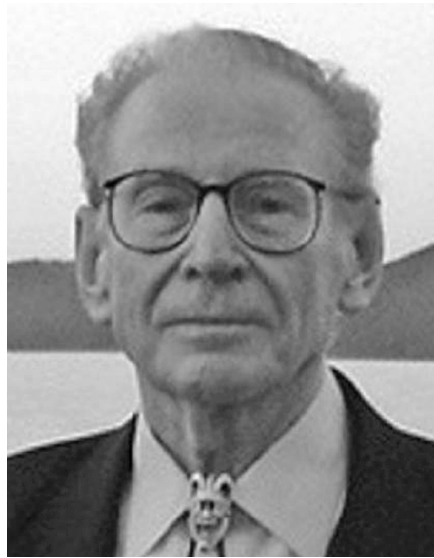
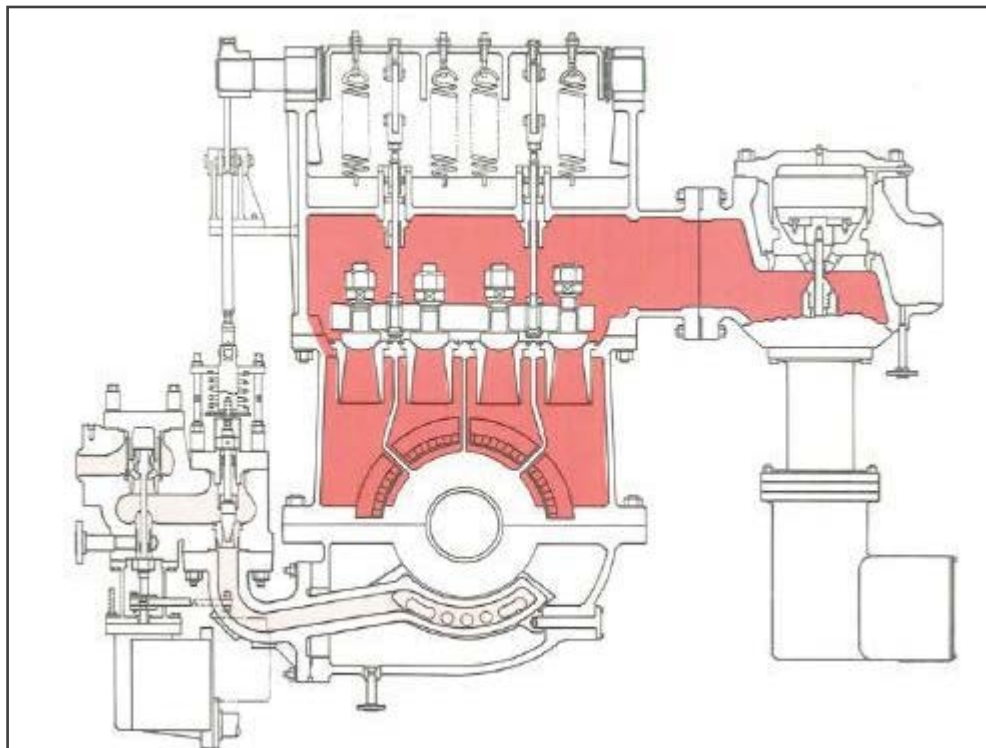




JOHN SOHRE
1927 - 2010



Dedication

It is rare that one comes across a person of the stature of John Sohre who contributed so significantly to the turbomachinery industry and positively influenced engineers for over three decades. A bulk of his seminal work was done at a critical juncture in the evolution of high speed turbomachinery when new problems and design concepts were being developed and machines were outpacing the analytical tools available at the time. John had the ability to blend superb theoretical work with practical and field engineering expertise. He made prolific contributions to turbomachinery design, practical troubleshooting and shaft current technology with his work being published at the Turbomachinery Symposium and at ASME forums. During his long and distinguished career, John worked as Design Section head at Elliott, and then at Terry Steam Turbines, where he was responsible for the thermodynamic and mechanical design of the JS line of turbines. He had a long and successful consulting career founding Sohre Turbomachinery in 1971, a leading company in the design and manufacture of shaft grounding brushes.

John was a man of great integrity and humility, and would always take the time to explain technical issues to me while I was a young inexperienced engineer. John's contributions have shaped and molded the consciousness of several turbomachinery specialists. He will long be remembered as one of the most distinguished engineers of our times. — *Cyrus Meher-Homji, Engineering Fellow and Senior Principal Engineer, Bechtel Corporation, Turbomachinery Symposium Advisory Committee Member*

John Sohre was a great engineer, human being and a leader in the Turbomachinery field.

In 1972 as chair of the First Advisory Committee we were looking for speakers to launch the First Turbomachinery Symposium. The name that was most prominent on the committees' mind was that of John Sohre, since most of us on the advisory committee had worked with him on many projects, and were familiar with his prodigious repertoire of technical papers and his ability to hold spellbound technical audiences. This being the First Symposium of its type in the field of Turbomachinery the speakers had to be excellent to ensure the success of the Symposium. Looking back from where we stand today, some forty years later, we can say without any hesitation that we achieved our goals, and thank our speakers at that first symposium like John Sohre that helped us reach our goal. He was not only one of our first speakers but was afterwards a regular speaker and a tutorial lecturer at the symposium from then on.

John and his wife Nora emigrated from East Germany after many trials and tribulations. He became a senior compressor designer at Elliott Company; subsequently he became chief engineer at Terry Steam Turbine of Windsor, Connecticut. There he was personally responsible for, and designed a state-of-the-art test facility, and also designed a new line of innovative, high-performance mechanical-drive steam turbines. Many of these machines are still in service, driving high speed compressors in refineries, or making electricity in power stations.

After retiring from Terry he became an internationally recognized turbomachinery consultant, and founded Sohre Turbomachinery, troubleshooting machinery worldwide, and writing technical papers still used and cited today. I was privileged to work with him on a number of consulting projects. He invented a high performance, high reliability bristle type fiber grounding brushes for use in shaft grounding (shaft earthing) of all types of stray electrical shaft currents. His invention makes steam turbines safer and more reliable. The company's products have been used on every continent, and are also on the international space station.

My wife Zarine and I had the pleasure of knowing John and his wife Nora for many years. They were good friends, and John was a good friend of the Turbomachinery Symposium and the Turbomachinery Laboratory. I also had the privilege of working with him on consultation projects and he was always a fountain of knowledge.

We have lost a great friend and engineer. The Turbomachinery community will greatly miss him but it has been enriched immensely by his many contributions. — *Meherwan P. Boyce, Chairman, The Boyce Consultancy Group, LLC, Turbomachinery Symposium Advisory Committee Founding Member*

TURBOMACHINERY ANALYSIS AND PROTECTION

by

John S. Sohre

Turbomachinery Consultant

Vernon, Connecticut



Born and educated in Germany. Graduated State Institute of Technology Chemnitz, Germany 1951. Professional Engineer (Pennsylvania). Member, American Society of Mechanical Engineers.

Mr. Sohre has had twenty-eight years experience in shop, operating departments, and design work. Eight years with Elliott Company (Section Head, Mechanical Design and Engineering Mechanics Section, Turbines and Compressors). Seven years with The Terry Steam Turbine Company as Chief Development Engineer. Had responsibility for development of Terry's type JS line of turbines for high-speed, high horsepower service. Since July 1970 in practice as an independent consultant for Turbomachinery Design, installation and problem correction.

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 overall 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

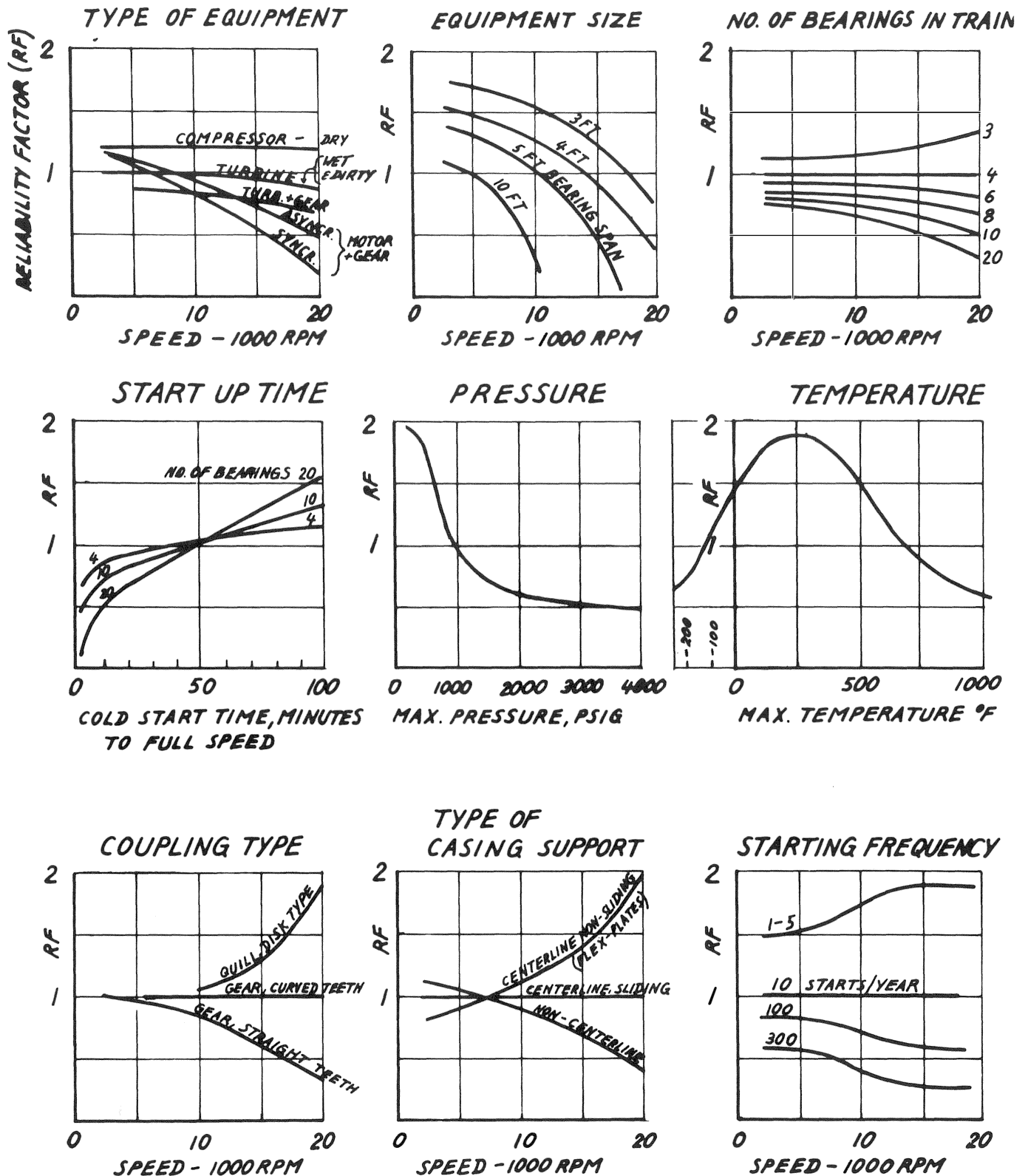


Figure 1.

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 num-

ber 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

PIPING

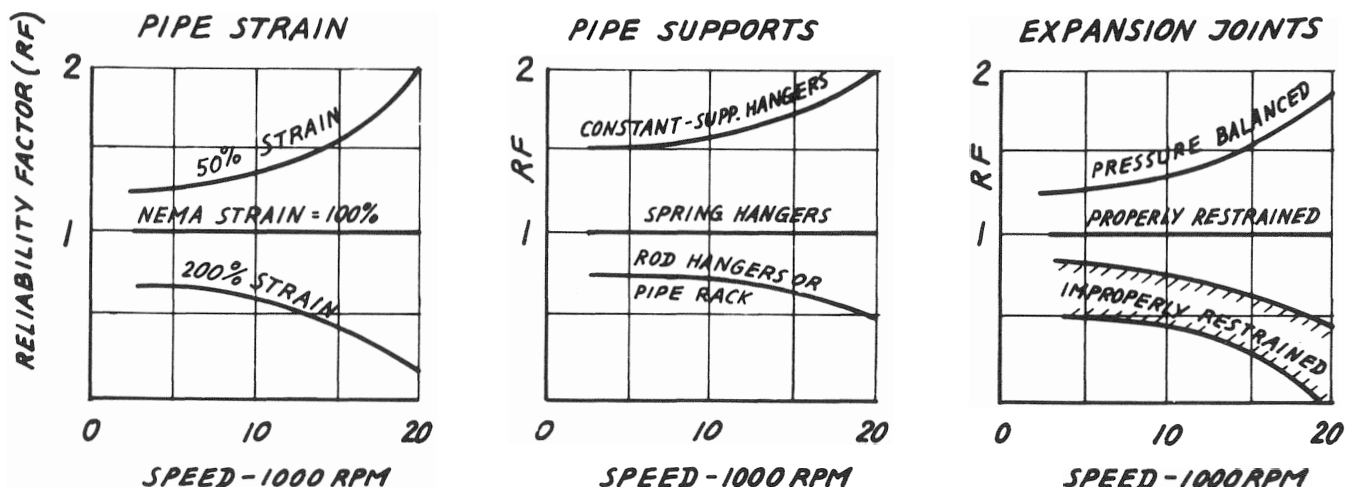


Figure 2.

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.

FOUNDATION CHARACTERISTICS

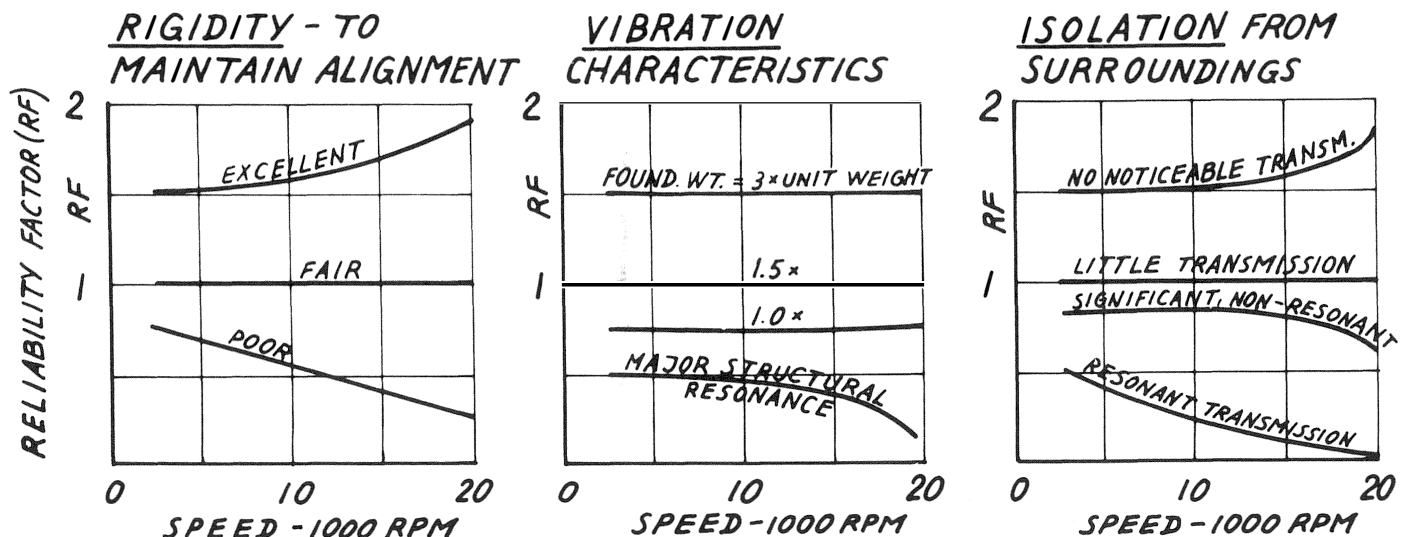


Figure 3.

SYMPTOMS AND DISTRESS MANIFESTATIONS

[illegible]

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

CHART NO. 2

OPERATIONAL EVIDENCE, CONT'D.

Cause of Vibration	Predominant Sound (DURING VIBRATION TEST RUN)								History of Machine				Damage or Distress (Signs)																								
	Low frequ. "Rumble"	Low "Roar"	"Hum"	Comes and goes periodically ("beat")	High pitched "whine"	Very high loud "scream"	Very high "squeal"	Ultrasonic (measure v. sound anal.)	Initial startup	1st year	1-10 years	over 10 years	Seals rubbed	Shaft bent	Bearings failed	Wiped	Fatigued	Rabbitt squeaked 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	Coupling bolts loose	Foundation settled or cracked	Soleplates loose or rusted	Sliding surface 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	
Unbalance		80	20						✓				10			10	10																				
Permanent bow or lost rotor parts (vanes)		80	20						✓		✓	✓	50	+	30	10	10																	20	20		
Temporary Rotor bow	10	70	10	10					✓	✓			90	10	50		20				20	20	20	5		5											
Casing distortion	10	70	10	10							✓	✓	90	10	50	10	10	+	10	20																	
Foundation distortion	10	50	30	10					✓	✓	✓	✓	90	10	50	30				50	30						+	30	30	10						50	20
Seal rub	20	40		10		10	10	10	✓	✓	✓	✓	+	15	15	10	10	10	20	10								10	10	10	10	10				20	10
Rotor rub axial	30	40		20				10	✓	✓			90	30	50						20							10	10	10	10			10	10		50
Misalignment		40	40	20					✓	✓	✓	✓	50	10	10	20		20	+	40				10		10		10	50	50	10	50					50
Piping forces	30	40	30						✓	✓	✓	✓	50	10	10	20											10	50	50	10	50					10	
Journal & bearing eccentricity		10	90						✓				50	10	10	20		40	40	40							10		10	20	50				50	10	
Bearing damage	20	40	10		10	10	10		✓	✓	✓	✓	90	10		+		10	40	20	20	10	10	20	20	20	20	20	20	20	40	10	10	10	10	10	
Bearing & Support excited vibration (oil whirl, etc.)	60	10		30					✓	✓			90	10	80	60	30				20	20	10	10	10											10	
Unequal bearing stiffness horizontal - vertical		10	60	20	10				✓				10		40	40																				10	
Thrust brg. damage	80	10	10						✓	✓	✓	✓	90	20	50		30																				
Insufficient tightness in Assy. of rotor (shrink fits)	60	20		20					✓	✓	✓	✓	90	10	60	60	40				80	40	40	20	50	10					60	30	60	60		50	
Brg. liner	60	20		20					✓	✓	✓	✓	90	10	60	60	40				80	40	40	20	50	10											
Brg. case	60	20		20					✓	✓	✓	✓	90	10	60	60	40				80	40	40	20	50	10											
Casing support	60	20		20					✓	✓	✓	✓	90	10	60	60	40				80	40	40	20	50	10											
Gear inaccuracy	20	10	10	20	20	10	10		✓	✓	✓	✓	20		40	60	10				20																
Coupling inaccuracy or damage	10		40	10	40				✓	✓			20		20	20					40																
Rotor & brg. system critical		50	30	20					✓	✓			50	10	50	50					30	20	10	5												10	
Coupling critical		80	40	20	20				✓	✓			30	10	50	50																				20	
Overhang critical		50	40	10					✓	✓			50	20	60	60	10				40																
Structural resonance																																					25
Casing	20		20	60					✓	✓			20		20	20																					
Supports	20		20	60					✓	✓	✓		20		20	20																					
Foundation		10	20	10					✓	✓	✓		20		20	20					20															10	
Pressure pulsations	40	20		40					✓	✓	✓	✓	30		30	30				30	30						30	20	10	20						10	
Electrically excited vibr.		10	80	10					✓	✓			30		30	30				20	30						30	20								10	
Vibr. transmission	20		20	60					✓	✓			20		20	20				30	10	10	5		5											15	
Valve vibr.	10	10				40	40		✓	✓			20		20	20				30	30															10	
NAME OF PHENOMENON																																					
Sub-harmonic resonance	80			20					✓	✓	✓		10		20	10					10	30	10	5		5	10	10									
Harmonic resonance			40	20	40				✓	✓			10		20	20					20	30	10	5		5	10	10									
Friction induced whirl	60	20		20					✓	✓	✓		90	10	60	60	40				80	40	40	20	50	10											
Critical speed		50	30	20					✓	✓			50	10	50	50					30	20	10	5													
Resonant vibr.	40		30	30					✓	✓			20		20	20				20																	
Oil whirl	60	10		30					✓	✓	✓		60	10	80	60	30				20	20	10	10	10		30	10	10	20							
Resonant whirl	60	20		20					✓	✓	✓		80	15	90	80	40				40	30	20	20	20	10	10										
Dry whirl						20	80		✓	✓	✓	✓	80	10	90	20	40				10	5	5	30	5	70											
Clearance induced vibrations	60	20		20					✓	✓			90	10	30	30	20				80	20	20	10	5	10											
Torsional resonance	10	20	20	30	10	10			✓	✓			5		5	20					40	80	80	40	5	40											
Transient torsional				60 (ROUGH)		20	20		✓	✓			5		5	5					30	80	80	50	10	50											

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

Figure 5. Continued.

CONDITIONS LIKELY TO BE RESPONSIBLE FOR DIFFICULTIES

[illegible]

Figure 6. Example housing “Trouble-Shooting Chart”—Operational Problems.

INSTALLATION		POSSIBLE CAUSES OF DIFFICULTY, CONTINUED																				COMMENTS	REFER- ENCES
CHART NO. 4	FOUNDAT. CONTD.	Piping	Expansion Joints not prop- erly installed	Not prop- erly aligned & 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 under- size	Drain lines under- size	Pipe expansion restricted by contact with foundation or other pipes	Branch lines restricting expansion	Dead ends not drained	Electrical	Reverse- current relay failed	Synchronous motor starting pulsations excessive for system	Starting cycle improperly timed	Process, Design or Manufacturing				
Cause of Vibration																				Not covered in detail in this paper. where high probability of error is indicated, contact manufacturer.			
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 AND OF COUPLING. (16)			
Permanent bow or lost rotor parts (vanes)																				Best 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 assembly, pump, or bases are lost or cracked CHECK FOR CORROSION, 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-archer every 5 min. until rub clears. Then resume slow run. THIS MAY TAKE 20 HOURS. 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)			
Casing distortion																				Usually caused by poor mat under foundation, or thermal stress (hot spots), or unequal shrinkage. Usually requires extensive and costly repairs. (2) (17)			
Foundation distortion																				Slight rubs may clear, but trip unit immediately if a high-speed rub gets worse. Turn by hand until clear. (2) (17)			
Seal rub																				Unless thrust bearing has failed this is caused by rapid changes of load and temperature. Machine should be opened and inspected. (2) (17) (18)			
Rotor rub axial																				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)			
Misalign- ment																				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)			
Piping forces																				Bearings may get distorted from heat. Make hot-check, if possible, checking contact. (18) (19) (20)			
Journal & bearing eccentricity																				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. (10) (11)			
Bearing damage																				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 PUMPING OR FOUNDATION RESONANCE AT WHIRL FREQUENCY. (16)			
Bearing & Support excited vibration (oil whirls, etc.)																				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)			
Unequal bearing stiffness horizontal - vertical																				Usually the result of slugging the machine with fluid, solids built up on rotor, or off-design operation, especially surging. (11) (12)			
Thrust brg. damage																				The frequency at rotor-support critical is characteristic. Disks and sleeves may have lost their interference fit by rapid temperature changes. Parts usually are not loose at standstill. Often confused with oil whirl because characteristics are essentially the same. Before suspecting any whirl make sure something in bearing assembly is absolutely tight with interference fit. This is rare, but should always be checked as a matter of routine. Usually involves sliding pedestals and casing feet. Check for friction, proper clearances, and piping strains. (11) (12)			
Insufficient tightness in ASST. of rotor (shrink fit)																				To get frequencies, tape microphone to gear case and record noise on magnetic tape. (11) (12)			
By liner brg. case casing support																				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 .002 in. at standstill, at most use hollow coupling spacers. MAKE SURE COUPLING AND BEARING SUPPORTS ARE PROPERLY OVERHAULING. (3) (4) (5) (6) (7)			
Gear inaccuracy																				Try field balancing, more viscous oil (Selenia), larger, longer bearings with extreme 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 3000 RPM. (3) (4) (5) (6) (7)			
Coupling inaccuracy or damage																				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). Can be exceedingly troublesome. Same 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)			
Rotor & brg. system crit- ical																				So called "case-drumming" can be very persistent but is sometimes harmless. Danger is THAT PIPES MAY COME LOOSE and fall into the machine, also the tailing interaction may be involved. Distress drumming is serious. Local drumming is usually harmless, but may be resonant, resulting in vibration of entire case or a unit are potentially dangerous because of possible rubs and component failures, as well as possible excitation of other vibrations. As above, but with the added complications of settling, cracking, warping and misalignment. Also may get into piping troubles and possible case warpage. Foundation resonances are serious and greatly reduce unit reliability. (2) (17)			
Coupling critical																				Can excite other vibrations, possibly with serious consequences. Should eliminate such vibrations by using restrainers, flexible pipe supports, guy braces, shock absorbers, etc., plus isolation of foundation from piping, building, basement and operating floor. (2) (10) (11)			
Overhang critical																				Mostly at 2 x line frequency (7000 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)			
Structural resonance of																				Can excite serious vibrations or cause bearing failures. Isolate piping and foundation and use shock absorbers and guy braces. (2) (10) (11) (17)			
Casing																				Fairly rare but sometimes very violent. These are aerodynamically excited. Change valve shape to reduce turbulence. Increase rigidity of valve gear. (12) (13) (14)			
Supports																				Vibration is 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 clearances effects. (3) (4) (5) (6) (7)			
Foundation																				Vibration at 2x, 3x, 4x, exciting frequency. Fairly common and troublesome. Treatment is the same as for direct resonance: Change frequency and add damping. (3) (4) (5) (6) (7)			
Pressure pulsations																				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 oil corrosion such as sulfur, lime, spacers and fluid, slugging. 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. (2) (10) (11)			
Electrically excited 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)			
Vibr. trans- mission																				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)			
Valve vibr.																				As above, but with additional resonance of rotor, stator, foundation, piping or externally excited. Find resonant frequency 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. 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. (10) (11)			
NAME OF PROBLEM																				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. As above, but encountered only during start and shutdown due to very strong torsional pulsations. Occurs with: Reciprocating machinery, synchronous motors. CHECK FOR TORSIONAL CRACKS. (8) (9)			
Sub-harmonic resonance																							
Harmonic resonance																							
Friction in- duced whirl																							
Critical speed																							
Resonant vibr.																							
Oil whirl																							
Resonant whirl																							
Dry whirl																							
Clearance in- duced vibr'g.																							
Torsional resonance																							
Transient torsional																							

Figure 6. Continued.

OPERATION AND MAINTENANCE

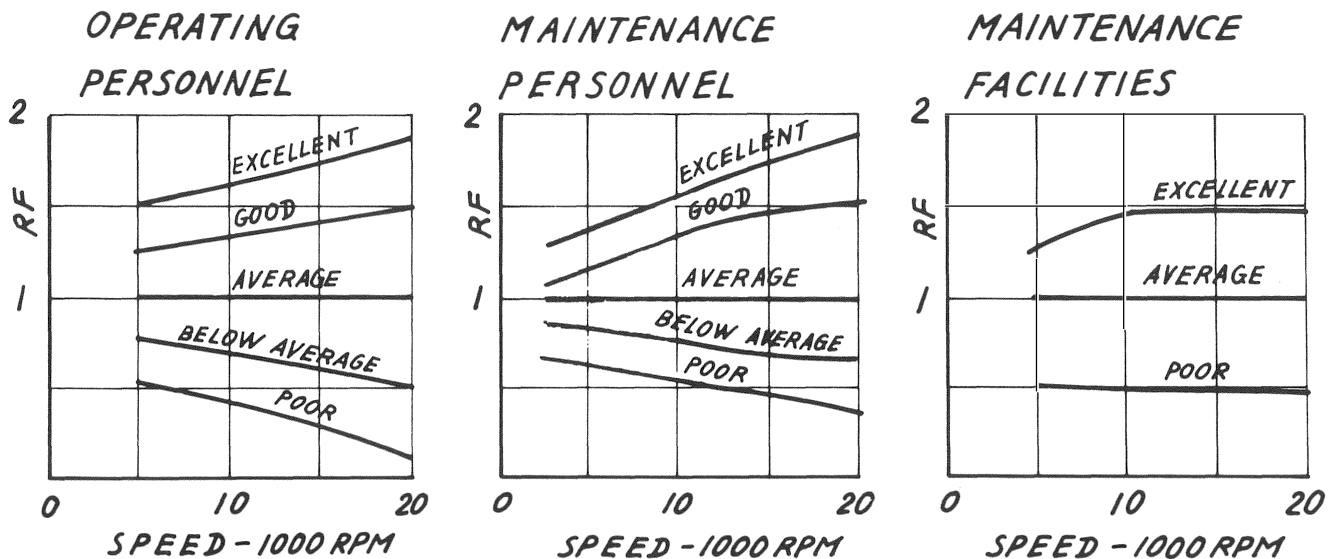


Figure 4.

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 whirled, 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 $\frac{1}{2}$ RPM with speed. A resonant whirl would suddenly appear either at $\frac{1}{2}$ 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.

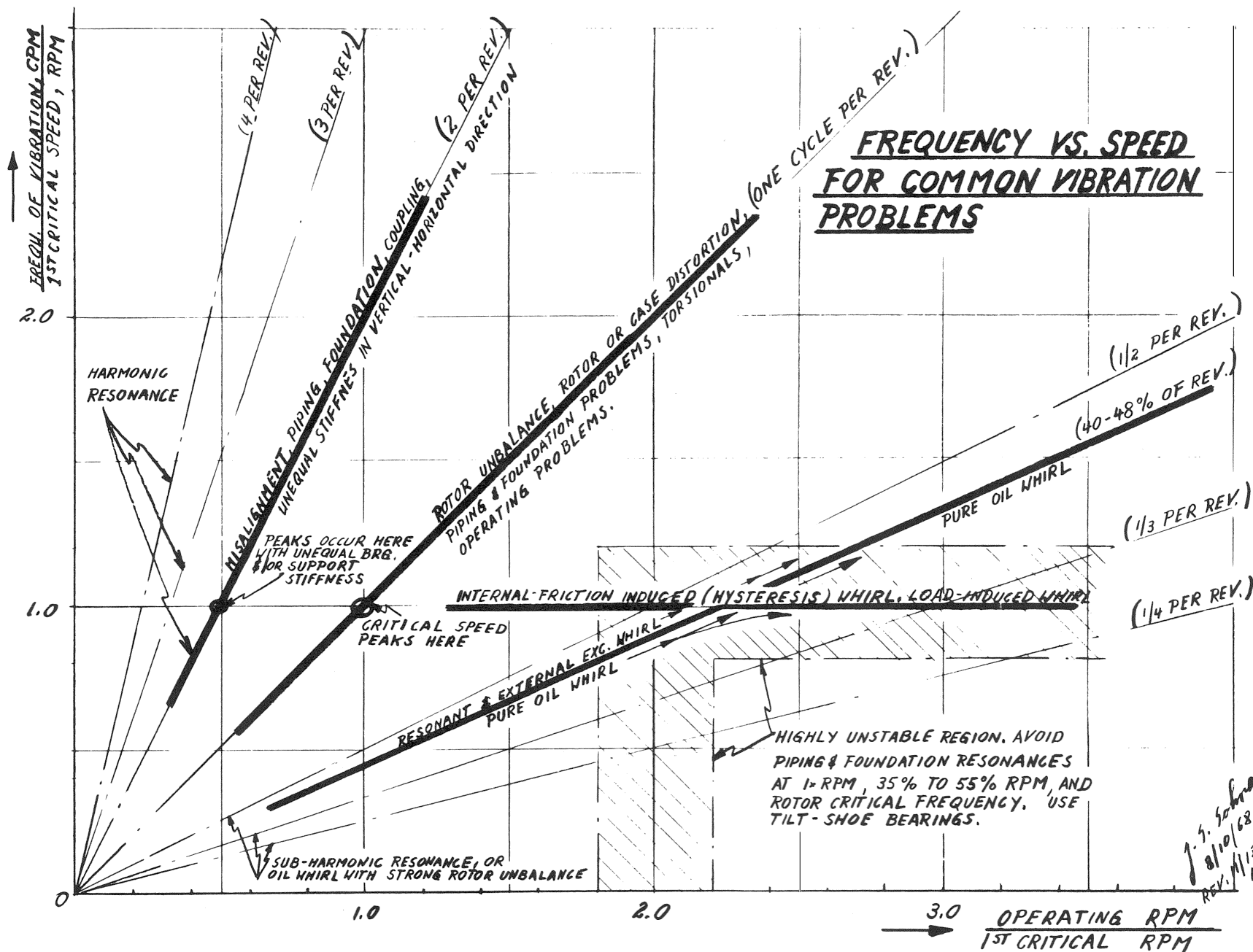


Figure 7.

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 to other causes, with the result that the failures recurred.

Fig. 8 shows the fatigue strength diagram for a typical 13% chromium 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.

EFFECT OF CORROSIVES ON
FATIGUE STRENGTH 13 CR STAINLESS STEEL

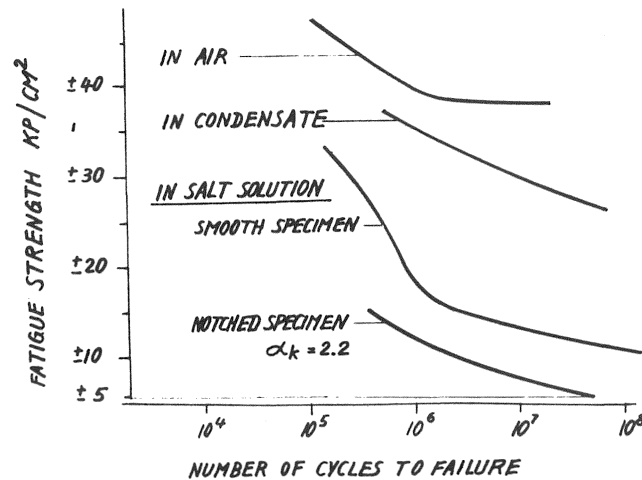


Figure. 8.

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 re-design 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

*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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