STEAM TURBINE BLADE FAILURES, CAUSES AND CORRECTION

by

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Mr. Sohre's experience includes 9 years of shop work as an assembly mechanic and as an instrument repairman. His engineering experience includes 8 years with Elliott Company as Section Head, Mechanical Design and Engineering Mechanics Section, Turbines and Compressors. He spent 7 years with the Terry Steam Turbine Company as Chief Engineer being responsible for development of Terry's type JS line of turbines for high-speed, high-horsepower service. Since July 1970 he has been an independent consultant for Turbomachinery Design, installation and problem correction.

ABSTRACT

This paper is meant to help with failure investigation, presenting a review of the causes and effects involved when blades fail. Included are descriptions of failure mechanisms (stress; resonance; environment) and of the conditions causing the failures (design; material; manufacturing; operation). Also included are chapters on failure analysis, evaluation of circumstantial evidence, failure history, and investigative procedures.

A table coordinates causes and effect.

An appendix explains methods used to compare stress levels against allowable values.

INTRODUCTION

The requirements of variable speed and of starting while under load impose much more severe conditions upon the blading of Mechanical-Drive (MD) turbines than those experienced with turbine generators. At this time, blade reliability represents a limit of technology in the development of large turbines, and it thereby introduces limiting factors into the design of process plants.

Vertical column:

1.0 Failure Mechanisms

- 1.1 Excessive stress
- 1.2 Resonance
- 1.3 Environmental effects

NOMENCLATURE

- c = Steam velocity; ft/sec
- c_o = Steam velocity; at nozzle exit
- D = Diameter. Pitch (mean) diameter of blading unless otherwise specified; inches
- E = Modulus of elasticity; lbs/sq. in.
- g = Gravity constant; or centrifugal force expressed as multiple of a parts weight (a 1 lb weight at 10,000 g pulls with 10,000 lbs)
- G = Steam flow; lbs/h
- Δh = Heat drop across a stage, usually isentropic; BTU
- H = Mode response factor for resonant vibration
- HP = Stage horsepower
- K_{Rs} = Root size factor; relative to foil base; K_{Rs} = 1 for a root of w = 1.2 s
- K_{Rt} = Root type factor; = 1.0 for top-quality, 3-land pine-tree root, axial or tangential entry (with restraining lips)
- ℓ = Blade length; usually from tip to base of foil (in)
- m = Mass; lbs sec²/in.
- M = Mach number at blade exit = velocity/acoustic velocity
- n = Rotor speed; RPM
- n = Mode of resonant vibration; cps.
- p = Pressure; psi p_1=nozzle inlet pressure, p_2=bucket exit pressure
- RPM = Revolutions per minute
- s = Blade foil width; chord or axial, at base
- S = Stimulus of excitation
- SF = A stress-level factor, for comparative purposes only; SF $\leq$ 1.0 would correspond to normal, conservative stress levels, 2.0 indicating possible high-stress conditions worth checking. Note that this is a very crude indicator and conclusions should not be based on it. All it is meant to indicate is whether or not a detailed check appears advisable.
- u = Pitch line speed; ft/sec., = D x RPM/229
- v = Specific volume; cu ft/lb
- V = Vibratory amplification factor
- w = Root width, axial; in.
- w_2 = Blade exit velocity, relative to airfoil; ft/sec.
- t = Blade-to-blade pitch; at pitch diameter or root
- Z = Number of pulses of excitation per revolution
- α = Load-dissipation doefficient for shroud and/or lashing wires
- β_2 = Foil exit angle; degrees
- γ = Specific weight; lbs/cu in
- δ = Damping coefficient
- ϵ = Admission ratio [180° admission arc $\triangleq$ (ϵ=.5)]
- η = Efficiency
- ν = Resonant frequency of blade; cps
- σ = Stress. See Appendix for definitions

Horizontal headings:

- *2.0 Engineering, Design, Service Conditions*
  - 2.1 Blade design
  - 2.2 Material
  - 2.3 Manufacturing effects
  - 2.4 Stage environment and operating conditions
  - 2.5 Maintenance effects
- *3.0 Symptoms and Evidence*
  - 3.1 Failure location
  - 3.2 Fracture analysis
  - 3.3 Surrounding evidence
  - 3.4 Failure history
  - 3.5 Operating symptoms
- *4.0 Investigative Procedures*
- *5.0 Remedial Action*

The tables attempt to show areas of possible interaction:

1=possible correlation

2=probable correlation

3=highly probable correlation

These indicators are not meant to reflect statistical probabilities, but rather to show us what areas to look into, and the degree of likelihood that a correlation exists. The indicators also show us in which areas we can expect the most effective improvements.

Only the most essential considerations are covered, in a simplified and condensed format, suitable for use during failure investigation. The procedures are not meant to be used for design analysis, being too crude for this purpose.

GENERAL CONSIDERATIONS

When a blade failure occurs, the first reflex reaction is usually "Resonance!", and off we go, into a lengthy and costly exploration of all the possible (and sometimes impossible) modes of resonant vibration. We are all aware of the exceedingly detrimental effects of resonance, but it may be a good idea to sit back a minute and to consider a few basic facts:

Operation on resonant frequencies, under load, and in relatively poor steam, is a fact of life with most mechanical-drive turbines. This is the reason we have such difficulties building machines over, say, 60,000 HP, while the generator drives are far bigger, having 30 in. blades running at 2000 ft/sec tip speed. The first resonant mode of a first-stage blade will be about 2000 to 6000 cps, while an 18″ last-stage blade will come out around 100 to 150 cps. The intermediate stages will cover the spectrum between these values. Considering the many modes of direct, harmonic and subharmonic excitation for each of these resonant modes, there can hardly be any doubt that we will run on some resonances while — and this may be more important — we will also have to pass through some of the most destructive modes during each start, accumulating quite a few thousand high-stress cycles each time. For constant speed and no-load starting we can tune blades to avoid resonances, but not so with variable-speed MD turbines.

Evidently, certain blades must be strong enough to operate under resonant conditions, and a few of these resonances will be severe. If a non-defective blade fails under these conditions, it was obviously too weak for this kind of service, and stresses must be reduced to make it survive. Getting out of a severe resonance (assuming this to be the main problem) is one way to do this, but it is by no means the only way and, as a rule, it is a very lengthy operation, involving many uncertainties which can result in repeat failures and plant shutdowns. The weakness of a blade — or its exposure to highly detrimental working conditions — is often quite obvious, and in most cases it is possible to reduce stress by such means as making the blading stronger, using better design and quality, and by adding damping by means of caulking strips and lashing, if resonance is suspected. Excitation can be reduced by smoothing the steam path. Load dissipation can be improved by means of shrouding and lashing, to name a few possibilities. Other important improvements may be obtained by using a different material; providing better, less corrosive steam; giving more attention to mode of operation (load, back-pressure, etc.). Many more immediately applicable measures are available.

It is important to remember that blading must have a *generous* safety margin, to survive the many abuses to which it will be exposed. If failures have occurred, it proves beyond argument that the actual margin was insufficient, for whatever reason. It is mandatory that stress, strength, load, or environment be improved by a known factor of at least 3, especially so in case of repeat failures. A minor improvement, say 20% — such as may be obtained by better surface finish, shot-peening, caulking, etc. — is not adequate to establish a definite minimum level of reliability. Such minor improvements may well work in some cases, and they are, of course, always desirable, but they should never be regarded as satisfactory solutions by themselves.

One curious situation can be observed more often than not: Money and time are spent generously on the analysis of the failed blade, but when the time comes to make the hardware stronger, people settle for the skimpiest kind of improvement — for cost reasons! Would it not be better the other way around? However, it should always be kept firmly in mind that blade strength is, roughly, a linear function of cost, all things considered. This includes wheels and turbine length, critical speeds, etc., etc. There is no sense in asking for a stronger blade unless you are willing to foot the bill. Furthermore, in many cases the problem originates with nozzles and flow irregularities. Good diaphragms are *very* expensive!

FAILURE INVESTIGATION

1.0 Failure Mechanisms

1.1.0 Excessive Stress

The total stress at any location of a blade consists of: Centrifugal tension + centrifugal bending + steady steam bending + alternating bending. See appendix for stress criteria. Alternating bending is the product of: steady bending x damping x stimulus of excitation x load dissipation (shroud) x resonant response.

1.1.1 Centrifugal Stress

In steam turbines, centrifugal stress is never the principal cause of a blade failure, except in the rare cases of turbine runaway (including reverse-rotation), or possibly with frequent start/stop service (low-cycle fatigue). Pure centrifugal stress failure would occur at 3 to 4 times normal design stress, or 75% to 100% overspeed. The failure appearance would be quite striking, being characterized by strong fracture elongation (about 20%) and area reduction in the critical areas, and/or by stripping of the root serrations, the way threads are stripped on overloaded bolts.

However, centrifugal stress is an important contributing factor with fatigue failures, corrosion fatigue failures, and stress

CHART NO. 1 OF 2

ENGINEERING, DESIGN, & SERVICE CONDIONS

BLADE DESIGN

1.0 FAILURE MECHANISMS	FOIL: Short	Medium	Long	Free-standing	Lashing only	SHROUD: Integral, abutting	Riveted segments	Tenon not round	Sharp corners at holes, fillets, tenons	ROOT: Inserting-gap or variable bucket pitch	With packers	With C.G. offset	Internal root without retaining rim	Lands not curved	Inserting slot too wide	Surface finish > 63 RMS	NOZZLE: Single nozzles	-Cast	-Split mismatched	-Part. admission	-Sheet-metal nozzle blades	Nozzle pitch variations	Ribs or braces	No. of nozzles ≈ No. of bkts. or ≈ even multiple	Strong afterexpansion	STAGE TYPE: First	Intermediate	Last, backp.	Last, condensing	Last bf. extraction	Last bf. split
1.1.0 EXCESSIVE STRESS																															
.1 Centrifugal stress		1	3						3		2	3	1	3	3												1	1	3	1	3
.2 Steam stress, steady		1	1	3	2	3		2	3		3	2		3	3										2	2	1	3	1	3	3
.3 Steam σ alternating	3	3	2	3	2	2	1	2	3		3	2	3	3	3	2	3	3	3	3	3	2	2		3	3	2	3	2	3	3
.4 Impact				3	2	2	1		3		3	1	2	2	2											3	1				
.5 Low-cycle fatigue	1	2	3						3		3	3		3	3	2									2	1	1	1	3	1	3
.6 Thermal fatigue							2	2	3																	3					
.7 Creep									3					2	2											2					
1.2.0 RESONANCE																															
.1 Mode of vibration:																															
Tangential	3	3	3	3	3	3	3		3								3	3	3	3	3	3	3		3	3	3	3	3	3	3
Axial, packet vibration	2	3	3		1	1	3		3	3	3	3	3				3	3	3	3	3	3	2		3	3	3	3	3	3	3
Torsional		1	3	3	2	1	2		2									3			3	1			1		2	2	3	2	3
.2 Mode of excitation:																															
Nozzle-passing	3	3	1	2	1	3	3	3	3		3	3	2			3	3	3		2	3	3			3	3	2	2	1	2	3
2, 3, and 4 x Rev			3	3	3	3	2			3						3		3	3	1	3	3	3		3		1	1	3	1	3
Wave reflection	3	2									3	2	2			3	3	1			2	2			3	3	1	1	2	3	2
Harmonic	2	3	3	3	3	3	3			2	3	3	2			3	3	3	3	3	3	3	3		3	1	3	3	3	3	3
Partial admission	3	3				3		3	3	3	3	3	2	2	2	3							3		3	3	3	1		1	
Multivalve control	3					3			3	3	3	3	3	3	2	3				3		3	3		3	3					
Flow non-uniformity		3	3	3	2	3	2			3	3	2						3	3	3	3	3	3		3	3	2	3	3	3	3
Blading discontinuities	2	3	3							3	3				3		2	3	3	3	3	3	3		3	3	3	3	3	3	3
Out-of-roundness	1	3	3															3	3		3	2			2	2	3	3	3	3	3
Obstructions	1	2	3	3						2									3	3	3	3	3		3	3	3	3	3	3	3
Disk vibration	1	3	1				3			3	3	2	3				3	3	3	3	3	3	2		3	2	3	3	1	3	2
Rotor vibration		1	2	2	1	1					1	1															1	1	2	1	2
Rotor torsionals		1	2	3	1	1					1																1	2	3	2	3
Condensation shock		2	3								2	2					3								3	1	3	3	3	3	3
1.3.0 OPERATING ENVIRONMENT																															
.1 Stress-corrosion		2	2																							1	2	2	2	1	2
.2 Corrosion fatigue		3	3					2	2		3	3	2	3	3	3											3	3	3	3	3
.3 Standby corrosion	3	3	3					3	3		3		2	3	3	3										3	3	3	3	3	3
.4 Erosion		2	3															3	3		3		3		2	1	1	1	3	1	1

1.0 FAILURE MECHANISMS	MATERIAL: Yield strength > 100 ksi	Yield strength < 70 ksi	Elongation > 18%	Impact ≤ 35 ft.lbs	MANUFACTURING: Surface finish poor, nicks, toolmarks, assembly tears	Not to tolerance	Blades not packed tight	Punched shroud holes	Shroud not down	Rivet heads cracked	Shroud forced on	Nozzle arc not concentric	Disk axial runout	Wrong assembly lubricant	Windage shields in wrong position	Bent or damaged nozzles	$M_2 > 1.15$
1.1.0 EXCESSIVE STRESS																	
.1 Centrifugal stress		3	3		3	3		3	3	3	2						
.2 Steam stress, steady		3				3	3	3	3								2
.3 Steam σ alternating		3			3	3	3	3	3	3			3		3	3	3
.4 Impact			3	3	2	1	3	3	3	3							
.5 Low-cycle fatigue		3	3	3	3	3	3	3	3	3	3						2
.6 Thermal fatigue		1	3		3	3		3	3	3	3						
.7 Creep						3		3	3	3	3						
1.2.0 RESONANCE																	
.1 Mode of vibration:																	
Tangential						3		3	3			3			3	3	3
Axial, packet vibration						3		3	3	3	3	3	3		3	3	3
Torsional						3		3	3	3						3	3
.2 Mode of excitation:																	
Nozzle-passing					3	2		3	3	3	3						3
2, 3, and 4 x Rev						2		3	3	3	2	3	3			3	3
Wave reflection								3	3	3	3					3	3
Harmonic						2		3	3	3	2	3	3		3	3	3
Partial admission						2		3	3						3		3
Multivalve control					3	3	3	3	3	3	3				3	3	3
Flow non-uniformity						3	3	3	3	3	3	3	3		3	3	3
Blading discontinuities						3					3				3	3	3
Out-of-roundness						3			3			3					2
Obstructions												3			3	3	3
Disk vibration						2	3				3		3		3	3	3
Rotor vibration							2	2	2								
Rotor torsionals							3	3	3								
Condensation shock																	
1.3.0 OPERATING ENVIRONMENT																	
.1 Stress-corrosion	3		3	3	3		3	3	3	3	3			3			
.2 Corrosion fatigue	3		3	3	3		3	3	3	3	3			3			
.3 Standby corrosion	3				3	3	3	3	3	3	3			3			
.4 Erosion		3													2	3	3

1.0 FAILURE MECHANISMS	STAGE ENVIRONMENT AND OPERATING CONDITIONS: Wide speed range	Dry/wet (100°SH to 5%)	Wet (> 5% moisture)	Frequent speed or load variat.	Frequent starting	Quick-starting	Varying inlet steam	Varying exh. press.	Exhaust or cross-over crowded	Exh. press. below design	Up-exhaust	Condenser leaking	Water slugging	Water washing	Boiler carryover	Desuperheater	Heat exchangers leak	More than one boiler	Boiler highly loaded	Sealing steam on during shutdown	Common casing drain or sewer	MAINTENANCE: Handling damage	Blades straightened	Nozzles dressed	Not all blades replaced aft. failure	Weather exposure during repair	Abrasive blasting
1.1.0 EXCESSIVE STRESS																											
.1 Centrifugal stress					2	3																3	3				1
.2 Steam stress, steady					2	3		3		3				3								3	3				1
.3 Steam σ alternating	3			3	3	3		3	3	3				3								3	3	3	3		3
.4 Impact					3	3	3				3	1	3	3	2	1		2	3			2	3		3		
.5 Low-cycle fatigue	3			3	3	3	3	3	3		3		3	3							3	3	3				2
.6 Thermal fatigue	3			3	3	3	3						3	2	3	3		2	2			3	3				1
.7 Creep																											
1.2.0 RESONANCE																											
.1 Mode of vibration:																											
Tangential	3			3	3				3	3				3								3	3	3			3
Axial, packet vibration	3			3	3				3	3				3								3	3	3			3
Torsional	3			3	3				3	3				3								3	3	3			3
.2 Mode of excitation:																											
Nozzle-passing	3			3	3			1		3				3								3	3	1			2
2, 3, and 4 x Rev	3			3	3			3	3	3				3								3	3	3			2
Wave reflection	3			3	3		2	3	1	3				3								2	3	3			2
Harmonic	3			3	3		3	3	3	3				3								3	3	3			3
Partial admission	3			3	3																						
Multivalve control	3			3	3		3																				
Flow non-uniformity	3			3	3				3	3												2	2	3			
Blading discontinuities	3			3	3				3	3												3	3	3			
Out-of-roundness	2			2	3				3	3																	
Obstructions	3			3	3				3	3												3					
Disk vibration	3			3	3																	3	3	3			
Rotor vibration	3	3	3	3	3	3	3				3		3	3		1	1	1	3	2							
Rotor torsionals	3	3	3	3	3																						
Condensation shock	3	3		3	3		3	3	2	3			2	3	3	3		2	2								
1.3.0 OPERATING ENVIRONMENT																											
.1 Stress-corrosion		3	3		3						3	3			3	3	3	1	3	3	3	3	3			2	
.2 Corrosion fatigue	3	3	3	3	3	3	3	3		3	2	3	3	3	3	3	3	2	3	3	3					3	
.3 Standby corrosion					3						3	3	3	3	3	3	3		3	3	3	3				3	
.4 Erosion			3	3	3	3	3	3	3	3	3	3	3	3	3		3	3	3			2	2	3			

VERY STRONG RESPONSE; RESONANT FREQUENCIES UNPREDICTABLE

RESONANT FREQUENCIES UNPREDICTABLE

CRACKED AND EXHAUSTED BLADES LEFT IN WHEEL

SHOT-PEENING REMOVED; CONTOUR CHANGED

CORRELATIONS OF CAUSES AND EFFECTS:

1 = Possible Correlation.
2 = Probable Correlation.
3 = Highly Probable Correlation.

REVISIONS	JOHN S. SOHRE TURBOMACHINERY CONSULTANT DESIGN • ANALYSIS • INSTALLATION PROBLEM CORRECTION 93 GRIER ROAD • VERNON, CONNECTICUT 06066 U.S.A.		
	TITLE BLADE FAILURE TROUBLE SHOOTING CHART		
	SCALE	DR. BY JSS	DRAWING NO.
	DATE Aug. 3/1979	CK'D. BY	

CHART #2 OF 2

SYMPTOMS & EVIDENCE

.0 FAILURE ECHANISMS	FAILURE LOCATION									FRACTURE ANALYSIS											
	Shroud assembly / Lashing wire	Foil, base	Foil, other	Root	Wheel root	Near inserting slot	90 or 180° from slot 60 or 120° from slot	At end or center of shroud bands	Lands sheared	Plain fatigue	Branched cracks	Transgranular fract.	Intergranular fract.	Residual mtl. brittle	Virgin crack surface: -Silver, or gray	-Light blue to purple / -Gold, yellow	Residual area of ductile failure -Less than ¼	-More than ¼	No. of cracked blades Per thrown blade:	-Several	-None
1.1.0 EXCESSIVE STRESS																					
.1 Centrifugal stress	2	/		3	3	3			3	/								3		3	/
.2 Steam stress, steady	/	2		3		3												3		3	/
.3 Steam σ alternating	3	3	3	3	3	3				3		3		3	3		3			3	/
.4 Impact	/	2		3		/			3	3					3					3	2
.5 Low-cycle fatigue	2	/		3	/	3				2		2			3			3		3	/
.6 Thermal fatigue	2	/		3		3				2		2			3					3	/
.7 Creep	2	/		3					/						3						
1.2.0 RESONANCE	3	3	3	3	3	/	3	3		3		3		3	3		3			3	/
.1 Mode of vibration:																					
- Tangential	3	3	3	3	/	/		/													
- Axial, packet vibration	3	3	/	3	3	3		3													
- Torsional	2	3	/	/		/															
.2 Mode of excitation:																					
- Nozzle-passing	3	3	3	3		3		2												3	
- 2, 3, and 4 x Rev	3	3	/	3		/		2												3	
- Wave reflection	3	3		3		/														3	
- Harmonic	3	3	/	3		2		2												/	
- Partial admission	3	3	/	3	3	3		3												3	
- Multivalve control	3	3	/	3	3	3		3												3	
- Flow non-uniformity	3	3	/	3	3	2		3												/	
- Blading discontinuities	3	3	/	3	3	3		3												/	
- Out-of-roundness	2	3		3		/		/												/	
- Obstructions	3	3		3		/		2												3	
- Disk vibration	3	2	/	3	3	3	3	3												3	
- Rotor vibration	/	2	/	3		/														3	
- Rotor torsionals	/	2	/	3		/														3	
- Condensation shock	2	3	/	3		/															
1.3.0 OPERATING ENVIRONMENT																					
.1 Stress-corrosion	3	/	/	3	3	/					3		2							3	
.2 Corrosion fatigue	3	2	/	3	3	3					3	3	3		/	3	3			3	/
.3 Standby corrosion	3	/	/	3	3	3				3	3	3	3		3	/				/	3
.4 Erosion	/	2	3	/	2					3					3					/	3

.0 FAILURE ECHANISMS	SURROUNDING EVIDENCE																
	Nozzles eroded	Nozzle edges failed in fatigue	Buckets eroded	Wheel-rim eroded	Blades bent forward	Blades bent backward	Blades bent axial	Blade assy. loose in wheel	Corrosion pitting / Turbine internals pitted	Rust covering	Deposits: -Sodium salt (=salty taste)	-Process-related	-Silica deposits	-Salt in root crevices	Wheel: Cracked radial	-Cracked in keyways	-Rim rubbed axially
1.1.0 EXCESSIVE STRESS																	
.1 Centrifugal stress								3									
.2 Steam stress, steady		3						3									
.3 Steam σ alternating		3						3									
.4 Impact	3			2	3	3	3	3									3
.5 Low-cycle fatigue	3			2		/		3									
.6 Thermal fatigue	3	3						2									
.7 Creep																	
1.2.0 RESONANCE	3	3						/							3	/	
.1 Mode of vibration:																	
- Tangential	3	2															
- Axial, packet vibration	3	3															
- Torsional		/															
.2 Mode of excitation:																	
- Nozzle-passing																	
- 2, 3, and 4 x Rev	3	3															
- Wave reflection	3	3															
- Harmonic	3	3															
- Partial admission																	
- Multivalve control		3						3									
- Flow non-uniformity	3	3															
- Blading discontinuities		3															
- Out-of-roundness																	
- Obstructions			3														
- Disk vibration	3	3	3					3							3	3	3
- Rotor vibration																	
- Rotor torsionals															/	3	
- Condensation shock	3	3	3	3													
1.3.0 OPERATING ENVIRONMENT																	
.1 Stress-corrosion	/	/	/	/					3	3	3	3	3	3	3	3	
.2 Corrosion fatigue	3	3	3	3					3	2	3	3	3	3	3	3	
.3 Standby corrosion									3	3	3	3	3	3	3	3	
.4 Erosion	3	2	3	3					3							/	

.0 FAILURE ECHANISMS	FAILURE HISTORY																	
	First failure	Three or more failures	Original startup	First year	1 to 10 years	> 10 years	Few starts (< 100)	Many starts (> 500)	Sitting idle without purge	After shutdown	After plant modificat.	After turbine was open	After turbine was cleaned	During starting cycle	During normal operation	During plant upset	During speed change	During [illegible] change
1.1.0 EXCESSIVE STRESS																		
.1 Centrifugal stress	/	2		2	2	/		2										
.2 Steam stress, steady	/	2	3	3	2	/	3	3						3	3	3		3
.3 Steam σ alternating	/	3	3	2	2	/	3	3						3	3	3		3
.4 Impact	3	/	/	/	2	2	2	3		3	2	3		3		3		/
.5 Low-cycle fatigue				/	2	3		3										
.6 Thermal fatigue					2	3		3								2		3
.7 Creep					/	3	2								3			
1.2.0 RESONANCE	3	3	3	3	/		3	3		3				3	/	3	3	3
.1 Mode of vibration:																		
- Tangential																		
- Axial, packet vibration																		
- Torsional																		
.2 Mode of excitation:																		
- Nozzle-passing																		
- 2, 3, and 4 x Rev																		
- Wave reflection																		
- Harmonic													3					
- Partial admission			/	3	/										3			
- Multivalve control			/	3	/	/	2							3	2	3	3	3
- Flow non-uniformity			/	3	/		/							/	3	/	/	/
- Blading discontinuities			/	3	3	/							3	/	3	/	3	/
- Out-of-roundness			3	3	/	/	3	/							3			
- Obstructions			/	2	3	3	2	3				2		2	3	2	2	2
- Disk vibration			/	2	3	2	/	3						3	3	3	3	/
- Rotor vibration								3						/		3		
- Rotor torsionals			/	3	3	/	/	3						2	/	3	2	
- Condensation shock				/	3	2	2	3			2			/	3	2	/	2
1.3.0 OPERATING ENVIRONMENT																		
.1 Stress-corrosion	3	/		/	3	3	3	/	3	3	2	2	2	/	3	2	/	
.2 Corrosion fatigue	/	3	/	2	3	3	3	3	/	/	3		2	2	3	3		
.3 Standby corrosion	3	/	/	2	3	3	/	3	3	3	3	3	3		3			
.4 Erosion	3	2	/	/	3	3	3	3			3			3	3	3		2

.0 FAILURE ECHANISMS	OPERATING SYMPTOMS							
	High-pitched whine / Whine at certain speeds only	High frequencies (> 100 cps) in shaft spectrum	Torsional vibrations (gear noise)	Governor hunting / Compressor surge	Occasional sudden, rapid governor action, or unit slows down suddenly	Water coming out of seals at standstill	High-pressure flanges & joints blow at times then re-seal / Had casing distortion / Had steam chest & 1st stage nozzle cracking / Had rotor bow	Had thrust failures
1.1.0 EXCESSIVE STRESS								
.1 Centrifugal stress								
.2 Steam stress, steady								
.3 Steam σ alternating		3						
.4 Impact				2	3		3	3
.5 Low-cycle fatigue				3	3		3	3
.6 Thermal fatigue				3	3		3	3
.7 Creep								
1.2.0 RESONANCE	3	3		3				
.1 Mode of vibration:								
- Tangential								
- Axial, packet vibration								
- Torsional								
.2 Mode of excitation:								
- Nozzle-passing								
- 2, 3, and 4 x Rev			3					
- Wave reflection								
- Harmonic								
- Partial admission								
- Multivalve control				3				
- Flow non-uniformity		3						
- Blading discontinuities		3						
- Out-of-roundness								
- Obstructions		2						
- Disk vibration	3	3		3				
- Rotor vibration				2				
- Rotor torsionals	3	2	3	2				
- Condensation shock				2	/			
1.3.0 OPERATING ENVIRONMENT								
.1 Stress-corrosion						3		
.2 Corrosion fatigue						3		
.3 Standby corrosion						3		
.4 Erosion				3	3	3	3	3

RUNAWAY - OVERSPEED

ANY MODE OF RESONANT VIBRATION

CHECK FOR DEFECTIVE HARDWARE

REVERSED-ROTATION

WATER SLUGGED OR THRUST FAILURE, EXCESS DIAPHRAGM DEFLECTION

THRUST FAILURE, DIAPHRAGM DEFLECTION

TRANSIENT RESONANCES

WATER SLUGGING

TURBINE AND/OR CONDENSER FULL OF WATER

WATER-SLUGGING, STEAM TEMP. VARIATIONS RAPID LOAD VARIATIONS

CHECK FOR POSSIBLE WATER-SLUGGING

CORRELATIONS OF CAUSES AND EFFECTS:

1 = Possible Correlation.
2 = Probable Correlation.
3 = Highly Probable Correlation.

REVISIONS	JOHN S. SOHRE TURBOMACHINERY CONSULTANT DESIGN • ANALYSIS • INSTALLATION PROBLEM CORRECTION 93 GRIER ROAD • VERNON, CONNECTICUT 06066 U.S.A.		
	TITLE BLADE FAILURE TROUBLE SHOOTING CHART		
	SCALE	DR. BY JSS	DRAWING NO.
	DATE Aug. 3/75	CK'D. BY	

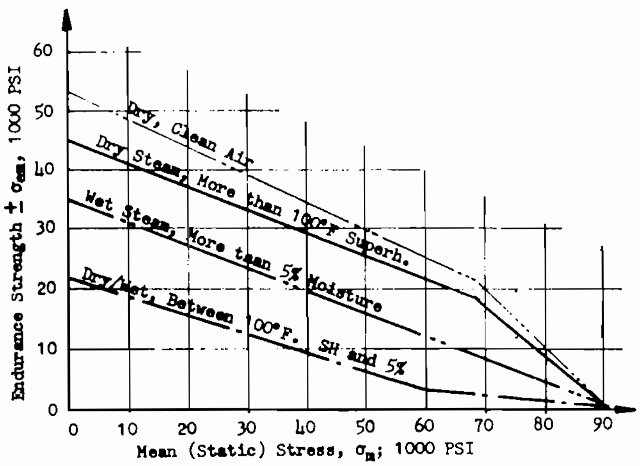

Figure 1a. Goodman Diagram, For Steam Turbine Blading, at Room Temperature. For Type 403 (13%Cr) steel, having 90,000 psi yield strength, assuming uncontaminated, demineralized, commercial quality steam, with minimum amounts of corrosive impurities, as encountered in a well-equipped, properly operated process plant.

Note: This diagram represents an attempt to provide some guidance concerning the effect of environment upon typical turbine blades. It is based on data from many sources. Corrections for temperature, surface finish, and stress concentrations must be applied.

The curves represent an estimate, as neither the type nor the concentration of the contaminants is known, and neither is the ratio of bending stress to pure tension, for a specific blade. Also, endurance data in corrosive environments is very scarce. It must be kept in mind that, in corrosive environment, an endurance limit does not really exist. Curves are estimated for 10^8 cycles. (7)

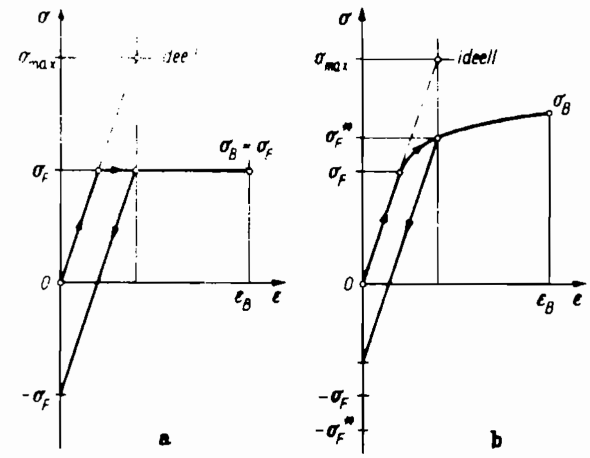

Figure 1b. Local yielding at stress concentrations and resulting residual stress.

a. Theoretical yield curve

b. Actual behaviour, leaving some safety margin (6)

corrosion failures, where it lowers the endurance strength of the material (see Fig. 1a), and with creep failures (metal temperatures over 650° for type 403 material, 750° for type 422). Creep is not normally a problem because, in the hot region of the turbine, centrifugal stress is usually low.

The centers-of-gravity of shroud, foil, root, and root lands should be located on a common, radial axis, within a few thousandths of an inch. This prevents centrifugally-induced bending stresses. Some of the cheaper blades — used primarily in single-stage turbines — are not so designed, and this may result in very high stresses, leaving no margin for alternating stresses, and possibly exposing the material at the root fillets to reversing yield (Fig. 1b) (tension at trip, compression when standing). This can cause local embrittlement and, ultimately, fatigue failures.

Some important facts and figures:

-Blades of different size but *of the same geometry*, running at the same tip speed, experience the same centrifugal stress at the base, as well as at all other sections in root and rim. This means: a 20″ blade on a 40″ wheel running at 4,000 RPM is stressed exactly like a 10″ blade of similar geometry on a 20″ wheel at 8,000 RPM, or a 5″ blade on a 10″ wheel at 16,000 RPM. However, the geometry must be *exactly* proportional, in all respects, especially ℓ/D. Material must be the same.

-Now assume existing blades of given length, mounted on wheels of varying diameter. Then run the wheels up to a speed where tip speed is the same for both the large and the small wheels. We will find that the blade stress for the small wheel — at high RPM — is considerably higher than for the blade on the large wheel — at low RPM —, although tip speed is the same for both. This explains why wheels and blading for high speed machines are so difficult to design. They must be exactly proportional in order to obtain the same tip speed as a larger machine. This leads to very delicate blades, requiring great manufacturing accuracy and extraordinary care in assembly and handling, as well as special maintenance procedures in the field.

The example also illustrates that it makes no sense to set a limit for tip speed without making it a function of RPM and, espacially, of blade geometry (ℓ/D, taper, root). Tip speeds of 2000 ft/sec at 3000 RPM are common, but try these same 2000 ft/sec at 15,000 RPM, unless blade geometry is *exactly* scaled!

The reason for this behavior is that, for a *given* blade (= given weight), only the strength of the centrifugal field is important, and this is proportional to m x r x ω^2, or mass x tip speed/tip radius.

-If, on the other hand, the type of blade, flow triangle, steam volume flow (cu. ft/sec.), blade exit angle, velocity ratio and RPM are given, then the *blade stress decreases as tip speed increases*. This means: The *larger* the disk diameter, the *lower* the blade stress. This seems paradox, but is explained by the fact that the stress is a function of blade length and pitch diameter, and ℓ/D decreases for a larger disk, as long as steam volume-flow is constant. This relationship is limited by the fact that:

- The disk stress goes up and ultimately becomes the limiting factor.
- The steam velocity leaving the bucket cannot go up indefinitely. Sonic conditions will be reached, and the flow relationship and efficiency will no longer be similar. Also, as the bucket becomes very short, the efficiency decreases due to steam friction losses at the blade surface. Nevertheless, this relation holds true over a wide range of practical applications, and therefore it is of considerable importance, because blading can be made more reliable where this rule is properly applied. The above leads directly to a

simple formula for centrifugal blade stress, at the airfoil base, as a function of steam flow parameters. This formula is not a rule of thumb, being mathematically correct for straight, unshrouded blades, and it can be closely approximated for other configurations.

$$\sigma_t = .0512 \frac{Gv \left(\frac{RPM}{1000}\right)^2}{w_2 \sin \beta_2}$$

-Formulas to estimate centrifugal stress, using dimensional input:

- Vane, at base: $\sigma_t = 4.03 \times \ell \times D \times \left(\frac{RPM}{1000}\right)^2 \times K_T$; for steel

  for Titanium: $\sigma_{t\ tit.} = 0.572 \times \sigma_{t\ steel}$

K_T = Taper factor

$$K_T = 1 - [0.05 \times (\ell/s) \times T]$$

T indicates degree of taper:

Straight: T=0
Linear taper: T=1
Parabolic: T=2

Centrifugal stress factor, vane: $SF_{tv} = \sigma_t/40{,}000$

This assumes 40,000 psi allowable stress (average), at normal operating speed. A stress factor of SF=1.0 represents 40,000 psi, SF=.5 represents 20,000 psi, and so on. Note that 40,000 psi centrifugal stress would not allow high alternating stresses, Fig. 1.

- Shroud stress:

This would be difficult to calculate quickly. A rule-of-thumb says that a good shroud assembly can stand 40,000 g, for a riveted shroud having a width of no more than 8 times its thickness. If we call this a stress-factor SF=1.0, we can tell how much higher or lower the actual stress is, by using the formula

$$SF_s = \frac{\text{shroud dia.} \times \left(\frac{RPM}{1000}\right)^2}{2817}$$

RPM is for normal operating speed

- Root Stress:

We can develop a root stress factor, to give us a rough idea of the level of root stress. Again, 1.0 is conventional stress at 100% load, 100% speed:

$$SF_R = SF_{tv} \times K_{RS} \times K_{Rt}$$

Where: K_{RT}=Root type factor, =3/No. of hooks (pairs). If the root has no retaining lips (Figs. 11 & 12a), use .7 x actual number of hooks.
K_{RS}=Root size factor, =1.2 s/w,
s is vane chord length or axial width (being about the same), w being root axial width, see Figs. 11 & 12a

This equation is based on the assumption that vane and root centrifugal stresses are about equal for a top quality, 3-land root having a width of about 1.2 times vane width. This is a rule of-thumb. The root type factor is based on the load capabilities per square inch of projected rim area as given in (6), for high-efficiency roots, plus corrections for neck geometry. Normal pitch-to-chord (t/s) ratios were assumed, as used for Rateau type blading. The above is for top-quality design, integrally machined roots, axial or circumferential.

If root and vane center-of-gravity are not the same radial axis, multiply root stress factor by 2.5 to 3.0, to account for centrifugal bending stresses.

1.1.2 Steam-induced stress, steady-state.

The bending stress generated by the "driving force," or "steam-load." This bending stress is superimposed on centrifugal tension. It is not normally a principal cause of failure. However, this stress is the basis for the vast majority of failure-causing alternating stresses, which are proportional to the steam load.

a. Effects of bucket geometry and operating conditions upon steam stress.

Because of the complexity of the stress situation — and especially of the dynamic or vibratory stress situation, for which the steam stresses furnish the basic "Stimulus" (see later) — it is important to develop a feel for the conditions which result in high steam stress levels, and also for the blade proportions and configurations which are suitable for highly loaded stages in various sections of a turbine.

For example: What happens if a low-pressure blade with a loading as shown in Fig. 1c were to be operated at twice the exit pressure, or at a higher steam flow (exit Mach number), or if the blade were to be shortened (a frequently used makeshift after a failure, and often not effective). Is it better to have many small blades, or rather fewer large ones?

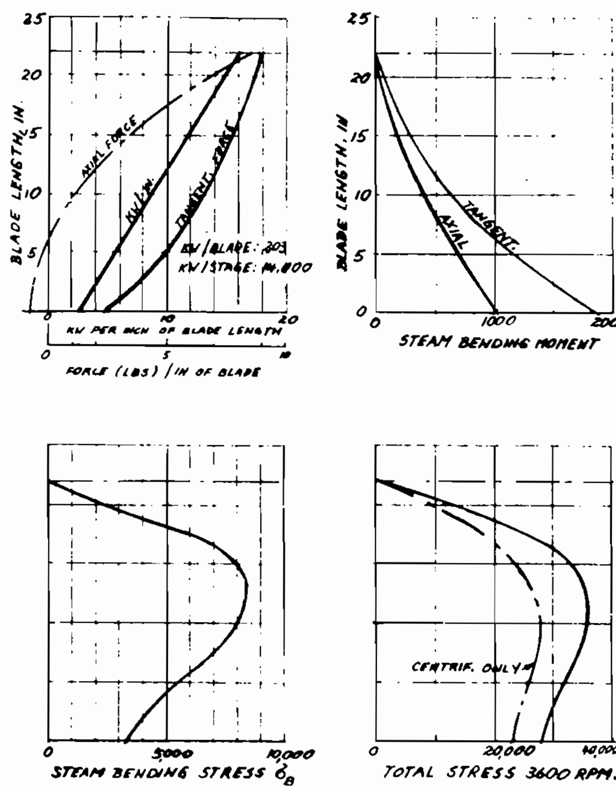

Figure 1c. Distribution of steam stress, bending moments, and energy in a 22.5" long, high-performance blade tapered and twisted, mounted on a 55" wheel. Running at 3,600 RPM at 8.0" H.G. abs. exit pressure.

Note: The stress levels shown would not necessarily be considered permissible. (7)

The following discussion will explain such *relative* influences. The *absolute* stress level will not enter the picture — it would have to be calculated or estimated separately.

Disregarding all constants and non-essential factors, the basic steam stress relation boils down to:

$$\sigma_b \triangleq p_2 \times M^2 \times \frac{D}{D_b} \times \left(\frac{\ell}{s}\right)^2 \text{; or: } p_2 \times \text{BTU} \times \frac{D}{D_b} \times \left(\frac{\ell}{s}\right)^2$$

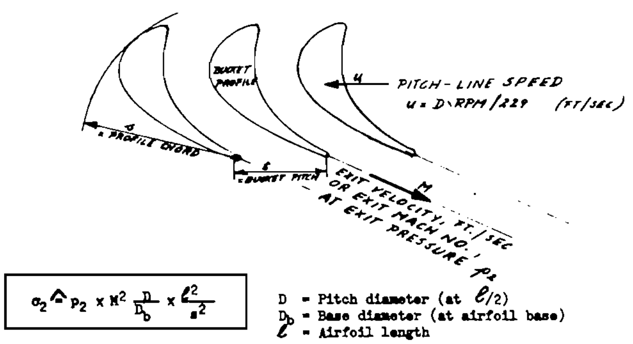

Figure 1d. Steam-bending stress as function of operating conditions and blade dimensions.

This non-dimensional relation holds only for stages having identical velocity triangles, but it can be used to estimate effects of blade size modifications and to estimate the effect of changing operating conditions. M^2 is proportional to the BTU drop across the stage, but stress will increase much more than M^2 when $M \approx 1.15$ is exceeded. (7)

Fig. 1d shows the essential parameters.

The sign $\triangleq$ is used here to indicate that the bending stresses are proportional to the factors on the right side, but that certain limitations exist which must be very carefully observed:

-The relation holds in its strict sense only for a given stage layout, having a fixed exit angle and other comparable thermodynamic characteristics (velocity ratio). However, the variables can be changed drastically, except for M, which see below.

-The profile geometry must be constant, although ℓ and s may be varied individually, s/t must be constant. Note that the chord dimension "s" (or axial width) is, by far, the most important factor where airfoil strength is concerned, since it enters the stress picture as a square function. Since M and s are both square functions, and the Δ h or HP is also a function of M^2, we derive the simple statement that:

"The heat drop (or Horsepower) a blade can handle is directly proportional to its axial width." This holds for the root width also.

-Flow must be sub-sonic (M<1.0), because stress at high Mach Number will be higher than indicated, because of increasing reaction (=pressure drop across buckets), this can cause very high additional stresses. See Fig. 7.

-If throttle steam flow changes, M will change proportionally in the last stage, while p_2 will remain constant. In all intermediate stages p_2 and M will change as a function of flow only.

-With long blades (as compared to wheel radius) the blade tips work much harder than the base sections. Maximum stress is not necessarily at the base, because of the high degree of taper ("parabolic taper") which such blades may have. See Fig. 1c.

b. Formula to estimate steady steam bending stress at blade root at top

$$\sigma_{bi} = 344 \; \frac{G \, \Delta h \, (0.6\ell + 0.3w)}{D\,(D-\ell)\,(\text{RPM})\,\epsilon \, w \, s} \times K_{Rs} \times K_{Rt} \quad \text{(psi)}$$

where K_{Rs} and K_{Rt} are the same as used for the centrifugal root stress estimate. This relation gives the bending stress for a free-standing blade in the vicinity of design speed, down to $\approx$ 50% speed. At zero RPM, σ_{bi} would become infinite which, of course, is not true. This results from neglecting the drop of efficiency at low speeds, and it can be corrected by multiplying σ_{bi} with the ratio of efficiency at actual speed over design efficiency. However, the blade stress at very low speeds and high flow is much higher than indicated by the shaft torque, because of unstable flow conditions, resulting from flow separations.

The higher driving force at the tip is included in the above formula by assuming the driving force to be acting at 60% foil height.

See the appendix concerning evaluation of stresses and application of stress concentration factors.

Stress at the vane base can be estimated from σ_{bi} at the root, by applying the following reasoning: If the root stress is high, the vane stress will also be relatively high. If the root has poor strength, vane stress will still be low, even with high root stress. But if the root is of a high-strength design (3 lands, efficient geometry), then the average vane base stress can be as high as the root stress, or even higher. However, the stress concentration at the root fillet is usually much greater than at the vane base, and once this is taken into account, vane stresses do not come out much higher than root stresses, even with very good roots.

1.1.3 Steam-induced stress, alternating

This is a principal contributor to most blade failures. The steam forces may be strong enough to cause failures of non-resonant blading — for example, where a series of single-jets is used in conjunction with high pressure-ratios (single-stage turbines; Curtis stages). Or where stage loading is extremely high, as with first stages of multi-stage, multi-valve turbines while operating at part-load (one valve open is the worst condition). More often than not, some form of resonant amplification is involved.

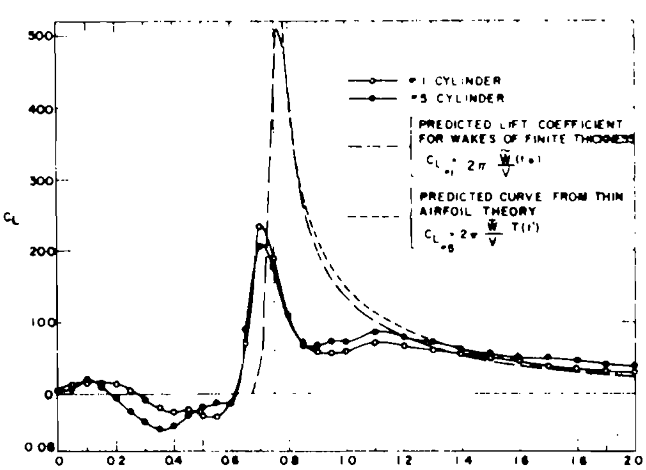

Figure 2a. Unsteady lift coefficient versus time. (15)

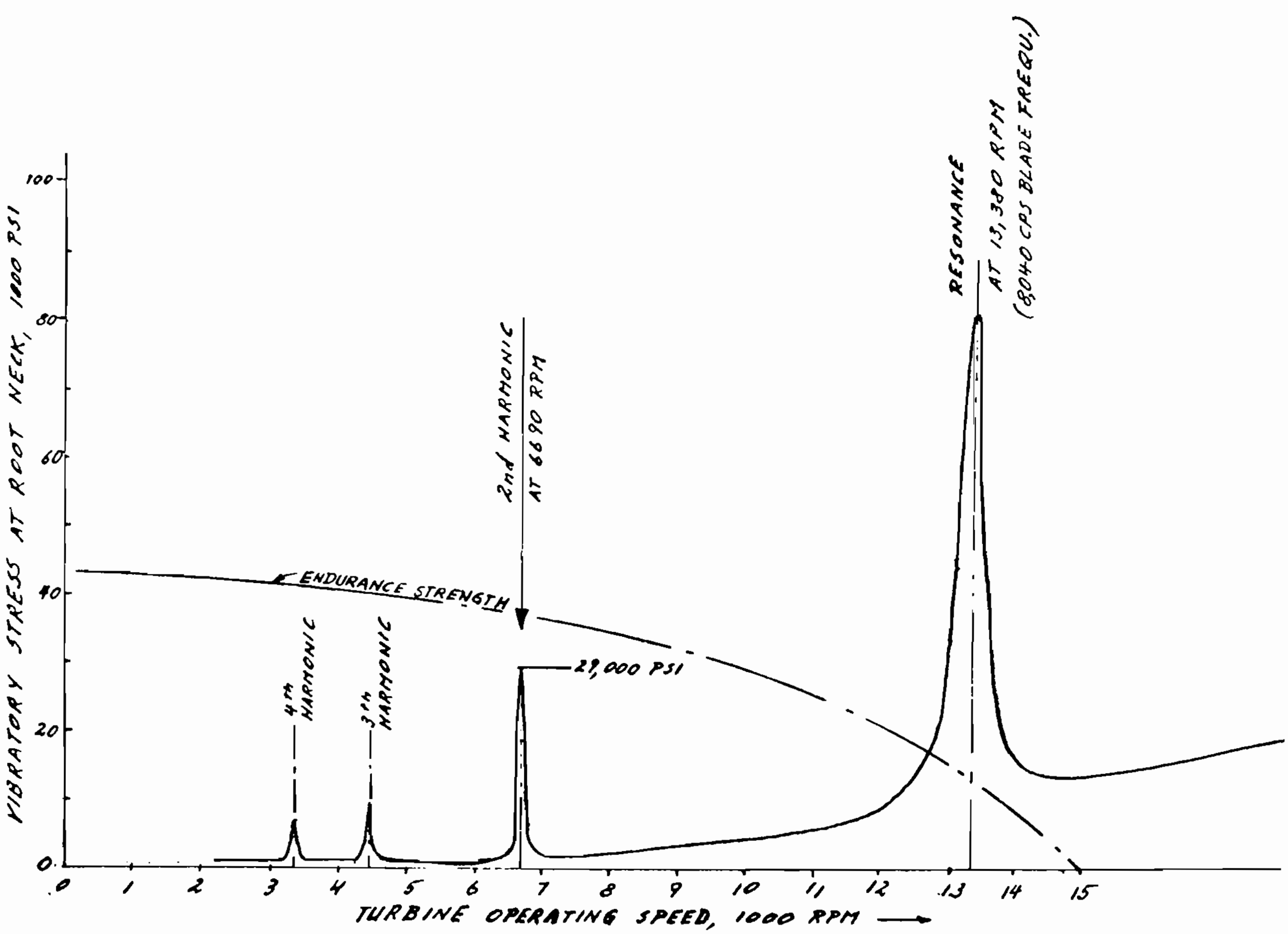

Figure 2b. Vibratory Stress During Starting Cycle. (Stress Shown Between Peaks is Approximate) (7)

The alternating stress can be induced by the following mechanisms:

-Interrupted arc-of-admission, or partial admission (harmonic excitation), see Fig. 4 Amplitudes encountered during start, at points of harmonic excitation, are shown in Fig. 2b.

-Nozzle-wakes (at nozzle-passing frequency). See Figs. 2a and 5.

-Mismatch of diaphragm nozzles at the horizontal split, and/or missing diaphragm blades at split (2 x Rev. exciting frequency.) See Fig. 3.

-Pitch variations of nozzles and/or buckets (harmonic excitation).

-Wakes from struts or braces ahead of stage (usually 2 x Rev to 6 x Rev). These wakes can propagate several stages downstream.

-"Bow wave" from struts or braces behind stage. Distance should be at a radius of at least one blade length from any point of the blade (2 x Rev to 6 x Rev, 4 x Rev being the most common offender, in low-pressure stages).

-Nozzle profiles of poor aerodynamic design (Figs. 5 & 6). Eroded or damaged profiles. Re-worked ("hand-faired") nozzles, especially at horizontal split.

-Insufficient nozzle area ratio, resulting in jet-deflection and buffeting of blades. Occurs at supersonic pressure — ratios

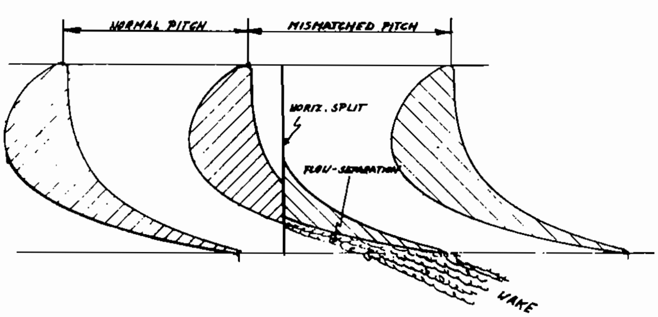

Figure 3. Nozzle mismatch at horizontal split (7)

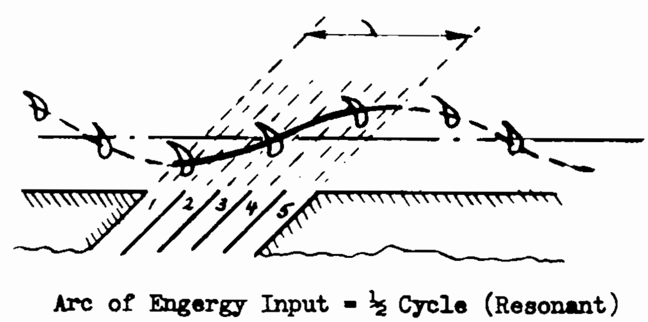

Figure 4. Excitation behind partial-admission diaphragms (7)

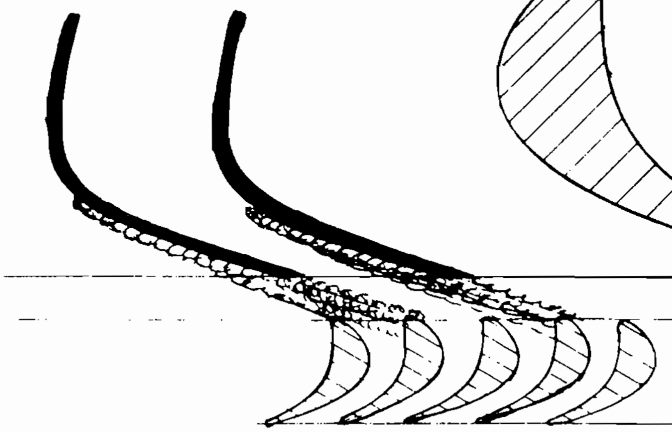

Figure 5. Poor efficiency, flow separation. Strong, wide wake (7)

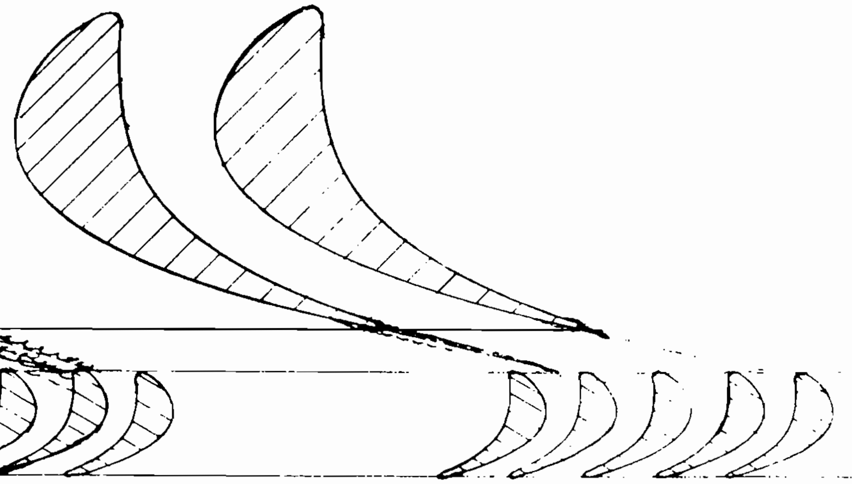

Figure 6. Good efficiency. Narrow, mild wake (7)

if nozzle is not designed correctly. Also with first stages operating at part-load, see Fig. 7. Forces may be high enough to cause non-resonant failure. This can also cause severe wave reflection, Fig. 8, with stimulus as well as frequencies load-dependent, and virtually impossible to predict (10).

-Disk vibration, which also results from steam-induced stresses, both steady and alternating. This has caused many blade failures.

The steam induced alternating stress is expressed as a multiple of the steady steam stress, by the "Stimulus Factor" S. In the case of resonance, S is always used in conjunction with damping, resonant response, and shroud dissipation. There is no easy way to predict stimulus factors, as they vary with each type of excitation source.

See 1.2.4 "Resonant Stress" for range of amplification factors.

1.1.4 Impact stress

Not a normal factor. Results from foreign body contact, such as impact with other failed blades. This may yield the famous "blade-salad," a chain reaction which can rip practically all downstream rotating and stationary blades out of a machine,

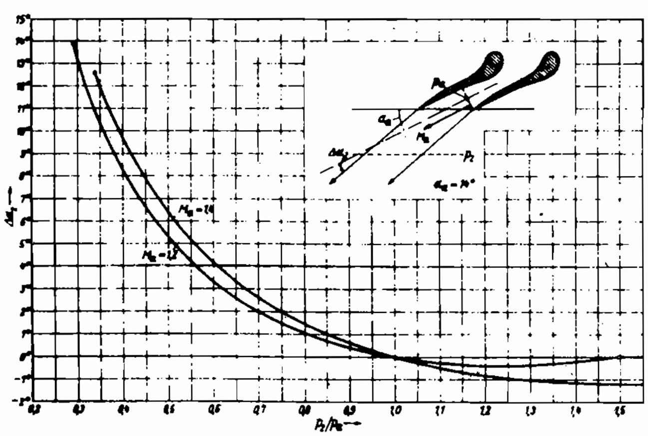

Figure 7. Straight Nozzle. Jet-deflection as function of pressure ratio (6)

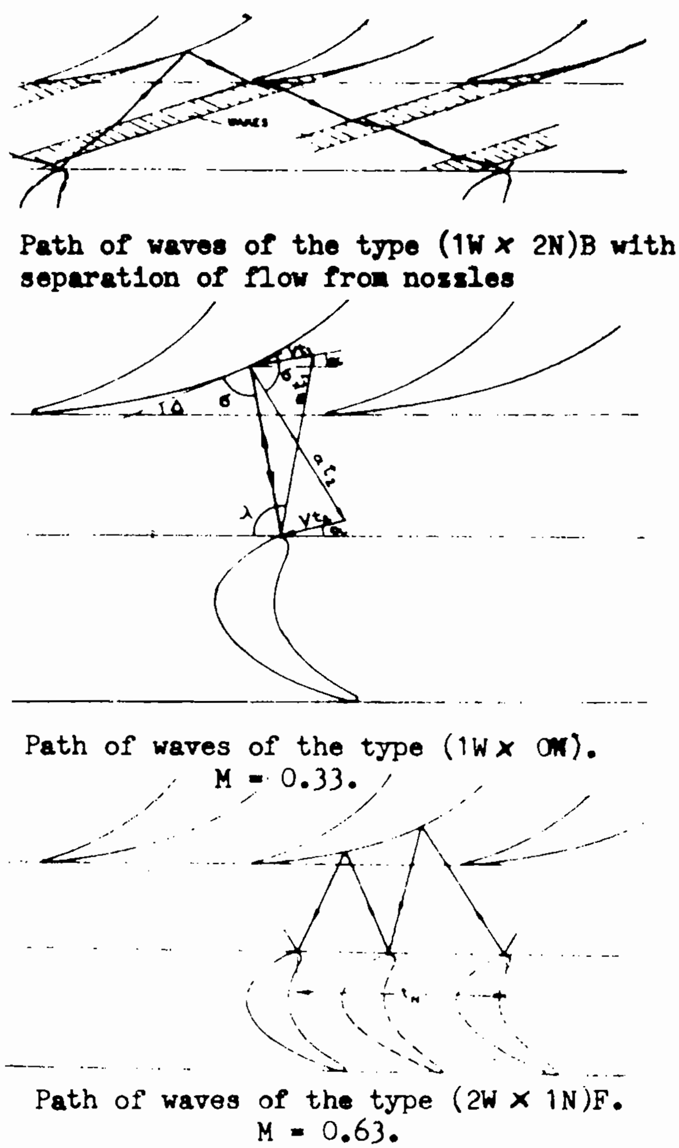

Figure 8. Various types of wave reflection in axial gap (10)

depositing them as a mangled mess in casing, piping, and condenser. This happens mainly with reaction turbines, where the stationary blades (if unshrouded) are not strong enough to stop the avalanche.

Impace stresses also result from water-slugging and from blade rubs (thrust bearing failures).

A high impact strength of the material (over 35 ft-lbs) is very desirable, to minimize the resulting damage if such accidents occur. But the main benefit of high impact strength is the reduced notch sensitivity and the much better over-all behavior of the material, especially in corrosive environment.

Impact results in a characteristic brittle-fracture, which is usually easy to distinguish from fatigue and from ductile fractures, both fracture location and appearance being different.

1.1.5 Low-cycle fatigue

A very important consideration, not always properly considered. It means that failure occurs within a few hundred to a few thousand stress cycles. The failure mechanism is explained in the Appendix.

Low-cycle fatigue or reversing yield can be the sole cause of a failure but, far more often, ultimate failure occurs from alternating steam stresses — however small they may be — after the material strength has been nearly exhausted by low-cycle fatigue. It may require considerable metallurgical effort to properly identify such failures, especially if corrosive action has also been involved.

Causes of low-cycle fatigue are: frequent start/stop operation (=centrifugal stress), the thermal cycling, frequent water-slugging or water washing, inadequate casing drainage (up-exhaust!).

1.1.6 Thermal fatigue

A low-cycle fatigue process activated by thermal stress. It is not a major consideration with present steam temperatures, but quick-starting, water-slugging, rapid and frequent load-changing (compressor surge), and steam temperature cycling can fatigue the high-pressure blades.

The blade foil is the hottest part of the blade, because of energy recovery in the boundary layer (about 70 to 85%). This means the stem in contact with the foil is near nozzle upstream temperature. Shroud, root, and disk are cooled by the much cooler "spent" steam behind the stage. Therefore the shroud will push-pull tangentially on the blade tips as operating conditions vary. If shroud segments are long, rivet tenons or shrouds may fail. Highest tenon stresses occur near the ends of the packets, highest shroud stress occurs near the center.

With the short blades, bending stresses at foil base and root can also be high, especially near the ends of the packet.

1.1.7 Creep stress

Becomes a consideration with metal temperature over 650°F for type 403 material, 750°F for type 422. This affects first stages running with inlet temperatures over 750°F. Pure creep failures are rare with present temperatures, but cracking at stress-concentrations may result if radii are insufficient. The rule of allowing 2 times theoretical yield as max. fillet stress does not apply in the creep range. Therefore, high-temperature blades either need much larger fillet radii at root and vane-base or, for a given blade, allowable stresses must be reduced by at least 50%. Otherwise the material strength at the fillet will be exhausted by local creep, with consequent loss of endurance strength. The blades will fail in pure fatigue.

1.2.0 Resonant Vibration

The theory of resonant vibration cannot be described here, for space reasons. See references for information on this subject. However, we will consider areas of practical importance and briefly recapitulate some basics.

1.2.1 Mode of vibration

-Tangential: Perhaps the most troublesome mode. First and second bending modes, excited by any of the steam forces, have probably caused more failures than all other modes combined. Affects all types of blades, shrouded and unshrouded, but a strong shroud assembly can greatly reduce stresses, by means of load dissipation.

Failures can occur at vane base, root, or shroud, but if the foil fails part way up, a higher mode was involved.

-Axial: Occurs as steam-excited packet vibration (Fig. 9 and Ref. 9 & 12), or as a result of disk vibration (Fig. 10). Both types of excitation may act combined. With steam excitation, failures occur at the end or center of packet. Disk vibration often causes circumferential failure patterns reflecting the disk mode, i.e., 90° and 180° for 2-nodal diameters (a frequent type); 60°, 120° and 180° for 3-nodal diameters (a rare type). Blades may also be loose in the wheel, "lifted," or worn at supports, in such patterns.

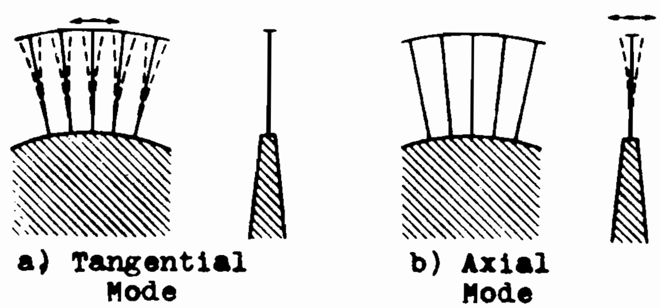

Figure 9. Tangential Mode & Axial Mode (6)

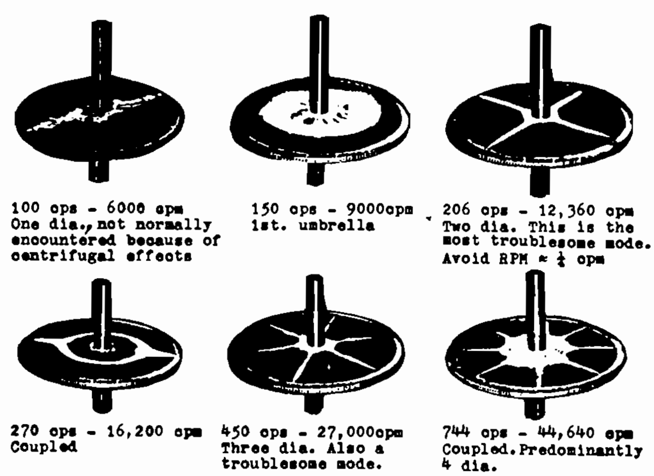

Figure 10. Coupled Disk and Bucket Modes, Example of Nodal Patterns, (made visible by a sand-layer: Electronic excitation). (7)

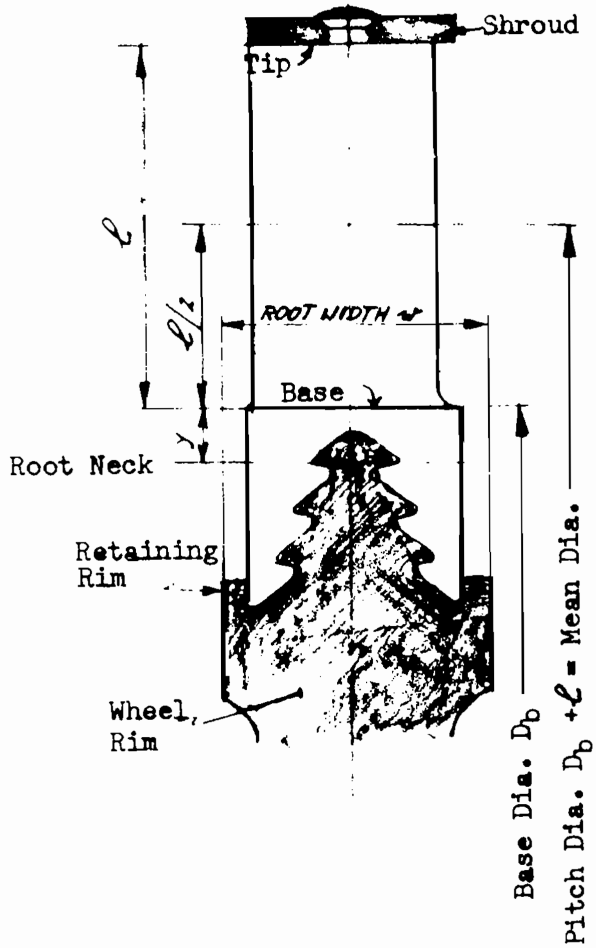

Figure 11. Blade with straddled root (7)

When exposed to any type of axial vibration, straddled type roots (Fig. 11) are obviously much stronger than internal roots (Fig. 12a), and they provide a higher resonant frequency. Very poor are internal roots without retaining-rims (Fig. 12a). With these, circumferential wheel cracking is frequently encountered, Fig. 18.

The failures can be located anywhere on the foil, root or shroud, and they can start on either side (upstream or downstream), sometimes on both. Root failures are, perhaps, the more frequently encountered type.

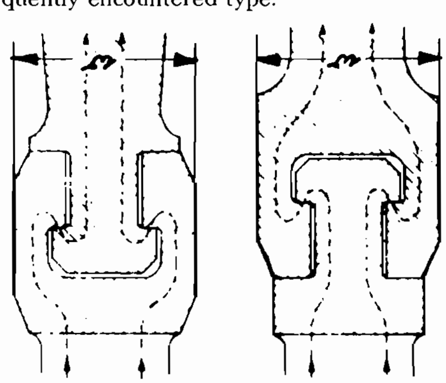

Figure 12a. Comparison of internal ("grooved") and straddled root. Note absence of retaining rims with both types, which would cause disk cracking with the internal type (Fig. 18) root failure with the straddled type. (6)

-Torsional modes: Occur mostly in longer blades. Response is generally lower than for bending modes.

1.2.2 Basic Relationship of Blade-Frequency. (Traupel, 6)

$$\nu_{en} = \frac{\chi_n' \times s}{\ell^2} \sqrt{\frac{E \times k_I}{\rho \times k_f}} \qquad \text{For bending modes.}$$

$$\nu_{etn} = \frac{\chi_{tn}'}{\ell} \sqrt{\frac{G}{\psi\,\rho}} \qquad \text{For torsional modes.}$$

Where:

ν_{en}=Resonant frequency for mode n, cps.

χ_n'=Mode-factor.

s=Chord-length of profile, in.

ℓ =Blade length, inches. Use effective length to point of root support (top land).

E=Modulus of elasticity, at operating temperature, psi.

ρ=Density of blade material

$\rho=\gamma/g=0.283/386=7.332 \times 10^{-4}$

$k_I=I/s^4$ Where I=cross-sectional moment of inertia, $in.^4$

$k_f=f/s^2$ Where f=cross-section, $in.^2$

G=Shear modulus.

ψ=Factor reflection blade cross-section geometry

Since we cannot perform an accurate frequency calculation by hand, the numerical value of these components does not interest us here. The structure of the equations, however, is very important. It tells us the following:

-Blades of different size, but with identical geometric proportions and identical material have χ'_n, E, ρ, and k_I/k_f in common, and also χ'_{tn}, G, and ψ.

This reduces the equations to:

$$\nu_{en} = K_n \frac{s}{\ell^2} \qquad \text{for bending}$$

$$\nu_{etn} = K_{tn}/\ell \qquad \text{for torsion}$$

Where K_n and K_{tn} contain all the constants for a given mode of vibration, including the mode factors χ'_n, and χ'_{nt} for the respective mode.

With this, we can find *all* resonant bending frequencies of any size blade if we know them for just one size,

$$\nu_{en} = \nu_{eno} \frac{\ell_o^2}{s_o} \frac{s}{\ell^2} \qquad \text{for bending}$$

Where subscript $_o$ refers to the original (known) blade.

This equation still contains s/ℓ, because in this form it can also be used as an approximation for blades of similar geometry, where s/ℓ may not be the same as for the reference blade. This results in good approximations for a wide range of turbine blades of the same general type.

For exactly identical geometry, the equation is still rigorously exact and, because in this case s/ℓ will also be identical for all sizes of blades, we derive:

$$\nu_{en} = \nu_{eno} \cdot \ell_o/\ell \qquad \text{for bending}$$

$$\nu_{etn} = \nu_{etno} \cdot \ell_o/\ell \qquad \text{for torsion}$$

Evidently, the scaling factors are the same for both bending and torsion, and therefore also for coupled modes. We can combine the two into one final *scaling equation*:

$\nu_{en} = \nu_{eno} \cdot \ell_o/\ell$ for bending and torsion, all modes, blades of identical geometry.

This relation tells us that the resonant frequency of a blade is inversely proportional to its length, as long as basic geometry is maintained. Since this is the case — within reasonable limits — for all steam turbine blades of a given category, such as, for example, tapered-and-twisted low-pressure blades — we can say that the frequencies of such similar blades are essentially a function of blade length. This is true, to a remarkable degree, regardless of design details and blade origin. The only major variation is the "slenderness" ℓ/s of the blade, and the previous equations permit correction of this factor. With this, basic frequency plots can be prepared, plotting frequency vs. length (7). These can give us an idea of the frequencies to be expected with any typical blade, regardless of manufacturer. Even a dimensionless Campbell diagram can be plotted. Unfortunately, space does not permit to include these graphs here.

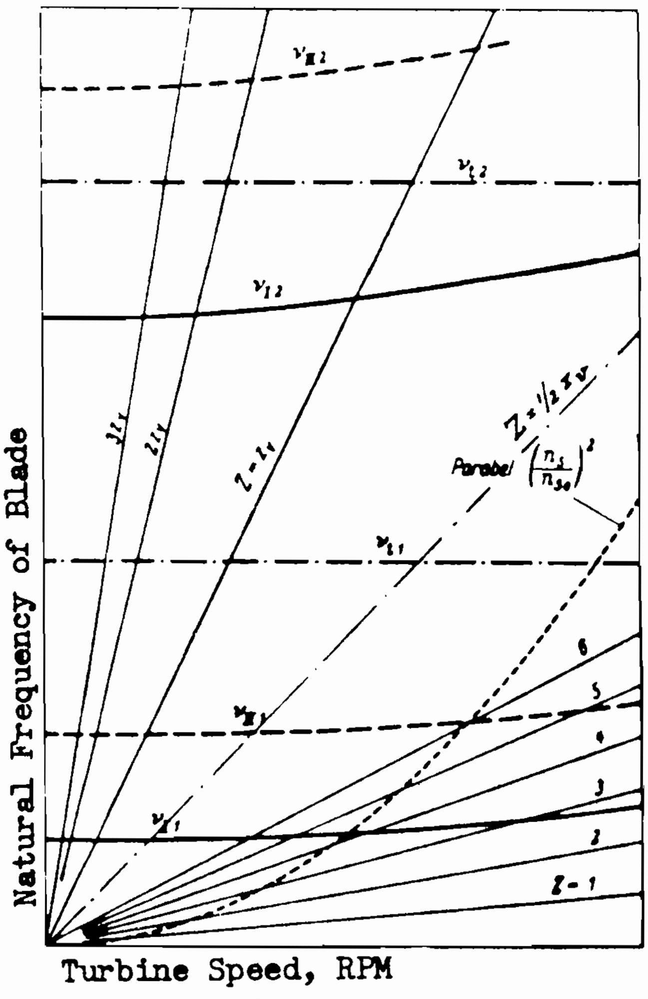

Figure 12b. Campbell Diagram.

Z-lines represent sources of excitation, as multiples of RPM. z_N is the nozzle-passing frequency. ν-lines are blade resonant frequencies. ν_t are torsionals. Note that pure torsional modes are not affected by speed.

Intersections are points of resonant operation. Only detrimental resonances should be shown. (4) (6)

1.2.3 Mode of excitation

See also previous chapters, especially 1.3, "Steam Stress, Alternating." Interaction of excitation with various resonant frequencies is represented in the Campbell Diagram, Fig. 12b.

-Nozzle-passing: Very strong. Direct resonance as well as 2nd harmonics and subharmonics can be devastating if blade structure is weak. Nozzle-passing frequency cps=RPM x 6/angle between nozzles. Note that, if nozzle pitch is non-uniform, several exciting frequencies will occur.

An extremely bad condition results if nozzle-pitch=bucket pitch or, to a lesser degree, with even multiples or fractions thereof (Ref. 17). Also, resonant frequencies will shift drastically. A 10 to 20% margin should be provided.

-Harmonic excitation: With control stages (first stage and extraction stage) and partial admission stages. Also where nozzle pitch is non-uniform. Fourier series is used to find harmonic factors (=severity of excitation for each frequency), and these are compared to blade resonant frequencies. Note that exciting frequency of multi-valve control stage differs for each valve position.

Flow non-uniformities such as windage shields, case openings (extraction, crossover passages, exhaust) also generate harmonic excitation.

-Blading discontinuities are strong sources of excitation. Missing diaphragm blades or pitch errors excite all kinds of resonances. Missing rotating blades at inserting slot excite disk resonances and diaphragm vibration. The interaction between rotating and stationary discontinuities can generate very severe vibrations in both rotating and stationary structures, most pronounced at 2 x Rev and 4 x Rev.

-Out-of-roundness and runout of diaphragm and bucket ID and OD. Gives strong excitation with 2 and 4 x Rev resonances, so does uneven blade tip clearance. Axial gap variations around the wheel cause strong excitation with all axial blade and disk modes.

-Obstructions such as struts, ribs, etc. generate excitation at RPM x 6/angle between obstructions (in cps). The most notorious troublemakers are ribs and struts in the exhaust casing, especially if spaced closer than one blade length downstream of long blades. 2 x Rev, 3 x Rev, and especially 4 x Rev resonances with blade frequency are serious (4 x Rev resonance means blade frequency =4 x RPM). Very bad are ribs and struts upstream of blading, and crowded (=high velocity) crossover and exhaust passages.

-Disk vibration: It is hard to believe that disks can vibrate so severely that blades are fatigued, but it is a fact, and it is fairly common, too. Most of the failures occur at ¼ of disk resonant frequency of the 2-nodal diameter type (Fig. 10c), and some at 1/6 of the 3-diameter type. Disturbances (windage shields or admission arcs) of 45, 90 and 180° (2 diameter type) and 30, 60, 120° (3 diameters) provide strong excitation, usually in conjunction with missing blades and/or pitch variations at the inserting slot. Failures are often 90° (for 2 diameter mode) or 60° (for 3 diameter mode) apart. Blades must be numbered before disassembly and magnetic particle inspection to find this failure pattern. A disk frequency test (Ref. 3) is a simple way to verify the resonances. See Fig. 13 for an idea of the range of frequencies normally encountered. This is sketchy data, but is is the best available.

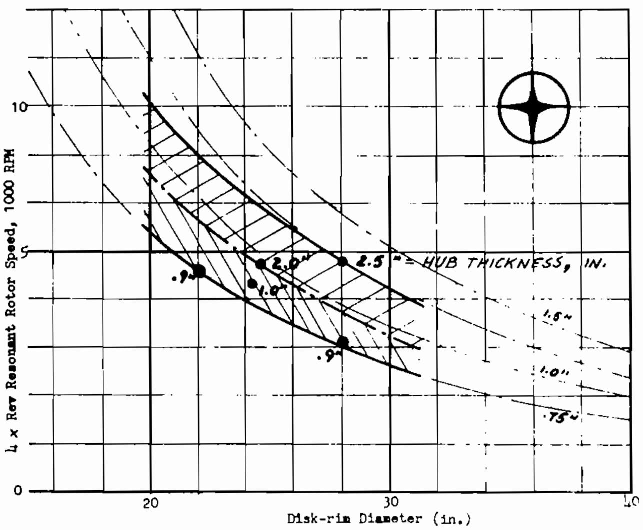

Figure 13. Order-of-Magnitude of Resonant Speeds with 4 x Rev. Disk Frequencies.

2-Nodal Diameter Mode. Bladed, Shrunk-on Disks.

.9" and 1.0" Thick disks are straight sided

2.0" and 2.5" Disks are profiled, thickness given is at hub.

Thin phantom lines are circular plates without holes. (7)

Disk vibration afflicts mostly slow machines (say below 6,000 RPM), having large, thin disks. The ¼ of 2-diameter mode is so troublesome because most MD turbines are small and fast, disks having a higher resonant frequency than generator drives. Therefore the dangerous disk critical speed at ≈ ½ of resonant frequency will not be reached, leaving ½ of disk critical speed (≈¼ of resonant frequency) as the only commonly encountered resonance.

Thick, highly profiled disks do not often cause blade failures. Circumferential cracking of the wheel rim is common if roots are of the internally grooved type, especially if without retaining lips.

Radial disk cracking may occur, starting at keyway corners, at balancing holes or at corners of bucket inserting slots. This is very dangerous because disintegrating disks will, in many cases, penetrate the casing.

-Rotor vibration: Unbalance vibration or any other 1 x Rev. rotor vibration (critical speed) cannot excite blading, because the shaft is orbiting with a bow, and not really vibrating. However, if the orbit, at the stage, is highly elliptical, this can excite resonances of long blading at 2 x Rev and 4 x Rev. Non-syncronous modes can excite blade resonances, if vibration is severe enough. Only long blades are affected, because their resonant frequencies can be low enough to become resonant at 2 to 4 cycles per revolution. One-per-rev resonance is not encountered with conventional M.D. turbines.

-Rotor torsionals: These can cause blade failures if the torsionals are at a high enough frequency to excite blade resonance and if the wheel is located near an anti-node. However, such strong torsionals would probably also manifest themselves by shaft failures, coupling failures, backside contact in teeth of gears and couplings, and by gear and coupling noises.

-Condensation shock: Where steam expands from dry to wet, condensation does not occur at the same location at all times, nor does it occur gradually. Undercooling occurs (Wilson Lines in i-s diagram), but not as a steady-state phenomenon. As a result, a "condensation front" or "shock" is formed, which pulsates in intensity as well as with respect to its location within the turbine, jumping back-and-forth across a stage or even two. Similarly, if the condensation occurs in an expanding (Laval) nozzle, the shock front pulsates back-and-forth along part of the expanding section. This will generate pressure fluctuations as well as wetness fluctuations, which can excite the blade. Furthermore, it is quite unlikely that the front is perfectly flat and perpendicular to the shaft centerline. This means the blade must pass the front two or more times per revolution. We know little about this process — and we could do nothing about it even if we knew — but its presence is another reason to make sure that the best available blading design is used in all stages which may come within the dry/wet region during any operating periods, normal or abnormal.

1.2.4 Resonant Stress

This is very difficult to predict accurately, and we can only present a few basic facts here, to give a feel for the structure of the problems.

$$\sigma_a = \sigma_{bi}\ (\pi/\delta_n)\ Sz\ \alpha z\ H_n$$

Where:

- σ_a = Alternating bending stress amplitude, vane or root, at resonant conditions.
- σ_{bi} = Theoretical (ideal) steam bending stress caused by the driving force. For the single, *free-standing* blade.
- π/δ_n = V_{max} Resonant amplification factor (ranges from ≈ 100 to ≈ 300)
- δ_n = Logarithmic decrement, containing all forms of damping.
- Sz = Stimulus factor, representing exciting forces, at exciting frequency Z (ranges from .1 to .3)
- αz = Load-dissipation factor, for groups of shrouded blades, at exciting frequency Z (ranges from 0 to 1.0)
- H_n = Resonant response factor including mode of vibration (n), type of fixation and shape of airfoil (taper). (ranges from 0.1 to ≈ 0.9).

For a major 1st mode nozzle-passing resonance, by the time all the factors are included, we get a α_a of 5 to 40 times α_{bi}. To get an amplification of 5 requires first class design and workmanship, for a heavily shrouded blade, with 3-land pine tree root and root damping provisions.

1.2.5 Some important facts and figures concerning resonant stress.

a. Resonant speed versus resonant stress.

Since we must accept some resonant conditions, should we prefer a blade with low frequency or one with a high frequency?

We know how load varies with speed in an existing stage, but when modifications must be made, where should we select the speed at which resonance will occur?

Traupel (6) states that, for a variable speed-drive, steam stress α_{bi} — and consequently vibratory stress α_a — varies with the square of the RPM:

$$\alpha_{bi} \approx \alpha_{bio}\ (RPM/RPM_o)^2$$
$$\text{and } \alpha_a \approx \alpha_{ao}\ (RPM/RPM_o)^2$$

The index zero refers to the design point of the blade.

This relation is useful in most cases, but it needs some qualification to avoid improper application:

-The first question concerns the steam stress (or coupling-torque) versus speed characteristic of the entire unit. Evidently, this varies with application. Basically, if we increase speed and hold horsepower or steam flow constant, torque moment — and thereby average steam stress — would *decrease* as a linear function of speed: M_t=63,000 HP/RPM. But in the normal case — with compressors, pumps, or ship-propulsion as a load — the horsepower required by the driven machine varies, more or less, as the 3rd power of speed, and we would need more steam to accomplish a speed increase. The two factors together result in a torque versus speed relation of approximately 2nd power, as was stated by Traupel. If we consider only blade resonance, the resonant stress would vary by this square ratio, as stated above. There are, however, many cases where this simple relation does not hold, for example when anti-surge controls take over, or where we have adjustable compressor blades (axial), or for starting (helper) turbines on trains using hot-gas expanders, and for many other cases. Such applications would have to be studied individually.

To get a feel for the situation, we will use the $\sigma_a \approx \sigma_{ao}$ $(RPM/RPM_o)^2$ relation, but we must keep a few other things in mind:

-Theoretically, the steam load would be zero at zero RPM. This is evidently not true for all stages. The first stage blading of a multivalve turbine experiences its most severe load condition at a very low speed (1st valve open). We can immediately conclude from this that, for first stages, resonant blade stress *decreases* greatly with *increasing* speed and load, and therefore we are far better off to encounter resonant conditions at high speed and high turbine load, when the first stage pressure is also relatively high, reducing the pressure drop across the stage. It can be seen that a wide blade with strong shrouding, and a strong root, offers really decisive advantages for 1st stage service.

-The last stage of a condensing or a non-condensing turbine idles — or even "windmills" — at low load, while at turbine overload it may be exposed to very high loading, especially if the valves have extra flow margin.

Consequently, steam load increases at least with the square of the speed, possibly with a higher exponent.

-The intermediate-pressure stages would experience a steam load variation with the second power of the speed ratio, essentially as assumed by Traupel.

Within the above limitations we can now summarize as follows:

-The resonant frequency of a blade of a given length varies as a direct function of blade width, and therefore the RPM at which resonance occurs is also directly propotional to blade width.

σ_{bi} and σ_a are proportional to $(RPM)^2$
-σ_{bi} and σ_a are proportional to $(1/s)^2$

And we can conclude:

"The resonant stress of a blade of given length will not vary if the resonant frequency (=resonant turbine speed) is varied by changing the bade width. This is essentially true for intermediate and last stage blading.

In other words, no matter at what speed we design the blade to be resonant within the operating range, the stress will be about the same. However, manufacturing errors, surface finish, handling damage, nicks, erosion, etc. will have a stress raising effect which is inversely proportional to blade cross-section. Also, a small blade will encounter resonance at a low speed, and higher modes may come into the operating range. Both factors are strong reasons to favor the larger blade section and thereby a high resonant speed. The reduced centrifugal stress at resonance is the only consideration in favor of a lower resonant speed. However, this should not be a controlling factor. If it is, then the blade design is most likely marginal to begin with.

For torsional modes, the resonant speed can only be varied by a change of taper or profile geometry. A change of s only has no effect on the frequency, but the vibratory stress at resonance is proportional to 1/s.

b. Effect of number of pulses/revolution (=number of nozzles/circle) upon resonant stress.

An increase in the number of nozzles/circle will lower the resonant speed, and thereby resonant stress:

"Resonant stress is inversely proportional to the square of the number of nozzles (or pulses) per revolution" —

$$\sigma_a = \sigma_{ao} \left(z_o/z\right)^2$$

The number of pulses/rev. for partial admission always refers to the theoretical number, as if the admission were 100%.

This is, of course, a very effective factor. However, the risk of exciting higher modes is again in the picture, as is the problem of manufacturing errors with the smaller nozzles. But for problem correction this possibility should always be considered, except for first stages, where a higher resonant speed is desirable.

This appears very attractive at first sight. But to get the necessary quality of workmanship in the smaller nozzles (thin exit edges, split-matching!) — and then to maintain it in the field (erosion, deposits, handling, maintenance) — is no minor problem. As we know, all the above benefits can be lost by relatively minor discrepancies of the aerodynamic profile of the nozzles, and then we are facing stimuli of unknown magnitude and frequency. So, evidently, a reasonable compromise must be found to minimize this problem. Diaphragm deflection, stress, and vibration are other factors in this picture (these vary with the 3rd and 2nd power of the number of nozzles).

However, one very important point emerges from this consideration: If a known resonance exists and it has to be shifted to a more convenient area within the operating range (assuming we cannot shift it above or below the operating range) by a change of nozzle pitch, the resonance should always be lowered — the number of nozzles increased — never the other way around, except if strong higher modes must be considered. If failures have occured and the resonant speed were to be increased, failures would then recur after a much shorter period of resonant operation, possibly even as a result of momentary resonant operation. This holds for intermediate and last stages of multistage turbines, and for single stage turbines. For first stages, the opposite is true.

1.2.5 Summary of Means to Adjust Resonant Frequencies and Stress.

a. Change blade frequency by changing cross-section (s).

b. Change exciting frequency, for example, number of blades per circle of the diaphragm. Resonant stress varies with square of nozzle pitch, for all but 1st stages.

c. Use airfoil damping (lashing) wires.

d. Use root damping wires.

e. Combine individual blades into groups, using shrouds and/or lashing wires. Use stronger shrouding.

f. Use better material (titanium)

g. Use axial pins through base. This will raise frequency and improve mode factor H (fixed at base) and it will provide significant damping. This has proven very effective after blade failures, especially of the types induced by disk vibration and shroud group vibration. This fix can often be made in the field, on the spare rotor, or with a minimum of delay at the factory.

h. Reduce exciting stimuli by avoiding ribs, out-of-roundness, mismatch at split. Avoid damaged or inaccurate nozzle profile and variations of nozzle geometry. Provide very precise manufacturing techniques for all contours in the steam flowpath. Avoid hand-blending and/or "touch-up" of steam-path contours. Providing first-class hardware to begin with.

i. For last stages, use exit diffusor (machined!) to reduce buffeting during off-design operation, and to obtain smooth flow behind the blade.

j. Last but no least: Provide a high degree of manufacturing accuracy and quality control to obtain dimensional accuracy, adequate surface finish, and absence of accidental damage.

k. Provide means and procedures to maintain this high degree of quality during operation and maintenance.

1.3.0 Operating Environment

1.3.1 Stress corrosion:

Rare with blades, but fairly common with rotors, where it has caused some catastrophic failures. Fracture would be intergranular and branched. Usually the part disintegrates into many pieces. It is caused in stressed parts by corrosive steam or by standby in corrosive environment (chlorine in moist atmosphere, for example; or leaving a turbine open with inadequate weather protection). At standstill, the necessary static stress component is residual stress, mainly at or near areas where local yield or creep has occurred during operation.

1.3.2 Corrosion-fatigue:

Perhaps the most common single factor in blade failures in the wet region of turbines, affecting especially the dry/wet area near the saturation line. It is well described by Barer & Peters, (Ref. 1):

"—It is commonly accepted engineering practice to consider that the "endurance limit" of steels is approximately half their ultimate tensile strength. Of course this is lowered by even minor imperfection in surface finish and by other variables which may affect the surface. *It is also generally accepted that there is no endurance limit when any corrosive action is present; that is, the level of stress which the sample will withstand continues to fall with increasing number of cycles.* To give some feel for the magnitudes involved, a steel of 200,000 psi ultimate tensile has an approximate endurance limit of 100,000 psi (in a smooth surfaced fatigue specimen) and a value of some 8,000 to 10,000 psi after several million cycles in sea water. This compares with some 30,000 psi for monel or 60,000 psi for titanium under the same number of cycles in sea water. Corrosion and cycling stressing acting jointly are more severe than might be expected cumulatively from each. One factor which influences this could be inability of the metal surface so affected to either develop a protective oxide film or even a partly protective corrosion product film. In fact, it seems likely that corrosion products would function in a wedging fashion to intensify stress at the advancing front.—"

"—*Corrosion fatigue is usually transgranular, displaying thereby little concern for the microstructure.* The transgranular behavior demonstrates the primacy of the mechanical or stress cycling component. It is also likely that some *intergranular action might accompany corrosion fatigue where the fatigue component is small or absent* for various periods. Along similar lines is the corrosive action proceeding at right angles to the corrosion fatigue crack.—"

This is a pretty grim picture indeed. To get any definitive safety factor at all, we must either keep the steam completely free of chemicals, or completely eliminate all alternating stresses. Otherwise we will be confronted with the fact that the material has no endurance limit, blade life being limited to a certain number of cycles, see Fig. 14. Considering that the nozzle-passing frequencies are in the order of magnitude of one half million per minute, and more, even a life span of 100 billion cycles is no consolation, because it would cause a plant shutdown about every half year.

Evidently we must compromise, keeping the steam as non-corrosive as possible, while using the strongest possible blade structure and the best quality available, in conjunction with a low steam load. The dry/wet stages cannot tolerate significant resonances. This may require limitations on operating speed and load, especially while passing resonances during the starting cycle. Standby (crevice) corrosion must be avoided at any cost.

Titanium alloy blading is far less sensitive to corrosion-fatigue, titanium being essentially a noble metal. This, in conjunction with its much lighter weight and much better strength and ductility, makes titanium an ideal solution for problems in the dry/wet and wet area. Improvements of safety factor between 5 and 15 are feasible, depending on conditions. Operating experience — although limited to a few hundred wheels — has reportedly been excellent.

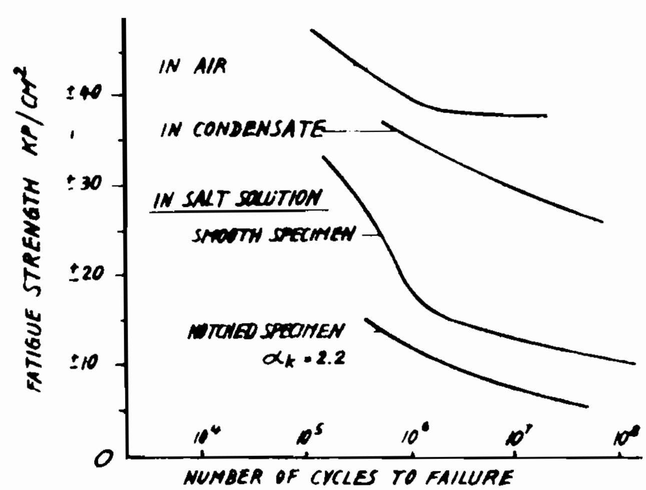

Figure 14. Effect of corrosives on fatigue strength. 13 Cr stainless steel.

With increasing material hardness, 12% Cr Stainless (types 403 and 422) becomes progressively more sensitive to corrosion fatigue. Therefore, yield strength levels much over 90,000 psi can be quite detrimental.

Silica in the steam forms deposits which can trap the corrosive chemicals, making the situation much worse. See "crevice corrosion."

The more common offending chemicals are:

-Chlorides, carbonates, CO_2, caustics, etc., from boiler feed water treatment or raw water. Gets into turbine by means of carry-over, priming, etc. May also be injected into steam in desuperheater, if boiler feed water or contaminated condensate is used. Boiler feed is often injected during plant start-up, which is especially bad, because this is when resonances are encountered in many stages. Also, the resulting contaminated condensate is being recirculated into the desuperheater. Seawater leakage into condensate is another source of contamination.

-Sulfides (Hydrogen sulfide) and syngas (ammonia), by way of leaking heat exchangers.

The weakening effect of corrosives has been included in the Goodman diagram, Fig. 1a. Both this figure and Fig. 14 are not very satisfactory because we have no distinction of chemicals or concentrations. However, it is based on the best data available, in an attempt to get some usable numbers reflecting conditions in an average process plant. It is felt that these stress levels will allow about one to ten billion cycles.

A few minutes of exposure to chemical action (=a few million cycles) can be enough to cause failures, if conditions are unfavorable.

The typical fracture appearance shows all earmarks of fatigue, including transgranular cracking. If no more corrosion symptoms are evident, it may take examination by means of a scanning electron-microscope to distinguish between fatigue and corrosion fatigue. However, quite often there are other indicators to make this unnecessary:

-Fracture-surface discoloration: Can range from deep purple to light blue, and from deep tan to a shiny gold. Virgin fractures must be examined, breaking cracked blades after cooling them in liquid nitrogen. Sometimes a good blow with a sledgehammer will do, without cooling. If beachmarks show boundaries of varying color intensity, the number, duration, and severity of chemical exposures can be estimated. Remember that the fractures cannot have blushed from heat. First of all there is not heat in the low-pressure end (light blue $\approx$ 600°F), secondly this would require air in the crack, which is also not present.

-Branched and/or (partial) intergranular cracks.

These symptoms permit positive identification. Another clue is the crack origin at a corrosion-pit, but this is not positive proof. To round out the picture of the corrosion-fatigue syndrome, here are some additional items of evidence:

-Boiler running at a high steaming rate.

-Problems started after steam system modification.

-Failures occur in certain seasons (cold winter!)

-Turbine internals corroded. Corrosion pitting on wheels, shaft, casing.

-High-pressure and intermediate blading coated white, with salt (taste) and/or silica deposits. Low-pressure blading is black, brown or clean (deposit washed off by moisture). Failures are in white-to-dark transition region.

-Repeat failures in the same stage even after blading has been improved. Especially if resonant frequencies have been changed.

-Failures have occurred in several rows in the wet end. These blades cannot all have been resonant at the same time (lengths are different).

-Turbine has been started frequently or has been operated at off-design conditions, especially if this has shifted the dry/wet line into the failure zone.

To detect the presence of chemicals, periodic sample testing is required. Conductivity of steam and condensate should be continuously recorded at several locations in the system.

1.3.3 Standby Corrosion

Blades fail from fatigue, initiated by crevice corrosion or surface pitting. Occurs during shutdown (a few days are enough, sometimes only hours) with air, moisture, and corrosives in the turbine. Typical settings are:

-Drains going to a common header or sewer, perhaps even into a compressor drain containing gas. All drains should be individual and visible.

-Sealing steam not turned off; inlet valves or extraction valves leaking.

-Moist climate, corrosives in air (chlorine, ammonia, hydrogensulfide, nitrogen oxides).

-Turbine filling with water. Up-exhaust turbines are notorious for this (on one job turbine filled with seawater leakage every week — to the top). Watch for water coming out of seals, and for water-level marking on internals.

-Trip and throttle valve stem leakage piped to gland condenser from there the vapors back up into the turbine.

-A common condenser serving more than one turbine. One machine running, the other shut down.

Appearance of all internals is rusted and pitted, with no preference for hot or cold section. But either top or bottom may be worse.

Crevice corrosion occurs if corrosives get into root crevices (this in itself is a prime piece of evidence). The process is well described by Barer and Peters (1):

"—An interesting, and in some ways insiduous type of action, is known as crevice corrosion. This is the corrosion that occurs under debris on a surface that is immersed, under washers or gaskets, or between sheets of similar metals. It is the effect which can be harnessed to cut a piece of immersed stainless steel with a rubber band. (The rubber band shields the metal beneath it, which then becomes an actively corroding area which in time crumbles and the rubber band slowly advances.)

"The action is related to the difference in oxygen available in the freely aerated surrounding solution and the more limited supply in the film of liquid trapped in a crevice. The resultant effect is particularly strong with those metals such as stainless steel and aluminum which depend on oxygen availability to repair weak areas in their protective oxide films. Under a deposit or other crevice, there is insufficient oxygen and, coupled as it is metallically to the rest of the metal which has access to plentiful oxygen, there is a strong tendency for the sheltered metal to become an anode and the remainder, a large cathode. A corrosion cell is thereby set up and the action can be intense.—"

The blade failure — which occurs much later, in service — shows the typical symptoms of corrosion-fatigue, or even stress corrosion. But there are strong signs of corrosive attack in crevices, and deposits of the offending chemical are present.

Standby-corrosion may cause stress-corrosion cracking of disks, especially inside the keyways. This is exceedingly dangerous, since often it cannot be seen without rotor disassembly.

Prevention, in order of preference:

-Nitrogen-purge. Start purging immediately after arrival of turbine on jobsite, because much irreparable damage occurs during storage and erection.

-Instrument-air purge.

-Hot air ventilation.

-Use of ventilation stack in conjunction with drains to set up air circulation. OK if air is clean and dry.

The steamline between shut-off valve and trip valve should be ventilated (stack and drain).

The gland condenser and its drain and water seal deserve special attention, as corrosives and moisture can be sucked into the machine via this route.

Watch any drains which are combined with others and/or not fully visible!

1.3.4 Erosion by Water Droplets

Causes failure by means of nozzle-erosion and consequent, strong blade excitation, at unpredictable frequencies.

Eroded buckets may fail from the effect of stress concentration generated by the erosion.

Failure appearance can be pure fatigue or corrosion fatigue.

Remedy: get rid of water by means of drainpots and drain in each wet stage, as well as by other erosion-preventing measures (check for steam quality, water leakage into up-exhaust turbines, etc.).

2.0 *Engineering, Design, Service Conditions*

General: The following explanations will concern only selected items from the table. Correlation of the other items will be self-evident when referred to.

2.1.1 to .3, Blade design:

Design features most frequently causing failures are:

-Insufficient radii at fillets.
-Square shroud holes, especially with sharp corners.
-Top of foil not contoured for shroud radius (Fig. 15).
-Inserting-slot too large, weakening adjacent blades.
-No root retaining lip (Fig. 12a).
-Root lands not curved, giving edge-contact (Fig. 16).
-Excessive tolerances and rough surface finishes.

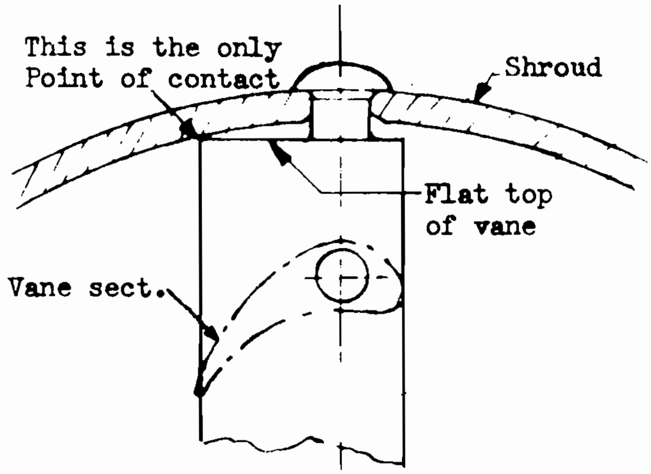

Figure 15. Incorrect Shroud Assembly (7)

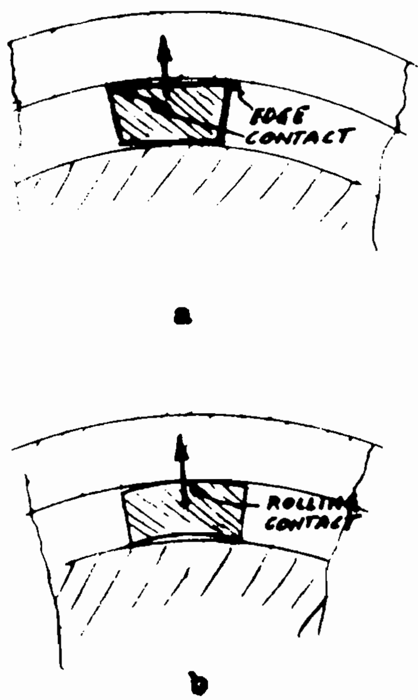

Figure 16.
a. Edge contact in straight roots
b. Reduction of stress concentration by means of undersized radius-of-curvature. This reduces risk of failure very significantly, especially with small wheels and/or larger blades. (7)

-Nozzle pitch variations.
-Ribs and braces in steampath.
-Excessive pressure-drop for nozzle area-ratio (expansion ratio).

2.1.4 Staging, common problem areas

The first and last stages "take the swing" when throttle flow changes: At low flow, the last stage no longer gets sufficient steam volume to build up an effective pressure drop, and it begins to idle, or "windmill" (i.e. to consume power instead of generating it). The intermediate stages now all operate at a lower pressure, but quite efficiently and with good pressure ratios, because the lower pressure gives them enough volume flow (cu.ft/sec.) to make up for the reduced throttle flow (lbs/hr). The first stage pressure (behind the first wheel) is greatly reduced, about proportional to throttle flow. The worst condition for the first stage buckets exists when the first valve of a multi-valve turbine is just wide open. Then the first stage pressure (p_2) is still quite low and there may exist an enormous pressure drop across the nozzles. The low blade exit pressure plus the high pressure drop (and corresponding BTU drop Δ h) cause a very high bucket exit velocity, and thereby a most severe stress condition — far worse than during normal operation. At turbine overload, the first stage is very lightly loaded or even idling. In addition, because the nozzle pressure-drop is so very high (for example 10/1 for 1,500 psi to 150 psi), we have hypersonic flow conditions with a high jet-deflection, afterexpansion, supersonic shock fronts (and their reflections), a shock front across the bucket inlet passages, and a high pressure-drop (=reaction) across the buckets, resulting in — among other things — a high pressure-drop across the shroud, as with a pressure vessel wall. This can simply lift a riveted shroud off the blade, fatiguing the tenon or the shroud itself.

An integral-plus-riveted double shroud is much stronger in this respect.

With compressor drives, this condition occurs at process startup, at reduced speed. Then M_2 is even higher, and with it the stress levels. Also, resonant frequencies will be encountered. In other words, the blades get an unbelievable beating under such conditions, and the machine may run like this for hours. Where this kind of operation is required, throttling with the trip-and-throttle valve would help, by reducing the nozzle inlet pressure and thereby M. A better solution would be to arrange the valve opening sequence so as to lift the first few valves in parallel.

The situation described above is, of course, a main reason for the many first stage blade failures encountered in the past. Where this condition exists, an analysis of first stage bucket stress under normal operating conditions is of little value. For turbines to be operated at part load, the one-valve-open condition must be checked. A first stage of a variable-speed turbine should be strong enough to operate continuously on any resonant frequency, for all predictable operating conditions.

Intermediate stages:

The intermediate stages are protected against overload by the first and last stages of the turbine. During off-design operation, steam stress cannot get out of hand in the same manner as with the first and last stage. Stages with partial admission may have a high oscillating steam load, but the main problem with intermediate stages is corrosion fatigue.

High steam stresses may occur in the last stage before a controlled extraction — which is essentially a last stage (an extraction turbine is composed of a high pressure turbine and a low pressure turbine in tandem), therefore such stages are exposed to the same hazards as last stages of back-pressure turbines. Especially operation with low extraction-pressure during start is risky.

Split-flow machines (such as turbines having double-flow or triple-flow exhausts) usually have a highly loaded stage before the flow division. At this point the blading is generally relatively long, and at a relatively high pressure. Often, this stage also has a high BTU drop because it is desirable to have the double-or-triple-flow division as far downstream in the turbine as possible, for reasons of cost, compactness of the turbine, and critical speed. This stage deserves special attention, especially if it lies in the dry/wet region. Because axial spacing is at a premium in a high-speed turbine, sometimes the exits of such stages are also obstructed, and perhaps even partially blocked with ribs, pipes or other structural items. If the crossover is internal (no external pipes) strong 2/Rev. and other harmonic excitation will result from the flow assymmetry.

It is important to keep in mind that all the steam discharged in one half of a stage must cross the horizontal splitline to get to the extraction-opening or cross-over opening in the other half of the turbine casing. It is a good idea to make a simple check of the steam velocity across the horizontal splitline, to see whether velocity (=pressure fluctuation) is excessive. Remember that the steam flow coming from one half of the case may run crosswise to the steam flow discharging from the blading in the other half. This can cause flow obstruction and pressure fluctuation at the bucket exit, and it can be a major stimulus for bucket vibration.

Multivalve re-admission stages behind a controlled extraction must be considered in the same way as first stages. At minimum flow to condenser the stresses can be very high, because of the combination of the relatively long blades and high Mach Number.

Last stage:

-Condensing. The hazards of overload caused by excessive vacuum have been mentioned before. Ribs in the exhaust case and crowding of steam flow across the split are also a very important factor. If the stage is properly designed, and run at normal or maximum operating conditions, the steam stresses on the blading are relatively low, because of the low exit pressure p_2. Centrifugal stresses are the main factor in this stage and, of course, vibratory stresses, because the net amplification factor is high.

However, the last stage may be crowded ($M>1.15$), even at normal operating conditions, if the blading is too short for the steam flow. Then, the exhaust pressure at the blade throat is considerably higher than the condenser pressure, and the steam will expand again after leaving the bucket throat, as it enters into the exhaust hood behind the blades. This is essentially the same situation as would result from operation with excessive vacuum, but it is even worse because the throat pressure in the bucket is higher than designed, in addition to a possibly very high Mach Number. Usually the exhaust casing is also built undersize with such machines. This can be verified by checking the steam velocities across the horizontal split and at the exhaust flange.

If failures are caused by this mechanism and/or overload, the exhaust pressure can be increased so as to get M_2 down to 1.15-1.25.

-Non-condensing turbines:

Conditions are similar to condensing turbines, especially with respect to operation at the proper backpressure. Too low a backpressure can cause the strongest buckets to fail. Warning devices for low backpressure would be very valuable, but are seldom used in practice. The last stage blading of backpressure machines is often relatively short and there is a danger of excitation at the nozzle-passing frequencies as well as harmonics or sub-harmonics thereof. Both steam stress and centrifugal forces are important. Corrosion fatigue is often a problem because the last stage may operate in the dry/wet region. If failures persist and temperatures are not too high, titanium blading may provide an answer in such cases.

2.2 *Material*

Quality is important, not just physicals and chemicals. Material of all failed blades should be identified. Not that high yield strength and hardness will not improve the endurance strength beyond a certain point, if corrosive environment is considered. If yield strength is much above 90,000 psi, material becomes susceptible to corrosion-fatigue.

12% Cr Stainless, types 403 and 422 (hot region) are standard, but do not overlook the possibilities of titanium in the lower temperature zones, especially below about 450°F. It is by far the best way to get out of serious corrosion-fatigue and erosion problems, since it does not require time-consuming design changes.

2.3 *Manufacturing*

About all items listed occur frequently, with careless assembly, surface imperfections, and dimensional errors heading the list.

2.4 *Stage Environment and Operating Conditions*

Most frequently, improvements can be made by avoiding operating upsets (don't "jockey" the machine at the slightest pretext), by providing clean steam and correct steam conditions. It is sometimes incredible how much actual operating conditions (vacuum!) vary from design.

2.5 *Maintenance*

Frequent problems arise from:

-Improper blade replacement procedures (locking pins not fitted right, inserting slot elongated).

-Not all blades of a wheel replaced after a fatigue failure. This is a frequent mistake. And then people are surprised when they have one failure after another!

-Abrasive blasting used to remove deposits. A very common problem. It can change the contours (trailing edges), leaving stress concentrations. Shot-peening layers will definitely be damaged, leaving the blades with residual stresses and minus the considerable benefit of the peening-layer.

3.0 *Symptoms and Evidence*

3.1 *Failure location (fatigue failures), on blade assembly*

This can provide good clues concerning the destructive mechanism.

-Failure in shroud or lashing is proof that the shroud or lashing assembly was too weak to perform its function to dissipate oscillating loads (by averaging the pulsations on each blade in such a way as to cancel the total for the entire packet — as far as possible). This could be a result of design weakness, unexpected resonances, corrosives, or manufacturing deficiencies (shroud not pulled down). Usually several of these factors are involved.

-At foil base or root: With long blades this indicates a probability of resonance. With shorter blades, it is a sign of insufficient shroud stiffness, or of a poor shroud dissipation factor (α), or of a weak blade design (too narrow, poor root). Always check for poor shroud design and/or assembly, and for fretting marks at all contact points.

-Foil, other than base: Higher mode resonance.

-At or near locking piece: Slot too long, pin improperly fitted, blade (if used) at end of shroud packet (=high stress), disk vibration (locking blade is source of excitation).

-Wheel root: Very rare with straddled blades. Common is circumferential cracking of wheel rims with internal roots not having restraining lips to hold the rim together (a low performance design). See Figs. 17 and 18. These failures indicate see-saw modes of the packets (which are encouraged by these weak rims), often in conjunction with disk vibration, corrosive action, poor shrouding.

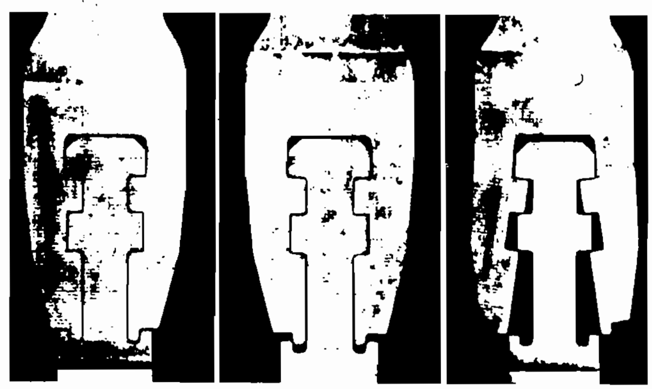

Figure 17. Pull-test showing effectiveness of retaining lips, preventing rim-bending. (6)

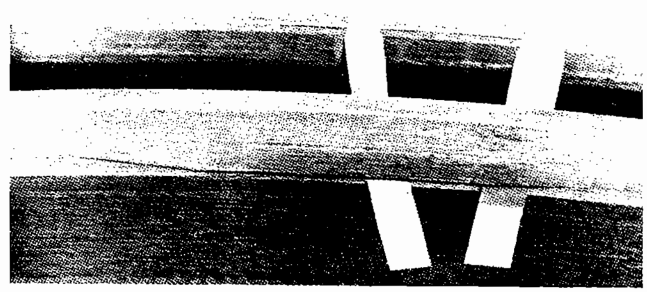

Figure 18. Typical rim failure caused by axial vibrations and lack of retaining lips (see Figure 17). Shims in crack are 4 mils thick. (8)

3.2 *Fracture Analysis*

See "Failure Mechanisms" for correlations. One can get an idea of fatigue damage accumulated in apparently healthy blades by holding the root of a blade firmly (!) in a vise, with top of jaws at failure location. Then bend blade with a long pipe and/or hit hard with a sledgehammer. This sometimes tells a lot about remaining ductility. Some of these blades may break like a poor grade of cast-iron (especially the ones having cracks), showing very little ductility and a very coarse fracture surface. This gives a clue about magnitude of stress-levels involved, and their distribution around the wheel and within the packets.

3.3 *Surrounding Evidence*

Most correlations will be self-evident from the table. Note that water slugging will bend the blades backwards.

If blades are loose in the wheel, look for a disk-vibration pattern, and for evidence of overspeeding (reverse rotation). Otherwise either poor fit or excessive stress, centrifugal or steam. Can be caused by water slugging or water washing.

Axial rim rubs not uniform around the circumference of rim, shroud, or blading may indicate heavy disk vibration. Look for pattern.

3.4 Failure History

Failure shortly after original start-up, or while bringing the plant to capacity, often indicate inadequate design and/or construction, but also steam system problems. Failures may occur after steam system modification (more load on boiler, new boiler onstream, pipe slope shifted and dumping water).

3.5 Operating Symptoms

Blade failures give little or no warning, but sometimes one can spot conditions which may lead to failure. A very high-pitched noise may indicate disk vibration. If so, it would disappear completely with a minor speed change (10-20 RPM), as these resonances have a very sharp peak.

Taking a vibration spectrum on shaft, case, or steam (noise) is probably the best way to get some data suitable for monitoring, See Fig. 19 and Ref. (2).

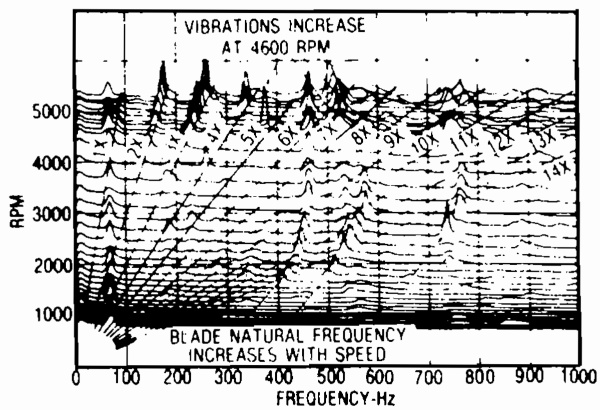

Figure 19. Excitation of lower blade natural frequencies as a function of speed (Hydrocarbon Processing, April 1974).

Compressor surging, if prolonged, can be murder for the blades if it causes speed cycling and/or governor action, because it is very likely to cycle one or more stages through some resonances.

Note that water slugging can be detected by watching the governor valves. As the water slows the unit down, the governor valves will open wide to maintain speed, then close, and finally come back to normal. A valve-position recorder (must be *fast* — no 16-point recorders!) can be used to verify this. A real fast thermocouple in the steam line can also be used, but it is not as foolproof.

Water slugging is also a prime cause of rotor-bow, casing distortion, nozzle-bowl-cracking and thrust failures (always on active side). This gives another clue.

4.0 Investigation Procedure

It is obviously not possible to cover all types of problems and possibilities. A few general rules are given below, for a grass-roots type of investigation. See also notes on chart.

4.1 Inspection

-Open turbine as soon as possible after failure. Be present when turbine is opened.

-Keep away people with rags, scraping knives, and an uncontrollable urge to touch things — such as failure surfaces, coatings, etc.

-Take photos, notes, samples of deposits. Make sure to get precise condition and position of nozzles, windage shields, flow obstructions. Wrap fractured blades in *clean* rags, then put in box with small bag of silica gel ("De Moist," available in hardware store), seal box with tape.

-Spray thin film of clear lacquer on exposed fracture surfaces in rotor (later to be removed with acetone).

-Remove rotor to shop for disassembly. Don't let people touch anything they don't have to handle! Don't allow blast-cleaning, oiling, or any surface treatment.

-Carefully inspect shroud assembly (check for clearance with feeler-gauge). Take sample of any deposits under shroud, check for looseness, rubs, unusual flow-markings, tightness of locking assembly. Check for tight packing in wheel (feeler gauge). Bump-test wheel (3) to get an idea of disk frequencies. Note sound of wheel and blades when tapped. If wheel sounds dull, make sure to magnetic-particle check for cracks, especially keyways, shaft-disk junction (if integral), inserting-slot area.

-Number each blade at top and bottom using electric pencil. Make map, to full scale, showing packet locations.

-Remove blades very, very carefully, paying special attention to inserting area. Be there to witness. Make sure to give complete instructions. Don't allow lubricants (penetrating oil) unless there is absolutely no other way. Inspect each blade as it comes out. Put in special, flat, clean box in same circle as original assembly, print of map on bottom. Check each blade for tightness of assembly before removal.

If blades are shrouded, cut foils in half in lathe or band saw (preferred). This is why we need two numbers per blade.

-Have all blades magnetic-particle inspected for cracks (Dye-check is no good for this). Mark location of cracked blades on map, with details. Check failure pattern for signs of wheel vibration (90 and 180° or 30-60-120°).

-Break cracked blades. First try to break at room temperature, noting signs of embrittlement and loss of strength, mark this on map. If too tough, cool blades in liquid nitrogen and try again, using impact.

-Immediately register (and photograph) any coloration of the fatigue portion of the fracture surface, other than silver, or grey. Preserve blades in dry-box, for immediate metallurgical inspection.

-Go through complete final inspection, including surfaces, dimensional accuracy, metallurgy, etc.

-Go through operating logs, inspect installation, boilers, feedwater apparatus, piping, condenser, drains, plant history, turbine history, history of other turbines and steam users.

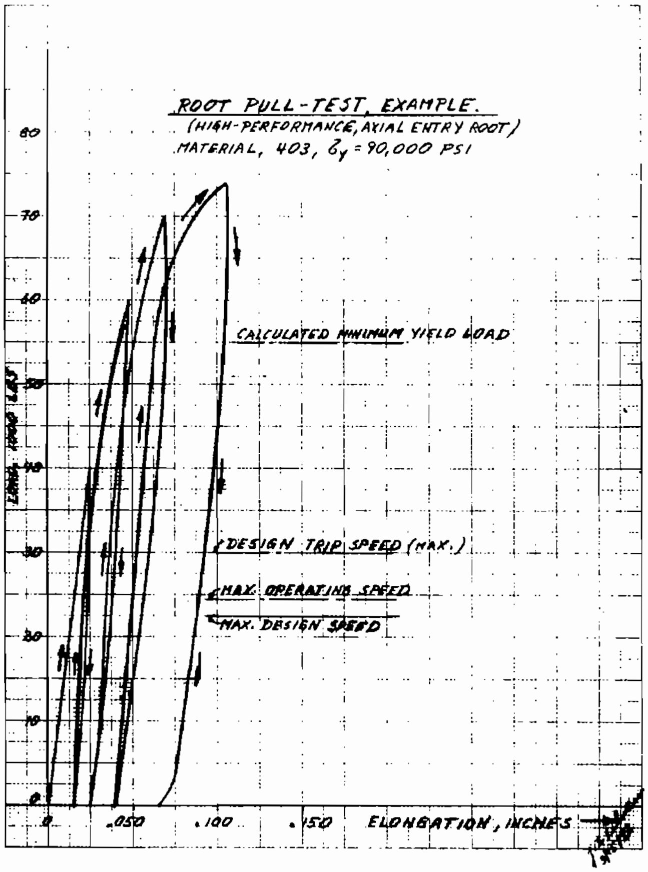

Figure 20. Pull test of axial-entry, 3-land pinetree root, mounted in a wheel rim section.

Note successive loading and unloading, for the purpose of finding yield point. Ref (3) (7)

5.0 Remedial Action

Regardless of the type of failure, a few improvements can usually be made. This list describes possibilities in sequence of effectiveness. The obvious improvements of quality of hardware, steam and operation are not listed. Neither are solutions requiring re-design. Of course, stresses must be checked and adequate procedures used.

a. Hardware

-If in cold end, use titanium blades, with vacuum-welded shroud.

-If blades are 1" or less in width, silver-solder shrouds to blade tips. New blading only, as it requires surface preparation. Requires careful metallurgical control. Steel temperatures not to exceed ≈ 1300° , cool very slowly to prevent air hardening.

-Use heaviest possible shrouding. Consider titanium shroud. Check centrifugal stress. Cannot make shroud heavier if g-load is over ≈ 35,000.

-If centers of gravity of root, foil, and shroud are not in a radial line, or with packer-type blades, get correctly proportioned, integral blades made, with same root, airfoil, etc., to fit the existing disk.

-Install additional lashing wires, if blade is long.

-Install damping wires or strips in root, possibly in shroud.

-On existing assemblies, install axial damper-pins (1/6 to 1/8" Dia) through root platforms, between blades.

b. Operational

-If first stage, throttle on T&T valve until governor is nearly wide-open. This increases steam consumption but lowers pressure-drop across first stage to the minimum. Make sure T&T is suitable for continuous throttling and that it will trip reliably in part-open position (some don't!). *Always* start on T&T and don't go on governor until full continuous operating speed is reached. Then open T&T very, very slowly, taking about ½ to one hour to open T&T, after full speed and load has been reached. Don't "spin the valve open!" It is bad in many ways.

-If last stage, check blade exit velocity, adjust backpressure to stay at a Mach Number of ≈ 1.15.

-If an intermediate stage is in trouble, go for lowest possible steam flow.

APPENDIX

BLADE STRESS EVALUATION PROCEDURE, AT ROOT OR FOIL

This method is based on Traupel (6), but has been extensively modified and condensed. The procedures for single nozzles were added.

1.0 Input Data

1.1 Stresses; all are without stress concentrations

σ_t = Caused by pure centrifugal tension, without any moments.

σ_{bc} = Centrifugal bending (c.g. offset)

σ_{bi} = Average, ideal steam bending stress for the free-standing blade, caused by steady driving force. Any load reduction caused by shroud or lashing, or vibratory amplification, will be reflected in σ_a.

σ_b = Average steam bending stress, including stiffening effect of shroud and lashing.

$\pm\sigma_a$ = Bending stress amplitude ($=\sigma_{bi}$ V S α H) for vibratory bending

1.2 Crush stresses on lands; all are average stresses.

$\bar{p}_t$ = Caused by pure centrifugal tension

$\bar{p}_{bc}$ = Caused by pure centrifugal bending (c.g. offset)

$\bar{p}_{bi}$ = Caused by pure, ideal steam bending, equivalent to σ_{bi}

$\bar{p}_b$ = Caused by pure steam bending, including stiffening effect of shroud and lashing, equivalent to σ_b.

$\bar{p}_a$ = Caused by vibratory bending stress amplitude σ_a.

1.3 Stress correction factors

K'_t = Geometric stress concentration factor, tension (Peterson [11] p. 113, Figs. 103 to 107)

K'_b = Geometric stress concentration factor, bending (Peterson p. 113 & Fig. 108)

q = Notch sensitivity factor (Peterson p. 9 & 10, Figs. 8, 9, & 10, Eqn. 14)

$$\left.\begin{array}{l} K_t=q(K'_t-1)+1 \\ K_b=q(K'_b-1)+1 \end{array}\right\} \text{Stress concentrations including notch sensitivity}$$

b = Surface roughness factor: (non-reversing stress)

RMS	b
125	.7
63	.85
32	.98

for 90,000=σ_y. If higher σ_y, subtract 0.015 for each 10,000 psi over 90,000.

K_p = Crush stress distribution factor. $p_{max} \approx 1.5$ See Fig. A-3

f_t = Temperature correction factor, applied to material strength

1.4 Material properties

σ_u = Ultimate strength, at opertaing temperature.

σ_y = Yield strength, at operating temperature.

2.0 Stress Evaluation; Full Admission Stages

Criterion #1:
To be determined for both trip speed (σ_{bi} $+\sigma_a= 0$) and for most unfavorable operating speed.

$$\boxed{\sigma_{max} \leqslant 2\, f_t\, \sigma_y}$$

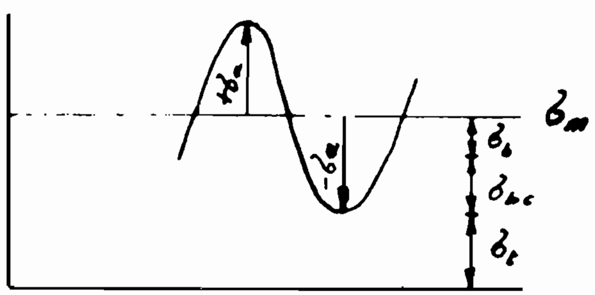

Figure A-1. Relationship of centrifugal and alternating stress.

"The peak stress, at the stress concentration, must not exceed twice the yield strength of the material."

This prevents reversing yield and consequent material embrittlement. Stress concentration factors are applied to all stresses, static and dynamic.

$$\sigma_{max} = K_t\, \sigma_t + K_b\, (\sigma_{bc} + \sigma_b + \sigma_a)$$

Criterion #2:
At most unfavorable combination of load, speed, vibration, and environment.

$$\boxed{\sigma_m + K_b\, \sigma_a \leqslant 0.65\, b\, f_t\, (\sigma_m + \sigma_{em})}$$

"Combined static and dynamic stress at the peak of the cycle must not exceed 65% of the combined static and dynamic endurance strength of the material."

This criterion protects against fatigue failure. Note that the 0.65 safety margin is applied to both static and dynamic strength, giving a considerably higher margin to the endurance portion.

$\sigma_m = \sigma_t + \sigma_{bc} + \sigma_b$ = Static mean stress, without stress concentration.

σ_{em} = Endurance strength of material, read at mean stress σ_m from the Goodman diagram, *for the applicable environment* and corrected for temperature. See Fig. 1.

Criterion #3:
At trip speed and at most unfavorable combination of load, speed and vibration.

$$\boxed{P_{max} \leqslant f_t\, \sigma_y}$$

"The maximum crush load on the root projections (lands) must not exceed yield strength." Since, with normal dimensioning, max. shear stress (in the 45° planes) is a direct function of max. crush, this protects against shear-induced failures (low-cycle fatigue, embrittlement caused by reversing yield, consequent reduction of fatigue strength).

$$p_{max} = K_p\, (\bar{p}_t + \bar{p}_{bc} + \bar{p}_b + \bar{p}_a)$$

Note: The crush-load criteria safeguard against excessive shear stress in the 45°, maximum shear plane A-C. This is only applicable if plane A-C has minimum dimensions as shown at right, for contour II. Contour III would be insufficient, requiring additional stress corrections. Contour I is as used in Petersons tests.

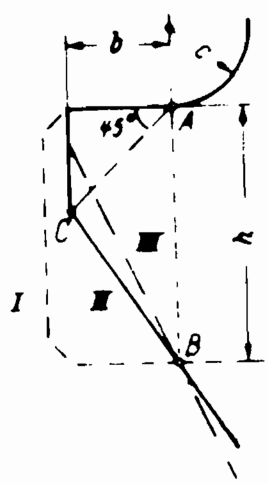

Figure A-2. Minimum requirements for shear area (6)

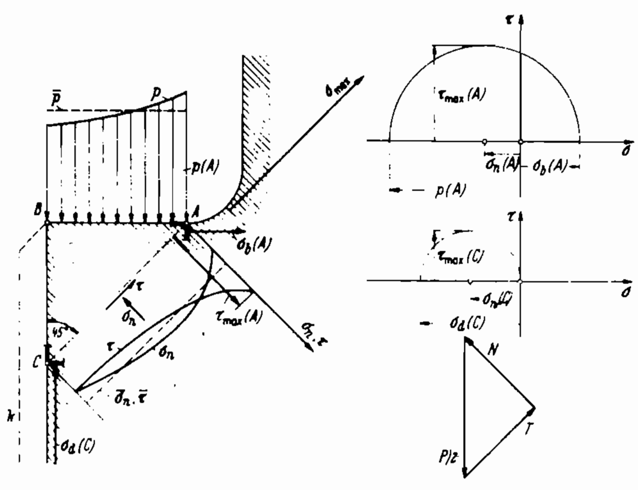

Figure A-3. Location of important stresses in a root projection (land). (6)

Criterion #4:
Both at trip speed and most unfavorable operating speed.

$$\boxed{p_{max} \leqslant b\, f_t\, (p_m + \sigma_{em})}$$

"The maximum crush load on the lands, at the peak of the cycle, must not exceed the combined static/dynamic endurance strength of the material." This protects against shear fatigue. An appropriate safety factor is inherent in the derivation of this relation.

p_{max} = same as for #3, including various operating conditions, but also including environment.

$p_m + \sigma_{em}$ is determined in the same way as $(\sigma_m + \sigma_{em})$ for #2, disregarding the theoretically negative sign (= compression) for p_m.

$$\bar{p}_m = K_p (\bar{p}_t + \bar{p}_{bc} + \bar{p}_b)$$

3.0 Stress Evaluation, Stages with Individual Nozzles and/or Partial Admission

If represented as under 2.0 the bottom of the non-resonant cycle would be below the centrifugal stress level which, for non-amplified conditions, is not reasonable. Evidently, the steam bending stress σ_b must be multiplied by K_b for this case, to find the correct mean stress.

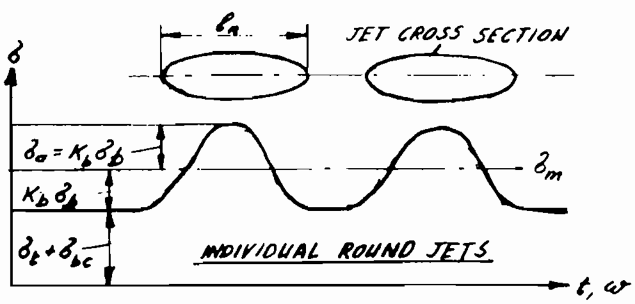

Figure A-4. Stress versus time, round jets. (7)

Procedure:

3.1 Partial admission

Criteria #1, 3, & 4: Not affected, but p_{max} must be calculated for top-of-cycle loading.

Criterion #2: Same relation, but $\sigma_m = \sigma_t + \sigma_{bc} + K_b \sigma_b$.

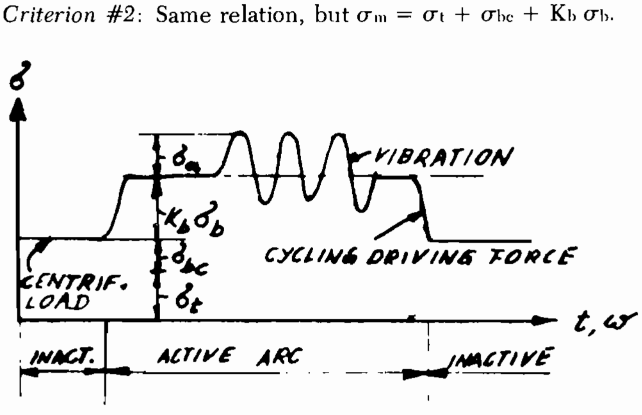

Figure A-5. Stress versus time for partial admission nozzle blocks and diaphragms. (7)

3.2 Individual jets, non-vibratory (Fig. A-5)

a. round jets (drilled-and-reamed), procedure:
- Get driving force per nozzle, assuming 100% blade efficiency.
- Find active blade/nozzle = nozzle ℓ_n/blade pitch, and get average driving force/blade.
- Max. driving force/blade = 1.3 x average, at center of steam-jet ellipse.
- The stress amplitude resulting from ½ of this force, multiplied by K_b, represents both $K_b \sigma_b$ and the non-vibratory cycling stress amplitude σ_a.

$$\sigma_m = \sigma_t + \sigma_{bc} + K_b \sigma_b \,; \qquad \sigma_a = K_b \sigma_b$$

Use equations in the usual manner. Vibratory amplification and/or load dissipation (shroud) will apply to the vibratory component of σ_a — especially shroud dissipation α.

For non-resonance $\sigma_b = \sigma_{bi}$ V S α H becomes σ_b, all other factors being 1.0. In case of vibration, the bottom of the cycle will extend below the stress existing between jets — $\sigma_t + \sigma_{bc}$.

The second row of the Curtis stage must be calculated the same way, because the jet from the nozzles will essentially permeate the first row and reversing row, giving the same basic load pattern, plus another exciting frequency from the reversing blades.

b. Square nozzles (milled or cast)

Use same procedure, but do not multiply average driving force/blade by 1.3.

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