Progress With the Solution of Vibration Problems of Steam Turbine Blades

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This Keynote paper discusses several vibration problems of steam turbine blades where the solution in each instance led to an improvement to the blade technology of that time. The sum of these solutions represents the state-of-art of today’s blading technology. The purpose of the paper is to collate this technology and its major results, and to present these results in summary form for future guidance of practicing analysts and blade designers. A guide such as this does not appear to exist in the current subject literature. The causes of a number of problems are identified and some successful solutions for problems are presented. A recent blade problem that was solved by use of modern technology and the guidelines presented is described.

1. Introduction

Excellent textbooks on steam turbines and blading such as Stodola [1] (1st edition 1904, US edition 1927) and Kearton [2] (1st edition 1922, last edition 1958) have been available for many years. Neither of these texts addresses modern blade vibration problems such as forced resonant response, life analysis, or blade flutter in any detail. This paper therefore contains a brief but current overview of modern blading problems and of the technology which has developed over the years for addressing such problems. Several present day problems requiring attention by designers, and solution by researchers, are identified.

The paper contains a discussion of several past problems and their solutions. These problems were selected because they led to advances in the blade technology of their time. Important lessons learned from these cases are summarized as conclusions at the end of the paper. Comments are made on current problems which await solution. The paper does not seek to be all-inclusive because of the vast scope of the subject technology which has developed over the past 80 or more years. Some important contributions may have been omitted due to size limitations. However, it is believed that those papers which have been mentioned will serve as useful guideposts for this subject.

2. Early Blading Problems and Solutions

The first reaction blades used by Parsons [3] were short brass cantilevers inserted around the circumference of a drum rotor. Up to 20 rows of such blades were used, with matching stationary blades. An early Parsons turbine from 1891 is shown in Figure 1. The impulse blade sections sketched by DeLaval [4] in Figure 2 were shaped to redirect the steam flow after the converging nozzle had accelerated this flow to a high velocity. Parson’s turbines and their Westinghouse counterparts grew quickly in output between 1891 and 1900. This growth was soon followed by that of machines from other manufacturers around the world. By 1920, turbines of 20 to 30 MW output were considered ‘huge’ machines.

The first serious blading problems appear to have developed around 1917. These were associated with the “bursting” of bladed turbine disks during operation. By that time, blade shapes had evolved from the relatively short blades of the 1880’s to longer more slender profiles of more or less constant section, around the circumference of a thin disk: see Figure 3. This slender blade and disk design led to the bursting failure problem discussed by Campbell [5]. As a result of Campbell’s work on bladed disk oscillations, and his explanation that the disk failures resulted from resonance between backward travelling waves and a standing pressure disturbance in the steam flow,
this type of bladed disk design was revised in favor of thicker disks and blades of greater axial width. Figure 4 shows the presentation of Campbell’s famous diagram.

The earliest type of blade vibration analysis procedure appears to have involved use of the simple single-degree frequency formula cited by Stodola [1]. An allowance for the stiffening effect of centrifugal force was later introduced using Lamb and Southwell’s [6] method, though this procedure appears to have been known earlier, judging by the dates involved. This “allowance” method is mentioned both by Campbell [5] and by Stodola [1] who gave a theoretical explanation for its validity. Apart from the design comments in Stodola’s book, Campbell’s paper apparently represents the first discussion of blading problems in the literature. The emphasis of Campbell’s paper was however on explaining the disk “bursting” problems that he encountered between 1917-1921. The same axial vibrations of the disk also caused the broken blades shown by Campbell in Figure 3.

Interest in blading revived in the mid-1930’s as machine output continue to increase and blade lengths grew longer. Kroon (1934) [7] described studies of the strength properties of lashing wires, in which Kroon’s calculated results were confirmed by measurements of deflections of laboratory beam models. In 1940, Kroon [8] performed a further series of experiments on first stage blades in a turbine operating under high temperature steam inlet
conditions. He used an optical method and a mirror system within the disk and rotor bore to observe blade vibrations as the turbine operated. Other damaging vibrations of blade groups were studied by Smith [9] in 1936-37 (but not reported until 1948), by which time similar problems had been encountered in the U.S. by Prohl. Studies were also made during the years 1941-1945 into the vibrations and blade failures encountered in the development of jet engine blades, in the U.K., in the U.S., and probably (unreported) in Germany: see Shannon [10].

3. Numerical Analysis of Blade Groups

Numerical analysis of blade vibrations became possible as a result of advances in machine computation made during World War II. Numerical analysis of blades began with the 1944 publication of Myklestad’s [11] method for calculating the vibrations of aircraft propellers. This method was further developed by Mendelsohn and Gendler [12] and others. An early (1951) adaptation of this method to steam turbine blade analysis was made by Jarrett and Warner [13], to investigate the natural frequencies of a last-stage steam turbine blade. The next major development was a finite difference scheme for calculation of blade group natural frequencies. Prohl’s [14] study with Weaver [15] in 1958 used finite difference equations for tangential motions for blades of symmetrical section, Figure 5. Axial motions were also considered by the same procedure, but separately for convenience, using modal separation. The paper with Weaver introduced the so-called “bouncing-ball” curves, shown in Figure 6, depicting the degree of modal detuning which occurs in accordance with the phase relation between the exciting frequency and the blade group natural frequencies.

Prohl confirmed his theoretical results using considerable data from field experience and laboratory studies of nozzle wake excitation. These problems were similar to the problem studied by Smith in 1936-37. Studies of this type were also made by other steam turbine manufacturers around the same time: See Rosard [16].

The frequency calculation of L-0R blades is complicated by the fact that these blades are tapered and twisted in shape. Often such blades are grouped together in packets of three to six blades, or today in a continuous ring. Such blades are mounted on the rim of a disk which also has a non-uniform radial section. Bending displacements between the tangential and axial directions are coupled by the stiffness properties of the airfoil shape and their radial variation. Further coupling results between bending and torsional motions of the blade group from blade-to-blade coupling due to the tiewires.
Coupled axial-tangential twist motions of last stage blade groups were first reported by Deak and Baird [17] in 1963. Bending-bending-torsion (BBT) vibrations of tapered-twisted blades with tip covers and tiewires in groups were again studied using the finite difference approach. Although this model also allowed for BBT root flexibility, true dynamic, i.e., mass-elastic flexibility of the disk was not included. It was beyond the ability of computers of that time to consider both blade group dynamics and disk dynamics simultaneously. Nevertheless, the efficient coding and the design improvements which this procedure introduced were major achievements for analysis of blade groups with tangential-axial-torsional coupling.

Deak and Baird started with the basic equations of flexural torsional motion. Assuming harmonic motions the authors reduced these expressions to 10 first order differential equations. Bending of the blade regions between the tiewires was then represented by a power series in terms of the moments and shears. This step allowed the differential equations to be replaced by difference equations. The resulting equations were then assembled in matrix form, and their determinant was solved by iteration for the natural frequencies of the blade group.

As an example, Deak and Baird obtained the mode shapes and natural frequencies of a 26-inch four-blade group operating at 3600 rpm, Figure 7. These results were plotted on a Campbell diagram, and compared with rotating rig test results obtained for such a blade group at speeds up to 3600 rpm, Figure 8. The Deak and Baird paper contains details of the difference procedure used. The correlation shown between results is impressive, even though disk flexibility has been omitted. LaRosa [17, discussion] of Westinghouse commented on the importance of disk axial flexibility, especially regarding the second mode. Other important factors are; how the attachment flexibility is considered (by hinge-factors in the paper), both at the blade-disk interface, and at the blade-tiewire interface. In those days no Interference diagram was in use for diagnosis. This paper continued the trend toward computerized blading design. The procedure described gave designers a valuable new tool for prediction of last-stage blade properties.

Additional complexities of this problem have been studied by subsequent authors who investigated the vibration properties of blade groups on a flexible disk of non-uniform cross section [18]. Many subsequent finite difference studies of blades and groups have since been made. These studies introduced simplifying refinements and greater efficiency and precision into calculations for higher modes [19] [20].

Finite element procedures were first applied to blade analysis around 1965 by Documaci et al. [21] and by Ahmed et al. [22], who used 20 node shell elements to find the first two modes of a free standing gas turbine blade. The natural frequency results were within 1 percent of experimental values for the first mode, but were inaccurate for higher modes. The effect of centrifugal stiffening was not considered. Several element types (bar, solid, shell) were proposed between 1965 and 1975 for blading analysis. Lalanne and Trompette [23] first used solid elements to study the steady stresses and dynamics of short gas turbine blades in groups. They compared their results with test data for the first three non-rotating modes. Rieger and Nowak [24] in 1976 used the 8 node 3D solid element in ANSYS to model the natural frequencies and steady stresses of a steam turbine 6-blade I.P. group on a segment of disk. These studies coincided with the appearance of several major commercial finite element codes, such as NASTRAN in 1968, and ANSYS in 1970. Several major practical blade problems occurred between 1968 and 1980, such as the Q.E.2 blade failures [25] (1968), development problems with blades in U.S. Navy turbines in the
1970’s, and problems in the LP rows of large utility turbines in the US during the 1970’s and 1980’s. These problems spurred the development and application of finite element methods to blading analysis. The circumstances of the utility blade problems are discussed by Dewey and Rieger [26].

Finite element blade analysis procedures have since developed more extensively than finite difference procedures because of the greater ease with which the blade and disk geometry can be modeled (and visualized), and the relative ease with which both linear and non-linear (geometrical and material) problems can be addressed. The time/cost advantage once offered by finite difference blade computation has ceased to be significant since powerful P.C.’s have become available. Steam turbine manufacturers and large electric utilities today use codes such as ANSYS and/or BLADE-ST in addition to their in-house tools for design and analysis of steam turbine blades.

4. BLADE-ST Computer Code

The earliest calculations with BLADE-ST [27] were performed around 1983, and the first commercial version of this code become available in 1986. Development of BLADE-ST was important because it was the first program to allow all needed calculations for blade modeling, stress, and life analysis to be made within a single code. Any grouping of blades, and any known attachment type and disk profile can readily be modeled, starting with a menu of common geometries. An entire blade-disk segment model can be created interactively from relatively input few dimensions, in 8 hours or less. Steady stresses for any blade in the group are then found at zero rpm, and at the required rpm, with either zero attachment tolerance mismatch, or with a specified 3D distribution of attachment tolerances. Similarly, all common cover types and numbers of tenons/blade can be represented, with the required tightness of tenon fit. A 360° blade disk model can next be formed on command from the segment model, which itself was created from a single blade disk sector. A special subroutine further allows any structural detail not considered by BLADE-ST to be addressed in ANSYS, and then incorporated back into the BLADE-ST model by direct transfer between the two codes.

BLADE-ST calculates steady stresses, natural frequencies, (animated) mode shapes, and dynamic stresses with 3D viewing upon command, for specified material properties, Rayleigh damping properties, and stimulus ratios. It also contains life prediction routines which employ the Local Strain method for crack initiation life, and Fracture Mechanics methods for crack propagation life. Complex load spectra are addressed using Miner’s rule. An advanced subroutine known as ALLEGRO [28] has recently been developed to consider cases where load time sequencing may be important for life estimation.

The multi-function concept of BLADE-ST is not unique. Other codes have been developed in recent years to address the needs to some degree similar to those which BLADE-ST serves, although BLADE-ST is probably the most comprehensive and user friendly code created for general purpose analysis of blading. General purpose codes for blade analysis have been written in the UK (Rolls-Royce), in Russia (1998), in France (EDF, Paris 1985) and in Germany (by Irretier 1991).

ANSYS can perform most of the tasks which BLADE-ST can address, i.e., linear and non-linear material analysis of blades, bladed disks, and blade groups, for the calculation of steady stresses, natural frequencies and mode shapes, and dynamic stresses. The solver routines are identical to those in BLADE-ST, and so the results are identical for identical loads, modes, and input data, as Lam and Rieger [29] have demonstrated. However, ANSYS does not have the same specialized generation routines for creation of a blade disk model which are in BLADE, nor does it possess the blade life estimation subroutine (Local-Strain LCF/HCF and fracture mechanics) which are in BLADE-ST/NT, with databases. The general modeling capability of ANSYS is of value in instances where geometrical complexities beyond the ability of the specialized generation routines in BLADE must be considered.

Maturation of blade technology has now reached the point where specific general purpose codes such as BLADE-ST have been developed, tested and thoroughly verified. BLADE-ST, for one, has now been put into practice at more than 28 locations (2002) worldwide. This shows the widespread need for an in-depth blading analysis tool for design and operational use in today’s turbomachinery industry. BLADE-ST is now being used routinely for design, problem analysis, maintenance planning, and for blade performance monitoring by power utilities and turbine manufacturers.
5. Field Demonstration of BLADE Technology

Recent blade-related failures at a station near Wallerawang, N.S.W., Australia led to application of both the experimental and analytical technology described in this paper. This work has been described by Hesler and Marshall [30]. Recurring cracking of buttstrap connections, shown in Figure 9, between the 12 and 13 blade groups of L-2 blades led to an extended investigation to determine the failure cause and to seek a remedy. A model of the 11.5 in blade was prepared using BLADE-ST, and a 360° bladed disk model was developed (with help from ANSYS in modeling the extended tenon and buttstrap assembly details). Free sliding contact between the buttstraps and tenons was assumed. Steady stresses in the buttstrap and in the tenons were calculated, including centrifugal growth at 3000 rpm, and full steam load. The corresponding buttstrap and tenon stresses depended on the degree to which the actual clearance agreed with the prescribed clearances. This is always an issue with field repaired tenons. Steady stresses, natural frequencies and modes, and dynamic stresses of the bladed-disk assembly were calculated. It was decided to calibrate the model for correlation of the natural frequencies using test data.

A program of strain gage tests was planned involving measurement of buttstrap steady strains with two buttstrap gages. Dynamic strains were also measured using eight dynamic strain gages located as shown in Figure 10. Two additional data channels were allocated for dynamic pressure sensors, which were placed at the leading edge of the rotating blades. After studying blade modes predicted by the finite element method the gages and pressure sensors were distributed in a randomized pattern around the disk, to ensure that all blade modes would be readily identified.

![Figure 9. Broken Buttstrap. Hesler & Marshall [30].](image1)

![Figure 10. Distribution of Strain Gages around L-2 Blade Row. Hesler & Marshall [30].](image2)

With the cooperation of the turbine manufacturer, and with supporting finite element data, it was decided to ‘snake’ the sensor wires down the blade surfaces, across the attachment region and down the disk face, and into a radial hole drilled in the shaft down to the rotor bore. The wires were then routed along the bore and out through another radial hole near the LP drive coupling. Radio telemetry transmitters mounted in the coupling bolt holes transferred the strain signals to the receiving antenna. This arrangement is shown in Figure 11. The sensor wires were protected against droplet erosion and moisture by an epoxy coating cured in-place on to prepared surfaces, armored with stainless steel cover strips over the wires, which were spot welded into place along the blades, attachment and disk surfaces.

The test results showed that measured static strains in the selected buttstrap locations were of the order of 350 microstrain while the measured dynamic strains were ±85 microstrain. Strains at the failure site would have been considerably higher due to stress concentrations. The corresponding strain gage natural frequency spectrum is shown in Figure 12. It is seen that not only is the strain response very strong at the 300 Hz, 6x harmonic, but the natural frequency of the 6 diameter mode occurs adjacent to this harmonic at 298.6 Hz. This is confirmed by the Interference diagram Figure 13 which shows that the 6x resonance was the cause. It is therefore evident that one significant cause of the buttstrap failures was the prevailing near-resonant operating condition between the strong 6x harmonic and the 6 diameter mode natural frequency at 298.6 Hz. Another contributing factor may have been the high stress concentrations at the buttstrap corners. With the near-resonant dynamic stresses, these stress...
concentrations appear to have caused buttstrap cracking to occur at these locations. Buttstrap dynamic stresses are shown in Figure 14. A third factor may have been weakening of the buttstrap material by “salt-zone” corrosion from the nearby Wilson line conditions.

Because the natural frequency of 6 diameter mode occurred somewhat below the 6th excitation harmonic it was decided as part of the fix to downwardly detune this blade disk mode, to 291 Hz, by the addition of a small mass to each blade tip situated beneath the coverband. The size of the required mass was obtained from a calibrated BLADE-ST calculation, and the required modifications were designed and made. A re-test using the same strain gage telemetry was performed, as most of the basic equipment was still in good working order. This confirmed that the detuned blade 6 diameter mode now occurred at 291 Hz, and with much lower vibration amplitude.

The entire L-2 blade row was then refitted with further modified (288 Hz) blades and with elliptical tenons having lower stress concentrations in the buttstraps. Seven additional rows in other rotors were also modified in the same manner. Since these modifications were installed there have been non subsequent blade or buttstraps failures in the L-2 rows of these units.
6. **Current Problems**

Problems experienced with steam turbine blades at the present time fall into three categories:

- Problems of older blades, typically damage-induced fatigue.
- New blade design problems: avoidance of resonance and flutter.
- Problems of re-bladed units: new blades, old designs.

For every blade which cracks or fails during operation there are two questions which should be asked, 1) what caused the failure to occur, and 2) what modifications are needed to prevent failure in the future? Experience has indicated that if 1) is neglected, then the fix in 2) has a 40% chance of failing again [24], whereas if 1) and 2) are both properly addressed, the chance of a second similar failure may be zero.

Blade failures in older machines are sometimes influenced by corrosive attack on highly stressed blade surfaces over a long period. Corrosion fatigue of blades cannot be quantified with any degree of certainty at present, but progress has been made in understanding the stress corrosion cracking mechanism [31] [32]. It is known that such failures have initiated due to a corrosion or corrosion fatigue mechanism. Once started, the crack eventually propagates to failure increasingly due to HCF, as the crack lengths. Typically both the corrosion mechanism and the fatigue loading involve complex intensity-time and load-time histories, which need to be considered when attempting to rationalize past damage, or in predicting future component life. In cases where corrosion is absent from the initiation/propagation processes, the prediction of blade life is much better understood, although the accuracy of any attempted HCF/LCF life prediction still depends on inclusion of blade load history in sufficient detail.

A recent blade-cracking problem involving the need for future life prediction has been solved with fracture mechanics technology throughout the cracking process, rather than by the customary Local Strain/Miner’s rule approach. Fracture Mechanics was applied from crack initiation through propagation and subsequent failure. A new program named ALLEGRO [28] was developed for this purpose, after Local Strain methods had proved imprecise in test calibrations. ALLEGRO is based on proven aerospace technology. Failure was predicted with good accuracy for a turbine blade that showed multiple cracking events after 30 years operation and 200 billion HCF cycles. No corrosion or other damage mechanism was involved. ALLEGRO does not use Miner’s rule. It addresses the effect of load sequencing as this is experienced in actual operation.

Flutter of LP blades is another recognized blade failure mechanism. Flutter may affect both old and new blades, and catastrophic failures due to flutter are believed to have occurred. Blade flutter in steam turbines occurs when the turbine exhaust pressure increases above 7 or 8 psia. Stall conditions may then occur locally in the flow around the blades (or groups), and as a consequence the blades may vibrate violently. Observations in test rigs have confirmed these statements. At present, instrumentation and equipment for use in monitoring steam turbine blade flutter is not widely used. The accepted countermeasure is to operate below the manufacturer’s stated back-pressure limit. This condition is sometimes difficult to maintain in heavily loaded units operating in warm climates, where limited resources of cooling water may be available for the condensers to sustain low back-pressure.

The design of modern steam turbine blades for failure-free operation has been assisted in recent years by the development of proven computer codes (ANSYS, BLADE-ST) of great scope and capability. The prediction of stresses and precise natural frequencies at operating speed, when aided by spin-pit verification during blade development is evidently a major step toward ensuring safe, long life operation of blades. Acceptance of the Campbell diagram as a design tool is now universal, and widening use of the Interference diagram has provided an additional refined tool for the avoidance of resonance. The fundamental need is to have an accurate understanding of the true location of the natural frequencies of the blade row. Inaccurate natural frequency predictions and an absence of actual test data at speed (actual blade design, actual groupings, and actual disk) each appear to have contributed to numbers of failed blade designs in the past. Resonance detuning from dangerous situations requires that the natural frequencies should be known precisely with adequate test confirmation. The final blade frequencies should be detuned by at least 10 Hz from all potential resonances.
7. Conclusions

1. Major causes of past blade failures have been HCF resonance (or near resonance), stress corrosion cracking, corrosion fatigue, and assorted stress concentrations.
2. Blade detuning from resonance is the best safeguard against HCF failure, but detuning must be undertaken appropriately, i.e., away from the excitation in the direction of the natural frequency, to avoid inadvertent detuning into the resonance.
3. Dynamic strain gage radio telemetry testing at operating speed is the currently the most proven way to absolutely verify natural frequency results.
4. Modal testing at zero rpm is a simple and practical way to obtain preliminary verification of natural frequency results for blade groups mounted on a disk. Use of Loctite to secure the attachment region may assist such modal testing.
5. Natural frequencies of the actual bladed disk must be calculated at zero rpm and at operating speed.
6. Natural frequencies are difficult to determine to within 2 percent accuracy by calculation alone, without calibration against test results.
7. Both the Campbell diagram and the Interference diagram are needed for the blade design process, to identify dangerous harmonic coupling conditions.
8. Tolerance variations in diaphragm passage dimensions is the major cause of diaphragm harmonic excitation.
9. Safe designs can be prepared without precise knowledge of damping and excitation. Use of typical values of these parameters known from experience is usually adequate for stress and life calculations.

8. References


### Table 1

**Lessons Learned from Past Blade Experience Blade Design**

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
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<tbody>
<tr>
<td>1</td>
<td>GIGO syndrome applies at all times for blading calculations.</td>
</tr>
<tr>
<td>2</td>
<td>Thorough detailed models are needed to obtain accurate natural frequency values. This applies even more emphatically for dynamic stresses.</td>
</tr>
<tr>
<td>3</td>
<td>Zero speed natural frequency values are different from at speed natural frequency values.</td>
</tr>
<tr>
<td>4</td>
<td>If blade disk stiffness effects are omitted from the model, inaccuracy will be introduced into the natural frequency calculations.</td>
</tr>
<tr>
<td>5</td>
<td>Blade groups which operate detuned from all resonances will not fail from HCF.</td>
</tr>
<tr>
<td>6</td>
<td>Strain gages do not give precise stresses. They give average stresses from over the area of gage contact.</td>
</tr>
<tr>
<td>7</td>
<td>Blades operating at steady stress levels above the material yield point can be expected to experience shorter lives to failure from dynamic stress cycling than blades with steady stress levels in the elastic range.</td>
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</table>
### Table 2
**Blade Damage Mechanisms**

<table>
<thead>
<tr>
<th>Mechanism</th>
<th>Location and Cause</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Low Cycle Fatigue</td>
<td>• Small number of large amplitude stress cycles</td>
</tr>
<tr>
<td></td>
<td>• May cause existing crack to grow in size</td>
</tr>
<tr>
<td></td>
<td>• From a high stress site</td>
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<tr>
<td>2. Stress Corrosion Cracking</td>
<td>• Electrochemical attack on exposed material surface</td>
</tr>
<tr>
<td></td>
<td>• Causes weakening of inherent material</td>
</tr>
<tr>
<td></td>
<td>• Steady applied stress</td>
</tr>
<tr>
<td></td>
<td>• Attack by chemical change, e.g., oxidation</td>
</tr>
<tr>
<td>3. Corrosion Fatigue</td>
<td>• Corrosive surface environment</td>
</tr>
<tr>
<td></td>
<td>• Cyclic applied stresses</td>
</tr>
<tr>
<td></td>
<td>• Typically HCF cycling rate</td>
</tr>
<tr>
<td>4. Erosion</td>
<td>• Destructive plastic wear of exposed surface</td>
</tr>
<tr>
<td></td>
<td>• Material degradation due to rapid energy transfer upon impact</td>
</tr>
<tr>
<td></td>
<td>• Water droplets or other small high velocity particles</td>
</tr>
<tr>
<td>5. Creep</td>
<td>• Degradation of material strength over time</td>
</tr>
<tr>
<td></td>
<td>• Causes weakening of material grain structure</td>
</tr>
<tr>
<td></td>
<td>• Prolonged exposure to high temperature under excessive applied steady stress</td>
</tr>
</tbody>
</table>

### Table 3
**Causes of Damage Mechanisms in Blades**

<table>
<thead>
<tr>
<th>Cause</th>
<th>Details</th>
<th>Reason</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Weakened Material</td>
<td>Inherent flaws, inclusions, voids, scratches, pits, corrosion, erosion, creep, embrittlement</td>
<td>Environmental attack or local damage weakens the ability of the material to resist applied loads.</td>
</tr>
<tr>
<td>2. Diaphragm Harmonics Excitation</td>
<td>Spacing variations between steam flow passages around circumference</td>
<td>Manufacturing tolerances, junction between halves of diaphragm, blockages.</td>
</tr>
<tr>
<td>3. Ineffective detuning between forcing and response</td>
<td>Insufficient frequency separation between adjacent harmonic forcing on blade surface and blade natural frequency.</td>
<td>Diaphragm harmonics, etc., or torsional stimulus acting on adjacent natural frequency with appropriate phase relationship for coupling.</td>
</tr>
<tr>
<td>4. Partial Admission Excitation</td>
<td>Control stage inlet flow nozzle segments separated circumferentially by dead no-flow regions.</td>
<td>Diaphragm harmonics arising from shape of cyclic (Fourier) flow pulses.</td>
</tr>
<tr>
<td>5. Stop/Start Operation</td>
<td>Low frequency, large stress range load cycles due to operating history.</td>
<td>Stop/start operation, initial overspeed, governor trip speed excursions, load rejection overspeed events.</td>
</tr>
<tr>
<td>6. Flutter (unstalled)</td>
<td>Passage of stall cells around rotating</td>
<td>Propagation of locally stalled flow</td>
</tr>
</tbody>
</table>
7. Flutter (stall)  
Development of flow separation from blade inlet surface.  
High exhaust back pressure causing decrease of inlet flow velocity and consequent stalling of flow on rotating blades.

8. Water Ingress  
Sudden surge of condensate into stage from circumferential drains.  
Temporary loss of function of condensate removal pump.

9. Seal Rub  
Rub between tip seal components or gland seal components  
Rotor bowing due to thermal bending and/or residual unbalance.

| Table 4  
Design Guidelines |
<table>
<thead>
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<tbody>
<tr>
<td>Detune blades by design. Obtain positive assurance of detuning by modal testing initial rows, and spin pit strain gage test.</td>
</tr>
<tr>
<td>Check blade natural frequencies with Loctite at zero rpm before measuring. Calculate frequency rise.</td>
</tr>
<tr>
<td>Reduce diaphragm excitation by close control of spacing differences during manufacture. Determine the magnitude of the excitation (or relative excitation) in advance by calculation. Minimize excessive harmonics by grinding problem spaces.</td>
</tr>
<tr>
<td>De-couple dangerous modes by shroud modifications (long arc effect) or by mistuning of alternate pairs of blades.</td>
</tr>
<tr>
<td>Minimize the effect of stress concentrations by the use of larger radii, by shot peening, by rounding corners, and by cryogenic freezing.</td>
</tr>
<tr>
<td>Frequency test blades in a rigid fixture before installation to ensure uniform natural frequencies of all blades.</td>
</tr>
<tr>
<td>Coat blades which operate in the Wilson line region to reduce corrosion in high stress regions.</td>
</tr>
<tr>
<td>Reduce droplet sizes to reduce rate of erosion. Use the thinnest possible trailing edges.</td>
</tr>
<tr>
<td>Check metallography before acceptance of blades. Determine nature of inclusions, flaw sizes, and spacing orientation.</td>
</tr>
<tr>
<td>Provide suitable seal clearance to minimize rotor rubs and steam blow-by.</td>
</tr>
<tr>
<td>Install Blade Vibration Monitor to monitor possible flutter vibrations in L-0 blade row.</td>
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