INTRODUCTION.

Turbine engineers and designers have made remarkable improvements in the efficiency and reliability of industrial steam turbines over the last 30 years. Remarkable improvements have been achieved for products that already had over 100 years of technical development behind them. For most of those first 100 years, the analysis of turbine blades had concentrated on the behavior of individual blades. A key change, and one of the most significant advances in turbine reliability, was the development and application of analytical techniques that make it possible to characterize and explain the behavior not simply of individual turbine blades, but of entire bladed disk assemblies.

Advancements in modal analysis and testing, fatigue analysis, creep analysis, fracture mechanics, aerodynamic theories, and the development of many new materials and manufacturing processes cleared the path for the design of more powerful, more efficient, and more reliable turbines. It became evident that design of blades is a multidiscipline activity. For a proper reliability assessment of a design, one needs to understand many fields of science and these must be applied as need be. These advancements helped designers to extend the capabilities of designs beyond past experience. This also helped to explain past successes and failures of components.

The simultaneous development of powerful and inexpensive computers has made it practical to quickly and efficiently carry out the calculations necessary to apply these advanced analytical techniques to the routine design of new and replacement blades and rotors for industrial steam turbines. Nowhere have these advances had a greater influence than on the design of critical service process compressor drives for the refining and petrochemical industries. Large drivers for ethylene and LNG processes exceeding 75 MW in power are in successful service. Older designs using double-flow exhausts with short, but very strong, blades have been supplanted in newer designs by single-flow exhausts with taller, but more reliable and aerodynamically sophisticated, stages. Inlet pressure and temperatures of 2000 psig/1000°F (140 barg/540°C) have become almost common in new process drive applications.

The purpose of this book is to introduce these advances in a concise volume and provide an easy-to-understand reference for practicing engineers who are involved in the design, specification, and evaluation of industrial steam turbines in general, and critical process compressor drivers in particular. This text has also attempted to present a unified view of concepts and techniques needed in the understanding of blade design. It includes some advanced concepts such as life estimation. One chapter is dedicated in introducing the reader and designers to the effect of uncertainty of input variables on the reliability of the design. Probabilistic-type analysis is introduced for reliability estimation, as it is said that every design decision has some risk associated with it and risk may be managed if it is known.

We would like to thank each person and the many industries whose works have been referenced in the book. We also take this opportunity to apologize to those whose

work might not have been referenced by mistake. Thanks to the many associates during our employment and consulting work whose thoughts guided the selection of many materials. We hope these will help readers in their work or at least make them think. Last but not the least, many thanks to Seema Singh for reading the manuscript word by word and making numerous suggestions for changes that made the work better and more readable.

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Importance of Blades in Steam Turbines

Structural integrity of all rotating components is the key for successful operation of any turbomachinery. This integrity depends on the successful resistance of the machine parts to the steady and alternating stresses imposed on them. The challenge with rotating equipment, such as turbomachinery, is often more severe due to the significance of the alternating loads that must be carried to satisfy their purpose.

One of the major classes of rotating machinery is the mechanical drive steam turbines, i.e., steam turbines that drive pumps and/or compressors. These steam turbines are differentiated from those that drive generators in that they operate at variable speeds. Steam turbines may operate from 1 to 5 hp up to several hundred thousand horsepower; they may operate with steam that ranges from vacuum to thousands of pounds per square inch; and blade tip speeds can exceed the force of the most severe hurricane (a large, last blade row with an 8-ft tip to tip diameter operating at 3600 rpm will experience tip speeds in excess of 1000 mph).

One of the causes of blade deterioration is static stress which is primarily the result of steam bending and centrifugal loads. Alternating stresses are imposed due to the vibration of the parts in question, e.g., blades and disks. If the combined loads become too large, vibration-induced fatigue of the rotor blades or disks is a major concern. In addition to the imposed loads, these forces are subject to resonant amplification caused by coincidence with natural frequencies. To put the scope of this problem into context, one must realize that there may be thousands of blades in a steam turbine. For example, there may be 10 to 20 rows of different blade designs with the possibility of each blade row having different dynamic characteristics.

Steam turbines have been in operation for more than 100 years and have always faced this problem. As may be imagined, the technology in engineering and physics to support these designs has grown dramatically over that time; tools have been enhanced and technological developments incorporated.

Brief Historical Perspective of Technological Development

The current state of design, as represented in the API standards for this class of machinery, sets a life of 30 years for all components. In many cases, this translates into a design requirement for infinite life and may exceed the needs of a specific installation. This requirement may be driven by the actual desire for infinite life, limitations in analysis techniques, tools that have existed over the years, and/or an incomplete understanding of the tools that have appeared in the recent past and are currently fully or partially available.

A common cause of vibration-related failure in steam turbine blading is resonant excitation of the blading occurring at an integral order, i.e., multiples of the rotational speed, nozzle passing frequency, and multiples thereof. The associated mode of failure is high cycle fatigue. A primary feature of resonant excitation is that dynamic stress amplitudes rise as the exciting frequency approaches the resonant speed and the response decreases after passing through the resonant speed. Hence, it is necessary to identify resonant frequencies of the system.

It is impossible to include all the work done by the numerous researchers and designers of steam turbine blades. Effort is made to include some of those that describe the progress and current methodology for steam turbine blade design. Many textbooks were published on steam turbines during the last century together with many technical publications dealing with all aspects of turbine design, specifically blade design. Early publications by Stodola (1905) and Kearton (1922) are worth mentioning because these two books are credited with setting the stage for detailed vibration and reliability analysis for blades. In many different ways designers followed the processes and methods outlined in these books. As the turbine design matured and manufacturers gained experience, methods were adjusted to include new technical methods and lessons learned from field experience and each manufacturer has evolved its own methods and criteria to achieve successful design. Hence, methods and criteria should not be expected to be consistent across manufacturers.

Blade design has evolved from the analysis of spring-mass systems to a single cantilever beam to a band of blades to a bladed disk. In addition, steam turbines have included bands of blades on a disk as a system. Throughout the years many effects of turbine speed to increase blade frequency were found, and it gave rise to the term *centrifugal stiffening*. Campbell (1925), while examining the failure (bursting) of disks, concluded that blades were broken due to axial vibration. This publication reported the results of an investigation conducted at General Electric to understand the wheel failures, mostly in wheels of large diameter, that could not be explained on the basis of high stress alone. About this time certain types of vibrations of standing waves were investigated by means of sand pictures. This test was conducted by scattering sand over the wheel surface. Wheels were then excited by means of a magnet exciter, and the turbine wheel was placed in a horizontal position.

An electromagnet was clamped with its poles close to the edge of the wheel, alternating current was passed through the coils of the magnet, and a series of pulls was exerted on the wheel. This resulted in deflecting the wheel in a transverse direction to the plane of the wheel. A variable-speed direct-current (dc) motor was used to drive the alternating-current (ac) generator and allowed the frequency of the pull of the magnet to be varied over a wide range. Frequency of excitation was varied until a sand pattern on the wheel appeared, and sand accumulated mostly in a radial line or pattern. When the frequency changed to some higher magnitude, a different sand pattern appeared on the wheel. These radial lines represented the location where the velocity of vibration was zero. The number of radial lines was always observed to be of an even number. These patterns, are known as nodal patterns, and two lines are taken as one diameter. It is now understood that the opposite radial lines might not be 180° apart. Frequencies at which these patterns are observed coincide with the natural frequency of the wheel in axial vibration associated with the mode shape represented by the sand pattern.

Figure 1 shows a picture of such a sand pattern. It is noticeable that sand has collected on certain portions of the wheel, and it forms a pattern showing four radial lines. This pattern is referred to as two nodal diameters mode. There are six radial lines in the pattern shown in Fig. 2. These modes are called three nodal diameters mode. Note that the radial lines pass through the balancing holes in the left picture while in the picture on right side these lines pass between the balancing holes.

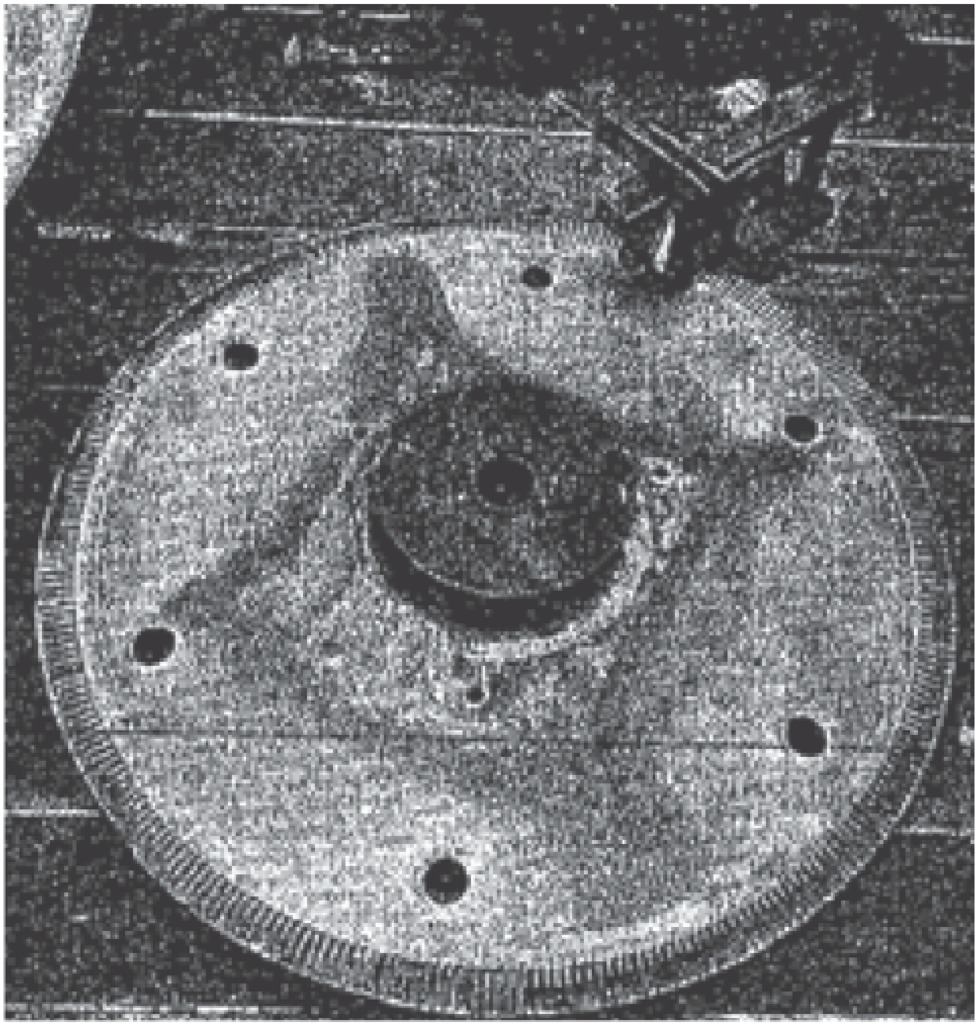

Figure 1. Four radial lines, two nodal diameters mode (Campbell, 1925).

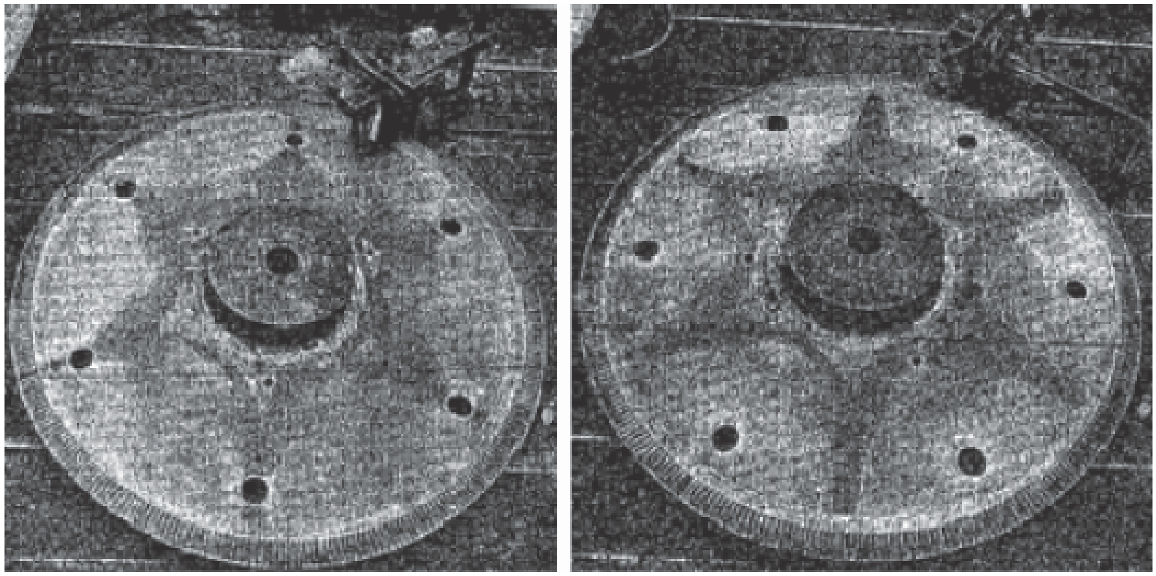

Figure 2. Six radial lines, three nodal diameters modes (Campbell, 1925).

Over time blades needed to be taller to accommodate the requirement of increasing power. This necessitated the blades to be joined together by a band of metal either at the tip or somewhere along the length of the blade. Kroon (1934) described a method to evaluate the effectiveness of such construction to reduce the dynamic response of the design under steam forces. Allen (1940) described design practices of blades in high-pressure and high-temperature stages. A detailed explanation for partial stage admission was included, as was one for full admission stage. Allen recommended limiting the number of blades per group to two for high-temperature service and argued that more blades in a group for high-temperature application tend to set up high stress. Two types of root attachment (axial entry vs. tangential entry) construction were explored, and the choice is dependent on the application, e.g., speed, power, and temperature. The effect of shrouding might be considerable for high-pressure blades.

A reduction of 25 to 60 percent in bending stress may be achieved. The natural frequency for high and intermediate stage blades should be more than 5 times the running speed of the turbine. For low-pressure stage blades the natural frequency more than 4 times might be sufficient. Trumpler and Owens (1955) provided a detailed discussion of factors that affect the strength of the blade for full admission stage. Damping was measured during a test on partial admission stage. Values of logarithmic decrement varied from 0.02 to 0.06 for 12 to 13 percent chromium steel, and the approximate material damping of these materials is 0.02. The effects of fixity due to centrifugal force, surface finish between mating surfaces, geometric tolerance, and the length of operating time were not explored, however.

Problems of blade damage were encountered during World War II. At this time advances had been made in computers to handle large calculations. Weaver and

Prohl (1956) demonstrated that blades are banded together, and blades in the band behave differently than single blades. Results showed that there were more frequencies and mode shapes of banded construction compared to what the analysis of a single blade provided. Large numerical calculations were performed for such a construction. The response of blades under excitation due to flow from nozzle, the natural frequencies of the banded blades, and the associated mode shapes were evaluated. In a companion paper Prohl (1956) described the numerical method and provided the equations that were used to estimate frequencies, mode shapes, and dynamic response of the banded blades. The basic beam equations for blades were developed where blades were coupled together through a band at the tip of the blades. To simplify the analysis and to make the results tractable, three key assumptions were made:

1. Series of identical blades were assumed to be parallel to one another, and they were uniformly spaced.

2. Principal axes of blade cross sections were assumed to be parallel to the tangential and axial direction of the rotor.

3. The shear center of the cross section was assumed to coincide with the center of gravity of the cross section.

It can be said that the above assumptions do not relate to the actual construction of bladed disks for steam turbines. For example, directions of blades' principal moments of inertia do not coincide with the tangential and axial directions of the rotor at least for reaction-type blades or exotic taper and twisted blades. However, they do coincide for impulse-type blades, which are not parallel to one another, but each is positioned in a radial direction on the disk. Spacing between blades changes from the base to the tip of the blade; however, the angle between them is equal. Shear center and center of gravity become coincident only for a circular cross section. Now with the development of finite element analysis (FEA), one does not have to make these assumptions. FEA made it possible to analyze the banded construction rather than a single blade for correct frequency and dynamic response determination. Notwithstanding these assumptions, results of this work provided the future direction for analysis. This was a milestone in the analytical development of blade vibration and the decision-making process for reliability.

Equations to estimate dynamic stress at the base of the blade due to loading from nozzles were provided. The static steam bending stress gets magnified during resonance. An assumption was made that energy supplied to the vibrating blade group by the exciting force is completely dissipated in damping.

$\sigma_v = K(\pi/\delta)S\sigma_b$

- where σ_v = resonant vibration stress at blade root

- K = resonant response factor

- δ = logarithmic decrement of damping
- $K(\pi/\delta)$ = amplification factor
- S = stimulus (always less than unity)
- σ_b = bending stress at root due to steam loading
- $S\sigma_b$ = exciting stress

It was assumed that the stimulus is uniformly distributed along the length of the blade and that the phase between stimulus and blade motion is constant along the length of the blade. Figure 3 summarizes the results of the analysis for tangential vibrations for seven modes. The first six modes are considered to belong to the first bending of a single blade, and the difference among them is the phasing among blades. The seventh mode is the second tangential mode.

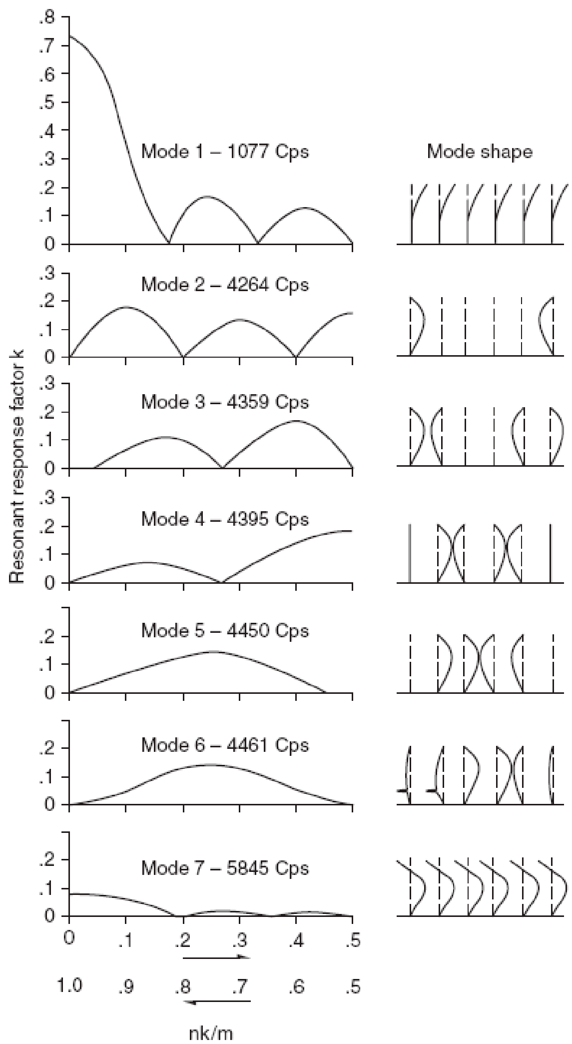

Figure 3. Resonant response factor (Weaver and Prohl, 1956).

The analytical result for the dynamic response of blades was shown to be a function of (1) the number of harmonic excitation, (2) number of nozzle openings, and (3) number of blades per 360°. The shapes of these curves, which resemble that created by a bouncing ball, are functions of mode shapes of the blade group. These mode shapes can be seen to differ from one another by the difference in their phasing of blades and the results correlated with field data and test data from laboratory study.

The result of this work created the need to analyze the blades as a group rather than as a single blade. The dynamic response of the group is due to the coupling between blades through the band. Frequency is a function of the relative stiffness of the band and the blade. The review of Fig. 3 indicates that with a proper selection of nozzle and blade pitching, it may be possible to considerably reduce the resonant stress even though operating precisely at the speed required to excite the natural frequency.

The next technological development was to include the disk in the analysis. It was recognized that a bladed disk, is a system and coupling between blades will be through the disk, and the consideration of the stiffness of the disk became important. In the tangential vibration of blades, stiffness of the disk may be considered very high, but it will be very small if there is any coupling through the disk. However, in the axial vibration, disk stiffness becomes a large contributor.

Including geometric variations among blades was the next logical advancement in the analysis of the dynamic response of blades. It has been shown that geometric variation influences mode shapes and frequencies and, in turn, the response of a bladed disk system. In a tuned system where each blade is identical, modes in general occur in duplicate. There are two modes that differ by a phase angle, but these modes have identical frequencies. However, when symmetry is disturbed through variation from blade to blade, these modes tend to split in two frequencies. Also, the shape of these modes gets distorted from the tuned case, which is a phenomenon called *mistuning*. Ewins (1969, 1970, 1973, 1976) has extensive discussions of this phenomenon, and the response of the mistuned case was found to be different from that of the tuned case. This was attributed mainly to a change in the mode shape and may pose a serious decision point for designers to account for variability among blades. Ewins (1976) specifically dealt with completely shrouded or unshrouded bladed disk constructions.

The next stage of advancement was made in the understanding and analysis of disks containing packets of blades. The blades of steam turbines more often are banded together in a packet. Singh (1982, 1988, and 1989) studied the dynamic behavior of packeted bladed disk construction. Figure 4 shows a comparison of a SAFE (Singh's advanced frequency evaluation) diagram for different types of construction for the same number of total blades mounted on the same disk. The vibration characteristic of packeted bladed disks is similar to that of the completely bladed disk, but it has some special features. This method of analysis is the focus of a large part of this book, and it is discussed in the appropriate sections.

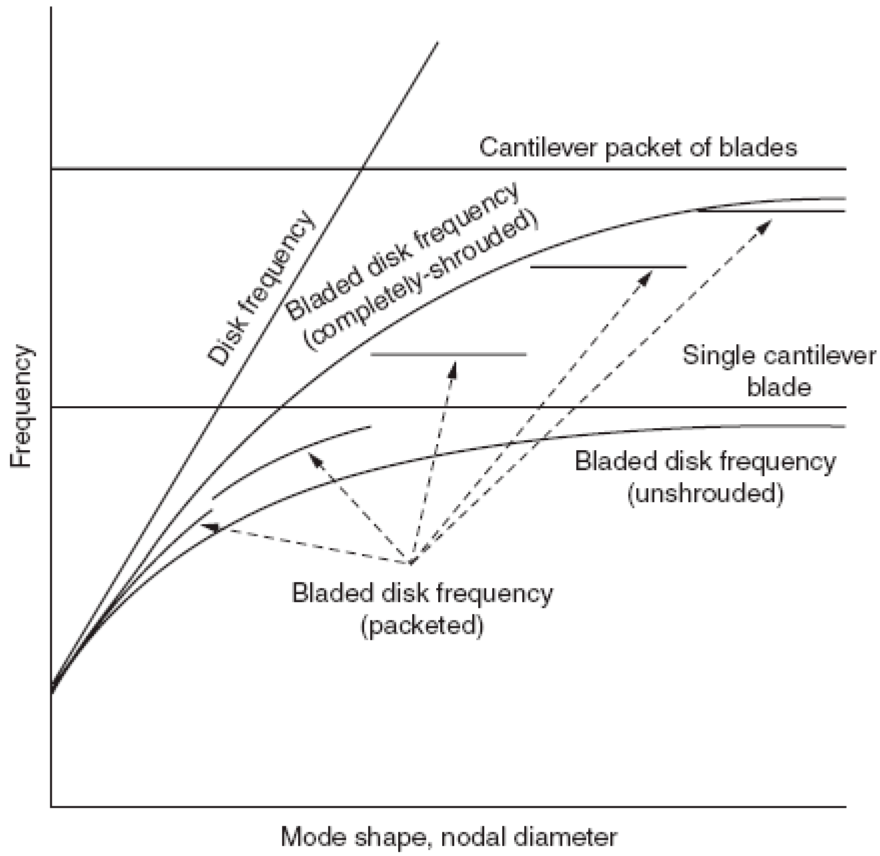

Figure 4. SAFE diagram for completely shrouded, unshrouded, and packeted bladed disk (Singh and Drosjack, 2008).

Reliability evaluation of a mechanical component becomes a multidiscipline activity and has greater relevance in the design of steam turbine blades because of the blades' contribution to the reliability of the turbine. In addition, many technological developments have been made throughout the years for the design of gas turbine blades. Many of these analysis methods are applicable to steam turbine blades and should be used.

To calculate the resonance characteristics of blades, one needs to deal with the following key issues, which are not all-inclusive:

- Unsteady aerodynamics over a wide variety of flow conditions that may exist within the operating range
- Structural vibration characteristics, i.e., frequencies and associated mode shapes within the operating range
- Damping in the system (structural as well as aerodynamic)
- Estimation of material properties (fatigue properties, yield strength, ultimate strength, modulus of elasticity, etc.) at the temperature in the operating range

- Manufacturing and quality assurance processes

Each of these issues may require a separate technical discipline to support an evaluation. In the last few decades, significant advances have been made in several of these issues, even though additional analysis and test development are still needed. Utilization of the current state-of-the-art technologies and tools will provide a better evaluation of the issues, resulting in more accurate and more optimal designs.

Unsteady aerodynamic analysis can produce two necessary pieces of data: aerodynamic damping and the magnitude of the pressure acting on the blade (pressure profile due to flow variation). During each revolution, every blade experiences a variation in pressure or force and will experience the same force in the subsequent revolutions. Each blade experiences a periodic force with a time lag with respect to its neighbor. These aerodynamic forces depend on blade geometry (profile, stagger angle, angle of incidence), cascade solidity, and flow conditions (subsonic, transonic, and supersonic). In case of shock or flow separation, aerodynamic analysis becomes complex, and so the unsteady force is estimated.

The blade structural dynamic analysis must consider an assembly of blades and the disk as a system. The characterization includes natural frequencies in the operating range, associated mode shapes, and damping, which are the required input for estimating the forced response of the bladed disk assembly. Since the blades vibrate in a flow medium, an interdisciplinary approach that includes structural dynamics and unsteady aerodynamic analysis is necessary.

The amplitude of the dynamic stress is proportional to the damping. Damping is provided by a material's internal resistance as well as by the flow medium (aerodynamic damping). More is known about material damping, but aerodynamic damping is dependent on the characteristics of the flow. For example, a blade will experience greater resistance to vibration in a dense gas than in a less dense gas. In addition, a taller blade will experience greater damping due to large amplitude of vibration than a shorter blade having a small amplitude of vibration; i.e., damping can be wildly nonlinear.

Many new materials are now available that have been used in blade manufacturing. The advent of many superalloys has allowed blade design for higher loads. Many materials are suited for different environmental conditions, e.g., erosive and corrosive environments. Now materials are available that can be used for higher speed, higher steam pressure, and higher temperature. Some materials provide better fatigue properties while others provide better creep properties while still others provide better resistance to crack propagation. The choice of material depends on the condition and type of operational loads being imposed on the blade. This has allowed designers to balance the properties for a given application to achieve higher reliability. Less dense material, such as titanium alloys, has allowed blades to be designed for higher speeds that might not have been possible in the past. Many types of surface treatments, e.g., coatings, shot peen, and laser peen enabled the achievement of increased blade life, even in some adverse conditions.

Current and former successful results have been achieved by applying experience-based rules and limits, which can cause designs that may seem to run out of space (the limits set design boundaries that may not be analytically sound). The assessment of the useful blade life is discussed with respect to damage from high cycle fatigue and may impose overdesign requirements that may damage the economic viability of equipment designs cascading to projects. As there is always a risk associated with any design decision, a probabilistic concept will be presented that permits a quantification of the risk of a proposed design. The potential to apply probabilistic analysis to extend design capabilities is included.