TURBOMACHINERY ANALYSIS AND PROTECTION

by

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ABSTRACT

a. Prediction and Analysis of the Reliability of Turbomachinery Installations—Influence of various components on over-all reliability, such as of turbines, compressors, gears, motors, piping, foundations. Critical items in reliability evaluation.

b. Identification of Problem Areas in Existing Installations ("trouble-shooting")—How to predict and identify problem areas and probability of shutdown or component failure. Application of protective devices and supervisory instrumentation to prevent or minimize losses.

c. Analysis of Turbine Operation to Predict Future Problems—Critical areas in turbine operation. Common failures caused by mal-operation. Instrumentation and protective devices to minimize risk.

PREDICTION AND ANALYSIS OF THE RELIABILITY OF TURBOMACHINERY INSTALLATIONS

During the last few years the size of petrochemical plants and chemical plants has increased considerably. This means that the turbomachinery which is used in these plants has also become larger. The machines operate at higher speeds, and they are now far more complex than before. In addition, the time between turn-arounds has now increased to two, three, and even five years. As a result of the increased output of product of these large plants, the losses which are experienced as a result of an accidental shutdown are very high, and numbers of 100,000 dollars per day and even more are not unusual.

As a result of this situation it became evident that there is a need for more systematic means to aid in trouble-correction and to reduce shutdown time. Also, means are required to predict reliability in advance, and to evaluate reliability versus initial cost, operating cost, and shutdown costs.

To indicate the reliability of a turbomachinery installation a series of curves can be developed, which are keyed to a common denominator, for comparative purposes. Fig. 1 shows a set of such curves. This approach has the advantage of giving a general idea of the interaction of the various factors. For example, it can be demonstrated why a perfectly good machine may not run well in a poor installation. Vice versa, it can be demonstrated why a poorly applied machine or poorly designed machine may run quite well if it is well installed, carefully operated, and capably maintained. The individual reliability factors are multiplied to find over-all factors for major components or for the entire installation. For a complete explanation of this method see References (1) and (2).

Of course there are many shades of gray between the two extreme examples quoted above. We need a method to determine which shade of gray is involved in a particular trouble job. The approach presented can also show which type of equipment is best for certain ranges of operating conditions, such as speed, pressure, temperature, and other operating parameters. In Reference papers (1) and (2) we used this approach to aid trouble-shooting. The procedure works as follows: Plotted versus speed is in all these curves a so called "Reliability Factor." This factor varies between zero and 2.0 in most of the plots. We assumed a reliability factor of ONE to represent an average reliability for a given application. In other words, in a petrochemical plant, a given machine is expected to run continuously between turn-arounds and to require a certain amount of maintenance in between. Also, average figures can be compiled for the so-called "normal" break-down rates for this type of plant. We have arbitrarily chosen to use the factor of ONE for an installation which has this break-down rate and reliability. A greater factor shows that the equipment is likely to run longer between break-downs. For example, a factor of TWO would indicate that the unit would run twice as long as a unit with a so-called "normal" break-down rate. The reliability factors for various items can be plotted versus speed, load, pressure—or any other critical parameter.

The first plot shows the type of equipment: A compressor will have a reliability rate somewhat different from a turbine. The motor and gear combination will again have different reliabilities. This situation will be influenced by speed—a turbine will usually be more reliable at high speeds, while a motor-and-gear combination can have very good reliability at low speeds, but it can have fairly poor reliability at the higher speeds. In the next plot the equipment size is shown.

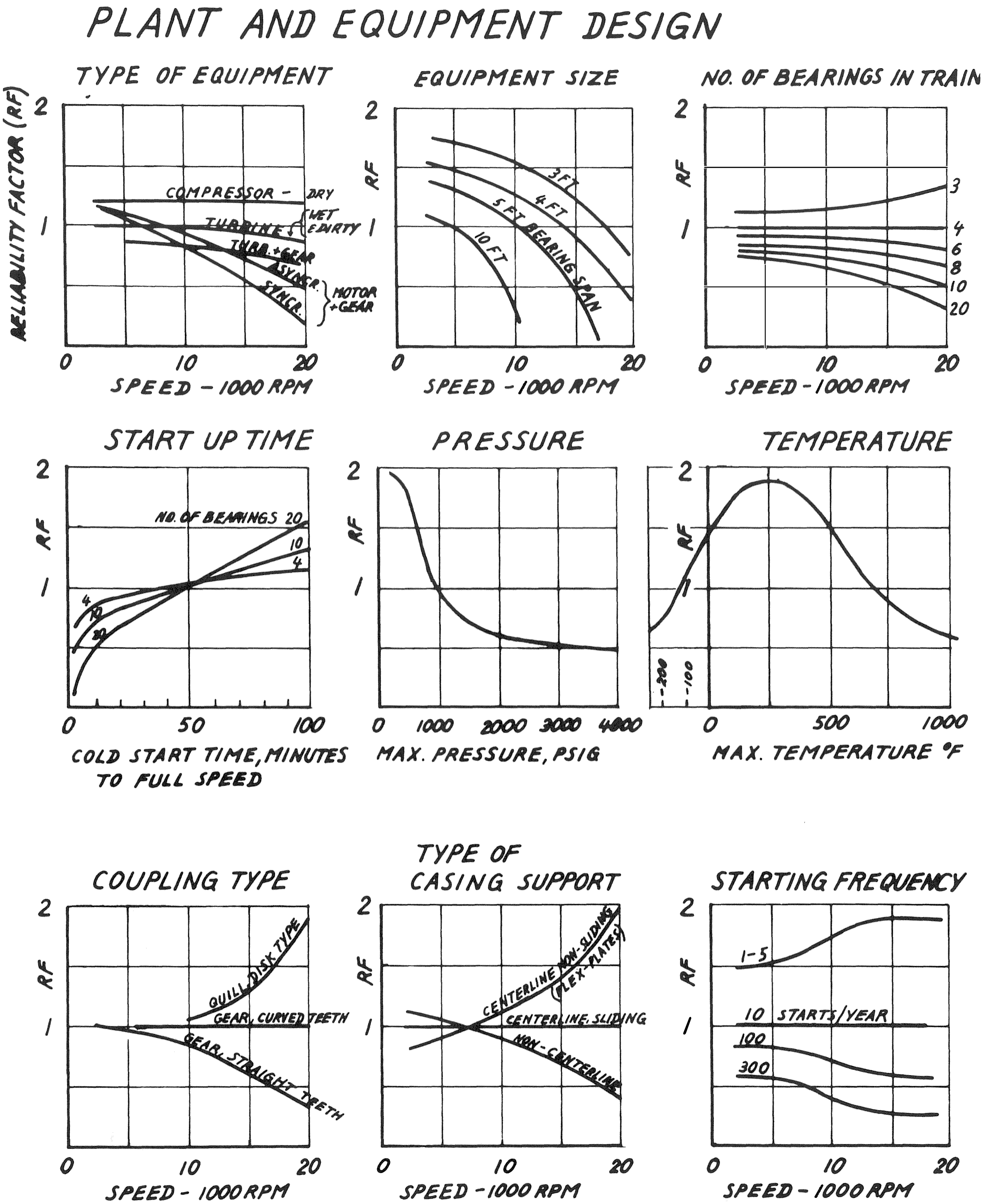
PLANT AND EQUIPMENT DESIGN
TYPE OF EQUIPMENT
RELIABILITY FACTOR (RF)
COMPRESSOR - DRY
TURBINE
WET
EDIRTY
TURB.+GEAR
ASYNCR.
SYNCR.
MOTOR +GEAR
SPEED - 1000 RPM
EQUIPMENT SIZE
3 FT
4 FT
5 FT BEARING SPAN
10 FT
SPEED - 1000 RPM
NO. OF BEARINGS IN TRAIN
SPEED - 1000 RPM
START UP TIME
NO. OF BEARINGS
COLD START TIME, MINUTES TO FULL SPEED
PRESSURE
MAX. PRESSURE, PSIG
TEMPERATURE
MAX. TEMPERATURE °F
COUPLING TYPE
QUILL, DISK TYPE
GEAR, CURVED TEETH
GEAR, STRAIGHT TEETH
SPEED - 1000 RPM
TYPE OF CASING SUPPORT
CENTERLINE NON-SLIDING (FLEX-PLATES)
CENTERLINE, SLIDING
NON-CENTERLINE
SPEED - 1000 RPM
STARTING FREQUENCY
10 STARTS/YEAR
SPEED - 1000 RPM

Figure 1.

The span between bearings was selected as the significant parameter. It can be seen, that the larger machines would not be very reliable at high speeds, because the faster a machine runs, the shorter and smaller it must be, mostly for reasons of critical speed and rotor dynamics. The next curve shows the number of bearings in the entire unit—or in the train as it is often called—and the curves are plotted here for three, four, six, eight, ten, and twenty bearings. The number of bearings is a good indication of the complexity of a unit. The more bodies and the more bearings a unit has, the more complex it must be and the more opportunity will it have for getting into trouble. These curves all represent data pertaining to the equipment itself.

On the same chart, Fig. 1, are curves showing operating conditions, such as pressures and temperatures. Also shown is start-up time and number of start-ups per year. Quite evidently, a unit which is started only once a year and then runs continuously will be far less likely to suffer a break-down than a unit which is started and stopped daily. This is reflected in this curve. The faster the unit runs, the more pronounced will this effect be.

One thing is of interest to note here: Once the operating conditions are given by the process designer, the designer of the rotating equipment has very little choice concerning the span he would like to use between bearings, or the number of bearings. This is essentially fixed, especially where the equipment approaches the limit of capacity or speed obtainable with rotating equipment. Such is the case, for example, with syngas compressors for ammonia synthesis, where turbines may be rated 35,000 HP at 11,000 RPM, and where extraction may also be required. In such cases, there are not many solutions, and a designer has to compromise and look for the best possible solution concerning the number of bodies in the train, the bearing spans, and the number of stages for each machine.

This brings us to a very important point, namely, that a design can often be greatly simplified—and the reliability greatly improved—by better cooperation between process designer and equipment designer. One example is the vacuum on a condensing turbine. For example, a small change of design vacuum on a condensing machine can sometimes result in a single-flow turbine instead of a double-flow machine, with a corresponding improvement of the reliability factor. The bearing span would get shorter, the initial cost would be lower, and often the efficiency is better under normal operating conditions. Quite often the reduced vacuum would only be needed during overload conditions such as during start-up or emergency, or during off-design conditions. During normal operating conditions the machine would operate with the design vacuum. But the way the specifications are often written, a designer has no choice but to quote his machine for the operating conditions as specified. The result may be considerable economic disadvantage and reduced reliability. This may serve as an example how curves as shown may be used to optimize an installation.

Fig. 2 shows the effect of piping on reliability. The piping affects the equipment by the forces and moments it creates at the connections to the equipment. The piping may be pushing the entire machine out of alignment and cause vibrations or rubs. Especially on hot machines—such as high temperature turbines—the piping can warp the casing. This may result in leaking flanges or internal rubs, causing serious vibration problems. Also, the piping may cause the casing guides to bind, or the casing may actually lift off its support feet. This, then, can create all sorts of vibration problems

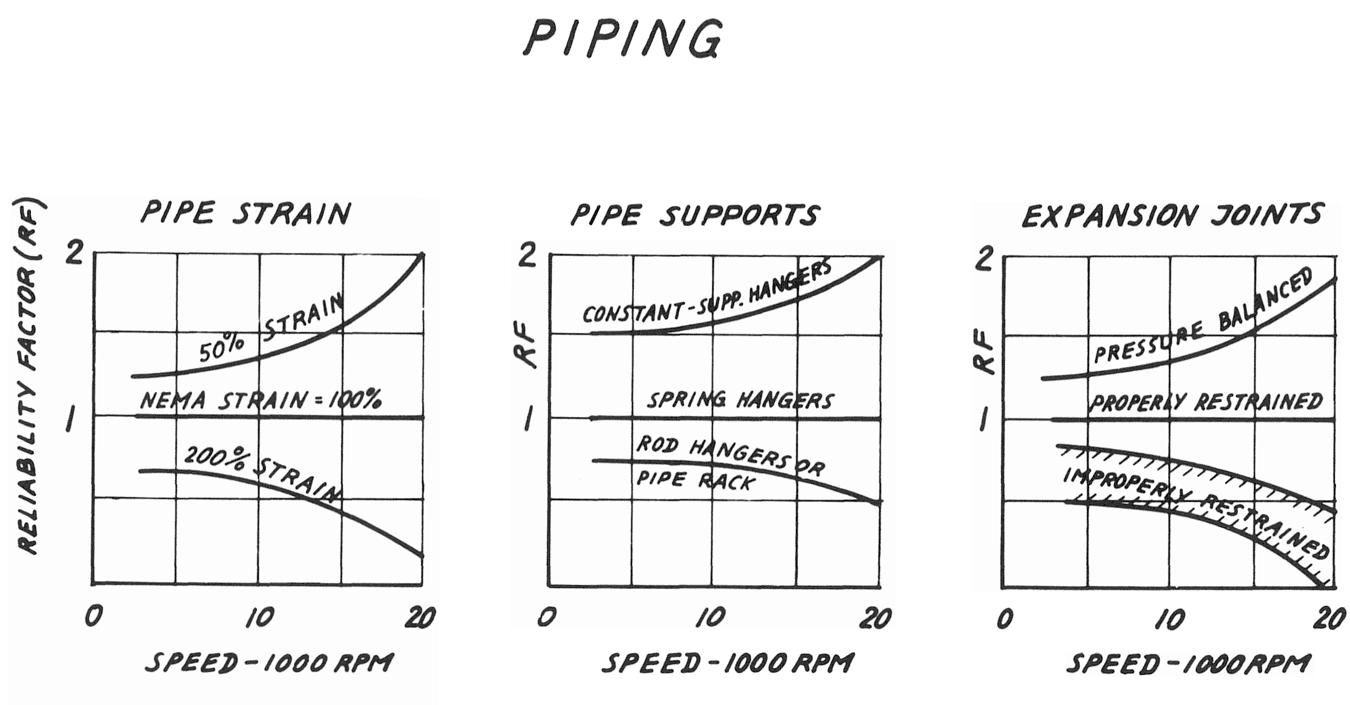

Figure 2.

and internal rubs. To correct problems of this type in the field can be very costly. Curves as shown may help to find the most economical solution.

Fig. 3 shows the influence of the foundation upon reliability of rotating equipment. Very briefly, the function of the foundation is to hold a unit in alignment, not only during normal conditions, but also during abnormal conditions. The forces the foundation has to absorb are the piping forces, and such forces as may be developed by expansion joints in the piping system or on the condenser of a turbine. Other factors which can cause misalignment are settling, shrinkage, or unequal heating of the foundation.

It is very important to prevent build-up or transmission of vibrations. The curves shown indicate the effect of foundation rigidity upon reliability. Evidently a high speed machine must be held in closer alignment than a low speed machine, and the effects of misalignment are more severe. Therefore the reliability factor drops off with speed for a foundation with poor rigidity. The vibration characteristics are expressed in terms of foundation weight. The normal relationship between foundation mass and unit mass is 1.5 times and this is used for a reliability factor of one. If the mass is less than that, the reliability will drop off, because the machine becomes more sensitive to small upsets and small operating problems, or, in case of emergency, the risk of serious damage is greater with a light foundation than with a massive one. The effect of major structural resonance is also shown. This, of course, will greatly reduce the reliability of the unit, especially during start-up conditions, upsets and emergencies. The faster a machine runs, the more pronounced will this effect be.

The third graph shows the isolation from the surroundings. Transmission of vibration from and to the unit has a significant effect upon the reliability of rotating equipment.

The effect of operational and maintenance practices is shown in Fig. 4. This effect also becomes more pronounced with increasing speed. A machine running at, say, 15,000 RPM is a very delicate piece of machinery, and it can easily be damaged by careless maintenance or operation.

Some time ago the magazine "THE LOCOMOTIVE," issued by the Hartford Steam Boiler Inspection Insurance Company, had a very interesting article, which ties in with this type of reliability evaluation. The article was headed "YOU CAN NOT ALWAYS BLAME THE OPERATOR." It points out that operator failure is in reality often management failure. When an operator fails, this may mean that he has not had sufficient training, or that he was not familiar enough with safety practices and, perhaps, that he was not properly selected to begin with. This article also points out that operator education and safety practice is not an out-of-pocket expense, but rather an investment which will yield returns. This same philosophy is what we want to demonstrate with the reliability curves: We want to show reliability as a function of investment, be the investment in the form of a training program, safety practices, or be it maintenance shops or, in an earlier stage of the game, investment in equipment, foundations, piping, and so on. I do not mean to say that just any investment is desirable, but rather that the investment should be made consciously, with a clear idea of the returns to be expected.

IDENTIFICATION OF PROBLEM AREAS IN EXISTING INSTALLATIONS

After we have seen how to predict and evaluate reliability, we must now provide the man in the field with tools to enable him to deal with a problem quickly and efficiently. Turbomachinery has a way of telling us when something goes wrong: It just starts to vibrate.

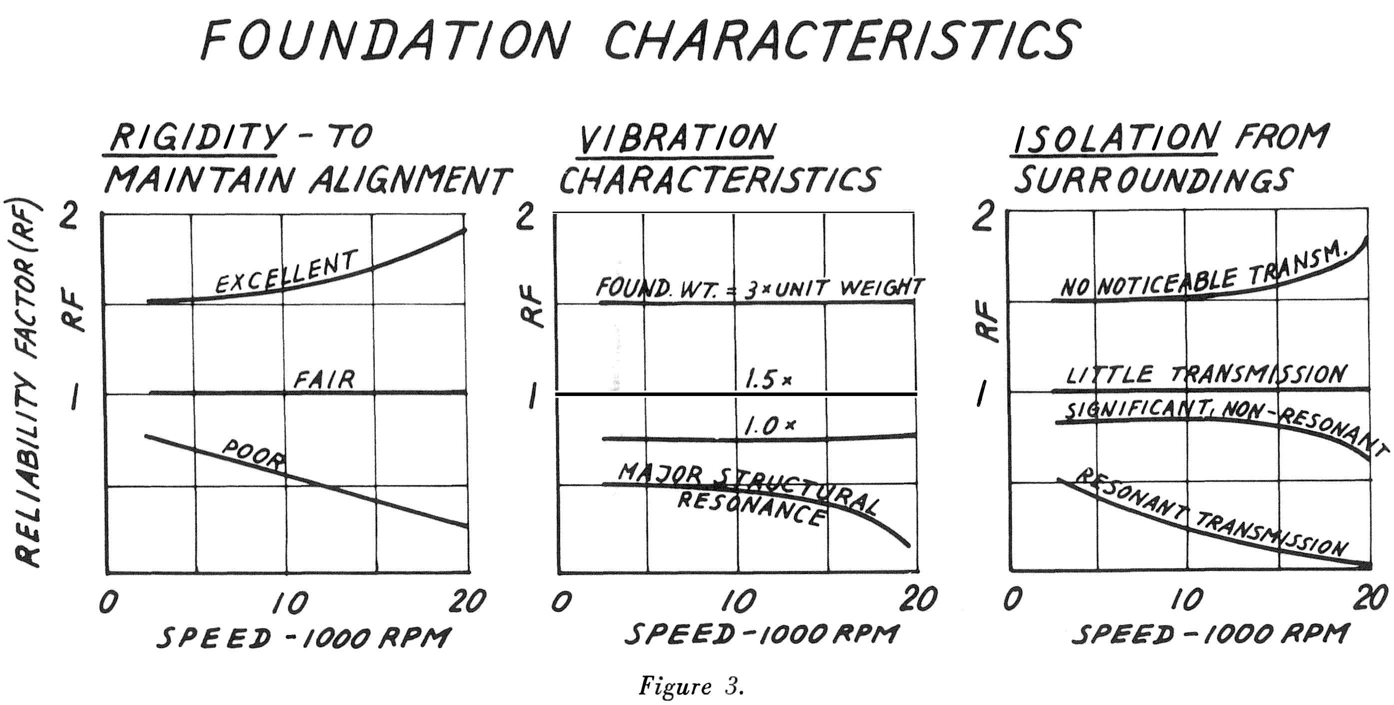

Figure 3.

SYMPTOMS AND DISTRESS MANIFESTATIONS

CHART #1

VIBRATION ANALYSIS

Cause of Vibration	Predominant frequencies: 0 - 40%	40 - 50%	50 - 100%	1 x running frequ.	2 x R F	Higher multiples	1/2 R F	1/4 R F	Lower multiples	Odd frequ.	Very high frequ.	Electrical: Line frequ.	Multiples of L. F.	Slip frequ.
Initial Unbalance				90	5	5								
permanent bow or lost rotor parts (vanes)				90	5	5								
Temporary Rotor bow				90	5	5								
Casing distortion Temp.	←	10	→	80	5	5								
Casing distortion Perm.	←	10	→	80	5	5								
Foundation distortion		20		60	20					10				
Seal rub	10	10	10	20	10	10			10	10	10			
Rotor rub axial	←	20	→	80	10	10			10	10	10			
Misalignment				40	50	10								
Piping forces				40	50	10								
Journal & bearing eccentricity				80	20									
Bearing damage	80 →			40	20						20			
Bearg. & Support excited vibration (oil whirls, etc.)	10	70					10	10						
Unequal bearing stiffness horizontal - vertical					80	20								
Thrust brg. damage	90 →										10			
Insufficient tightness in assy. of:	PREDOMINANT FREQUENCY WILL SHOW AT LOWEST CRITICAL, OR RESONANT FREQUENCY													
rotor (shrink fits)	40	40	10							10				
rotor Brg. liner	90 →									10				
Brg. case	90 →									10				
Casing & support	50 →									50				
Gear inaccuracy						20				20	60			
Coupling inaccuracy or damage	10	20	10	20	30	10								
Rotor & brg. system critical				100										
Coupling critical				100	ALSO MAKE SURE TOOTH FIT IS TIGHT !									
Overhang critical				100										
Structural resonance of:														
Casing		10		70	10		10					30	30	30
Supports		10		70	10		10					30	30	30
Foundation		20		60	10		10					30	30	30
Pressure pulsations	MOST TROUBLESOME IF COMBINED WITH RESONANCE →													
Electrically excited vibr.												←	100	→
Vibr. transmission														
Valve vibr.											80			
NAME OF PROBLEM	THE SECTION BELOW IS MEANT TO IDENTIFY BASIC MECHANISM													
Sub-harmonic resonance		RARE-LOOK FOR AERODYNAMIC ORIGIN (SEALS)					←	100	→			POSSIBLY ELECTRIC ORIGIN		
Harmonic resonance					←	100 →							100	
Friction induced whirl	80	10	10											
Critical speed				100										
Resonant vibr.				100										
Oil whirl		80	WATCH FOR AERODYNAMIC ROTOR-LIFT (PARTIAL ADMISSION ETC.) 10 5 5											
Resonant whirl		100												
Dry whirl											100			
Clearance induced vibr'ns.	10	80	10											
Torsional resonance				40	20	20				20		←	100	→
Transient torsional				50						50		←	100	→

Cause of Vibration	Direction and Location of predominant amplitude: Vert.	Horiz.	Axial	Shaft	Brgs.	Casing	Foundation	Piping	Couplg.	Amplitude Response to Speed Variation during vibration-test runs — Coming up: Stays Same	Increases	Decreases	Peaks	Comes suddenly	Drops out suddenly	Slowing down: Stays same	Increases	Decreases	Comes suddenly	Drops out suddenly
Initial Unbalance	40	50	10	90	10						100		PEAK					100		
permanent bow or lost rotor parts (vanes)											100							100		
Temporary Rotor bow										30	60	5		5		30	5	50	5	10
Casing distortion Temp.										30	50	5		5	10	30	5	50	5	10
Casing distortion Perm.										40	60					40		60		
Foundation distortion				40	30	10	10	10		20	80					20		80		
Seal rub	30	40	30	80	10	10				10	70		PEAKS AT CRITICALS	10	10	10		70	10	10
Rotor rub axial	30	40	30	70	10	20				10	40	10		20	20	10		50	20	20
Misalignment	20	30	50	80	10	10				20	30	10		20	20	20		40	20	20
Piping forces	20	30	50	80	10	10				20	40			20	20	20		40	20	20
Journal & bearing eccentricity	40	50	10	90	10					40	50	10				40	10	50		
Bearing damage	30	40	30	70	20	10				10	50	10		20	10	10	10	50	10	20
Bearg. & Support excited vibration (oil whirls, etc.)	40	50	10	50	20	20	20				10			90				10		90
Unequal bearing stiffness horizontal - vertical	40	50	10	40	30	30					40		50	10				40		10
Thrust brg. damage	20	30	50	60	20	20				20	50	10		10	10	20	10	50	10	10
Insufficient tightness in assy. of:																				
rotor (shrink fits)	40	50	10	60	20	20								90	10				10	90
rotor Brg. liner	40	50	10	80	10	10								90	10				10	90
Brg. case	40	50	10	70	20	10								90	10				10	90
Casing & support	40	50	10	50	20	30								90	10				10	90
Gear inaccuracy	30	50	20	80	10	10				20	20	20	20	10	10	20	20	20	10	10
Coupling inaccuracy or damage	30	40	30	70	20				10	10	20		20	(LOOSE SLEEVE, FRICTION, OR DIRT IN TEETH) 40	10	10		20	10	40
Rotor & brg. system critical	40	50	10	70	30						20		80					20		
Coupling critical	20	40	40	10	10				80		20		80	←LOOSENESS→				20	50% IF LOOSE	
Overhang critical	40	50	10	70	10				20		30		70					30		
Structural resonance of:																				
Casing	40	50	10		40	40	10	10			20		80					20		
Supports	40	50	10		20	50	20	10			20		80					20		
Foundation	30	40	30		10	40	40	10			20		80					20		
Pressure pulsations	30	40	30	CAN EXCITE WHIRLS OR RESONANCE		30	30	40		90	10%-DEPENDING ON ORIGIN OF DISTURBANCE					90	10 →			
Electrically excited vibr.	30	40	30			40	40	20		90						90				
Vibr. transmission	30	40	30			40	40	20		90						90				
Valve vibr.	30	40	30			80	10	10		80				10	10	80			10	10
NAME OF PROBLEM																				
Sub-harmonic resonance	30	30	40	20 IF BRG.-EXCIT.	20	20	20	20			20		20	30	30			20	30	30
Harmonic resonance	40	40	20	20	10	10	30	30		20	20		60			20		20		
Friction induced whirl	40	50	10	80	20									90	10				10	90
Critical speed	40	50	10	60	40						20		80					20		
Resonant vibr.	40	40	20	20	10	20	30	20			20		80					20		
Oil whirl	40	50	10	80	20									100						100
Resonant whirl	40	50	10	20	20	20	20	20						80	20				20	80
Dry whirl	30	40	30	40	20	20	10		10					80	20	80				20
Clearance induced vibr'ns.	40	50	10	70	10	10			10					80	20	20			20	60
Torsional resonance	TORSIONAL			100	LATERAL 40	40	AMPLITUDE		10		20		30	30	20	20			20	30
Transient torsional				TORSION. 100	40	40			10				50	30	20				30	20

OPERATIONAL EVIDENCE

Cause of Vibration	Effect of Operating Conditions — Comes in or Goes out at: No effect	Full load in	Full load out	No load in	No load out	Part load in	Part load out	Start up only	After load dump or surge	When other machine starts or shuts down	Effect of Oil p & t and Flow — Increase p: better	Increase p: worse	Decrease p: better	Decrease p: worse	Increase t: better	Increase t: worse	Decrease t: better	Decrease t: worse
Initial Unbalance	100	LOAD-INDUCED WHIRL 100%									90	10	10	90	10	90	90	10
permanent bow or lost rotor parts (vanes)	100										Slight changes only, due to variation of oil-film damping							
Temporary Rotor bow		10	20	20	10			20	20									
Casing distortion Temp.		10	20	20	10			20	20									
Casing distortion Perm.	40	10	10	10	10			10	10									
Foundation distortion	40	10	10	10	10			10	10									
Seal rub	10	10	10	5	10			20	20	5								
Rotor rub axial	5	30	10	5	10			10	30									
Misalignment	10	30	10	5	5	5	5	20	10									
Piping forces	10	20	10					20	20	20								
Journal & bearing eccentricity	100																	
Bearing damage	90																	
Bearg. & Support excited vibration (oil whirls, etc.)	30	5	5	5	5	20	20	10 MOSTLY MULTI-BEARING - SOLID COUPLED	10						90	10	10	90
Unequal bearing stiffness horizontal - vertical	100																	
Thrust brg. damage		30	5	5	30				30									
Insufficient tightness in assy. of:																		
rotor (shrink fits)	30	30	5	5	10			30	10	20								
rotor Brg. liner	90							10										
Brg. case	90							10										
Casing & support	10	5	5	5	5	30	30	10										
Gear inaccuracy	30	10	10	20	10	10	10											
Coupling inaccuracy or damage		(80% ← IF TOOTH FRICTION OR DIRT IN TEETH) 10	10	30	10	10	10	10	20									
Rotor & brg. system critical	90																	
Coupling critical	90		IF LOOSE	IF LOOSE				IF LOOSE	IF LOOSE									
Overhang critical	90																	
Structural resonance of:																		
Casing	90												NO EFFECT					
Supports	90																	
Foundation	90																	
Pressure pulsations	20	5	5	5	5	20	20			20								
Electrically excited vibr.	50	40								10								
Vibr. transmission	50									50								
Valve vibr.	10	30	10	10	10	10	10	10										
NAME OF PROBLEM																		
Sub-harmonic resonance	40					20	20			20	TRY SEAL PRESSURE VARIATION							
Harmonic resonance	80	10								10								
Friction induced whirl	40	(80% IF COUPLING FRICTION, OR LOAD-INDUCED WHIRL) 20	5	5	10			10	10		90	10	10	90	90	10	10	90
Critical speed	100										10	SMALL DAMPING EFFECTS ONLY		10		10	10	
Resonant vibr.	100																	
Oil whirl	60					20	20				90	10	10	90	90	10	10	90
Resonant whirl	50	50 →								50	90	10	10	90	90	10	10	90
Dry whirl	90	10							10		90						50	50
Clearance induced vibr'ns.	40					20	20				SMALL DAMPING EFFECTS ONLY							
Torsional resonance	30	50			10	10												
Transient torsional								100										

NUMBERS INDICATE % OF CASES SHOWING ABOVE SYMPTOMS, FOR CAUSES LISTED IN VERTICAL COLUMN AT LEFT

Figure 5. Example showing "Trouble-Shooting Chart" — Vibration Analysis.

CHART NO. 2

OPERATIONAL EVIDENCE, CONT'D.

Predominant Sound (DURING VIBRATION TEST RUN) and History of Machine

Cause of Vibration	Lo frequ. "Rumble"	Loud "Roar"	"Hum"	Comes and goes periodically ("beat")	High pitched "whine"	Very high loud "scream"	Very high "squeal"	Ultrasonic (measure w. sound anal.)	Initial startup	1st year	1-10 year	over 10 years
Initial Unbalance		80	20						✓✓			
permanent bow or lost rotor parts (vanes)		80	20						✓		✓	✓
Temporary Rotor bow	10	70	10	10					✓✓	✓		
Casing distortion	10	70	10	10							✓	✓
Foundation distortion	10	50	30	10					✓	✓	✓	✓✓
Seal rub	20	40		10		10	10	10	✓	✓	✓	✓
Rotor rub axial	30	40		20				10	✓✓	✓		
Misalignment		40	40	20					✓	✓	✓	✓
Piping forces	30	40	30						✓✓	✓	✓	✓✓
Journal & bearing eccentricity		10	90						✓✓			
Bearing damage	20	40	10		10	10	10		✓	✓	✓	✓
Bearg. & Support excited vibration (oil whirls, etc.)	60	10		30					✓✓	✓		
Unequal bearing stiffness horizontal - vertical		10	60	20	10				✓✓			
Thrust brg. damage	80	10	10						✓✓	✓	✓	✓
Insufficient tightness in assy. of rotor (shrink fits)	60	20		20					✓	✓✓	✓	✓
Brg. liner	60	20		20					✓✓	✓		
Brg. case	60	10		30					✓✓	✓		
Casing support	60	20		20					✓✓	✓	✓	✓
Gear inaccuracy	20	10	10	20	20	10		10	✓✓			
Coupling inaccuracy or damage	10		40	10	40	ALSO "METALLIC" SOUND			✓✓	✓		
Rotor & brg. system critical		50	30	20					✓✓	✓		
Coupling critical		80	40	20	20				✓✓	✓		
Overhang critical		50	40	10					✓✓	✓		
Structural resonance												
Casing	20		20	60					✓✓	✓		
Supports	20		20	60					✓✓	✓	✓	
Foundation		10	80	10					✓	✓	✓	
Pressure pulsations	40	20		40					✓	✓✓	✓	✓
Electrically excited vibr.		10	80	10					✓✓	✓		
Vibr. transmission	20		20	60					✓	✓		
Valve vibr.	10	10				40	40		✓	✓		
NAME OF PROBLEM												
Sub-harmonic resonance	80			20					✓✓	✓	✓	
Harmonic resonance			40	20	40				✓✓	✓		
Friction induced whirl	60	20		20					✓✓	✓✓	✓	
Critical speed		50	30	20					✓✓			
Resonant vibr.	40		30	30					✓✓	✓		
Oil whirl	60	10		30					✓✓	✓	✓	
Resonant whirl	60	20		20					✓✓	✓	✓	
Dry whirl							20	80	✓	✓	✓	✓
Clearance induced vibr'ns.	60	20		20					✓✓	✓		
Torsional resonance	10	20	20	30	10	10			✓✓	✓		
Transient torsional				60 (COUGH)		20	20		✓✓	✓		

Damage or Distress Signal

Cause of Vibration	Seals rubbed	Shaft bent	Bearings failed: Wiped	Bearings failed: Fatigued	Bearings failed: Babbitt squeezed out	Case distorted or cracked	Out of alignment	Coupling burned or pitted	Gear teeth broken or pitted	Gear teeth marked on back side	Shaft cracked or broken	Galling or fretting marks under disks or hubs
Initial Unbalance	10		10	10								
permanent bow or lost rotor parts (vanes)	50	+	30	10	10							
Temporary Rotor bow	90	10	50		20			20	20	20	5	
Casing distortion	90	10	50	10	10	+	10	20				
Foundation distortion	90	10	50	30			50	30				
Seal rub	+	15	15	10	10	10	20	10				
Rotor rub axial	90	30	50					20			10	
Misalignment	50	10	10	20		20	+	40				
Piping forces	50	10	10	20		40	40	40				
Journal & bearing eccentricity			30	60								
Bearing damage	90	10	—	+	—	10	40	20	20	10	10	20
Bearg. & Support excited vibration (oil whirls, etc.)	90	10	80	60	30			20	20	10	10	10
Unequal bearing stiffness horizontal - vertical	10		40	40								
Thrust brg. damage	90	20	50		30						10	
Insufficient tightness in assy. of rotor (shrink fits)	90	10	60	60	40			80	40	40	20	50
Brg. liner	90	10	30	30	40			80	20	20	10	5
Brg. case	50	5	20	20	30			20	20	20	10	5
Casing support	20		40	60	10			20				
Gear inaccuracy			20	20					40	20	10	
Coupling inaccuracy or damage	20		20	20				40				
Rotor & brg. system critical	50	10	50	50				30	20	10	5	
Coupling critical	30	10	50	50				30				
Overhang critical	50	20	60	60	10			40			5	
Structural resonance												
Casing	20		20	20								
Supports	20		20	20				20				
Foundation	20		20	20			30	30				
Pressure pulsations	30		30	30			20	30				
Electrically excited vibr.			20	20				30	10	10	5	
Vibr. transmission	20		20	20			30	30				
Valve vibr.												
NAME OF PROBLEM												
Sub-harmonic resonance	10		20	10				10	30	10	5	
Harmonic resonance	10		20	20				20	30	10	5	
Friction induced whirl	90	10	60	60	40			80	40	40	20	50
Critical speed	50	10	50	50				30	20	10	5	
Resonant vibr.	20		20	20				20				
Oil whirl	60	10	80	60	30			20	20	10	10	10
Resonant whirl	80	15	90	80	40			40	30	20	20	20
Dry whirl	80	10	90	20	40			10	5	5	30	5
Clearance induced vibr'ns.	90	10	30	30	20			80	20	20	10	5
Torsional resonance	5		5	20				40	80	80	40	5
Transient torsional	5		5	5				30	80	80	50	10

Cause of Vibration	Coupling bolts loose	Foundat. settled or cracked	Soleplates loose or rusted	Sliding surfaces binding	Thermal expansion restricted	Fluid marks on internals	Rotor components eroded	solids accumulated on vanes & rotor	Salt deposits on internals	Main flanges leaking	Seals leaking
Initial Unbalance								20	20		
permanent bow or lost rotor parts (vanes)						20	20	60	60		10
Temporary Rotor bow	5										20
Casing distortion				30	30					50	20
Foundation distortion		+	30	30	10					20	10
Seal rub		10	10	10	10	10		10	10		50
Rotor rub axial	10			20	40						60
Misalignment	10	50	50	10	50						10
Piping forces	10		10	20	50					50	10
Journal & bearing eccentricity											
Bearing damage	20	20	20	20	20	40	10	10	10	10	10
Bearg. & Support excited vibration (oil whirls, etc.)											10
Unequal bearing stiffness horizontal - vertical											
Thrust brg. damage	10					60	30	60	60		50
Insufficient tightness in assy. of rotor (shrink fits)	10					40	10				30
Brg. liner	10										30
Brg. case	10										10
Casing support											10
Gear inaccuracy											
Coupling inaccuracy or damage											10
Rotor & brg. system critical											20
Coupling critical											
Overhang critical											25
Structural resonance											
Casing											10
Supports		30	20	10							10
Foundation		30	10	10	20						10
Pressure pulsations		30	20								16
Electrically excited vibr.	5										
Vibr. transmission											10
Valve vibr.											
NAME OF PROBLEM											
Sub-harmonic resonance	5	10	10								
Harmonic resonance	5	10	10								
Friction induced whirl	10					40	10				
Critical speed											
Resonant vibr.		30	10	10	20						
Oil whirl											
Resonant whirl	10	10									
Dry whirl	70										
Clearance induced vibr'ns.	10										
Torsional resonance	40										
Transient torsional	50										

LOOK FOR LOST BLADES & CRACKED DISKS, ETC. DUE TO CORROSION FATIGUE OR STRESS CORROSION

FIGURES INDICATE THE LIKELIHOOD IN % THAT THE SYMPTOMS (TOP, HORIZONTAL) WILL OCCUR WHERE CAUSES (LEFT, VERTICAL) ARE INVOLVED.

REV. 11/20/68 3/25/68

Figure 5. Continued.

POSSIBLE CAUSES OF DIFFICULTY

CHART NO. 3

CONDITIONS LIKELY TO BE RESPONSIBLE FOR DIFFICULTIES

Operation (columns "Started too quickly" to "Phase fault") — Installation: Foundation (columns "Resonant" to "Cross-beams, key-bars, or horizontal stiffness insufficient")

Cause of Vibration	Started too quickly	Loaded too quickly	Sudden temp. variation (8°F/min. = max.)	Sudden press. variation	Fluid slugging	Carry over	Boiler priming	Chemicals in steam	Case filled with fluid	Fluid sucking up from drains	Trapped fluid flashing up	Dirt in oil	Drains not closed after warmup	Drains closed too early	Sealing steam left on standg. rotor	Purge gas left on standg. rotor	Hot leakage getting on standing rotor	Cooling air on hot standing rotor	Exhaust was overheated	Excessive vacuum	Not operated at design speed	Over loaded	Off-design p & t	Surging	Shaft currents	Incorrect leakoff pressure	Fluctuat. leakoff pressure	Hollow rotor filling w. fluid (for boroscope hole)	Syncronized out of phase	Mad short circuit	Phase fault	Resonant	Settling or shrinking	Unevenly heated (hot lines too close)	Not separated from building	GROUT SWELLING OR SHRINKING, RUST OR DETERIORATION UNDER SOLE PLATES	CROSS-BEAMS, KEY-BARS, OR HORIZONTAL STIFFNESS INSUFFICIENT
Unbalance	✓	✓	✓	✓	✓	✓	✓	✓	✓												✓	✓	✓	✓				✓	✓	✓	✓						
Permanent bow or lost rotor parts (vanes)	✓	✓	✓		LOOK FOR LOST BLADES, ETC. ✓ DUE TO CORROSION FATIGUE OR STRESS CORROSION	✓✓	✓✓	✓✓	✓	✓	✓			✓	✓		✓	✓	LOOK FOR LOST BLADES ✓ DUE TO EXCESSIVE BLADE LOAD	✓✓	✓	✓✓	✓✓	✓					✓	✓	✓					✓	✓
Temporary Rotor bow	✓✓	✓✓	✓✓	✓	✓✓	✓	✓		✓✓	✓✓	✓✓	✓		✓	✓✓	✓✓	✓✓	✓✓	✓					✓		✓		✓	✓	✓	✓					✓	✓
Casing distortion	✓✓	✓✓	✓✓	✓	✓✓	✓	✓		✓✓	✓	✓		✓	✓					✓✓				✓										✓	✓		✓	✓
Foundation distortion																													✓✓	✓✓	✓		✓	✓	✓	✓✓	✓✓
Seal rub	✓✓	✓✓	✓✓	✓	✓✓	✓	✓		✓	✓✓	✓	✓	✓	✓	✓	✓	✓	✓	✓					✓✓		✓	✓	✓	✓	✓		✓	✓✓	✓✓	✓	✓✓	✓✓
Rotor rub axial	✓	✓✓	✓✓	✓	✓✓																			✓✓	✓							✓	✓	✓	✓		
Misalignment	✓	✓	✓	✓	✓				✓✓										✓✓				✓✓	✓					✓✓	✓✓	✓		✓✓	✓✓	✓	✓✓	✓✓
Piping forces				✓																			✓	✓✓									✓✓	✓✓	✓		✓✓
Journal & bearing eccentricity																																					
Bearing damage	✓✓	✓✓	✓✓	✓	✓✓							✓✓							✓		✓		✓	✓	✓✓			✓	✓	✓		✓	✓✓	✓✓		✓✓	✓
Bearg. & Support excited vibration (oil whirls, etc.)	✓	✓	✓	✓	✓																✓	✓	✓					✓				✓✓	✓	✓	✓✓	✓✓	✓✓
Unequal bearing stiffness horizontal - vertical																																				✓✓	✓✓
Thrust brg. damage	✓	✓		✓✓	✓✓	✓	✓				✓	✓✓								✓	✓	✓	✓✓	✓✓	✓✓	✓	✓										
Insufficient tightness in assy. of rotor (shrink fits)	✓	✓	✓✓	✓✓	✓✓	✓✓	✓		✓	✓✓	✓✓			✓		✓		✓	✓		✓	✓✓	✓✓	✓			✓		✓	✓							
Brg. liner																								✓					✓	✓							
Brg. case																																					
Casing & support	✓	✓	✓		✓	✓			✓										✓✓				✓✓	✓					✓✓	✓✓		✓	✓			✓✓	✓✓
Gear inaccuracy					✓							✓✓												✓					✓✓	✓✓	✓						
Coupling inaccuracy or damage					✓✓	✓✓						✓✓		✓							✓	✓✓		✓✓	✓✓				✓✓	✓✓	✓	✓	✓	✓		✓✓	✓✓
Rotor & brg. system critical																																✓✓				✓✓	✓✓
Coupling critical																																				✓	✓
Overhang critical																																					
Structural resonance																																✓✓					
Casing																																				✓	✓
Supports																																				✓✓	✓✓
Foundation																																				✓✓	✓✓
Pressure pulsations									✓				✓										✓	✓		✓	✓										
Electrically excited vibr.																													✓	✓	✓	✓✓			✓✓		
Vibr. transmission																																✓✓			✓✓	✓✓	
Valve vibr.				✓	✓	✓	✓															✓	✓	✓													
MAKE OF [illegible]																																					
Sub-harmonic resonance																																✓✓			✓✓	✓✓	✓✓
Harmonic resonance																																✓✓			✓✓	✓✓	✓✓
Friction induced whirl	✓	✓	✓	✓								✓✓																								✓✓	✓✓
Critical speed																																				✓✓	✓✓
Resonant vibr.																																✓✓			✓✓	✓✓	✓✓
Oil whirl				✓																																✓✓	✓✓
Resonant whirl																																✓✓			✓✓	✓✓	✓✓
Dry whirl	✓	✓	✓																																		
Clearance induced vibr'ns.	✓	✓	✓	✓																												✓			✓	✓✓	✓✓
Torsional resonance																					✓✓	✓	✓	✓					✓✓	✓✓	✓✓					✓	✓
Transient torsional	✓✓	✓✓																			✓	✓	✓	✓					✓								

✓ LIKELY TO OCCUR ✓✓ FREQUENT CAUSES OF DIFFICULTY

Figure 6. Example howing "Trouble-Shooting Chart"—Operational Problems.

POSSIBLE CAUSES OF DIFFICULTY, CONTINUED

CHART NO. 4

	INSTALLATION — FOUNDAT. CONT'D. →		PIPING — Piping →														Electrical →				Process, Design or Manufacturing		
Cause of Vibration	Soleplates loose	Rigidity Insufficient	Excessive forces & moments	Expansion joints not properly installed	Not properly supported	Not properly sloped & drained	Resonant	Not taking off at top of headers	Water accumulates over valves	Casing drains run to common sewer or common header or into water	Traps not working	Drain pots undersize	Drain lines undersize	Pipe expansion restricted by contact with foundation or other pipes	Branch lines restricting expansion	Dead ends not drained		Reverse-current relay failed	Syncronous motor starting pulsations excessive for system	Starting cycle improperly timed	Not covered in detail in this paper. Where high probability of error is indicated, contact manufacturer.	COMMENTS	REFERENCES.
Unbalance																					✓	Long high-speed rotors often require field-balancing at full speed to make adjustments for rotor deflection and final support conditions. MAKE CORRECTIONS AT BALANCING RING OR COUPLING.	(16)
permanent bow or lost rotor parts (vanes)						✓✓		✓✓	✓✓	✓	✓	✓	✓	✓	✓	✓			CHECK FOR TORSION CRACKS ✓✓	✓	✓	Bent rotors can sometimes be straightened by "hot-spot" procedure, but this should only be regarded as a temporary solution because bow, will come back in time, and several rotor failures have resulted from this practice. IF BUCKETS, VANES, OR DISKS ARE LOST OR CRACKED CHECK FOR CORROSION FATIGUE, STRESS-CORROSION, RESONANCE AND OFF-DESIGN OPERATION.	(15)
Temporary Rotor bow						✓✓		✓✓	✓✓	✓✓	✓✓	✓✓	✓✓	✓	✓	✓✓		✓	✓	✓	✓	Straighten bow slowly, running on turning gear or at low speed. If rubbing occurs, trip unit immediately and keep rotor turning 90° by shaft-wrench every 5 min. until rub clears. Then resume slow run. THIS MAY TAKE 20 HOURS.	
Casing distortion		✓✓	✓✓	✓	✓✓	✓		✓	✓	✓	✓✓	✓	✓✓	✓✓	✓✓	✓		✓			✓	Often requires complete re-work or new case, but sometimes a mild distortion corrects itself with time (requires periodic internal and external re-alignment). Usually caused by excessive piping forces or thermal shock.	(18) (19) (20)
Foundation distortion		✓✓	✓	✓	✓									✓	✓						✓	Usually caused by poor mat under foundation, or thermal stress (hot spots), or unequal shrinkage. Usually requires extensive and costly repairs.	(2) (17)
Seal rub	✓	✓✓	✓✓	✓✓	✓✓	✓✓		✓✓	✓✓	✓✓	✓✓	✓✓	✓✓	✓✓	✓✓	✓✓		✓	✓	✓	✓	Slight rubs may clear, but trip unit immediately if a high-speed rub gets worse. Turn by hand until clear.	
Rotor rub axial		✓	✓	✓	✓	✓✓		✓✓	✓✓	✓	✓	✓✓	✓✓	✓	✓	✓✓		✓	✓	✓	✓	Unless thrust bearing has failed this is caused by rapid changes of load and temperature. Machine should be opened and inspected.	
Misalignment	✓	✓✓	✓✓	✓✓	✓✓									✓✓	✓✓			✓			✓	Usually caused by excessive pipe strain and/or inadequate mounting and foundation, but sometimes also by local heat from pipes or sun on base and foundation.	(2) (17) (18) (19) (20)
Piping forces		✓✓	+	✓✓	✓✓									✓✓	✓✓						✓	Most trouble is caused by poor pipe supports (should use spring hangers), improperly used expansion joints, and poor pipe line-up at casing connections. Foundation settling also can cause severe strain.	(18) (19) (20)
Journal & bearing eccentricity																					✓✓	Bearings may get distorted from heat. Make hd-check, if possible, checking contact.	
Bearing damage	✓	✓✓	✓	✓	✓	✓✓	✓	✓✓	✓✓	✓	✓	✓	✓	✓	✓	✓		✓	✓	✓	✓	Watch for brown discoloration which often precedes re-curring failures. This indicates very high local oil-film temperatures. Check rotor for vibration. Check bearing design and hot clearances.	
Bearg. & Support excited vibration (oil whirls, etc.	✓✓	✓✓	✓	✓	✓		✓							✓✓	✓						✓	Check clearances and roundness of journal, as well as contact and tight bearing fit in case. Watch out for vibration transmission from other sources and check frequency. May require anti-whirl bearings or tilt-shoe bearings. CHECK ESPECIALLY FOR PIPING OR FOUNDATION RESONANCE AT WHIRL FREQENCY.	(10) (11)
Unequal bearing stiffness horizontal - vertical	✓	✓																			✓✓	Can excite resonances and criticals and combinations thereof at 2 x running frequency. Usually difficult to field-balance because as horizontal vibration gets better, vertical gets worse and vice versa. It may be necessary to increase horizontal bearing support stiffness (or mass) if the problem is severe.	(16)
Thrust brg. damage						✓✓		✓✓	✓✓	✓	✓✓	✓✓	✓✓			✓✓					✓	Usually the result of slugging the machine with fluid, solids built up on rotor, or off-design operation, especially surging.	
Insufficient tightness in assy. of rotor (shrink fits)						✓✓		✓✓	✓✓	✓✓	✓✓	✓✓	✓✓			✓✓		✓	✓	✓	✓✓	The frequency at rotor-support critical is characteristic. Disks and sleeves may have lost their interference fit by rapid temperature change. Parts usually are not loose at standstill.	(11) (12)
Brg. liner			✓	✓	✓									✓	✓			✓	✓	✓	✓✓	Often confused with oil whirl because characteristics are essentially the same. Before suspecting any whirl make sure everything in bearing assembly is absolutely tight with interference fit.	
Brg. case			✓	✓	✓									✓	✓						✓✓	This is rare, but should always be checked as a matter of routine.	
Casing support			✓✓	✓✓	✓✓									✓✓	✓						✓	Usually involves sliding pedestals and casing feet. Check for friction, proper clearance, and piping strains.	
Gear inaccuracy																					✓✓	To get frequencies, tape microphone to gear case and record noise on magnetic tape.	
Coupling inaccuracy or damage		✓	✓✓																✓	✓	✓✓	Loose coupling sleeves are notorious troublemakers! Especially in conjunction with long, heavy spacers. Check tooth fit by placing indicators on top, then lift with hand or jack and note looseness. Should not be more than 1-2 mils at standstill, at most. Use hollow coupling spacers. MAKE SURE COUPLING HUBS ARE TIGHTLY MOUNTED	
Rotor & brg. system critical	✓	✓✓																			✓✓	Try field balancing; more viscous oil (colder); larger, longer bearings with minimum clearance and tight fit; stiffen bearing supports and other structures between bearing and ground. Basically a design problem. May require additional stabilizing bearings or solid coupling. Difficult to correct in field. Sometimes adding mass at bearing case helps considerably, ESPECIALLY AT SPEEDS EXCEEDING 5000 RPM.	(3) (4) (5) (6) (7)
Coupling critical																					✓✓	These are criticals of the spacer-teeth-overhang sub-system. Often encountered with long spacers. Make sure of tight fitting teeth with slight interference at standstill, and make spacer as light and stiff as possible (tubular) Consider solid coupling if problem is severe. Check coupling balance!	
Overhang critical																					✓✓	Can be exceedingly troublesome. ~~Long~~ overhangs shift the nodal point of the rotor deflection line towards the bearing, robbing the bearing of its damping capability. This can make criticals so rough that it is impossible to pass through the critical. Shorten overhang or put in an outboard bearing for stabilization. Use lighter coupling & spacer	(3) (4) (5) (6) (7)
Structural resonance of																							
Casing				✓	✓✓		✓✓							✓✓	✓✓						✓	So called "case-drumming". Can be very persistent but is sometimes harmless. Danger is that parts may come loose and fall into the machine. Also, rotor casing interaction may be involved. Diafram drumming is serious.	
Supports	✓✓			✓✓	✓✓		✓✓							✓✓	✓✓						✓✓	Local drumming is usually harmless, but major resonances, resulting in vibration of entire case as a unit are potentially dangerous because of possible rubs and component failures, as well as possible excitation of other vibrations.	
Foundation	✓✓	✓	✓	✓✓	✓✓		✓✓							✓✓	✓						✓✓	As above, but with the added complications of settling, cracking, warping and misalignment. Also may get into piping troubles and possible case warpage. Foundation resonance is serious and greatly reduces unit reliability.	(2) (17)
Pressure pulsations				✓✓			✓															Can excite other vibrations, possibly with serious consequences. Should eliminate such vibrations by using restraints, flexible pipe supports, sway braces, shock absorbers, etc., plus isolation of foundation from piping, building, basement and operating floor (2).	(2)(10) (11)
Electrically excited vibr.							✓✓							✓					✓	✓	✓✓	Mostly at 2 x line frequency (7200 CPM), coming from motor and generator fields. Turn fields off to verify source. Usually harmless, but if foundation or other components (rotor !) are resonant the vibrations may be severe. In such cases one must count with possible damage in case of short circuit or other upsets.	(2)
Vibr. transmission		✓		✓✓	✓✓		✓✓							✓✓	✓						✓	Can excite serious vibrations or cause bearing failures. Isolate piping and foundation and use shock absorbers and sway braces.	(2) (10) (11) (17)
Valve vibr.																					✓✓	Fairly rare but sometimes very violent. These are aerodynamically excited. Change valve shape to reduce turbulence. Increase rigidity of valve gear.	
NAME OF PROBLEM																							
Sub-harmonic resonance	✓✓	✓✓		✓	✓		✓							✓	✓						✓	Vibration at exactly 1/2, 1/4, 1/8 of exciting frequency. Can only be excited in non-linear systems, therefore, look for such things as looseness and aerodynamic or hydrodynamic excitations. It may involve rotor "Shuttling", then check seal system, thrust clearances, couplings, and rotor-stator clearance effects.	
Harmonic resonance		✓		✓	✓		✓							✓	✓						✓	Vibration at 2x, 3x, 4x, exciting frequency. Fairly common and troublesome. Treatment is the same as for direct resonance: Change frequency and add damping.	
Friction induced whirl		✓				✓✓		✓✓	✓✓	✓✓	✓✓	✓✓	✓✓								✓✓	Can be very serious. If intermittent look into temperature variations. Usually rotor must be re-built, but should try to increase stator damping first: larger bearings (tilt shoe), increased stator mass and stiffness, better foundation. This problem is usually caused by mal-operation such as quick temp changes and fluid slugging.	(12) (13) (1
Critical speed		✓	✓	✓	✓																✓✓	Basically a design problem but often aggravated by poor balancing and poor foundation. Try to field balance rotor at operating speed, lower oil temperature, use larger and tighter bearings.	(3) (4) (5) (6) (7)
Resonant vibr.	✓✓	✓✓		✓✓	✓✓		✓✓							✓✓					✓	✓	✓✓	Add mass or change stiffness to change frequency. Add damping. Reduce excitation. Improve system isolation. Reducing mass or stiffness can leave the amplitude the same even if resonant frequency shifts. Check "mobility."	(2) (3)
Oil whirl																					✓	Stiffen foundation or bearing structure, add mass at bearing, increase critical speed, or use tilt-shoe bearings which is the best solution. CHECK FOR LOOSE BEARINGS FIRST!	(10) (11)
Resonant whirl	✓✓	✓✓		✓✓	✓✓		✓✓							✓✓							✓	As above, but with additional resonance of rotor, stator, foundation, piping; or externally excited. Find resonant members and sources of excitation. Tilt shoe bearings are best. CHECK FOR LOOSE BEARINGS!	(10) (11)
Dry whirl			✓																		✓	The "squeal" of a bearing or seal. May be ultra-sonic and very destructive. Check for rotor vanes hitting stator, etc., especially if clearances are smaller than oil film thickness plus rotor deflection while passing through critical speed.	
Clearance induced vibr'ns.	✓✓		✓✓	✓	✓✓		✓							✓	✓						✓	Usually rocking motions and beating within clearances. Serious especially in bearing assembly. Frequencies are often below running frequency. Make sure everything is absolutely tight, with some interference. Line-in-line fits are usually not sufficient to positively prevent this type of problem.	
Torsional resonance																				✓✓	✓✓	Very destructive and hard to find. Symptoms: Gear noise, wear on back side of teeth, strong electrical noise or vibration, loose coupling bolts, fretting corrosion under coupling hubs. Wear on both sides of coupling teeth, torsional fatigue cracks in keyway ends. Best solution: Install properly tuned torsional vibration dampers.	
Transient torsional																			✓✓		✓✓	As above, but encountered only during start and shutdown due to very strong torsional pulsations. Occurs with: Reciprocating machinery, syncronous motors. CHECK FOR TORSIONAL CRACKS.	(8) (9)

Figure 6. Continued.

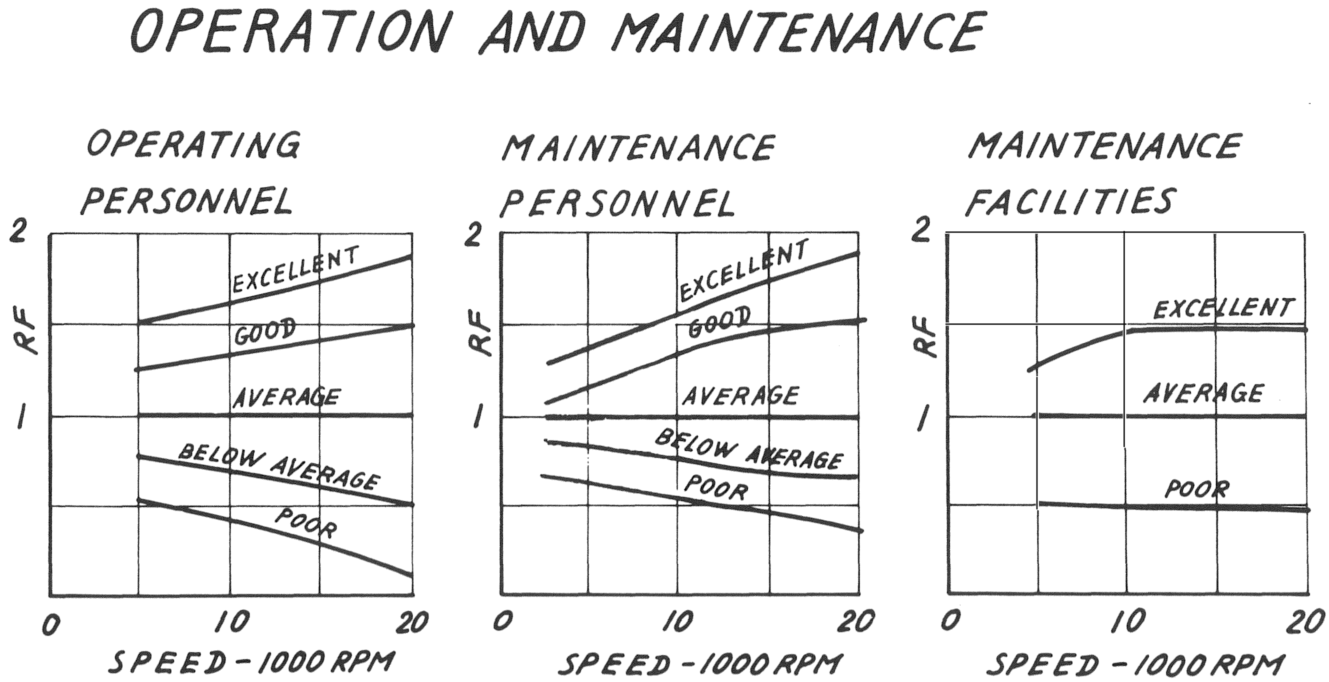

Figure 4.

But vibration can mean many things, and if we know how to interpret vibration phenomena, we can get a very good idea of what is wrong.

Of course, much has been written and said about vibration, but much of this is highly technical information—written by vibration experts for other vibration experts—and little information gets to the man on the job. The main reason for this is the fact that vibration phenomena are highly complex and require a great deal of mathematics. Unfortunately, one rarely has the time or opportunity to read a mathematical dissertation when he is faced with a serious field problem. In other words, there is still a wide gap between theory and practical application, although much progress has been made in the theoretical interpretation of vibration phenomena during the last few years. There is a great need to transmit the essence of this theoretical information to the man on the operating floor. Naturally, generalizations must be made, and many explanations may have to be oversimplified to accomplish this. But what we need is a cookbook—to simplify the first steps when trouble occurs—and such a tool has to be stripped of all excess baggage which is not absolutely required to do the job. To do this, we can develop a set of charts, showing the correlation between causes and effects, so as to show what type of vibration to expect from a certain cause—or the other way around—what type of problem may arise for certain faults in installation or operation (Fig. 5 and Fig. 6) (Ref. 3). Numbers shown indicate the percentage of units which will develop the symptoms shown at the top of the chart. These illustrations show a part of such a tabulation. The vertical column lists various types of vibrations, or causes of vibration, while the horizontal heading shows the symptoms or evidence which the trouble shooter sees. For example, it can be seen that a rotor unbalance has a typical frequency at 1 x RPM, misalignment frequencies are 1 or 2 x RPM, loose rotor parts cause frequencies at the resonant frequencies of the rotor—which may be any odd frequency in the lower-frequency group. This chart then shows which components are suspect, and which components could not cause a specific problem. The charts are extended horizontally to include installation problems and operation problems.

Fig. 7 shows the frequency characteristics of some major problems. This should help to identify the basic mechanism involved. The vertical axis indicates frequency of vibration, the horizontal axis shows speed, both expressed as multiples of critical speed. Plotted versus speed is a line showing 1 x RPM, which is characteristic of unbalance. Also shown are oil whirl frequencies, characteristic frequencies of internal friction whirls, harmonic and subharmonic resonances, and misalignment frequencies. For example, if a machine with a hysteresis whirl problem were to be run up to speed, the frequency would come in somewhere along the resonant frequency line and would remain constant as speed increases. The frequency of a pure oil whirl would go up at about ½RPM with speed. A resonant whirl would suddenly appear either at ½ frequency or at resonant frequency, and so on. With this data one can then enter the charts and make a list of the things to look for, starting with the item having the greatest probability, as indicated by the percentage figures shown in Fig. 5 and 6.

This way, the choice can be narrowed down to some half dozen possibilities, and these must then be explored in detail, which may involve getting more data and checking into references.

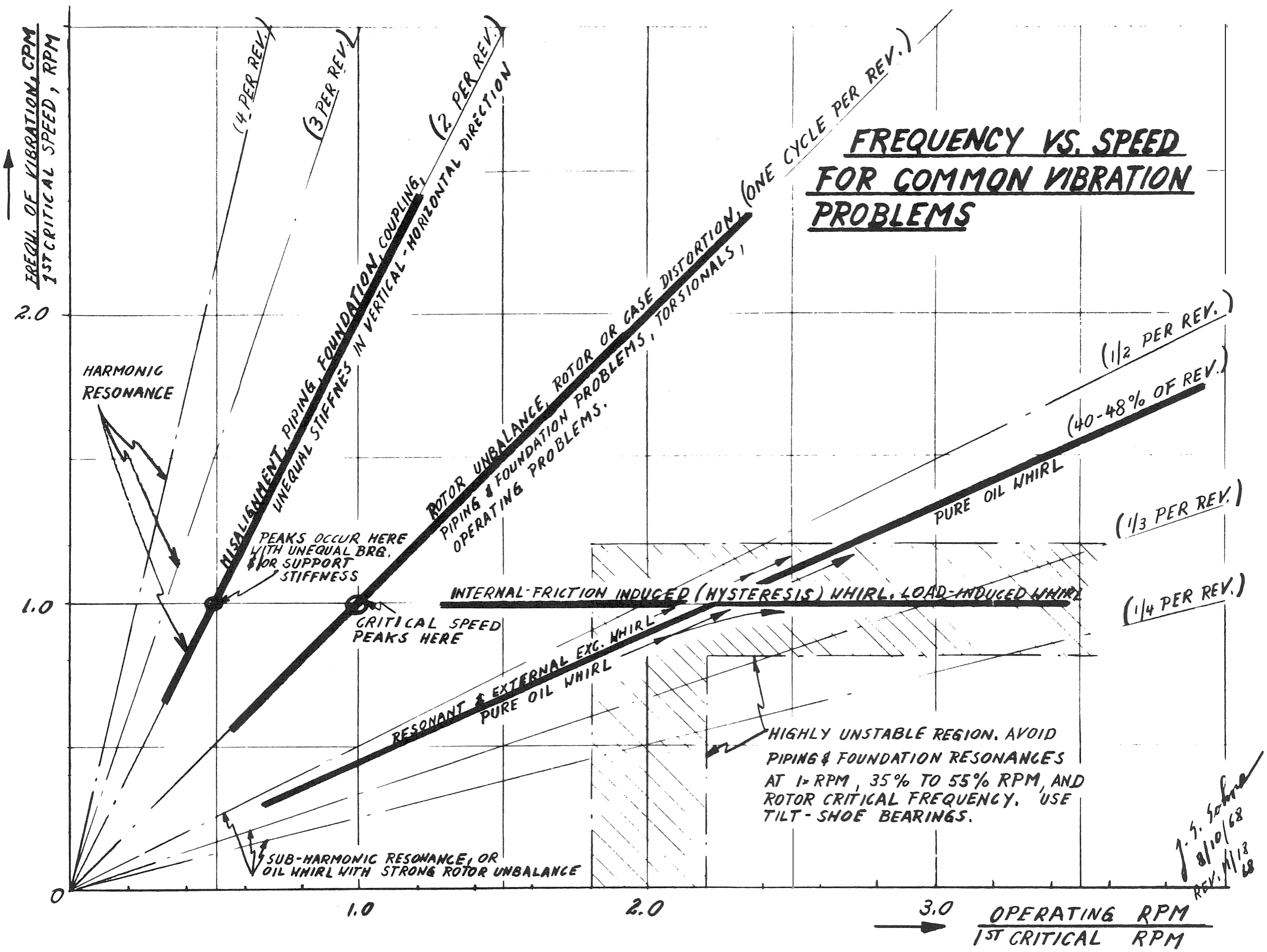
FREQUENCY VS. SPEED FOR COMMON VIBRATION PROBLEMS
FREQU. OF VIBRATION, CPM / 1ST CRITICAL SPEED, RPM
OPERATING RPM / 1ST CRITICAL RPM
0
1.0
2.0
3.0
(4 PER REV.)
(3 PER REV.)
(2 PER REV.)
(ONE CYCLE PER REV.)
(1/2 PER REV.)
(40-48% OF REV.)
(1/3 PER REV.)
(1/4 PER REV.)
MISALIGNMENT, PIPING, FOUNDATION, COUPLING, UNEQUAL STIFFNESS IN VERTICAL-HORIZONTAL DIRECTION
ROTOR UNBALANCE, ROTOR OR CASE DISTORTION, PIPING & FOUNDATION PROBLEMS, TORSIONALS, OPERATING PROBLEMS.
HARMONIC RESONANCE
PEAKS OCCUR HERE WITH UNEQUAL BRG. &/OR SUPPORT STIFFNESS
CRITICAL SPEED PEAKS HERE
INTERNAL-FRICTION INDUCED (HYSTERESIS) WHIRL, LOAD-INDUCED WHIRL
PURE OIL WHIRL
RESONANT & EXTERNAL EXC. WHIRL
PURE OIL WHIRL
HIGHLY UNSTABLE REGION. AVOID PIPING & FOUNDATION RESONANCES AT 1× RPM, 35% TO 55% RPM, AND ROTOR CRITICAL FREQUENCY. USE TILT-SHOE BEARINGS.
SUB-HARMONIC RESONANCE, OR OIL WHIRL WITH STRONG ROTOR UNBALANCE
J. S. Sohre
8/10/68
REV. 11/18/68

Figure 7.

SOME PROBLEMS CAUSED BY IMPROPER OPERATION

Many vibration problems and shaft failures are caused by loose fit of wheels and sleeves on the shaft. This may be a result of design weakness, poor quality control, or faulty maintenance, but one should remember that quite frequently a loose sleeve or even a loose disk can be the result of careless operation, mainly chilling of a hot turbine rotor with cold steam or water. A quick drop of inlet temperature causes the sleeves and wheels to contract. Since the shaft does not cool as quickly as the sleeves or disks, the cool sleeves squeeze the shaft, and a very high tensile stress at the bore of the sleeve is the result. If the temperature difference between the sleeves and the shaft is great enough, the sleeve will yield in tension.

When the temperature of sleeve and shaft equalizes again under normal operating conditions, the sleeve will be permanently loose on the shaft and this may lead to a friction-induced whirl or it may initiate a shaft failure. The temperature difference between shaft and sleeve to get yielding is around 250°F, and with 700°F inlet temperature or more, a slug of water can easily produce this temperature difference. So can a sudden drop of steam temperature, especially when the turbine carries substantial load.

The same problem can result from improper operation of the sealing steam system. If water can collect in improperly installed leak-off or drain lines—or if these lines go into a common tank which may fill with water—cold water may be sucked up or flashed up into the hot turbine during load changes or after trip. After a condensing turbine trips, the entire turbine is exposed to the condenser vacuum, and this can suck the water up into the turbine, or flash the water to produce cool steam which then enters the turbine. This has often caused severe damage such as loose sleeves and bent rotors.

Especially to be deplored is the practice of combining drain lines and leak-off lines into one long common header, or combining any leak-off or drain lines. The sealing system on a turbine offers many possibilities to damage the machine, and one should always be extremely careful with its installation and operation. Steam and water in this system can flow back under certain conditions. It can very well flow back up into the shaft seal if no provisions are made to prevent it from doing so. Also, sealing steam lines which are shut off during normal operation may fill with water, and when the turbine is unloaded or tripped, the sealing steam regulator opens to supply steam to the glands. This may shoot cold water into the seal and against the hot shaft, where it can loosen sleeves or cause the rotor to bend. These problems are usually not under the control of the manufacturer, they are often caused by oversight in installation or operation.

Another item deserving close attention is the problem of corrosion fatigue. This is caused by corrosives in the air, or gas, or steam, as the case may be. Even very small amounts of certain corrosives can have a serious effect on material properties, especially on fatigue strength. This has caused many failures on compressor blading, with both air and process gases. Operating personnel can do a lot about this, for example by observing the air going into a compressor. An observant person can often see a compressor take a gulp of nitrous acid fumes, or ammonia, or hydrogen sulfide, or other process gases which may be blown off or leak out of the system, especially during a plant start-up. With the wind blowing in the right direction, the gases can easily get into the inlets, starting failures. The blades or impeller-vanes may fail at once, or they may fail years later—and in this case it can be extremely difficult to establish the cause of the failure and to provide the proper corrective action. This situation is even more difficult to detect where gas compressors are involved and process gases are contaminated with corrosives, or where corrosives get into the steam of a turbine. Many bucket failures have been caused by this, but those failures have often been attributed t● other causes, with the result that the failures recurred.

Fig. 8 shows the fatigue strength diagram for a typical 13% cromium stainless steel, as used for turbine buckets. Shown is the strength versus the number of cycles for the usual, polished sample, in air. Note how the curve flattens out between one million and ten million cycles. This is called the endurance strength of the material. The material has an infinite life for stresses below this line, in other words, increasing the number of cycles will not produce a failure.

The next curve shows the same fatigue strength data in pure condensate. A significant reduction in fatigue strength is evident.

The same polished example in a salt solution is also shown. The fatigue strength is only a fraction of its original value, but even more important is the fact that the curve no longer levels out, but continues to drop. This means, the part no longer has an endurance limit—only a limited life. The part will fail ultimately, no matter how low the applied stress may be, provided enough cycles are applied—and this is usually the case in an industrial application.

A turbine blade is not a polished specimen. It has many geometric stress concentrations, for example where the blade joins the root or in the root itself. The lower curve is for a sample with a similar stress concentration factor (about 2.2). It can be seen that the material has lost 90% of its fatigue strength at 100 million cycles, and that the curve is still going down. In other words, the part must fail, no matter how low the actual applied stress might be. This is the situation we face in a turbine if we have chemicals in the steam. How severe the situation actually is depends on the type and concentration of the corrosive, as well as upon static and vibratory stress level. Failures can occur within minutes, because the blades are exposed to very high frequencies of excitation as they pass the stationary nozzles. To get a million cycles may take from five minutes to several hours—depending on the number of nozzles and on the speed—but sometimes the failure occurs after years, because of cumulative effects. A blade may get some chemicals today and some more next week and finally, after enough cycles have accumulated while under the influence of chemicals—the blade fails.

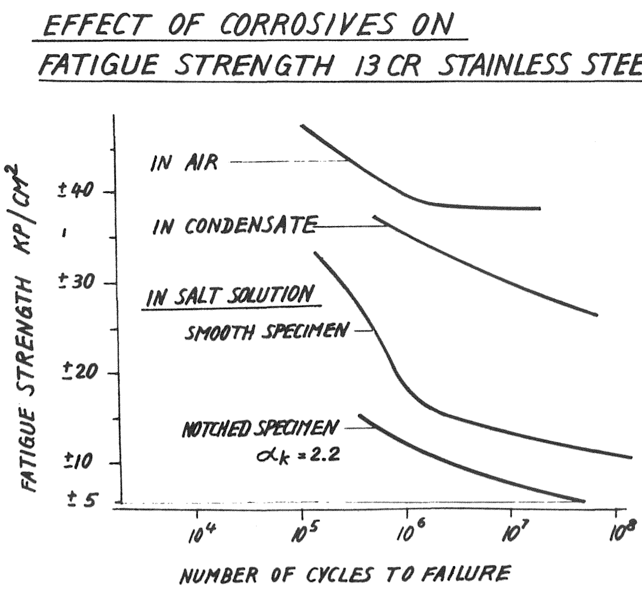

Figure. 8.

Such failures sometimes have a peculiar appearance, which clearly identifies them as corrosion fatigue failures.

Fig. 9* from Ref. (4) shows the fracture surface of a simple pull test in air, oxygen, hydrogen sulfide, and hydrochloric acid vapor. Below this are fatigue test specimen for the same gases, hydrochloric acid vapors, and hydrogen sulfide. At the top is a fractured turbine blade showing the same type of fracture discoloration as the samples.

Fig. 10* shows a fractured blade from a turbine which operated only 70 hours during plant start-up: It shows the type of colors of the hydrogen sulfide contamination. There were about 16 cracked blades in this wheel, two blades were actually lost. The surface of this fracture was exposed by breaking a cracked blade open. The white fracture area where the blade was cracked open in the vise can be clearly seen. The golden area is the corrosion fatigue failure which progressed during operation.

Fig. 11* shows a corrosion-fatigue failure caused by chlorides. Again, it can be seen where the blades were broken in the vise, as contrasted to the corrosion fatigue failure area. About 60 blades were lost or cracked in five out of ten stages of this machine before the situation was corrected. The machine ran for seven years before the first failure occurred. Apparently, boiler feed water, containing make-up, was getting into the desuperheater through a leaking valve.

If this type of problem occurs, no amount of redesign or bucket strengthening will prevent future failures with any degree of certainty. At best the machine may run a little longer between failures, but since the material actually has no endurance limit, only a limited life, the failures will re-cur sooner or later, unless the corrosives are eliminated.

Finally I would like to point out another source of turbine blade failures, especially in the last stage. If a turbine is operated at full throttle with excess vacuum, the pressure drop across the last row of blades becomes very great and the steam loading on the blade will become excessive. This causes cycling stresses in the blade which may exceed the fatigue limit, causing fatigue failures. If the cooling water gets cold in the winter and vacuum increases above the design limit, something should be done about it either by throttling the circulating water supply, or by bleeding air into the condenser. Many people believe that they get more power out of a turbine when they have a vacuum exceeding the design point. This is not necessarily true, because the steam velocities in the last row of blading may become supersonic, and no more power is really extracted from the steam. In fact, efficiency is often reduced. The result of excess-vacuum operation is an extremely heavy buffeting of the blades which in many cases has led to blade failures.

This situation actually presents an excellent opportunity to improve turbine water-rate under normal and part-load conditions. If a vacuum-controller is used in such a way as to increase vacuum at part-load and to reduce vacuum at overload, the blade exit velocity (Mach. No.) can be held constant. The resulting improvement in water rate, overload capacity, and reliability can be very significant.

It goes without saying that, if chemicals are admitted while operating at excess vacuum, blade failures are very likely to occur.

Of course, we realize that there are also other factors which can cause blade failures, such as excessive design stress at a blade resonant frequency, poor surface

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*Color slides. Not reproduced in the proceedings.

finish, inaccuracies in the blade roots, and so on. But these are design problems, while we are concerned with operating problems and with methods to prevent failures and breakdowns by proper installation and operation of the equipment.

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