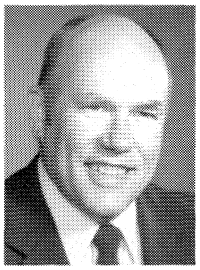
TAKE TRANSIENT STARTUP AND SHUTDOWN DATA—RULE NUMBER ONE

by

Charles Jackson

Turbomachinery Consultant

Texas City, Texas



Charles Jackson is a Turbomachinery Consultant with 43 years experience, the first 35 were with Monsanto Co., retiring as a Distinguished Fellow reporting to Corporate Engineering in St. Louis, Missouri. He is a Charter Founder Board Member of the Vibration Institute established in 1972. He is a Charter Founder and Emeritus Advisory Member of the Turbomachinery Symposium with 21 years duty. He is a Charter Member of Texas A&M's Academy of Distinguished Engineering Graduates. He is an ASME Fellow. He is a member of Pi Tau Sigma and Tau Beta Pi. He served on the API Subcommittee on Mechanical Equipment for 22 years. He is the author of over 100 technical papers, one book, The Practical Vibration Primer, *and wrote a chapter in the* Pump Handbook, *and coauthored two chapters in the* Handbook of Rotordynamics. *He continues as an Instructor for the Vibration Institute.*

ABSTRACT

The thrust herein is to encourage more people to take transient data during startups and rundowns. The common complaint is that it takes too much time and effort to prepare for this information. That argument is "weak" within today's instrumentation capabilities. Further, the more *forms* of information on the same transient data gives the person saddled with the responsibility of eliminating the problem a better confidence level in the final conclusion.

The rotor system, whatever its configuration, forces all the system's responses during the runup and rundown. The ability to look at plots of time waveforms, orbits, polars, Bodê plots, cascades, eccentricities, spectra, raw *vs* synchronous *vs* compensated information can be valuable. The ability to define accurately the "resonances" in the system and the corresponding ability to direct *logical in-situ* balancing. The ability to identify *rubs* when they occur. The initiation of "instabilities" is near breathtaking. The ability to correlate running speed vibration values *vs* subrotative (whirl or whip) values can be very meaningful and normally with unexpected results, i.e., qualifying the "damping" or "lack of" in a system. In all this, the FM tape recorder should never be overlooked; however, the digital storage capability with computer attached is extremely powerful and simple to manipulate. Speed tracking offers the ability of saving memory *during operational delays*, as the next speed threshold has not been reached.

Three case histories are used to illustrate the advantages of transient measure. The three descend in "actual data" presented to simplify the process. The more complicated/involved issue is addressed first, and it is a 22 MW compressor turbine gas generator, power turbine, generator system. This equipment suffered from overfiring the combustors, over acceleration, and over heating the gas turbine system, rubs on compressor blades and heavy destructive rubs on hot turbine shroud bands. The unit whirls and whips. For about 18 hr after repairs, it suffered other problems of "surge/stall" due to IGV control problems (not covered in this text).

The second case discusses a simple low horsepower, medium speed, five stage back pressure steam turbine with problems which existed for over one and one-half years. The basic trending, which showed varying amplitudes and phase, just simply *did not define the problem.* One set of transient startup/shutdown data on the soloed turbine brought to bear, "quickly," the main problem. Several options were open for correction; the simplest was performed, and the problem totally eliminated. Further, stress corrosion cracking failures were better understood under high stress.

The third is a *very* complicated hot gas turbine single shaft, expander compressor steam turbine, unit. It goes through three (two rigid and one bending) criticals (= resonances) to reach an operating speed of ~18,000 rpm. The failure of the first wreck is better investigated via transient data. The rebuilding and rebalancing involved is better understood. The comparison through a "resonant whirl" region that allowed correction is better understood. The original OEM went out of business in 1984.

FIRST CASE—22 MW GAS TURBINE GENERATOR

A cross section can be seen in Figure 1 of the gas generator rotor power aerodynamically coupled to the power turbine that drives a generator through an epicycloid gear reducer. The concern was the gas generator, better seen in Figure 2. This rotor has 10 rows of axial compressor blades supported between two bearings at 160/150 mm, and two overhung rows of hot turbine blades following the combustors. The total weight of this rotor is over 2300 lb and operates in the 9600 rpm range at full load. The separate power turbine rotor is held in bearings three and four operates in the 7700 rpm range, and is coupled to a gear box driving a generator.

The concern was the gas generator rotor, which had high vibration tripping seismic vibration sensors at 18 mm/sec (raised from ~13 mm/sec). The corresponding shaft relative vibration

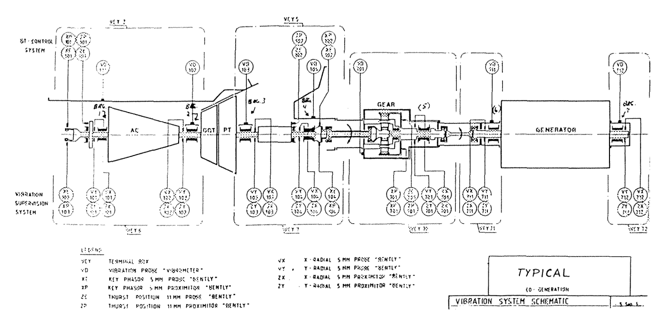


Figure 1. Gas Turbine Generator Layout—Gas Generator (Left), Power Turbine, Gear, and Generator (Right).

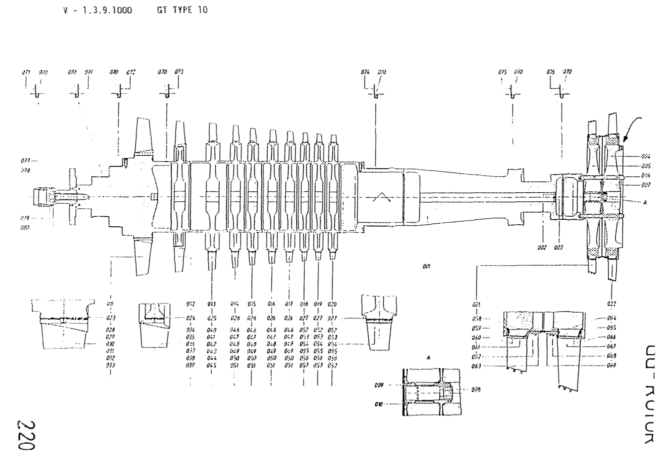


Figure 2. Gas Generator—10 Rows of Free Standing Axial Blades, Two Bearings, Two Turbine Rows, Three Balance Planes.

on this number two bearing reached 5.0 mils p/p, as measured by proximity probes. Over 19 false starts occurred in December 1993. Further, the rotor is "whip" excited at its second resonance (3450-3690 cpm) at a point when the power turbine is taking over load to be synchronized at the generator. Whip (resonance excitation) continues as load is applied.

Transient data recording was arranged in December, and the unit was declared unsafe for service with expected damage in the number two turbine bearing region. The conclusion after several startup attempts with trips and one run to higher speeds with load; but after the seismic vibration trips were increased ~5.0 mm/sec:

- The vibration was excessive at operating speed and part load.
- Whirl existed prior to reaching $1/\omega_s$ X N_{c2}; then, whip @ 1/2 × (resonant excitation of rotor, mode = ?) at about 3690 cpm occurred, which was much higher than running speed vibration.
- Rubs were indicated from reverse lagging of the high spot on the rotor on runup polar plots.

Only part of these data are shown here, just to help solidify the position taken to not run this unit again until opened for inspection. Repairs, inspections, and diagnostics are under contract by the vendor and have been since commissioning in 1986. Further, the vendor (like many) has little respect (probably due to little exposure) for shaft relative measure either from diagnostic advantages or protective shutdown values. Example: Four rows of blading were "corn-cobbed" off the compressor due to a stator blade tap failure in 1991; yet, the seismic vibration shutdown setting of 28 mm/sec (>>1 inch/sec) was not reached.

A *startup* cascade plot is shown in Figure 3 (a) along with a *shutdown* cascade plot (b) of attempt run two under user contracted data taking and user/owner consultant review. This run tripped out under lowered (1991) seismic shutdown values. It is clear that whip excitation of the rotor occurs and that the vibration continues to rise until shutdown, via seismic level (>1/2 inch/sec peak) is reached. Figure 4 is inserted to show the Bodé level at the number one bearing, which is higher than at the damage end bearing number two during the successful number three (trip increased) runup. Further, it shows in the unfiltered orbit, number two from bottom, that the rotor is in 1/2 × "whip" at 7330 rpm of the rotor (1/2 × = 3665 cpm). The polar plots of number two bearing and bearing number one ("figure 8" appearance) up to ~9100 rpm are shown in Figures 5 and 6. The full scale of number two bearing is 5.0 mils p/p taken on *startup* and

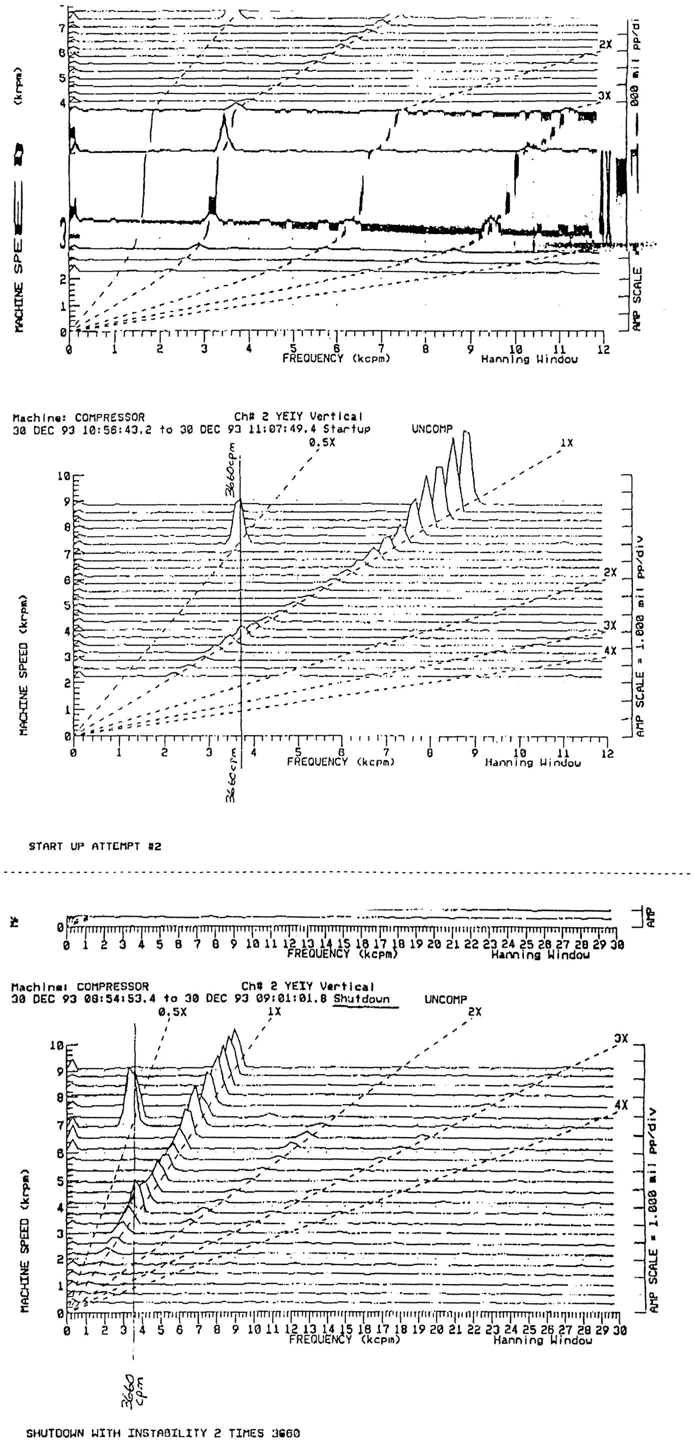


Figure 3. Cascade of Number Two Runup, Bearing 2 (2Y) with Whip @ 3660 CPM (a). Rundown Same Point, Number Two Run (b).

the number one bearing full scale is 2.0 mils p/p, taken on *rundown.* The interruption the plot is during the 1/2 × instability.

INSPECTION ON DISASSEMBLY

On inspection, several things were uncovered. Highlighted should be that the shutdown and disassembly is justified by the user/owner.

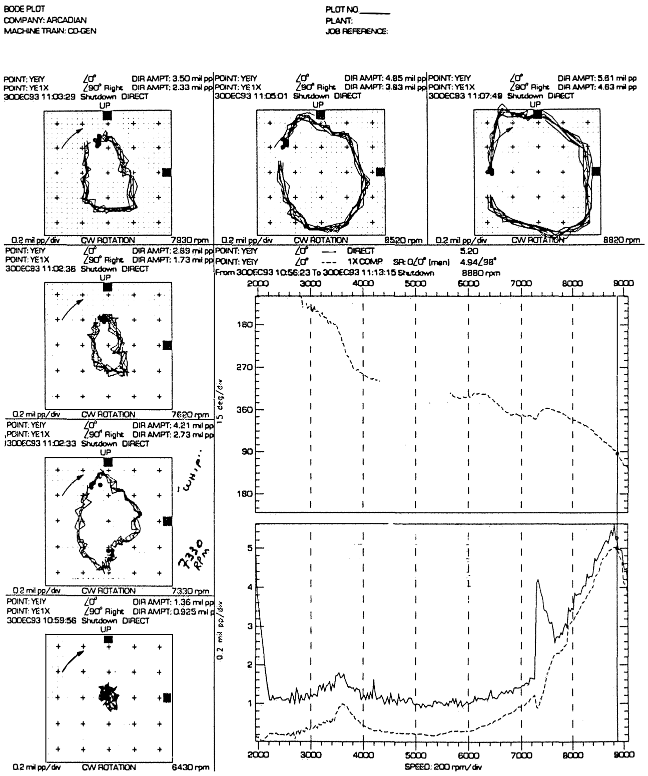


Figure 4. Third Startup After the Seismic Trip Reset to 18 mm/sec. This is the inlet bearing number one, probe 1 X, The vibration at full speed exceeds 5.0 mils p/p. The shaft "whips" @ 1/2 × @ 7330 rpm. The responsive resonance, second mode of gas generator is excited at its resonance of 3690 cpm. Bodé confirms orbit.

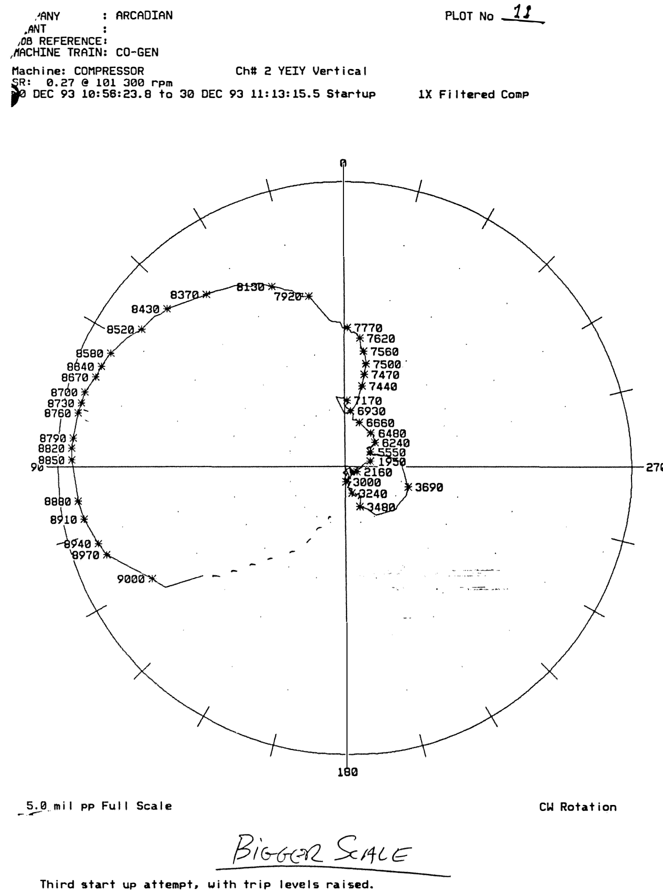


Figure 5. Polar Plot of Bearing 2, 2Y, during the Runup Number Three to Full Speed with Higher Trip Setting. (Seismic @ 18 mm/sec). Full Scale Polar = 5.0 mils p/p.

- Several later stage free axial blades of the compressor rubbed, i.e., light scrape offs from the stator.
- The "honeycombed" stator/shroud packing of the number two turbine stage was "destroyed."
- The number two shroud bands of the buckets were damaged >30 pieces as big as "quarters."
- Several buckets were replaced on the number one turbine row.
- The compressor blades were ice blasted and MPI inspected and found sound.

Note: The ultimate problem was the fuel gas transmitter/stepper switch control to the combustors which was over firing the combustors.

RECOMMISSIONING GAME PLAN FOR DATA AND FIELD BALANCE

Review

The detection pattern has proved out on data taken before overhaul. Unbalance was indicated at the number two bearing. Unbalance was *not indicated* at the number one bearing, but it had the highest shaft relative vibration. Rubs at number two were indicated and did occur. Problems in the air compressor region roughly 1/2 of span did not appear to be suspect. They were okay. Bearing damage or excessive clearance was suspect. The bearings were not *distressed in the babbitt (white metal)* the clearance "argument" continues. It is felt that 1.5 mils/in is a better number than slightly over 2 mils/in.

Goals

The resonances had been roughed in as ~2300 cpm first mode and ~3690 cpm second (responsive) mode and probably a heat cycle mode near 8700 cpm center, which is *not a resonance*. The vendor offered little help on resonances are bearing configurations; so, the spare bearings were inspected and one rundown in 1986 on a waterfall was reviewed.

- Record resonances of the rotors (including gas generator and power turbine). Generator is 875 cpm.
- Record decent runout compensated polar diagrams to assist field balancing.
- Reconfirm, again, that runup resonances are higher in frequency and lower in amplitude than rundown resonances that are lower in frequency and higher in amplitude.
- Observe and record any whirl and whip which is still expected as the first new machine startup in late 1986 had shown this "instability"—how serious is it?
- Determine the phase trigger relationships of the proximity probes *vs* the phasor the weight hole angles, weight hole numbering scheme, the accessible (3) field balance planes, orientation of the proximity probes (number one bearing has "y-x" probes @ 10:30 and 1:30 o'clock)..(number two bearing has "y-

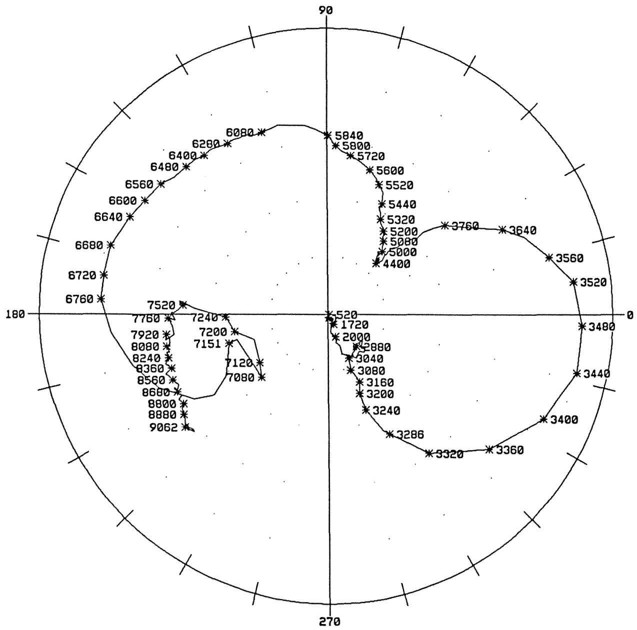


Figure 6. Polar Plot of Bearing Number One During Rundown, Run Number Three, Probe 1X Horizontal. Note second mode @ 3440 cpm, i.e., lower on rundown. High spots 180 degrees out.

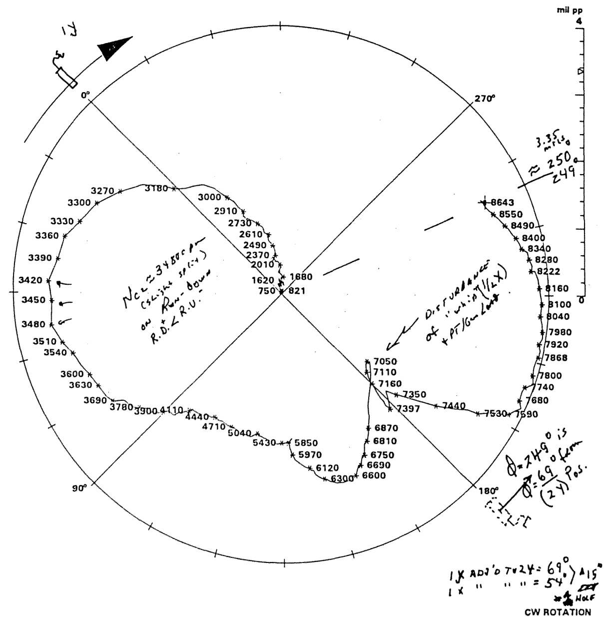


Figure 7. Bearing Number One, 1Y Shutdown, 2/9/94, to 8643 RPM. Full scale circle = 5.0 mils p/p. 1× phase angle corrected for 2Y position. Disturbance 7000 rpm is "whip" at the power turbine takeover point. Whirls then "whip."

x" probes at 4:30 and 7:30 o'clock), determine the influence coefficient for in-situ balancing.

PROBLEMS UNCOVERED ON FIRST RESTARTS—DATA RELATIONS

The polar plots are shown in Figures 7, 8, and 9 of the first (unbalanced) runup for bearings number one and number two. This run made it to 8643 rpm, but load was limited to 9.0 MW. So what do they show?

- Bearing number one does not see the first rotor mode at ~2400 cpm, but bearing number two does see it.
- Bearing number one wants corrections at opposite locations for the first mode *vs* the final rotor position, i.e., unbalance is not strong here . . . work on the other end!
- The vibration and unbalance response is strongest at the number two bearing with a field balance plane in the number two turbine disc. Determine balance from this location. This says to the author to add a weight at hole number 72 which is 150 degrees against rotation from the 2Y probe, *when the rotor is indexed-key phasor at trigger*. There is an existing weight of 24 grams at hole number 68, when this data was taken. The rotor position at triggering can be seen in Figure 11, where the weight hole locations are shown from the 2Y vertical probe. It also traces the vendor's three balance steps to find location (as defined on this first run).
- The location of balance *trial weight* ~24 grams suggested at hole number 72 along with the *existing 24 grams weight at hole number 68*, effectively gives about 42.7 grams at hole number 70 (120 degrees from probe 2Y). Further, the phase angle of the high spot (should be effectively 180 degrees shifted from heavy spot at this speed) and the horizontal probe, 2X, agrees within seven degrees with the vertical (warm feeling). In addition, if one reflects the number one bearing probe 1Y angle for *trial shot attempt*, it agrees within 12 degrees (warmer feeling). Finally, it is apparent that the rotor is about 96 degrees (150-54) number one phase *vs* number two phase at 8643 rpm, i.e., not to full speed at this point. One can do some "predestination predictions" (typical of Presbyterians) with some experience.

Note: Later tab run of all runs here show 233-14 or Δ219 degrees close enough to 180 degrees out, i.e., true pivotal rotor deflections (for Texas Aggies).

Hint: The final run had 24.7 grams at hole number 69, which is within 15 degrees.

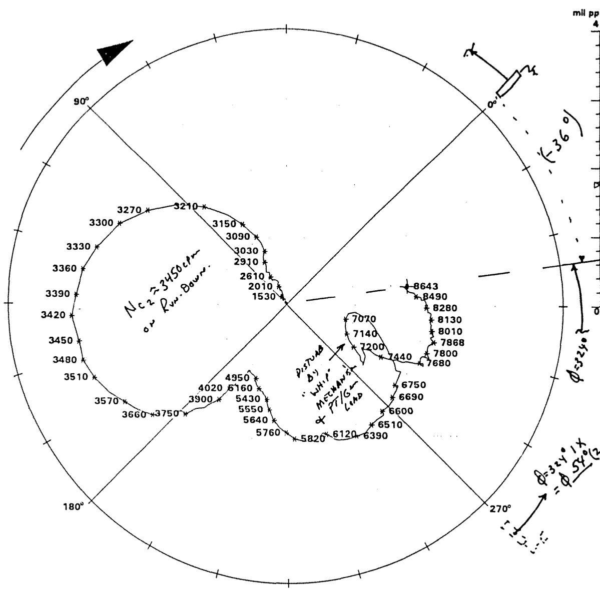


Figure 8. Polar Plot of Bearing Number One, 1X Shutdown. Bearing number one does not show Mode 1 ~2400 cpm response. Second mode is @ 3450 cpm. ϕ @ 324 degrees = 54 degrees @ 2Y pt. Unbalance ~Bearing 2, but highest vibration @ Bearing 1.

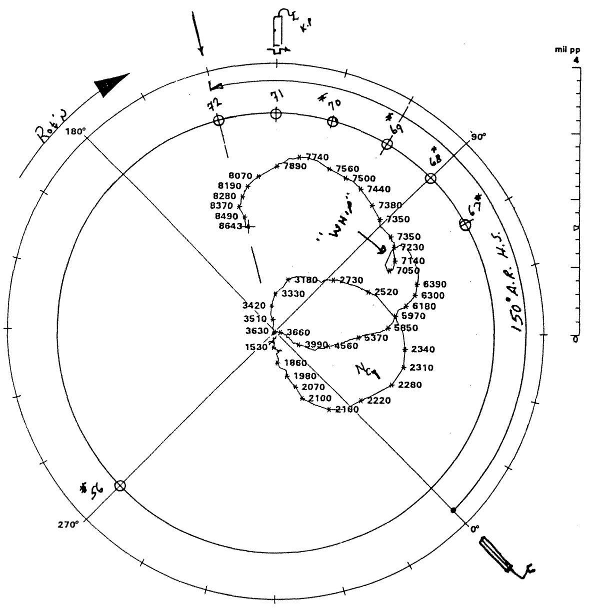


Figure 9. Polar Plot of Bearing 2, 2Y, Shutdown. At nine MW load, logic would add trial weight ~24 gm @ 150 degrees. This data recorded with "existing 24 gms" @ number 68 hole.

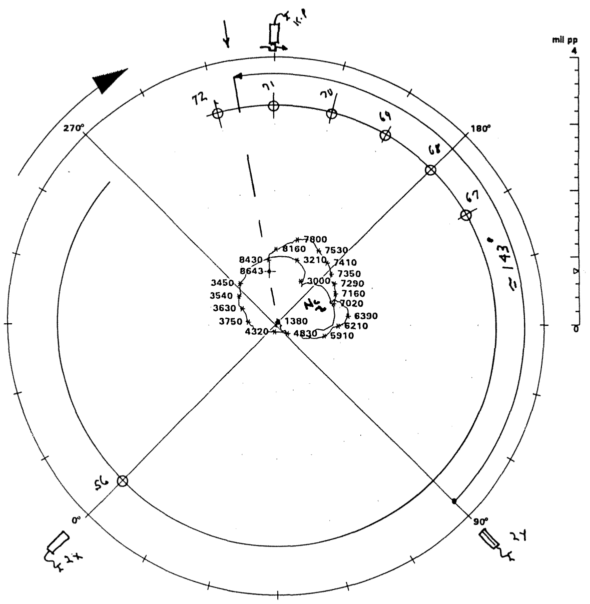


Figure 10. Polar Plot of Bearing 2, 2X, Shutdown. Phase logic agrees with 2Y within nine degrees (< Amplitude). Weights @ 68 and 72 vector sums to ~43 gms @ #70.

Now, the vendor *removed weight* where we would have *added weight*, then doubled the weight and went 30 degrees the wrong direction, then came within target two days later. The polar plots can help with logic. Secondarily, we could see the results of either "surging" or "stalling" as loads were attempted on the first two days, i.e., load and speed could not surpass a specific condition. This "suspect by some of gas preignition (backfiring)" later was confirmed as improper movement of the variable inlet guide vanes (IGVs) in that they were going "left" when called on to go "right" thereby "choking" flow of air. This was

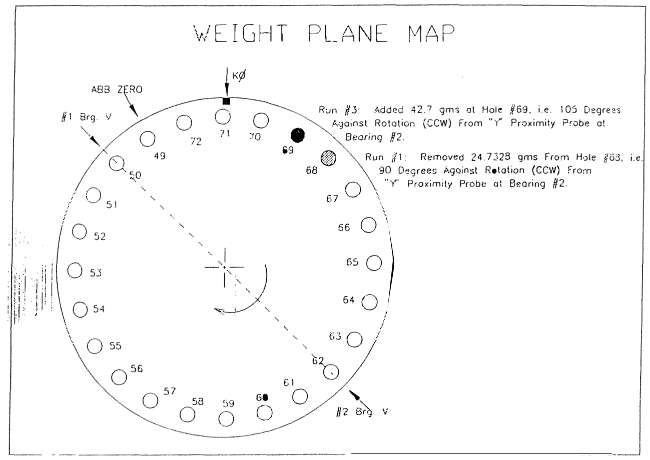


Figure 11. Layout of Field Balance Plane Number Three Past Bearing Number Two. There are three balance planes of 24 holes (15 degrees apart) with numbers against rotation (consecutive in rotation).The phasor notch is at holes 71, 47, and 23 (planes 3, 2, 1). Number two bearing, 2Y, was used for phase.

corrected and the final run to load of 21 MW was made on the third day.

CASCADES OF RUNUP/DOWN ARE VERY VIVID AND INFORMATIVE

The cascades of 1Y and 2Y for the first runup/down are shown in Figures 12 and 13. For those of you *who only believe in taking FFT spectra*; this is a whole bunch of spectrums taken in time/speed format throughout the startup and rundown of first run to 8643 and 9.0 MW load. It is fascinating how much information is here.

CONCLUSIONS

Taking transient data on this unit has enabled us to determine these conclusions for the owner and also offer some help in the future (vibration analysis, modes or balancing):

• The runup and rundown resonances of the rotors were determined without question and after a repair with computer weigh-matched bucket placements.

Criticals (Resonances)	*Frequency*	*Proof*
Nc_1 gas generator	2310-2400 2310-2400	Fig. 14 Bodé Fig. 9 Polar
Nc_2 gas generator	3690 up 3450 dn	Fig. 14 Bodé Fig. 8/7 Polar
Nc power turbine	3000-3200	Fig. 15 Bodé
Nc generator	875 cpm	86 Comm. Data

• The balance influence coefficients for weight added in plane number three versus the correction (- response) at bearings number one and number two via probes 1Y and 1X and 2Y and 2X. This influence coefficient calculated was determined to be:

Bearing number two (balance plane number three) ☞ 12.67 grams/mil ∠347°

Bearing number one (balance plane number three) ☞ 6.36 grams/mil ∠48°

• The amplification factor, AF, gives one a measure of the *damping* of the rotor in its supports through the relationship of

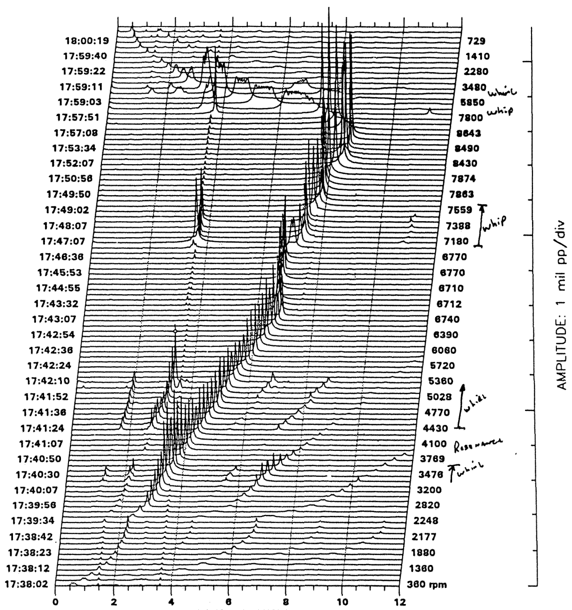


Figure 12. This is the Waterfall Plot of the First Run to 8643 RPM and Nine MW Load. It includes the rundown.

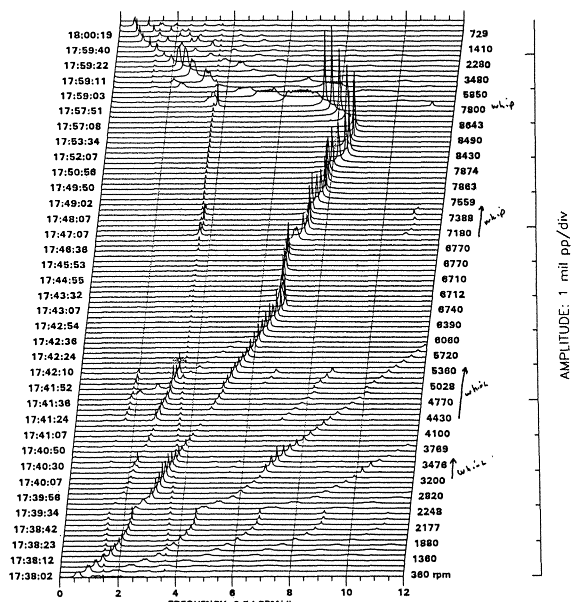


Figure 13. Same as Figure 12, but of the Second Bearing. These "waterfalls" are very vivid on the instabilities.

[AF = 1/2ξ], where ξ = C/Cc (damping ratio of damping to *critical* damping). That is determined from compensated Bodé plots on rundown (Figures 14 and 15).

• The confirmation of "whirl" and "whip" are determined from the Cascades 12 and 13 but more *accurately from the orbits* (Figure 16) where the inside loop and two key phasor marks are seen per period. Hint: the completion of the orbit loop equals one period . . . best seen in the "blanked" timewave (not shown here to reduce figures). Further, the fact that the orbit is frozen, i.e., not moving in either forward (normal) or *reversed precession* (typical of heavy rubs); means that exactly 1/2 running speed

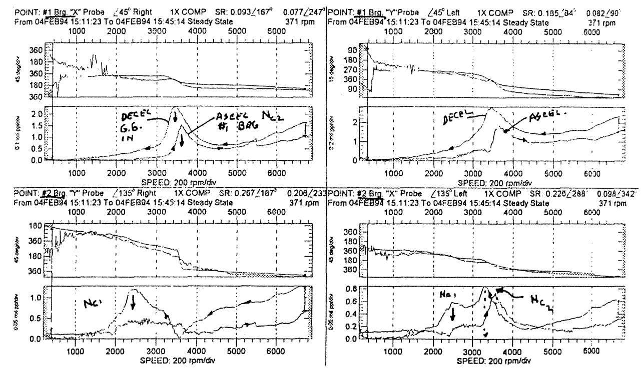


Figure 14. Bodé Runups and Rundowns of Bearings 1 and 2. First mode is 2400-2500 cpm; second mode 3600+ up/3400+ down, cpm. Bearing 2 sees both modes. Bearing 1 sees only the second mode.

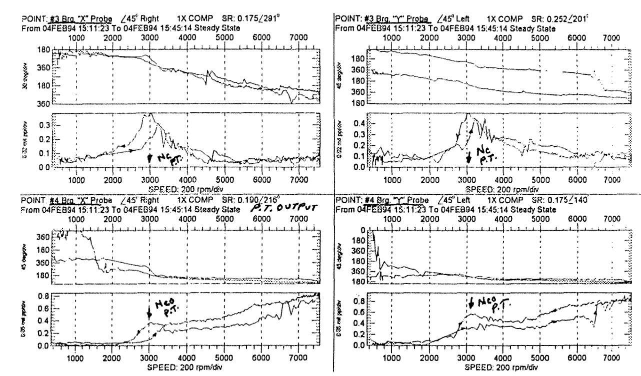


Figure 15. These are Bodé Runups and Rundown of Bearings 3 and 4. Power turbine rotor has critical @ 3000 to 3200 cpm. Mode is better defined at hot turbine end (Bearing 3).

vibration (1/2 ×) exists. The cascade before repair (Figure 3 (a and b)) and after repair (Figures 12 and 13) confirm this. As mentioned earlier, the Figure 3 (a and b) cascades show that the response, at resonance, *far exceeds the vibration at running speed.*

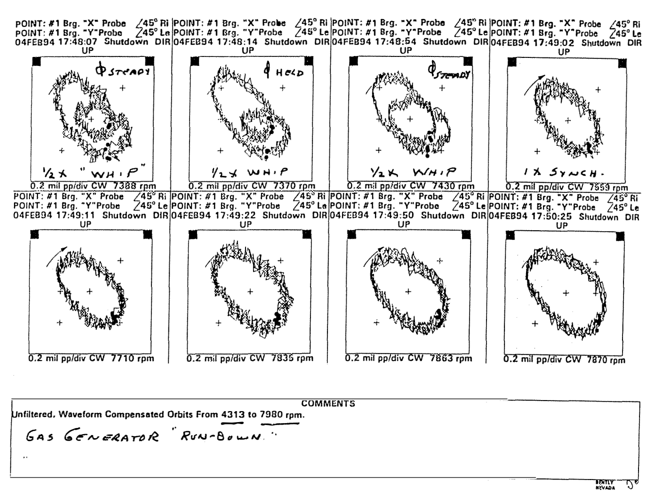


Figure 16. Unfiltered Orbits of Bearing 1. Rotor is "whipping" @ 1/2X in first 3 7388 to 7500.

• The time cycle of the thermal effects on a gas turbine hot urbine blade end,i.e. after combustion, has been *very interesting to me personally*, as I have witnessed this time cycle on a 42 ЛW turbine/generator sets and this 22 MW gas turbine turbine/ ;enerator set under severe and normal conditions.

• The effects in limited air flow and "surging" or "stalling" vere noted prior to learning the variable inlet guide vanes IGVs) were stroking the wrong direction. The orbit visibly bursts" inside about three orders, when this happens. Perhaps a nedical "trauma" team would say, "WE HAVE A *PULSE*!"

Final Figures

Amplitude/phase information records of the four runs are hown in Figure 17. The column labeled *adjusted* means the phase of bearing one data were adjusted *in phase only to the 2Y probe position.*

. Phase Angles and Weight Placement Angles are Referenced From the "Y" Transducer, h the Shaft Positioned Such That the Keyphasor Probe is Aligned With the Hole in the Shaft — *Data Referenced t Vert. Probe at Brg.

RUN	DATE	TIME	SPEED OR LOAD	BALANCE WEIGHTS BP	AMP (Grams)	LOC	±	COMPENSATED 1X DISPLACEMENT: [x] Relative [ ] Absolute	1Y	1X	2Y	2X	1Y* Adjusted	1X* Adjusted	2Y	2X
0	4 Feb 94	17:57	8643 rpm, 6 MW Original Vibration Response	—	—	—		1X@φ	3.36 249°	1.76 324°	1.67 160°	0.786 231°	3.36 89°	1.76 144°	1.67 162°	0.786 231°
								1X@φ								
1	5 Feb 94	10:06	6810 rpm, No Load	3	24.7328	80°	-	1X@φ	3.27 159°	2.61 269°	2.31 67°	1.17 155°	3.27 339°	2.61 74°	2.31 67°	1.17 155°
2	5 Feb 94	13:22	6873 rpm, No Load	3	42.7	76°	+	1X@φ	1.80 208°	1.01 285°	1.73 152°	0.848 338°	1.80 28°	1.01 105°	1.73 157°	0.848 278°
3	6 Feb 94	09:32	8640 rpm, 6 MW	3 3	42.7 42.7	76° 105°	- +	1X@φ	1.81 237°	0.874 298°	1.23 188°	0.678 241°	1.81 57°	0.874 118°	1.23 188°	0.678 241°
3	6 Feb 94	12:33	9680 rpm, 21 MW	—	—	—		1X@φ	0.433 184°	0.308 268°	0.370 223°	0.238 323°	0.432 14°	0.308 88°	0.370 223°	0.238 323°

Figure 17. There Were Four Runs with Data Recorded. Run 3 was delayed then the full load data to 21 MW were recorded. Final @ 0.3-0.43 mils p/p.

The unfiltered and filtered orbit and timewave information at ull speed, 9170 rpm, and load, 21 MW is shown in Figures 18 and 19.

The polar plots on towards full speed and load and lower vibration are shown in Figures 20 and 21 after *thermal equilibrium is near completed.* Few people have the *patience* to wait this out, mostly poor "in-situ" balancers. The reversal of phase lag is a condition that I have learned to watch for, as it may indicate a "rub;" however, in this case, it indicates the changing of response and phase with thermal settlings.

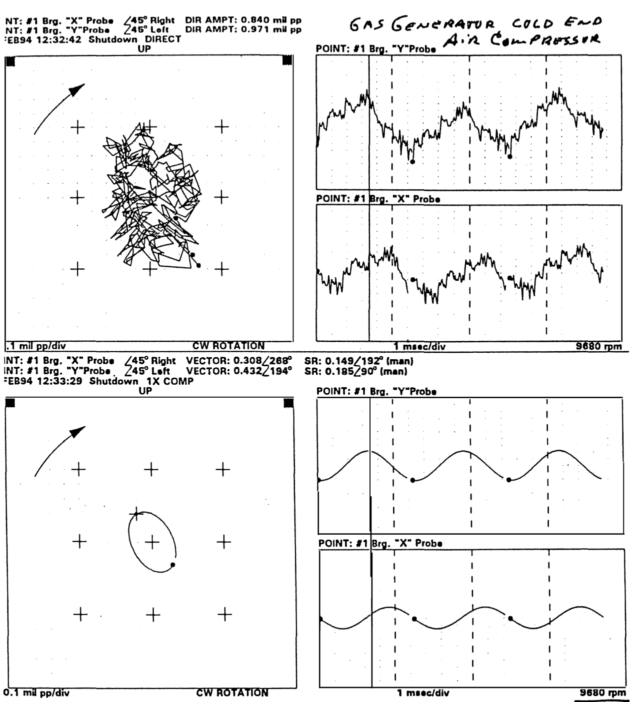


Figure 18. Unfiltered and Filtered Orbits @ Bearing Number One, Compressor Inlet, @ 21 MW (0.309/0.437 Mils p/p).

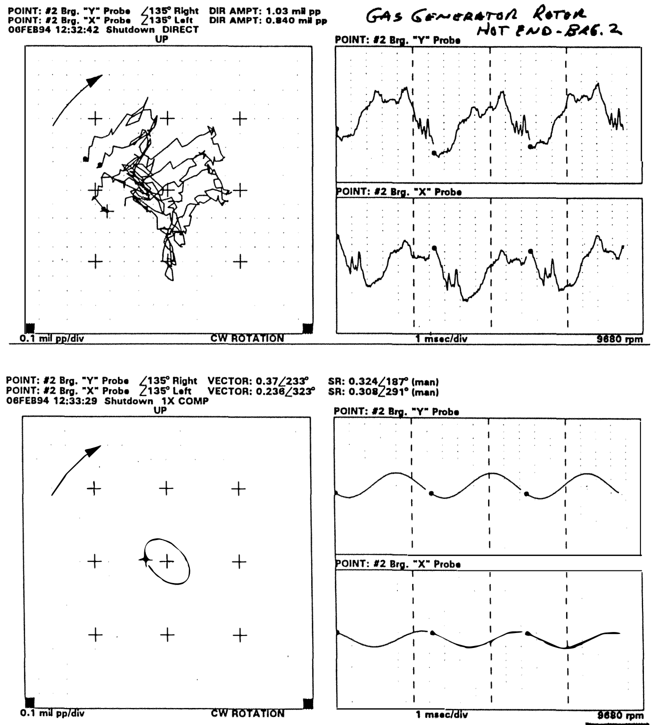


Figure 19. Same information (as Figure 18), but @ Bearing 2, Synchronous Vibration @ 0.37/0.24 Mils p/p @ 21 MW Load.

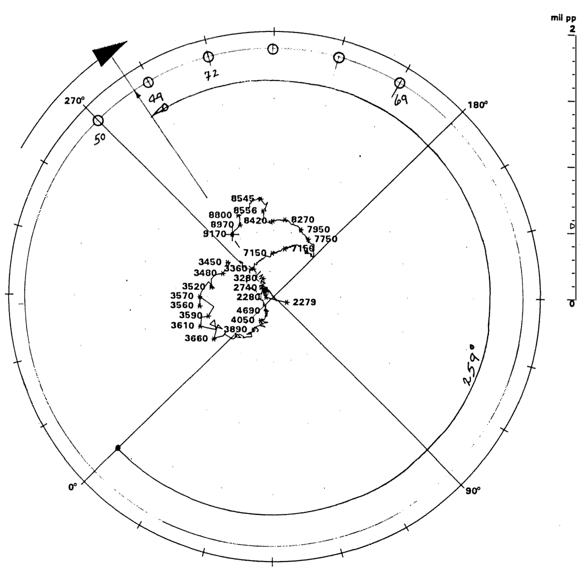


Figure 20. Polar Plot on Running to Full Load and 9170 RPM. Full Circle Is 2.0 Mils p/p. The effects of the thermal transient can also be seen.

This completes the discussion of Case number one. It is current to 1994. I hope it is interesting?

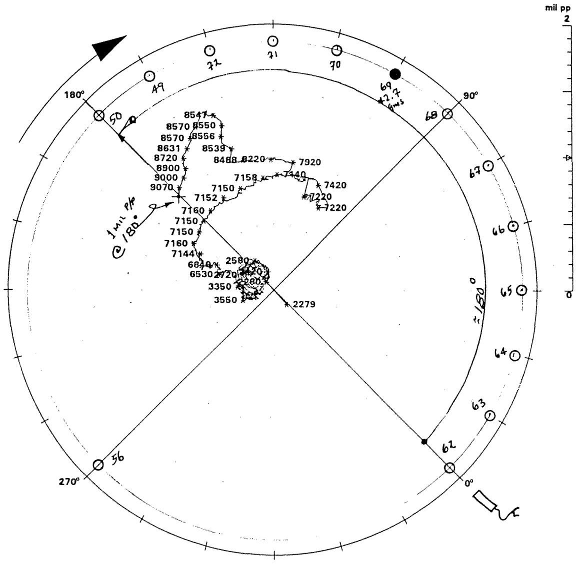


Figure 21. Polar Plot of 2Y, Bearing Number Two, Showing that the Number 70 Hole Was a Better Choice than 69 and the Reversal of Lag Angle Normally "Rub" Is Thermal.

SECOND CASE—STEAM TURBINE WITH PROBLEMS OVER ONE AND ONE-HALF YEARS

This case will be much shorter, as it will be obvious what the problem is after the transient startup and shutdown data are taken on this turbine. This turbine had two rotors put in over a span greater than one and one-half years, where the vibration level was so bad that all sorts of improvements were underway. Only to give a flavor of that problem; the rotor had been rebuilt, extra care in balancing had been exercised, bearing had been modelled for maximum damping, and a separate group was working of stiffening the floor (which was the wrong direction entirely, as the transient data will bring out).

The five stage back pressure rotor using low pressure steam can be seen in Figure 22. It operates at low-to-medium speed of 4950 to 5680 rpm. It is a forged disc rotor. It drives a refrigeration compressor through a coupling that could easily be reduced about 35 lb center of gravity weight, though that is not the problem.

The stiffness map can be seen in Figure 23 and the rotor mass model can be seen in Figure 24 for perhaps better illustration. Therefore, it should be operating *above* the first mode and *below* the second mode.

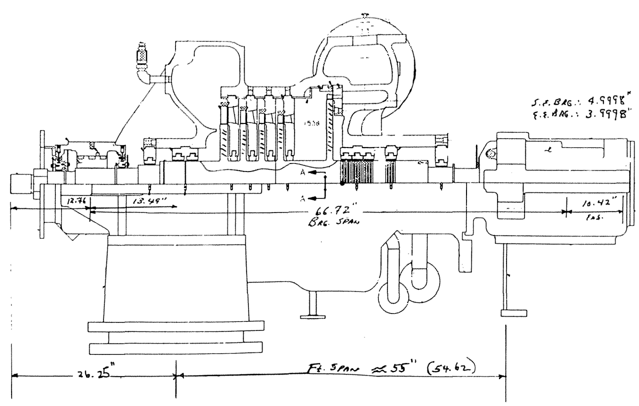


Figure 22. Cross Section of Turbine.

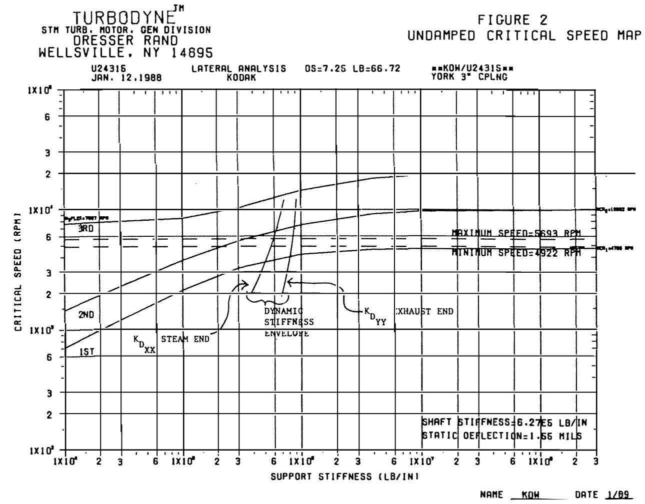


Figure 23. Undamped Critical Speed Map—Five Stage Turbine.

Trend data were taken continuously and when I became involved in this problem, those were the only data that I could obtain. With this data came endless questions on why one day or hour was worse than others, and why was the phase and amplitude varying continuously? Trend data took three different forms, but the typical large group took the forms as illustrated in Figures 25 (daily) and 26 (monthly +).

On my knees, I was begging for *any* transient startup or shutdown (often the best) data. Finally, the request to go onsite came forth, for the turbine was down again for repairs. Data were to be taken by others and the witching hour for at least a no load (solo) run of the turbine was near. This was "great" but having been bitten by this problem before, I took at least a tracking filter and a plotter in a shoulder harness on the airplane (fits regulation carry-on size . . . and given the seat request of row 15 or greater one can often board before *all* the carry-on baggage is stored).

In Figures 27 and 28, one can see the polar plots of the first runup. If one is not familiar with polar plots, the passing of the resonance (= critical) is seen as a complete circle. The illustration of this can be seen in the Single Plane Balancing paper for

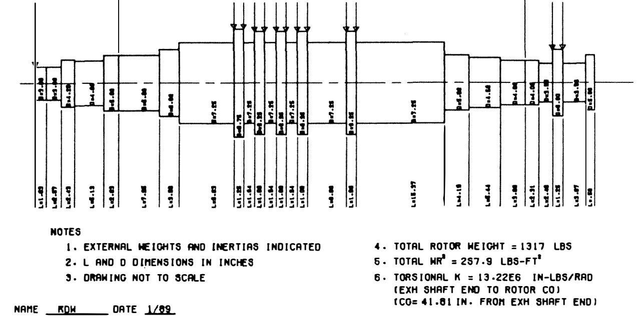


Figure 24. Rotor Mass Elastic Model.

TREND LIST Trend 7 day Rack 1 Monitor 1 Page 1

	1X AMPLITUDE mils	TURB OB VERT 1X PHASE LAG degrees	R1KP1 RPM
6AUG91 10:20	0.04/0.01	253/245	1201/1198
6AUG91 8:20	0.03/0.01	259/251	1204/1198
6AUG91 6:20	0.03/0.01	249/253	1203/1196
6AUG91 4:20	0.03/0.01	243/248	1203/1197
6AUG91 2:20	0.04/0.01	242/260	1201/1198
6AUG91 0:20	0.04/0.01	251/239	1204/1197
5AUG91 22:20	0.04/0.02	252/257	1205/1195
5AUG91 20:20	0.04/0.02	244/249	1204/1196
5AUG91 18:20	0.04/0.01	245/245	1202/1198
5AUG91 16:20	0.04/0.01	248/264	1205/1195
5AUG91 14:20	0.03/0.02	240/255	1205/1195
5AUG91 12:20	0.04/0.01	252/256	1205/1195
5AUG91 10:20	0.04/0.01	249/257	1202/1197
5AUG91 8:20	0.04/0.02	248/251	1201/1198
5AUG91 6:20	0.04/0.01	246/220	1203/1197
5AUG91 4:20	3.78/0.00	64/NA	5365/0 — TRIP
5AUG91 2:20	3.61/3.35	64/63	5368/5358
5AUG91 0:20	3.50/3.42	65/63	5365/5358
4AUG91 22:20	3.52/3.43	61/63	5365/5358
4AUG91 20:20	3.60/3.38	62/60	5365/5358
4AUG91 18:20	3.60/3.52	62/61	5365/5358
4AUG91 16:20	3.64/3.55	61/60	5365/5358
4AUG91 14:20	3.58/3.28	61/60	5370/5355
4AUG91 12:20	3.62/3.51	61/60	5365/5355
4AUG91 10:20	3.64/3.47	62/61	5368/5355
4AUG91 8:20	3.63/3.54	62/61	5365/5358
4AUG91 6:20	3.62/3.35	61/61	5393/5360
4AUG91 4:20	3.54/3.38	63/64	5395/5380
4AUG91 2:20	3.52/3.40	65/62	5395/5383
4AUG91 0:20	3.46/3.25	63/62	5395/5383
3AUG91 22:20	3.45/3.14	56/58	5390/5117
3AUG91 20:20	3.43/3.29	55/53	5124/5117
3AUG91 18:20	3.49/3.29	56/52	5126/5117
3AUG91 16:20	3.49/3.32	56/55	5131/5117
3AUG91 14:20	3.41/3.28	55/54	5124/5117
3AUG91 12:20	3.52/3.26	57/54	5126/5115
3AUG91 10:20	3.51/3.34	57/56	5126/5117
3AUG91 8:20	3.47/3.31	57/53	5124/5117
3AUG91 6:20	3.39/3.08	55/54	5124/5117
3AUG91 4:20	3.13/2.91	54/52	5126/5054
3AUG91 2:20	3.33/2.81	56/56	5063/5054
3AUG91 0:20	3.33/3.15	56/56	5063/5054
2AUG91 22:20	3.21/3.05	59/58	5163/5054
2AUG91 20:20	3.22/3.03	58/58	5165/5152
2AUG91 18:20	3.18/2.97	58/57	5163/5154
2AUG91 16:20	3.32/3.09	60/58	5163/5156
2AUG91 14:20	3.29/3.05	57/57	5161/5129
2AUG91 12:20	3.26/2.99	57/56	5135/5129

Figure 25. Typical Trend Data Submitted (>>Months).

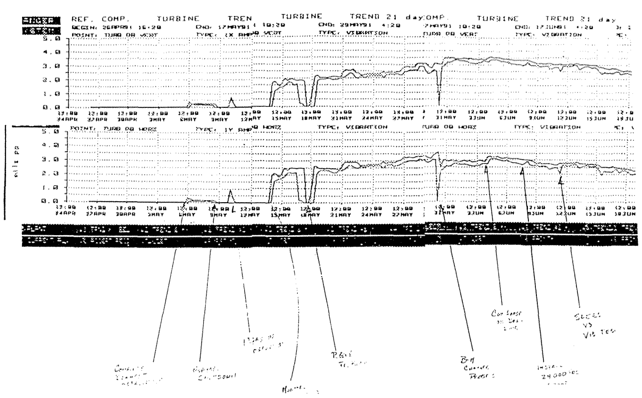


Figure 26. Typical Trend Data Submitted over Months of Problems Extending One and One-Half Years (Plus).

the Eighth International Pump Users Symposium (1991) and the tutorial demonstration of balancing at the Twenty-first Turbomachinery Symposium (1992). Nonetheless, when one only goes 1/2 way around (~+); it means that the rotor is basically on/near resonance. If you will note the movement with amplitude and phase changing, as speed is more or less held, then you can get a feeling of the uncertainty of this precarious position, which is worse under full load (full steam flow). The turbine was brought up to full speed to see how far "over the critical speed envelope peak" the vibration data responded. However, neither this maximum speed nor the trip speed (110 percent of max continuous speed) allowed the turbine to escape from the resonance response.

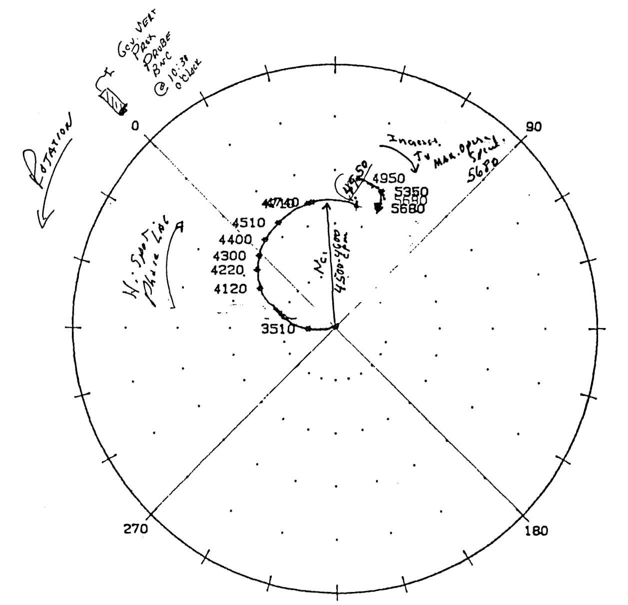


Figure 27. First Transient Data Taken. Polar of Startup to Operating Speed, ~4950 rpm, then slowly to 5680 rpm>5680 rpm. Note the amplitude/phase variations @ operating speed. The Critical Speed is 4500 cpm. Operating Speed ~N_c.

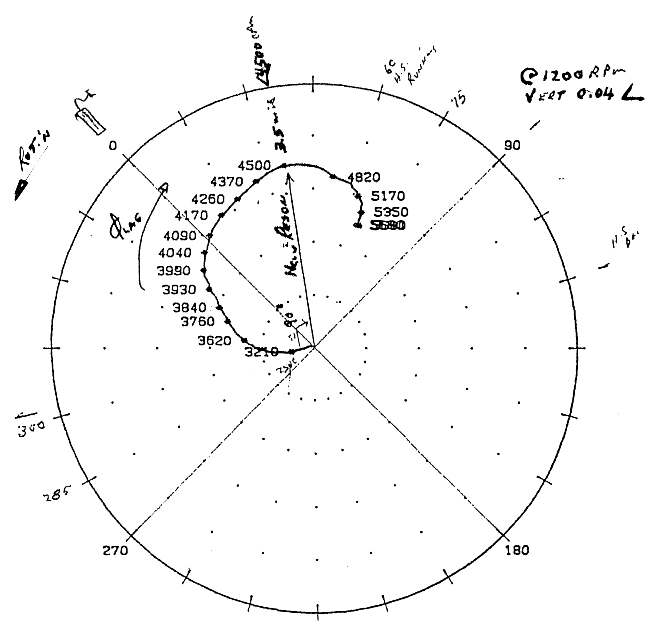


Figure 28. Polar Plot of Shutdown from Overspeed Trip. Plot here does not have amplitude/phase variations during continuous rundown. 4500 is peak. Going to full speed and trip is within N_c envelope.

To confirm this, the Bodé plot from the automated diagnostics for rotating equipment system was predicted and fell true as is shown later in Figure 29. The owner understood this plot better, so why not?

CONCLUSIONS:

The rotor could not be changed on the bearing span. The bearing had been optimized. Since a forged rotor was involved, an added mass could not be installed between the first stage and

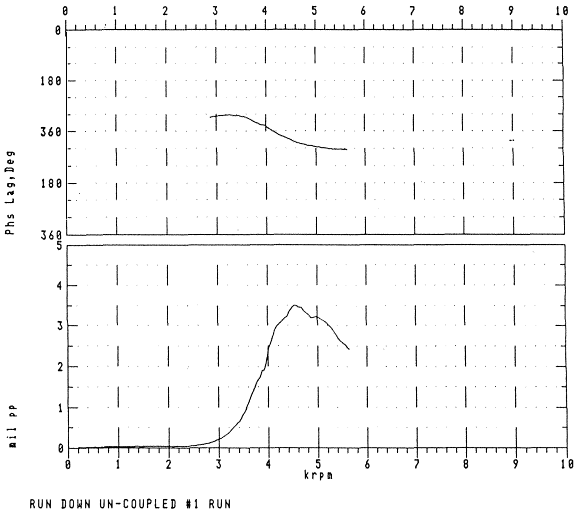


Figure 29. This is a Bodé Plot of Figure 28 Data.

second stage rows. Therefore, the rotor shaft was turned down about 1.5 in on the diameter and a second wobble plate was installed on the steam end bearing pedestal. Further, stress corrosion cracking due to high stress at resonance operation had cracked some 80 to 90 percent through the first disc, so a built-up rotor was incorporated.

The unit now has a lower resonance and goes through that quite successfully and operates with less than 0.5 mil p/p vibration levels. *Take transient startup and shutdown data—rule number one.* As an added suggestion, the user had installed a data manager system. In my opinion, a transient data manager system is the only way to go, if one needs to take transient data on critical equipment.

THIRD CASE HISTORY—HOT GAS EXPANDER AT ~18,000 RPM SUFFERS SEVERAL WRECKS

This case involves a hot gas expander that has an overhung radial inflow gas expander impeller at one end as the final (primary) driver. It has two three dimensional open vane air compressor impellers between the bearings, and a three-stage (very poorly designed) steam turbine starter-helper turbine overhung on the opposite end of the rotor. The rotor has all exotic alloy, i.e., no steel at all, and utilizes seven curvic couplings to hold the rotor together. The company that designed these units went out of business, circa 1984.

The model (by others) of the rotor is shown in Figure 30. It goes through the first two rigid modes without incident. As a matter of fact, the vibration data cannot detect these two modes. The third mode (bending with the nodes, near the bearing), is very responsive. The undamped critical speed stiffness map can be seen in Figure 31. It is desirous to keep the bearing stiffness soft and with a lot of damping to not converge the first three modes. On the first transient data taken by the owner shown in Figures 32 and 33, one can see that a subrotative excitation occurs about 16,000 rpm in runup and after hot gas has been admitted to the expander. This excitation has shut the unit down with the primary excitation coming at the subrotative speed. In my opinion, this is an excitation of the third mode. Others say the first mode. Either way, it is excited.

In the diagnosis of the first failure, the polar plot phase lag *reversal similar to the polar plots in Figure 21 of the gas turbine (rub indication)* was the tip off to the curvic coupling slipping at that bearing end. In failure diagnosis the bearing had wiped badly, the seals rubbed and this curvic had poor contact and was

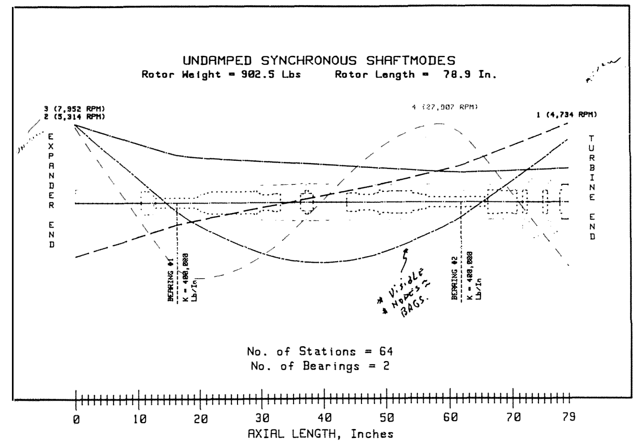


Figure 30. Model of Hot Gas Expander—First → Fourth Modes. Operating Speed ~18,000 RPM with Seven Curvic Couplings.

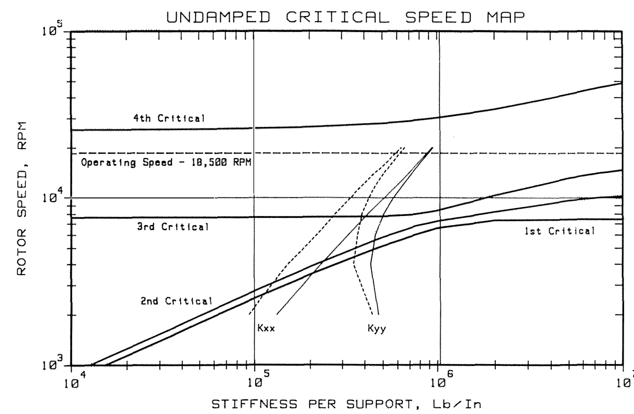


Figure 31. Undamped Critical Speed (Stiffness) Map with the Original Bearings. Load on Pad (LOP).

reground along with the others in repair. A curvic is a radial spline coupling (similar to a poker chip, given some imagination and adding through bolts of inconel).

The primary point is that the transient data helps define the problem. The waterfall (cascade) definitely helps define the *initiation* and the *severity*. Further, it reconfirms the position with two different data sets. This is not oil whirl. I was involved in a case of severe misalignment of a 17,000 hp steam turbine driving a boiler feed pump, wherein the exact 1/2 × subrotative frequency was *falsely interpreted* as oil whirl leading to an improper bearing conversion exacerbating the problem.

The repairs on this unit were extensive and many very important details had to be readdressed for this type of equipment to run successfully. My former company operates equipment at three times this size, which is an asset, and is the only company operating *both* the air compressor and NO_x compressor, since the original OEM went bankrupt.

This incident is reported in this *Proceedings* (Pardivala, et al.), the details of which I am leaving out and only some data are shown here. The point is that transient data is extremely important to this class of equipment, "in-situ" balancing, and many other confirmations of proper condition and operating practices. That is the message here.

A successful runup is shown in Figure 34 of this equipment, with a very low < 2.0 mils p/p response at the responsive resonance and an operating vibration at 18,000 rpm in the 0.6-0.8 mil p/p range. Figure 35 is submitted for help in any way to

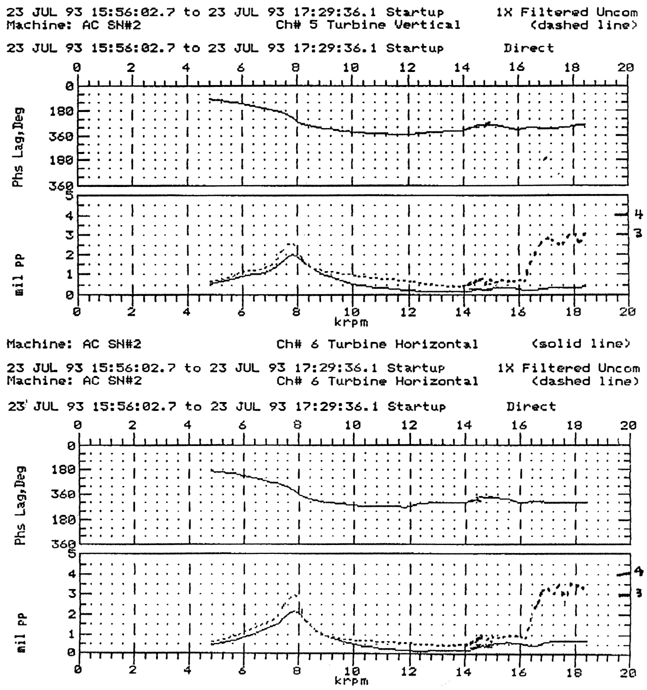


Figure 32. Bodé Plot of Exp-Compressor Reaching an Instability about 16,000 RPM with High Vibration.

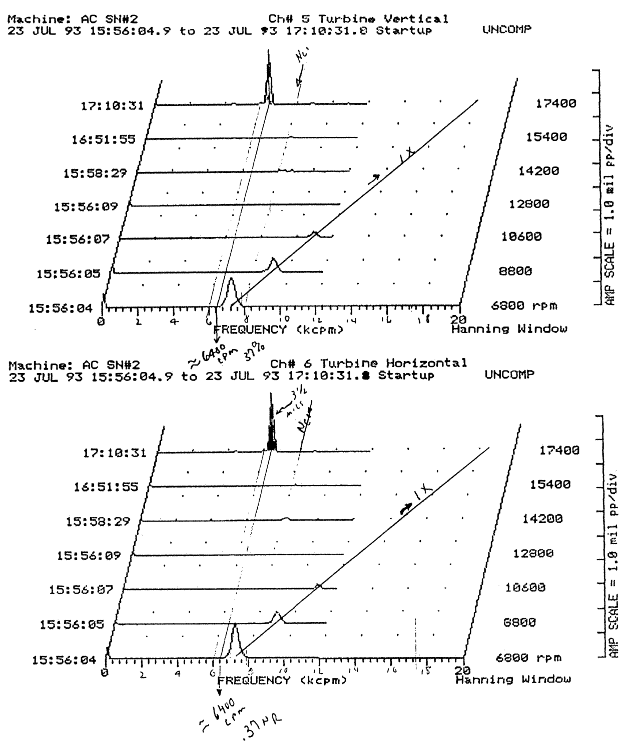


Figure 33. Partial Cascade (Waterfall) Showing the Subrotative Speed (~0.37N_r) Resonant "Whip" (= Trip-out).

better understand the lift measurements required for a five pad bearing, LBP or LOP. In this third case, the bearing vendor had to refit the shell/retainer twice to accurately hold the proper clearance to prevent *clearance whirl*, at 1X.

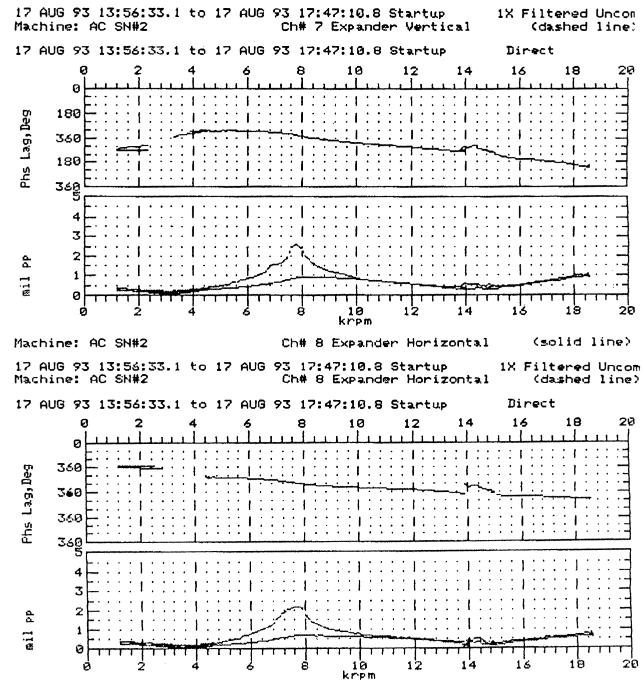


Figure 34. Bodé Plot after Progressive Balance of Individual Components, Regrinding Curvic, Replacing Some Rotor Elements, and Great Care to Bearing Lift. NO INSTABILITY OCCURS! Critical speed amplitude is low.

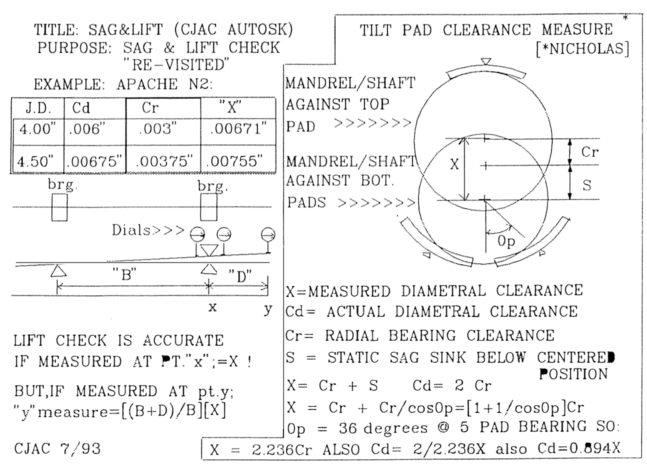


Figure 35. Lift Check Chart Prepared for the Owner and Repair Shop, and Bearing Vendor. Helped very much. Lift values for five pad bearings from J. Nicholas.

It is hoped that these three cases, selected with care, do illustrate the need for transient data. Most all phenomenon are excited during the runup and shutdown of a machine. It is best to observe this information.

ACKNOWLEDGEMENTS

The author wishes to acknowledge the contributions of Richard Thomas (measured data), Case 1, Malcolm Leader, (modelling), Case 3, and LeEllen Phillips, (supplied data plots), Case 3.