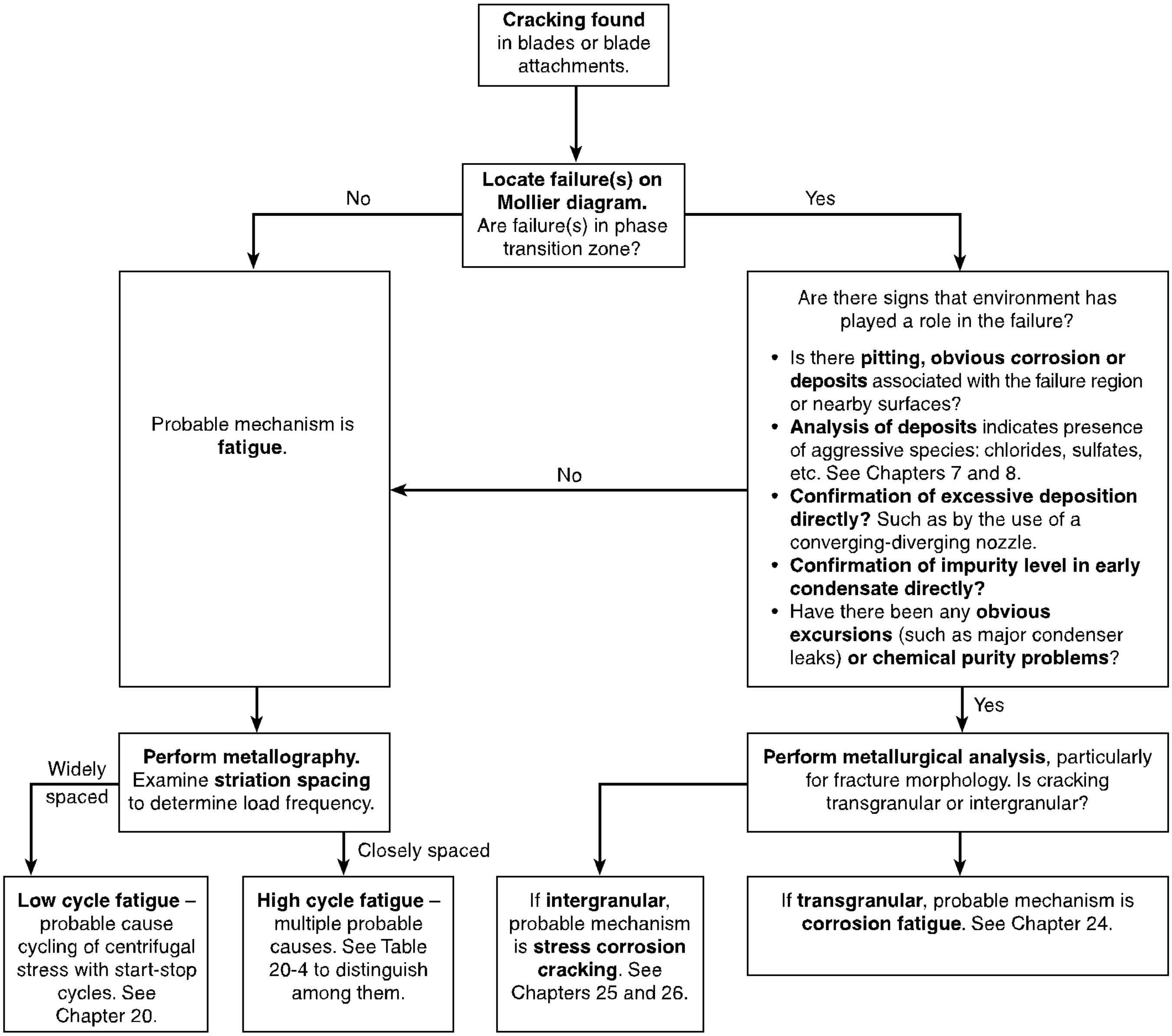

Figure A-5. Determination as to Whether Steam Path Damage Mechanism Is Either Fatigue, Corrosion Fatigue, or Stress Corrosion Cracking Flowchart.

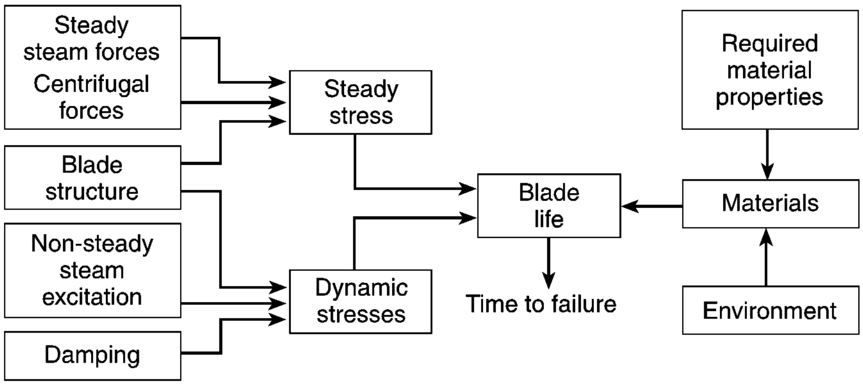

Figure A-6. Contributions of Dynamic/Static Stresses, Structure, Excitation, Damping, Material Properties, and Steam/Water Chemistry Environment on Blade Life.

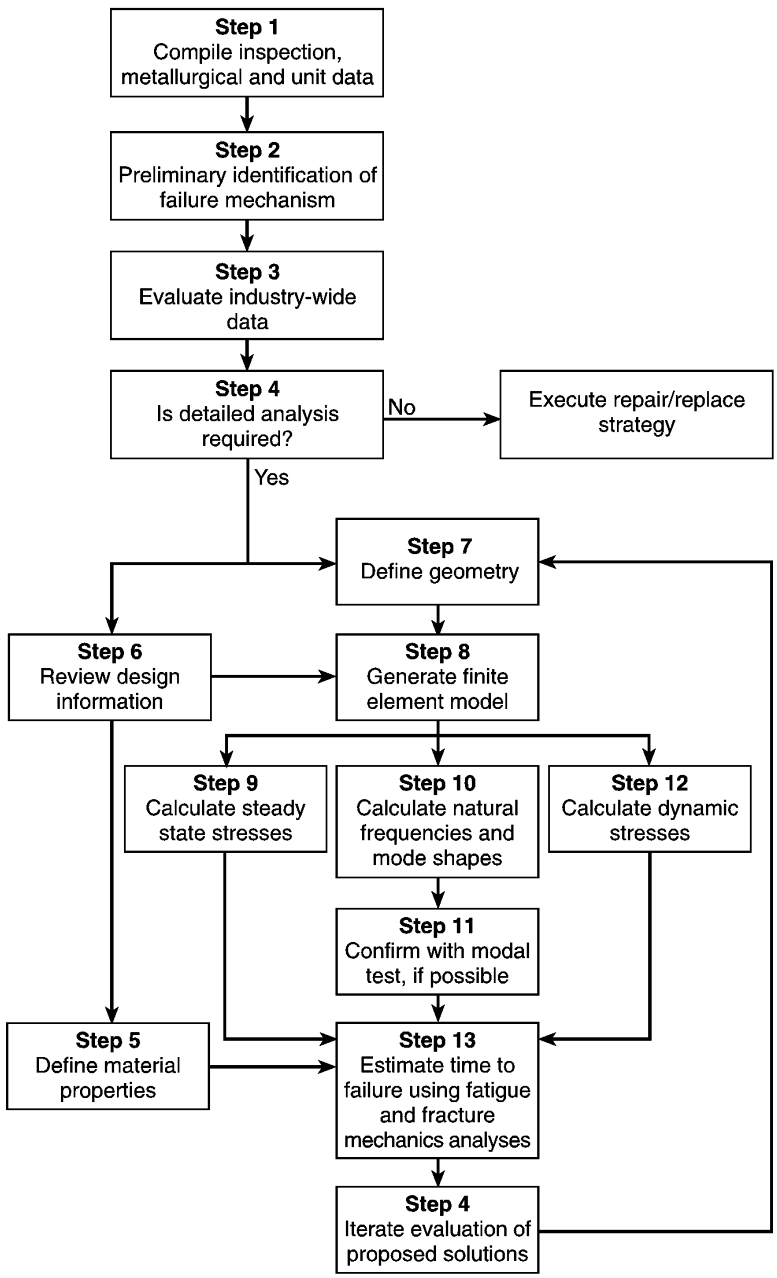

Figure A-7. Generic Flowchart for Turbine Blade Remaining Life Assessment.

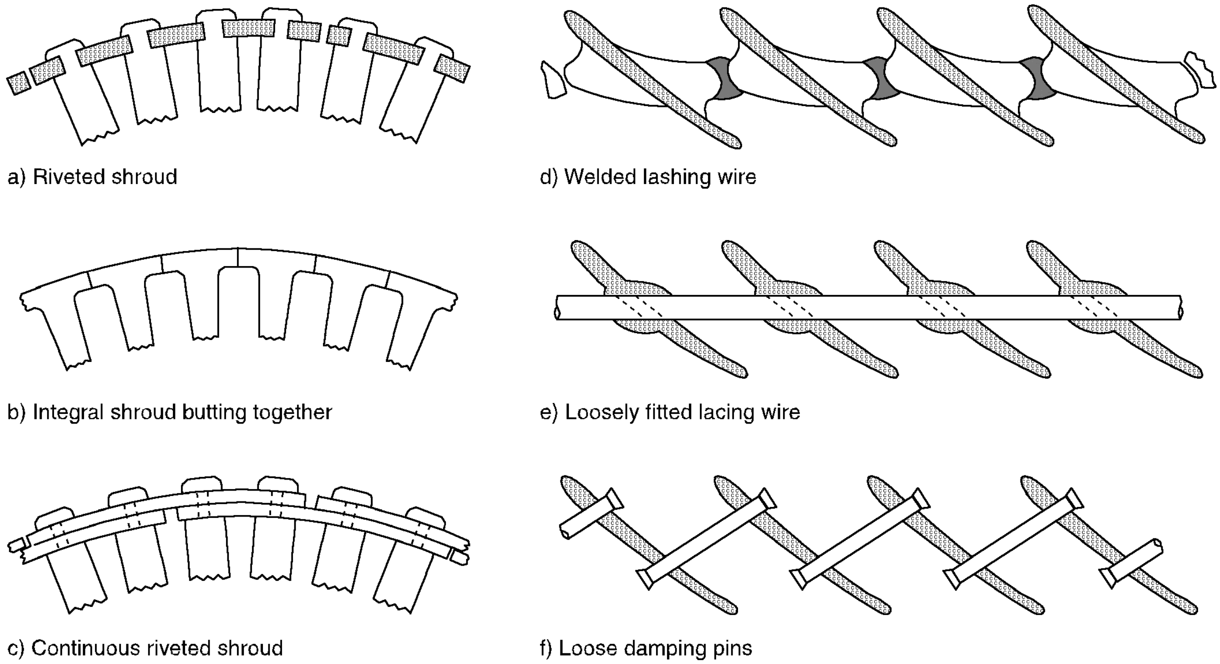

Figure A-8. Typical Tenon and Lashing Wire Rotating Blade Configurations.

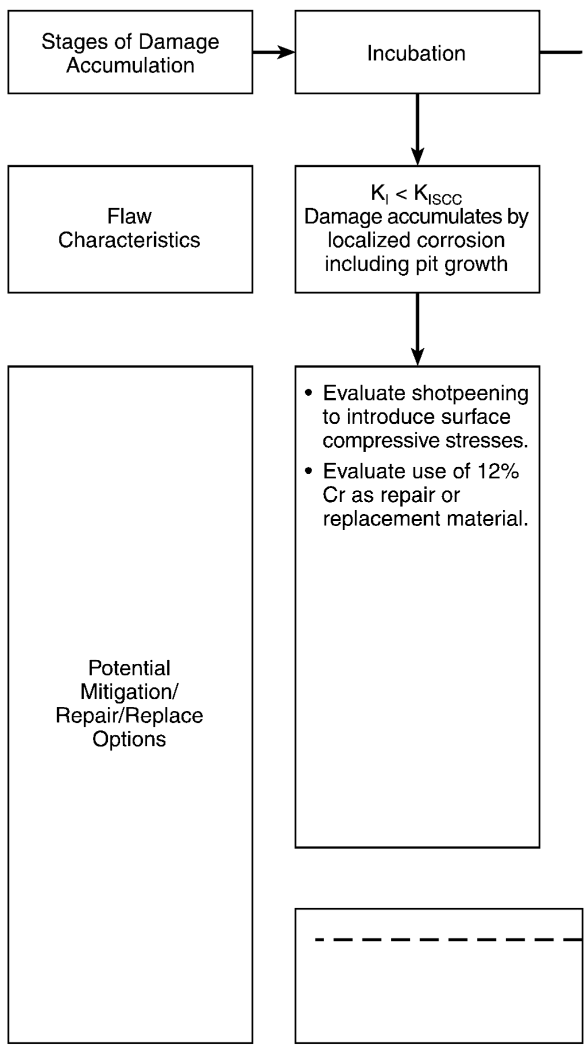

Figure A-9. Five Stages (Steps) of Damage Accumulation by Stress Corrosion Cracking in Discs and Blades (A).

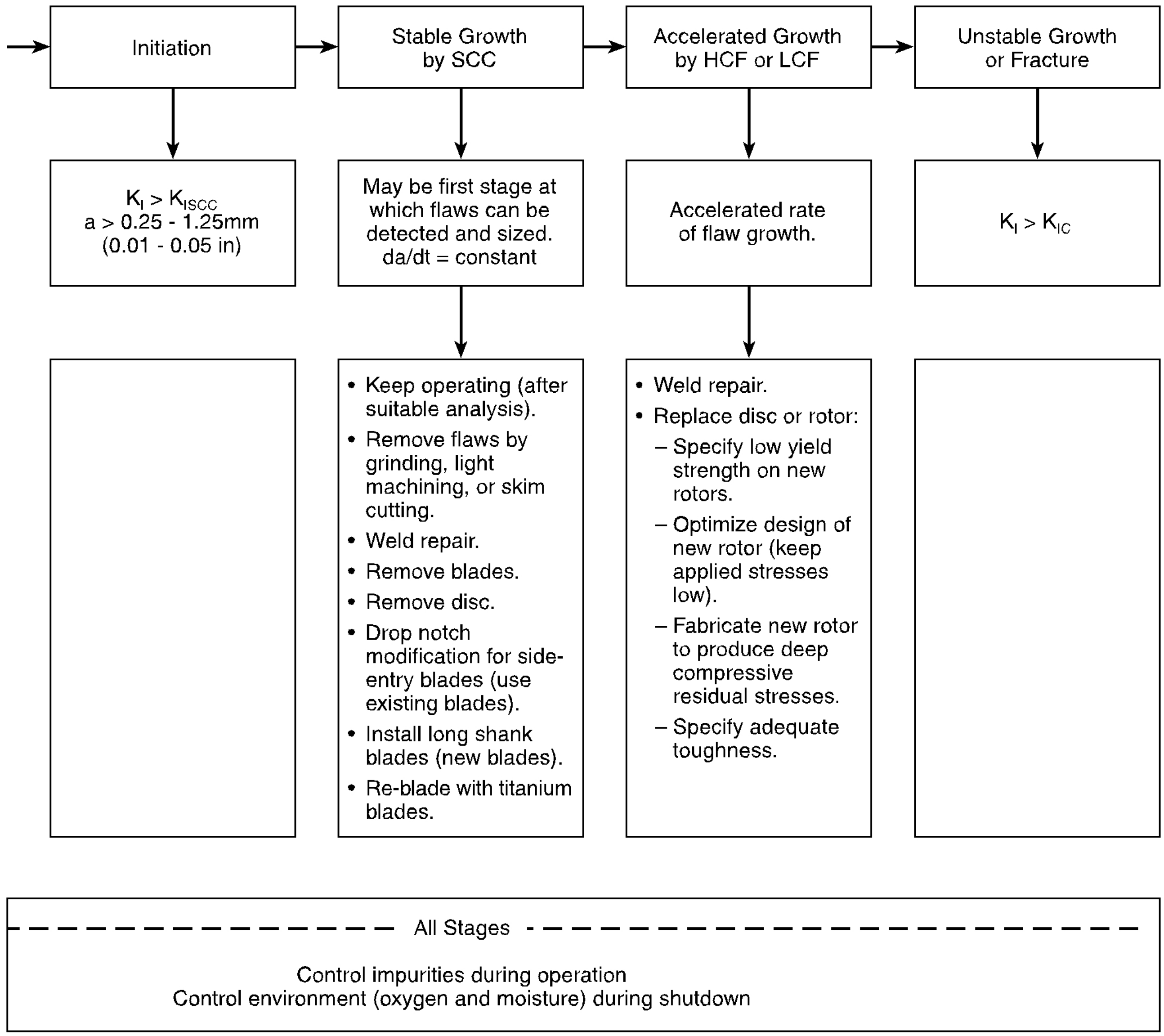

Figure A-10. Five Stages (Steps) of Damage Accumulation by Stress Corrosion Cracking in Discs and Blades (B).

REFERENCES

Atrens, A., Meyer, H., Faber, G., and Schneider, K., 1984, "Steam Turbine Blades," *Corrosion in Power Generating Equipment*, New York, New York: Plenum Press, pp. 299-328.

Atrens, A., Müller, M., Meyer, H., Faber, G., and Speidel, M. O., 1983, "BBC Experience with Low-Pressure Steam-Turbine Blades," *Corrosion Fatigue of Steam Turbine Blade Materials*, New York, New York: Pergamon Press, pp. 4-50 - 4-76.

Bates, R. C. and Cunningham, J. W., 1980, "Corrosion Fatigue of Turbine Blading Materials in Operational Environments," Second Annual Report of EPRI Project RP912.

Bates, R. C., Cunningham, J. W., Dowling, N. E., Heymann, F. J., Jonas, O., Kunsman, L. D., Pebler, A. R., Swaminathan, V. P., Willertz, L. E., and Rust, T. M., 1984, "Corrosion Fatigue of Steam Turbine-Blading Alloys in Operational Environments," Final Report CS-2932, EPRI, Palo Alto, California.

Bernal, J. D., 1970, *Science and Industry in the Nineteenth Century*, Bloomington, Indiana: Indiana University Press.

Campbell, W.,1924, "The Protection of Steam Turbine Disk Wheels from Axial Vibration," ASME Proceedings, Cleveland, Ohio.

Denk, J., 1994, "Low Pressure Steam Turbine Integrity," *Materials for Advanced Power Engineering Part I*, Dordrecth, The Netherlands: Kluwer Academic Publishers, pp. 157-170.

Dewey, R. P., McCloskey, T. H., and Rieger, N. F., 1983, "Analysis of Steam Turbine Blade Failures in the Utility Industry," Paper 83-JPGC-PWR-20, American Society of Mechanical Engineers, New York, New York.

Ebara, R., et al., 1983, "Influence of Dissolved Oxygen Content and Stress Concentration Factor on Corrosion Fatigue Strength of Turbine Moving Blade Materials," 1981 Workshop Proceedings, Palo Alto, California, *Corrosion Fatigue of Steam Turbine Blade Materials*, New York, New York: Pergamon Press.

Evans, D. H., 1993, "Comparison of Unstalled Flutter Predictions and Field Measurements for Steam Turbine Blades," Stress Technology, Inc., Proceedings of the Steam and Combustion Turbine-Blading Conference and Workshop—1992, Orlando, Florida, Proceedings TR-102061, EPRI, Palo Alto, California, pp. 4-85 - 4-99.

Ford, F. P. and Combrade, P., 1985, "Electrochemical Reaction Rates on Bare Surfaces and Their Use in a Crack Prediction

Model for the Low Alloy Steel/Water System," Proceedings of the Second IAEA Specialists' Meeting on Subcritical Growth, Sendai, Japan.

Galvele, J. R., 1987, "A Stress Corrosion Cracking Mechanism Based on Surface Mobility," *Corrosion Science*, *27*, (1), pp. 1-33.

Hesler, S. H. and Marshall, J. E., 1993, "Diagnosis and Correction of Recurring Failures in L-2 Turbine Stage," Stress Technology, Inc., Proceedings of the Steam and Combustion Turbine-Blading Conference and Workshop—1992, Orlando, Florida, Proceedings TR-102061, EPRI, Palo Alto, California, pp. 1-61 - 1-80.

Hickling, J., 1990, "Strain Induced Corrosion Cracking: Relationships to Stress Corrosion Cracking/Corrosion Fatigue and Importance for Nuclear Plant Service Life," Proceedings Third IAEA Specialists' Meeting on Subcritical Crack Growth, NUREG/CP-0112, ANL-90/22, *II*, Moscow, Russia.

Holdsworth, S. R., Nougaret, M., Roberts, B. W., and Thorton, D. V., 1997, "Laboratory Stress Corrosion Cracking Experience in Steam Turbine Disc Steels," Proceedings: Steam Turbine Stress Corrosion Workshop, Final Report TR-108982, EPRI, Palo Alto, California, pp. 45-59.

Jonas, Inc., et al., 1998, "Steam, Chemistry, and Corrosion," Final Report TR-108184, EPRI, Palo Alto, California.

Kundur, P., 1993, *Power System Stability and Control*, New York, New York: McGraw-Hill, Inc.

Kussmaul, K. and Iskluth, B., 1990, "Environmentally Assisted Crack Growth in a Low Alloy Boiler Steel in High Temperature Water Containing Oxygen," *Nuclear Engineering and Design*, *119*, pp. 415-430.

Kratz, J. L., Ortolano, R. J., and Kramer, L. D., 1987, "Corrosion-Resistant Coatings for Low-Pressure Steam Turbines," Final Report CS-5013, EPRI, Palo Alto, California.

Laird, C. and Duquette, D. J., 1972, "Mechanisms of Fatigue Crack Nucleation," Corrosion Fatigue: Chemistry, Mechanics and Microstructure, Houston, Texas: National Association of Corrosion Engineers, pp. 88-117.

Latcovich, J. A., Jr., 1997, "Steam Turbine Generator Inspection Outages—An Insurer's Perspective and Approach," in Fifth EPRI Steam Turbine/Generator Workshop, Lake Buena Vista, Florida, EPRI, Charlotte, North Carolina.

Macdonald, D. D. and Cragnolino, G. A., 1989, "Corrosion of Steam Cycle Materials," The ASME Handbook on Water Technology for Thermal Systems, The American Society of Mechanical Engineers, New York, New York, pp. 659-1029.

Macdonald, D. D., Szklarska-Smialowska, Z., and Moccari, A., 1985, "Stress Corrosion Cracking and Corrosion Fatigue in Steam Turbine Materials," Interim Report NP-4074M, EPRI, Palo Alto, California.

Magnin, T., 1983, "An Approach to the Mechanisms of Corrosion Fatigue Damage," Memoires et Etudes Scientifiques Revue de Metallurgies, LS7857/7Feb90/js.

Mayer, K. H. and Besigk, C., 1983, "Influence of Design, Manufacture and Operation on the Useful Life of Corrosion-Stressed Turbine Blades," 1981 Workshop Proceedings, Palo Alto, California, *Corrosion Fatigue of Steam Turbine Blade Materials*, New York, New York: Pergamon Press, pp. 5-19 - 5-36.

McCloskey, T. H. and Pollard, M., 1995, "Development and Implementation of a Turbine-Generator Outage Interval Extension Strategy," ASME International Joint Generation Conference, Minneapolis, Minnesota.

McCloskey, T., Dooley, B., and McNaughton W., 1999, "Turbine Steam Path Damage: Theory and Practice," EPRI Report, TR-108943 Volumes 1-2.

Nedeljkovic, S. I., Rozelle, P. F., and Seth, B. B., 1991, "Design and Verification of Mixed-Tuned Titanium Freestanding Low Pressure Steam Turbine Blades," Design, Repair, and Refurbishment of Steam Turbines, 1991 International Joint Power Generation Conference, San Diego, California, PWR-Vol. 13, American Society of Mechanical Engineers, New York, New York, pp. 203-208.

Newman, R. C. and Procter, R. P. M., 1990, "Stress Corrosion Cracking: 1965-1990," *British Corrosion Journal*, *25*, (4), pp. 259-269.

Ortolano, R. J., 1987, "Guide for the Use of Corrosion-Resistant Coatings on Steam Turbine Blades," Final Report CS-5481, EPRI, Palo Alto, California.

Parkins, R. N., January 1972, "Stress Corrosion Spectrum," *British Corrosion Journal*, *7*.

Rieger, N. F. and McCloskey, T. H., 1988, "Turbine Blade Life Assessment," International Conference on Life Assessment and Extension, The Hague, The Netherlands, pp. 219-233.

Roemer, M. J., Dewey, R. P., Atkinson, B., Mauney, D. A., and McCloskey, T. H., 1997, "Turbine-Generator Maintenance Outage Optimization: Probability/Risk Assessment of Net Present Value," Proceedings of the International Joint Power Generation Conference: Volume 2: Power, Denver, Colorado, PWR-Vol. 32, American Society of Mechanical Engineers, New York, New York.

Rozelle, P. F., Evans, D. H., and Seth, B. B., 1989, "Development and Application of a Blade Vibration Monitor," Latest Advances in Steam Turbine Design, Blading, Repairs, Condition Assessment and Condenser Interaction, 1989 Joint Power Generation Conference, Dallas, Texas, PWR-Vol. 7, American Society of Mechanical Engineers, New York, New York, pp. 37-45.

Rungta, R. and Begley, J. A., 1980, "Effect of Cathodic Potential on Corrosion Fatigue Crack Growth Rates of a Ni-Cr-Mo-V Turbine Disc Steel in Room Temperature Caustic Environment," *Metallurgical Transactions*, *11A*, (5), pp. 821-830.

Rust, T. M., Seth, B. B., and Thompson, T. L., 1990, "Application of Titanium to Low Pressure Transition Stage Turbine Blading at TU Electric," *Titanium Steam*.

Rust, T. M. and Swaminathan, V. P., 1983, "Corrosion Fatigue Testing of Steam Turbine Blading Alloys," 1981 Workshop Proceedings, Palo Alto, California, *Corrosion Fatigue of Steam Turbine Blade Materials*, New York, New York: Pergamon Press, pp. 3-107 - 3-134.

Sanders, W. P. and Southall, W. R., 1993, "A Guide to the Mechanical Condition Assessment in the Turbine Steam Path," Proceedings of the Steam and Combustion Turbine-Blading Conference and Workshop—1992, Orlando, Florida, Proceedings TR-102061, EPRI, Palo Alto, California, pp. 2-3 - 2-20.

Somm, E., 1976, "Problems in the Development of Last Stage Blades for Large Steam Turbines," Trans. Inst. of Fluid Flow Machinery of the Polish Academy of Science, *70-72*, Third Scientific Conference on Steam Turbines of Great Output, pp. 55-79.

Speidel, M. O., 1977, "Corrosion Fatigue in Fe-Cr-Ni Alloys," *Stress Corrosion Cracking and Hydrogen Embrittlement of Iron-Base Alloys*, Houston, Texas: National Association of Corrosion Engineers, pp. 1071-1103.

Speidel, M. O., 1983, “Corrosion-Fatigue of Steam Turbine Blade Materials,” 1981 Workshop Proceedings, Palo Alto, California, *Corrosion Fatigue of Steam Turbine Blade Materials*, New York, New York: Pergamon Press, pp. 1-1 - 1-23.

Stodola, A. and Lowenstein, L.,1905, “The Steam Turbine,” D. Van Nostrand Company, Boston, Massachusetts: Stanhope Press.

Tsundoa, E., Mimuro, H., Hisa, S., Sakakida, H., and Mori, S., 1989, “Torsional Vibration Evaluation Technique for Large-Capacity Steam Turbine,” Proceedings of the American Power Conference, *51*, Illinois Institute of Technology, Chicago, Illinois, pp. 109-114.

Viswanathan, R., Wittig, J. E., and Syrett, B. C., 1983, “Corrosion Fatigue of 17-4 PH Turbine Blade Steel in Chloride Environments,” 1981 Workshop Proceedings, Palo Alto, California, *Corrosion Fatigue of Steam Turbine Blade Materials*, New York, New York: Pergamon Press, pp. 4-202 - 4-214.

Wei, R.P. and Speidel, M. O., 1972, “Phenomenological Aspects of Corrosion Fatigue, Critical Introduction,” *Corrosion Fatigue: Chemistry, Mechanics and Microstructure*, Houston, Texas: National Association of Corrosion Engineers, pp. 379-380.

BIBLIOGRAPHY

Burton, W. and Ortolano, R., 1992, “Field Telemetry Testing of Long-Arc, Low-Pressure Turbine Blading,” Final Report TR-100216, EPRI, Palo Alto, California.

Ford, F. P., 1979, “Relationship Between Mechanics of Environmental Cracking and Design Criteria,” ICM3, 2, Cambridge, England.

Ford, F. P. and Emigh, P. W., 1985, “The Prediction of the Maximum Corrosion Fatigue Crack Propagation Rate in the Low Alloy Steel De-Oxygenated Water System at 288°C,” *Corrosion Science*, *25*, (8/9), pp. 673-692.

Gandy, D., et al., 1997, “State-of-the-Art Weld Repair Technology for Rotating Components,” Final Report TR-107021 V1&V2, EPRI, Palo Alto, California.

Hickling, J. and Blind, D., 1986, “Strain-Induced Corrosion Cracking of Low-Alloy Steels in LWR Systems—Case Histories and Identification of Conditions Leading to Susceptibility,” *Nuclear Engineering and Design*, *91*, pp. 305-330.

Höxtermann, E., December 1979, “Blade Damages in Steam Turbines: Evaluation of VGB Statistics from 1973 to 1977 and Basic Comments,” *VGB Kraftwekstechnik 59*, (12).

Kilroy, R. E., Jr., Hershey, T. G., Nowak, E. J., Alice, J. A., and Peterson, R., 1997, “A 12% Chrome Weld Repair Increases Stress Corrosion Cracking Resistance of LP Finger Type Rotor Dovetails,” Proceedings of the International Joint Power Generation Conference: Volume 2: Power, International Joint Power Generation Conference, Denver, Colorado, PWR-Vol. 32, American Society of Mechanical Engineers, New York, New York, pp. 243-250.

McCloskey, T. H., et al., 1997, “Risk Assessment of LP Turbine Dovetails—Probabilistic Modeling of Stress Corrosion Cracking,” International Joint Power Generation Conference, Denver, Colorado.

Omprakash, V., Lam, T. C. T., Gruwell, D., and McCloskey, T. H., 1994, “Life Extension Strategies of Cracked Disk Attachment for Low Pressure Steam Turbines,” Advances in Steam Turbine Technology for the Power Generation Industry, 1994 International Joint Power Generation Conference, Phoenix, Arizona, PWR-Vol. 26, American Society of Mechanical Engineers, New York, New York, pp. 159-164.

Sarlashkar, A.V. and Lam, T. C. T., 1996, “Blade Root Attachment Evaluation Low-Cycle Fatigue Estimates Based on Probabilistic Approach,” Proceedings of the 1996 International Joint Power Generation Conference, Volume 2: Power, American Society of Mechanical Engineers, New York, New York, pp. 511-516.

Speidel, M. O. and Magdowski, R., 1997, “Major Influences on the Growth Rates of Stress Corrosion Cracks in Steam Turbine Rotor and Blade Materials,” Proceedings: Steam Turbine Stress Corrosion Workshop, Final Report TR-108982, EPRI, Palo Alto, California, pp. 157-165.

Stodola, A. and Lowenstein, L., 1945, *Steam and Gas Turbines*, 1927, New York, New York: McGraw Hill Company, Reprinted 1945, New York, New York: Peter Smith.

Suhr, R. W., 1983, “Corrosion Fatigue of a 12CrNiMo Alloy in Steam and the Effect of Oxygen Content,” 1981 Workshop Proceedings, Palo Alto, California, *Corrosion Fatigue of Steam Turbine Blade Materials*, New York, New York: Pergamon Press, pp. 4-99 - 4-110.