FOUR COMPRESSOR TRAINS OF A LARGE ETHYLENE PLANT DESIGN, AUDIT, TESTING AND COMMISSIONING

by

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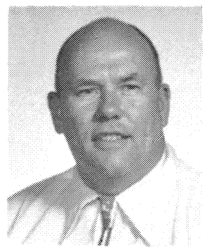

Charles Jackson is a distinguished fellow reporting to the director of Manufacturing Services in St. Louis, Missouri. He has thirty-one years experience with Monsanto and is located at the Texas City Plant. Having been involved in sixteen plant start-ups plus experience in Engineering, Operations and Maintenance, he founded a Mechanical Technology Department in 1960. He serves as a consultant in machinery and mechanical/method related problems. He is an API committeeman on the Committee on Refinery Equipment, a director of the Vibration Institute, an advisor to the University of Virginia's Turbomachinery Dynamics Research Lab, an adviser to Texas A&M's Turbomachinery Symposium, and a member of ASME, Tau Beta Pi and Pi Tau Sigma.

INTRODUCTION

Each new project that a company initiates should allow the mechanical developments up to that time to be applied at the proper point when the design premise is established. This paper will review what is considered a successful start-up of four main compressor trains to limit length and considerations. This project required four years from project design starting until plant start-up on October 14, 1980. This plant is jointly owned by two petrochemical companies, with engineers in both companies contributing greatly to the success of the venture.

This was an 870 million pound per year ethylene plant. The four compressor trains were the cracked gas compressor at 30,000 HP and 4900 RPM; the propylene refrigeration compressor at 20,000 HP and 3800 RPM; the ethylene refrigeration compressor at 3000 HP and 10,950 RPM; and the fuel gas compressor at 1250 HP and 12,400 RPM. The cracked gas compressor is driven by an extraction condensing steam turbine with 1200 psig (900°F) inlet, a 450 psig extraction, a 28 in. Hg vacuum condenser. The propylene compressor is driven by a same frame turbine, but using a 150 psig extraction. The ethylene and fuel gas compressors are driven by four-pole induction motors through increaser gears (i.e., a fixed speed design). The compressor trains and plant are shown in Figures 1 through 6.

The designs of the trains will be discussed, with the design audits, along with the shop tests and the commissioning test at the site being presented. The start-up was mechanically very sound and stable with shaft relative vibration levels of ¾ mil peak-to-peak (p/p) at one point and the balance of ½ mil p/p or less for the displacement probes. Seismic data on the bearing caps of the motors were ½ g peak maximum and the low speed gears were below 1g peak. Bearing metal temperatures were in the 140°F to 190°F range.

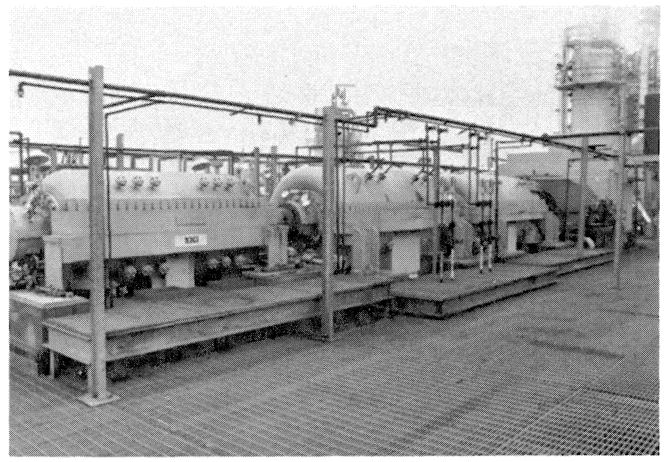

Figure 1. Steam Turbine Driven Cracked Gas Compressor, Three Compressor Casings, at 30,000 HP and 4900 RPM.

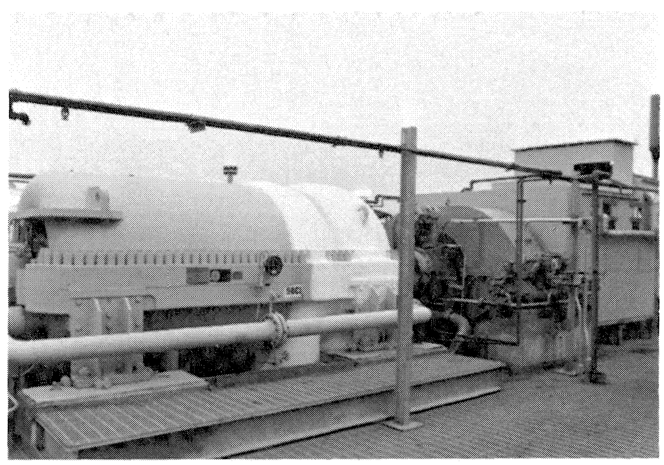

Figure 2. Steam Turbine Driven Propylene Refrigeration Compressor (Four Section Casing) at 20,000 HP and approximately 3800 RPM.

DESIGN PREMISE

Initial design preferences were about 95% accomplished and were:

- Steam turbines, centrifugal compressors, gears, couplings, and lube systems would be in accordance with API 612, 617, 613, 671 and 614, respectively.
- Vibration sensors and monitors would be in accordance with API 670 and latest drafts of API 678.

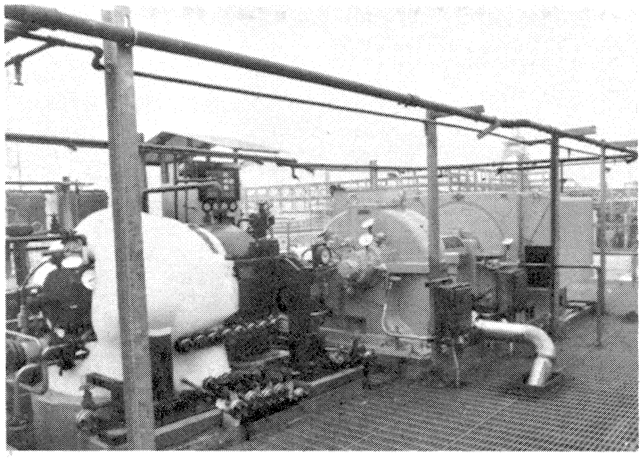

Figure 3. Motor-Gear Driven Ethylene Refrigeration Compressor at 3000 HP and 10,950 RPM.

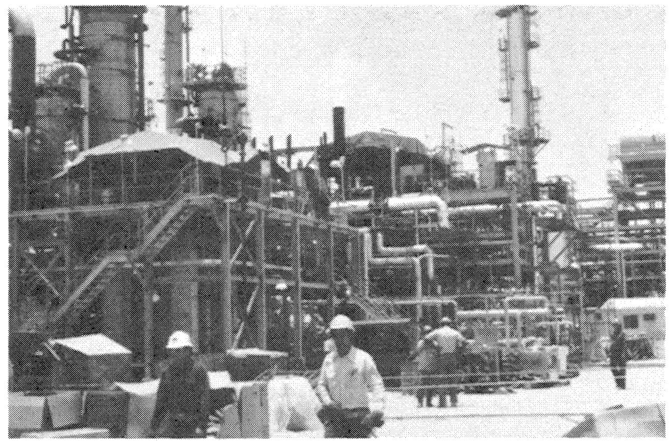

Figure 6. Mezzanine Mounting of the Compressors on Sole Plates with Down Connected Piping.

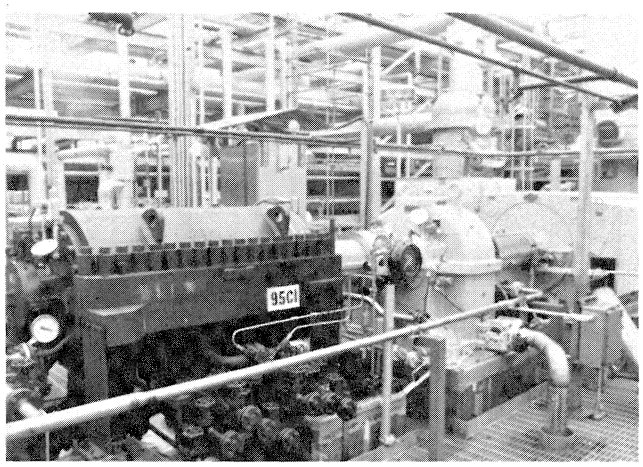

Figure 4. Motor-Gear Driven Fuel Gas Compressor at 1250 HP and 12,400 RPM. Note: API 678 style conduit protected accelerometer on low speed motor and gear bearings.

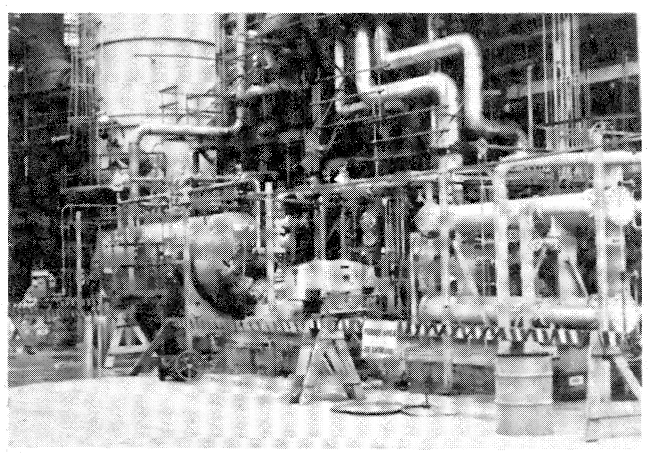

Figure 5. Lube Console for the Cracked Gas Compressor per API 614 with All Stainless Construction, Dual Coolers, Twin Ten Micron Filters and Five Oil Control Valves.

- Equipment would be installed with down connected piping and on epoxy grouted sole plates with 1/4 in. stainless steel shims.
- The lube console would incorporate stainless supply lines and reservoirs and, if economically sound, total stainless steel allowing steel backup flanges.
- Oil filters would be pleated paper of 5 or 10 microns filtration with 50 micron backup strainers used against rupture, bypass, or improper maintenance.
- Screw pumps would be used with flexible stainless pipe couplings on the inlet to reduce pipe strain.
- Drain oil piping from bearing would incorporate flexible stainless pipe couplings and the drain line was to enter the drain header(s) at 45° in the direction of flow.
- Turbine governors would be hydraulic relay and preferably electro-hydraulic controllers using redundant speed probes operating from a tach ring.
- Dry membrane flexible couplings would be used at no less than 18 in. between shaft ends and would be fitted either to integral flanges or by hydraulic dilation at 2 mils/inch interference.
- Steam turbines for main oil pumps were to incorporate hydraulic relay (NEMA 10) governors as a minimum, with bladed discs fitted no less than 1½ mils/inch; hand valves must have back stops and taper blade shrouds were disallowed.
- Oil pump couplings would also be a stainless shim pack type.
- Centrifuges were to be provided for the two larger reservoirs and one mobile unit for the balance.
- Spare rotor packages (rotor, seals, bearings and labyrinth) were to be provided.
- Spare rotors are to be vertically stored in rotor cans immersed in a nitrogen environment.
- Alignment targets (L-shaped) were to be provided on each horizontal side of each bearing and the sole plates were to be increased by 6 in. on three sides to accommodate water stands and to ease space for x-y-z alignment bolts.

- Sole plates would be coated by the vendor(s) with epoxy catalyzed primer immediately after manufacturing.
- The motor and gear vendor was to provide 2½ in. spot faces at specified horizontal positions for accelerometer mounting on site per API 678.
- Each high-speed rotor was to have x-y proximity probes and key phasor probes on the compressor or driver (if the same speed).
- Each tilting pad radial bearing on the high speed shaft was to have embedded dual tip thermocouples (TC) of stranded copper construction at the ¾ pad length and 30 mils behind the babbitt.
- Each thrust bearing was to be a self-leveling tilting pad type with embedded thermocouples as above at the ¾ pad length and arc point.
- Fifty percent of the active thrust bearing pads were to be provided with embedded TC's, along with two diametrically opposite pads on the inactive pads with TC's.
- Each pressure dam or plain journal bearing would utilize spring loaded probe type thermocouples at the minimum arc load line.
- Oil supply lines to each coupling would have plugged tees at the coupling flex plane for future oil spray at 3 GPM each (6 GPM/coupling).

Drillings in each case were to allow slip joint continuously lubricated coupling guards if needed later without field machine work.

- Backup flexible gear couplings, to be rate selected, would be available should conversion be necessary later.
- Coupling guards were not to be totally enclosed in order to reduce heat and prevent negative pressures at the external casing oil seals.
- Turbine shaft end integral flanges were to utilize adapters, if necessary, to prevent the pulling of bearings to inspect the coupling or to perform coupling maintenance.
- Thermal growth (axial, vertical, horizontal) was to be specified by each vendor.
- Rotor and bearing data was to be submitted to the two owning companies for internal analysis.
- Full mechanical tests per API 617 or API 612 were to be performed on all rotors, the first being classed as the spare rotor.
- Performance tests were to be performed on the side stream compressors, i.e., the propylene and ethylene refrigeration compressors.
- Full torque, reduced speed, tests would be performed for one hour on gears for a contact check followed by the called four-hour unloaded full speed test.
- Plug and ring gauges were to be provided for tapered shaft ends along with hydraulic installation and removal tooling on hydraulic (dilation) fitted coupling hubs.
- Governor shafts, cans, pins, bars, linkage, and hardware were to be hardened and free from salty, humid Gulf Coast environment corrosion.
- Vibration shop limits would be 1 mil p/p and the electrical run-out on all probe sensed areas was to be 25% or 0.25 mil p/p for the combination of electrical and mechanical runout.
- Instrument panels would be free standing NEMA 4 panels (i.e., not attached to any machine or mounting plates).
- All thrust displacement monitors were to shutdown automatically at the second limit position.
- Turbine electrical overspeed trips would be set at 10% ± 1% and the mechanical trips at 12% ± 1%.
- Electronic governors were to provide an "emergency manual" control system to allow the turbines to be operated without extraction manually from the control room panel.
- Governor and extraction visible indicators (rotary and linear) were to indicate valve position on the turbines and estimated steam flows of each valve open position.

Items not obtained, but preferred by the author, were:

- Ventilated roofs with part side walls placed over the compressors with a manually operated bridge crane and drop bay.
- Spare linear variable differential transformers (LVDT) feedback positioners on governor and extraction valves to indicate actual position in the control house.
- The standard "Hand Off - Auto" switch on the standby (auxiliary) oil pump could not be modified to a "manual-automatic reset" switch and a separate on-off switch. The Murphy "off" position has been proven to be an easy way for an operator to test the standby pump and leave the switch in the "off" position, thereby rendering the protection nil and void. As a compromise, an annunciator light was added to the control room clusters to warn operators that the switch is not in "automatic". The "off" position could be eliminated since the electrical code (safe practices) requires an "off" switch within 10 feet of a motor.

DESIGN AUDIT

In a design audit, an opportunity is available to investigate the designs on each rotor prior to design completions in most cases. This is definitely possible well before shop testing.

The following calculations and data were prepared on this project:

(1) A stiffness critical map was prepared on each rotor. Variable conditions were investigated, as can be seen from Figures 7 through 12. The clearance of each bearing was varied ± 2 mils. The preload of each bearing was varied from minimum machining to maximum machining tolerances. The offset factor of pivot points on tilting pads was varied from 0.5 to 0.55 (50% pad length pivot to 55% pad length pivot point). These changes all vary the stiffness of the rotor. In addition, "on pad" versus "between pads" positions were investigated for their effects. This is an easy conversion for tilting pads as the bearing can simply be turned upside down in the bearing housing.

(2) Interference Diagrams were plotted for each train. This allowed the lateral criticals, torsional criticals, and coupling axial resonances to be plotted versus the operating speed range and orders of speed — ½X, 1X, 2X, 3X, 4X (Figure 13).

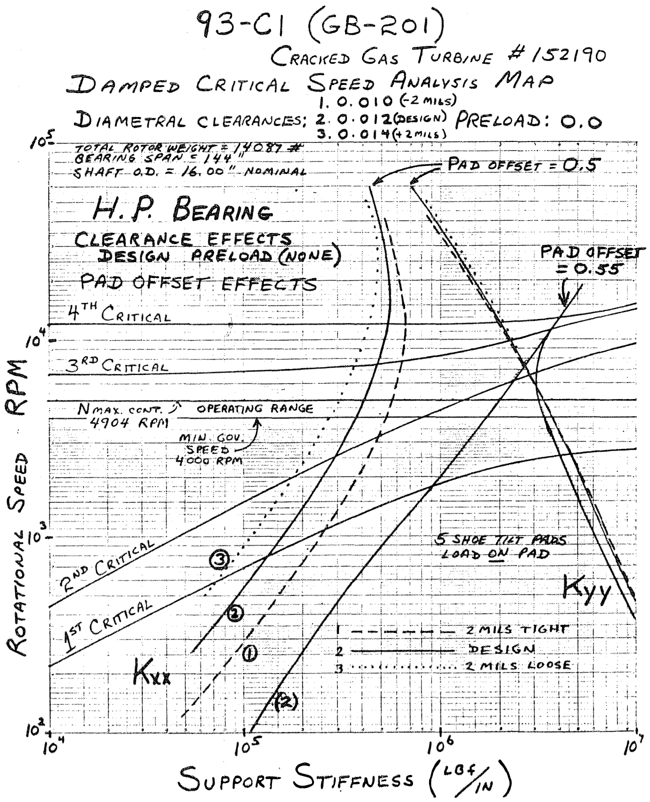

Figure 7. Stiffness Critical Speed Map for 8 in. Diameter Turbine Tilting Pad Bearing With Zero Preload and Design ± 2 Mils Diametrical Clearance and the Effect of Each. In addition, the effect of an offset pivot 0.5 to 0.55 is shown.

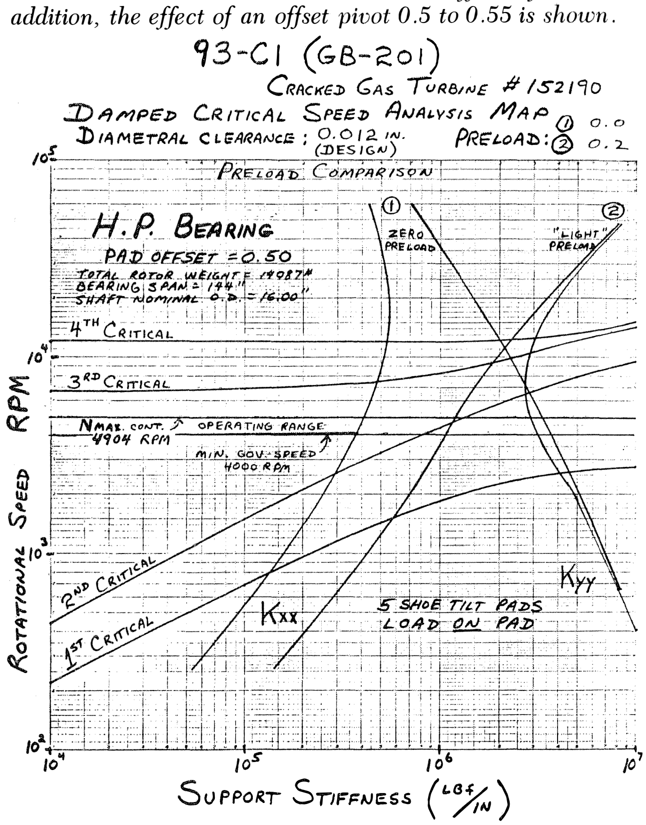

Figure 8. Stiffness Critical Speed Map For 8 in. Turbine Tilting Pad Bearing With Design Clearance and 0 to 0.2 Preload, Load on Pad Configuration of a Five-Pad Bearing.

Cracked Gas Turbine – H.P. Bearing
Damped Critical Speed Analysis
Summary Sheet – Load on Pad

Diametral clearance	Critical speed number and direction		Preload 0.0		Preload 0.2	
Inches	#	Type	Critical speed RPM	% MC	Critical speed RPM	% MC
0.012 design	1	X	790	16	1500	31
	1	Y	2600	53	2580	53
	2	X	2430	50	4650	95
	2	Y	6550	134	6450	132
	3	X	7450	152	9400	192
	3	Y	9800	200	10400	212
0.014 2 mils loose	1	X	580	12	1230	25
	1	Y	2600	53	2580	53
	2	X	2000	41	3850	79
	2	Y	6550	134	6350	129
	3	X	7300	149	8500	173
	3	Y	9850	201	9850	201
0.010 2 mils tight	1	X	1090	22	1760	36
	1	Y	2600	53	2580	53
	2	X	3200	65	5800	118
	2	Y	6550	134	7000	143
	3	X	7800	159	11000	224
	3	Y	9750	199	12750	260

Nmc = 4904 RPM

Figure 9. Matrix of Possibilities of Critical Speeds Based on the Previous Variable Clearance and Preload.

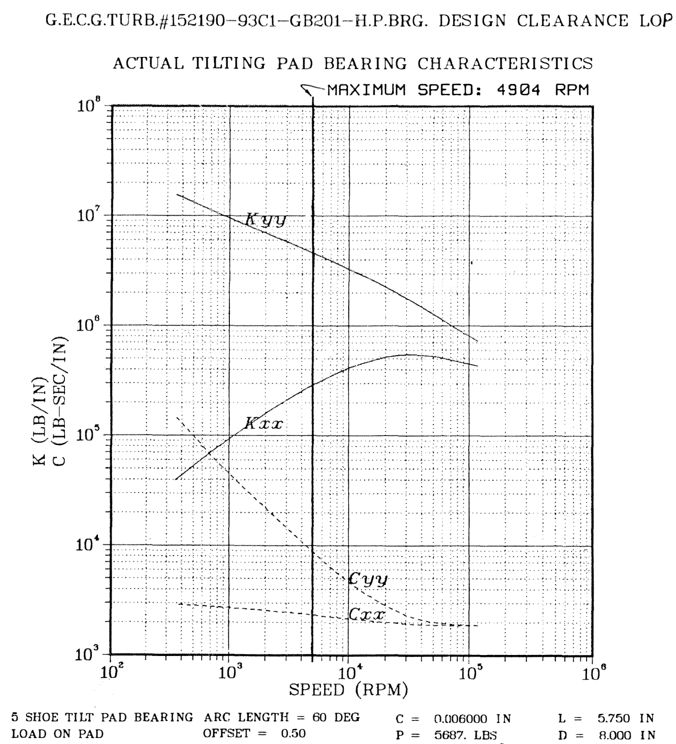

Figure 10. Stiffness — Damping Bearing Characteristics Map for 8 in. Bearing, Load-On-Pad, Zero Preload, *0.5 Offset.*

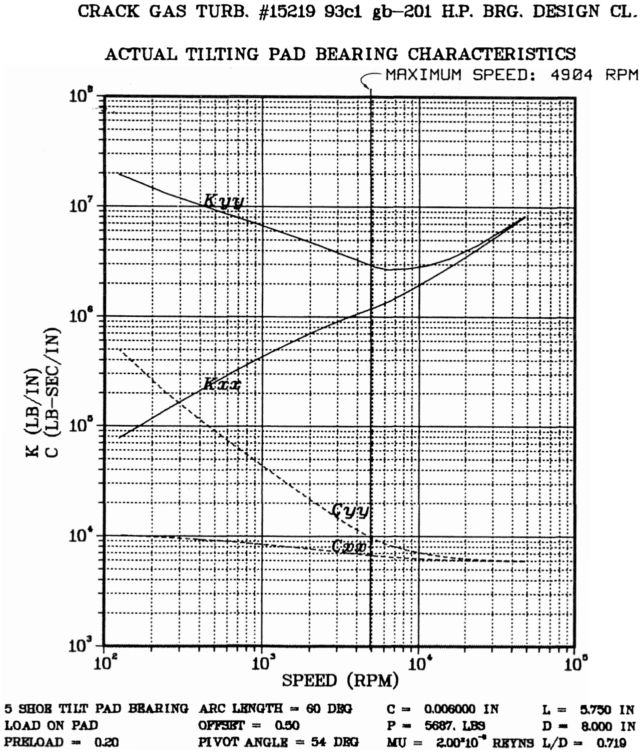

Figure 11. Stiffness — Damping Bearing Characteristics Map for 8 in. Bearing, Load-On-Pad, but 0.2 Preload.

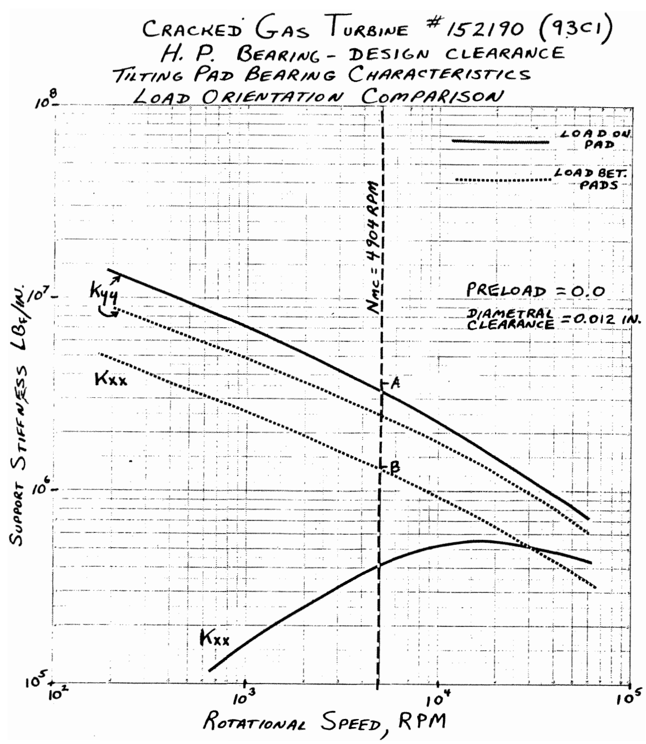

Figure 12. Stiffness — Damping Bearing Characteristics Plot for Load-On-Pad Versus Load-Between-Pad for Design Clearance and Zero Preload, 8 in. Bearing at 0.72 L/D Ratio.

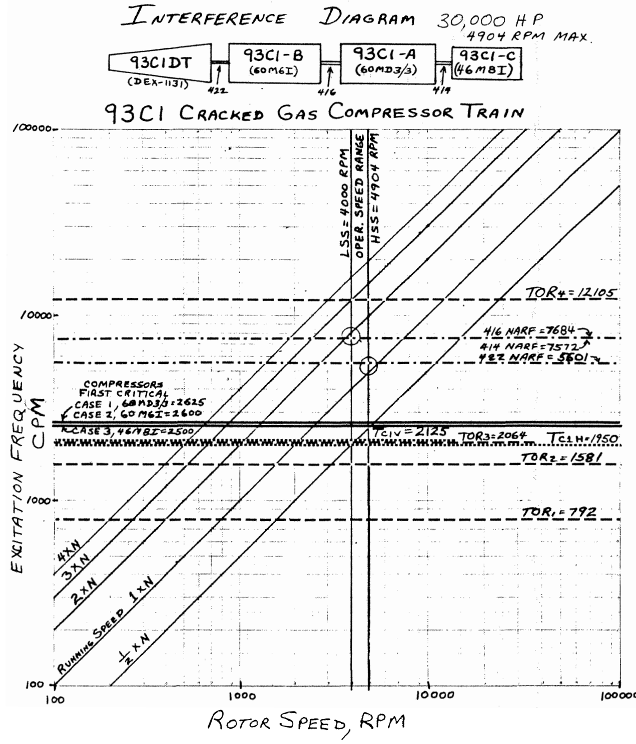

Figure 13. Interference Diagram of the Cracked Gas Compressor Train (Four Bodies). All the resonances (torsional, lateral, axial) emanate as horizontal lines from the ordinate. The operating speeds translate vertically from the absissa (4000 to 4904 RPM) and the ½x, 2x, 3x, . . . are harmonics of running speed, 1x. An interference of a resonance line with, for example, the 1x line in a design speed range, would be bad. The circle on the 1x line indicates a 14% offset with the coupling axial resonance.

(3) The critical speeds and their mode shapes can be plotted (Figures 14 through 18).

(4) The unbalance response or even the "at speed" deflection diagrams could be plotted. The unbalance response was different from a critical speed program in that actual engineering units of unbalance could be inputted and extracted from the program and investigated at any particular station (or rotor location).

(5) Stability analysis could be made on units of concern. Three rotors were investigated on this job. Both high speed, 8-stage, light rotor designs (the ethylene refrigeration and fuel gas compressors) and the propylene compressor, with a high weight, long span flexible rotor.

The propylene compressor analysis showed a stable rotor even though the preloads at the bearings were high and the rotor stiffness lines were on the flat asymptote of the stiffness maps (Figure 17). The first two mode shapes are shown animated in Figures 18 and 19.

Both the ethylene and fuel gas rotors implied they could go unstable with reasonable low aerodynamic cross-coupling. The gas loads (aerodynamic cross-couplings) were determined using some guidelines from Alford [1].

In the end, neither rotor indicated any sensitivity

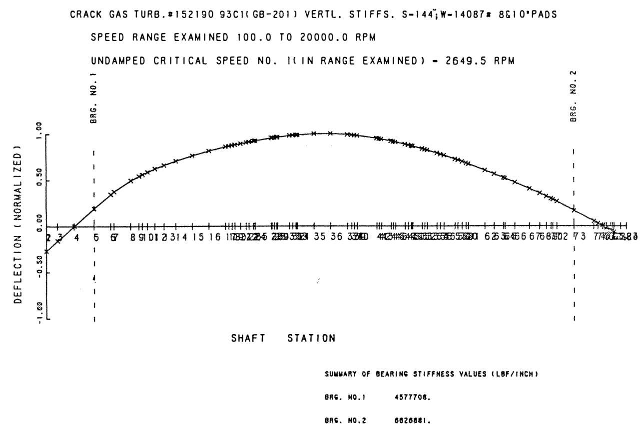

Figure 14. Mode Diagram Normalized to Show Maximum Deflections, 1.0, and the Undamped First Critical Speed (Translational Mode) of a 30,000 HP Steam Turbine at 2650 RPM. Span is 144 in., weight is 14,087 lb., bearing 1 is 8 in. and bearing 2 is 10 in. in diameter; bearing stiffnesses are 4.57 × 10^6 lb/in and 6.63 × 106 lb/in, respectively.

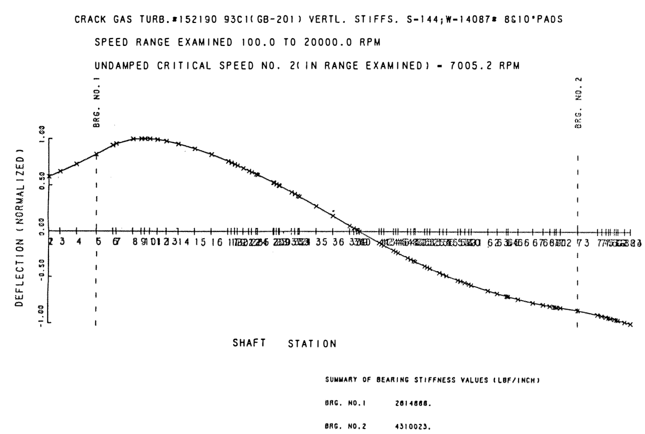

Figure 15. Mode Shape of the Second Critical Speed at 7005 RPM Which is a Conical Mode with Some Bending. This mode was not reached.

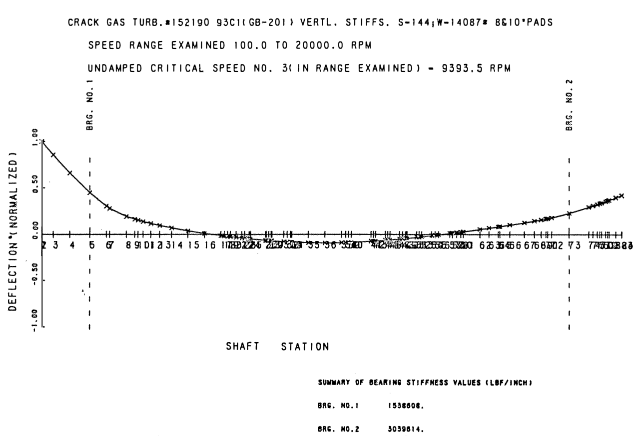

Figure 16. Mode Shape of the Third Undamped Bending Mode at 9394 RPM. Note the center axis is crossed three times. Later in this paper, these modes will be animated to illustrate full rotation.

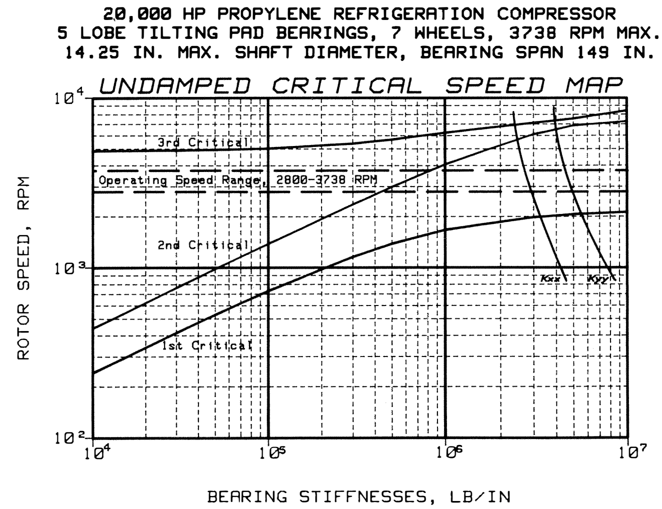

Figure 17. Undamped Critical Speed Map of the 20,000 HP Propylene Drive Turbine. Having the stiffness due to preload, 0.4, up on the asymptote is not a comfortable situation, i.e., any increase in the critical speed would become rotor stiffness dependent.

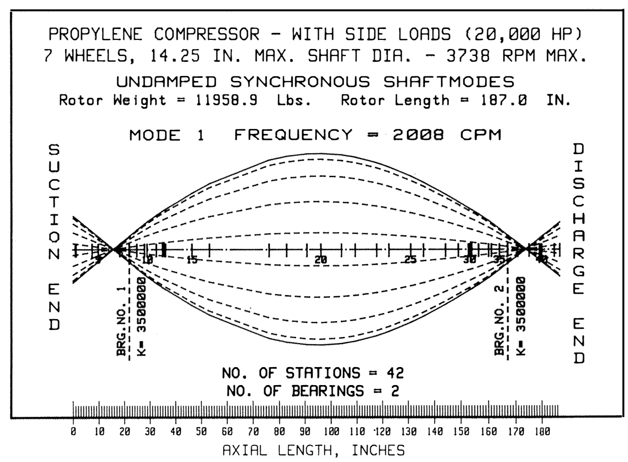

Figure 18. Undamped Critical Speed Mode Diagram of the First Critical Speed Resonance at 2008 RPM. This mode has been animated in plotting to see the full rotational generation. Also, the rotor length has been programmed in actual inches.

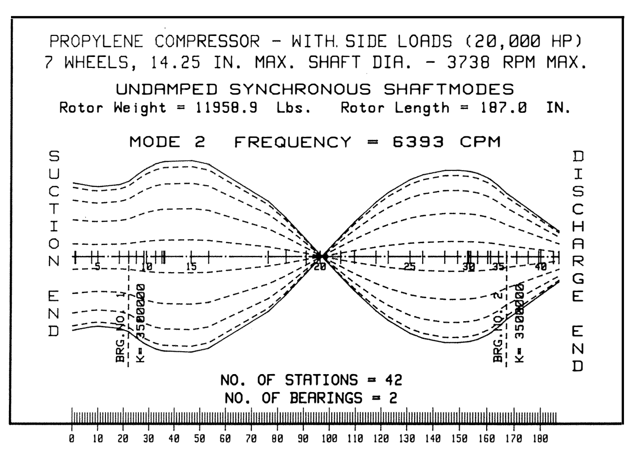

Figure 19. Propylene Compressor's Second Conical Mode Which can be Easily Seen with Animated Plotting.

towards hysteretic whirling and with the criticals below 50% speed, little probably could have been done short of a total rotor redesign. One compressor was increased from 4 in. to 4.5 in. at the journals from discussions with the vendor's design group. A larger diameter does raise the critical which had been at 46% speed. Further, there is more viscous damping in a larger bearing.

The stability plots for the propylene and fuel gas compressors are seen in Figures 20 and 21.

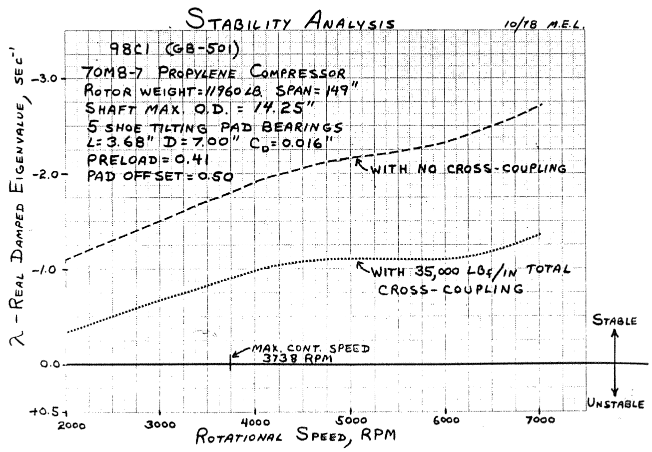

Figure 20. Stability Plot in Damping Factor, 1/sec., versus Rotational Speed for the Propylene Compressor. Note this compressor is stable with significant gas loading and becomes more stable with increased speed.

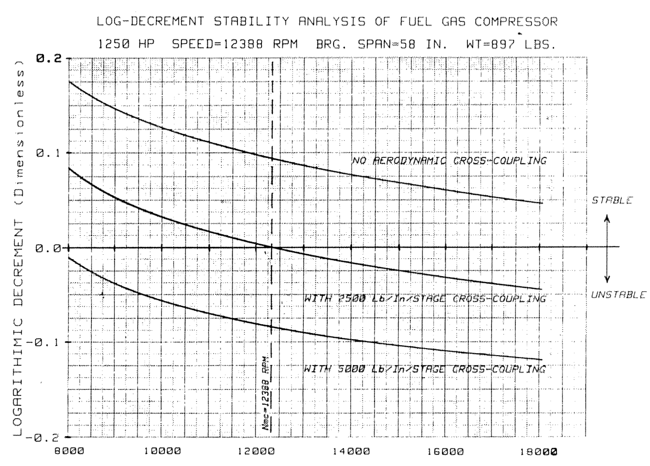

Figure 21. Stability Plot of the Fuel Gas Compressor, Which is Similar to the Ethylene Refrigeration Compressor and the Propylene Compressor. It indicates that light gas loading should drive this compressor operating at 12,388 RPM unstable. Note log decrement versus speed is plotted here.

SHOP TESTING

Unfiltered and filtered data was taken on each vibration sensor both on run-up (acceleration) and run-down (deacceleration). Recordings were made on x-y-y′ plots. These should have been speed, vibration, and phase. The vendors were not prepared for this, thus, only speed, vertical vibration, and horizontal vibration were recorded on run-up. Points were switched before run-down. Some turbine rotors had to be reworked due to damage to the probe area by improper procedure in probe length settings in assembly.

Some eight compressor bearings were rejected due to "dimpling" on the run test. This practice was greatly criticized as this vendor put the thermocouple in contact with the babbitt which can form depressions at 150°F to 200°F due to the babbitt flowing into air voids. This depression causes turbulence and false high temperature readings, and will tear the babbitt on the trailing edge of the dimple and eventually draw a gouge to the trailing edge of the pad. This vendor has since indicated that the method of embedment would be changed for placing the thermocouple tip in the backing approximately 30 mils deep.

Turbine data on the displacement sensors showing elliptical orbits (horizontal trace not 90° in-phase with the vertical trace) were confirmed. Velocity sensors at the same locations (as probes) also confirmed that the bearing cap had the same degree of separation. Typically, this would show on the exhaust end of a condensing turbine rather than the steam end, for tilting pad bearings.

A low frequency modulation of the running speed can be seen in Figure 22. This was taken from one compressor drive

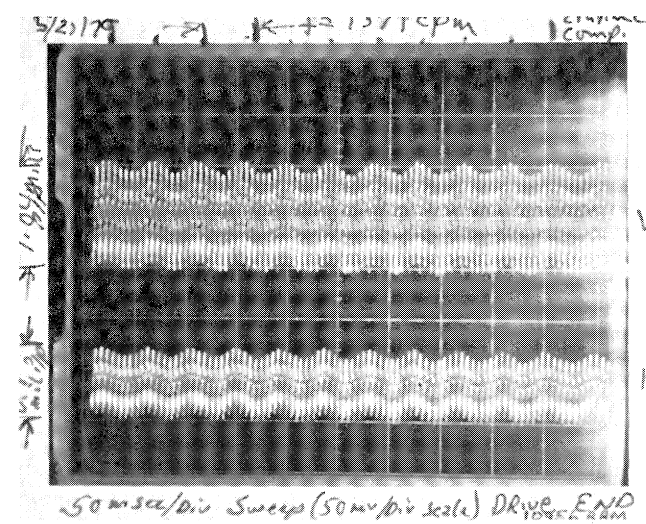

Figure 22. Time Trace Oscilloscope Photograph of a Low Frequency Signal at 1369 RPM Modulating the High Frequency Running Speed Frequency of 10,950 CPM. The flexplate (coupling end) at the test stand was in resonance during API 617 testing.

end bearing. The 1369 CPM (⅛ speed) modulation of the running speed (10,950 CPM) was thought to be generated from the flex plate which was in structural resonance. The shop mounting was not to win a Nobel Prize; however, each flex plate was rung on installation (Figure 23) to determine the mounted resonance after the compressor and piping were mounted and the spring hangers adjusted. (No problem existed in the field, fortunately).

Stability of the switch-over from the main turbine oil pump to the motor driven standby oil pump caused tense moments on the cracked gas and propylene/ethylene oil consoles. These oil consoles are large, handling 400 to 500 GPM flows at 400 psig. Several problems existed:

> The oil pump discharge provided relief valves, not at the immediate pump discharge, but, in fact, at the reservoir. This allowed this line to become air laden as it exists above the oil level and with a pipe length of 20 to 30 feet. Strip chart data showed that even though the standby motor pump started in one second, a period of 4 to 10 seconds was required to build the pressure sufficiently to open the pump discharge check valve. As one might expect, the time was excessive, leading to a low oil pressure shutdown or an extremely violent unstable system. Control valves were cycling, relief valves were popping, lines were shaking, pressure gauges broke, etc.
>
> A secondary factor was the need to snub (reduce) the direct oil supply to the lube oil controller valve diaphragm. There were five control valves on the cracked gas compressor lube console.
>
> Even though modulating type relief valves were used on this system, each had to be removed, cleaned, and recalibrated.
>
> The corrosion protection offered by the turbine builders on governor-extraction valve hydraulic cylinders was insufficient and each cylinder plus servo controller was removed, cleaned, and one cylinder relined with four cylinders rehoned.
>
> The oil filters did not arrive with the 50 μ backup strainer for the 10 μ pleated paper filters. These were provided to protect against (a) filter bypassing, (b) filter rupture, and/or (c) improper maintenance. As a result, item (c) was a problem both in the fact that filters were not drained prior to filter changes and the cartridge end seats were improperly engaged. Most of these bad things happened during flushout.

The oil accumulator capacity at 25 gal. was changed to 50 gal. after several weeks of troubleshooting the lube oil control instabilities. The proper charging techniques for bladder type accumulators also needed to be studied. The author finally resolved to personally charge each and check bladder pressure periodically by partial draining. This insured against a bladder leak. Actually, the author was on site for five months and had the opportunity to check many things personally.

There seems to be a need for a decent quick response direct acting pressure control valve for oil pressure regulation in the larger sizes, i.e., above 2 in. In the past, direct acting stainless steel diaphragm regulator type valves in smaller sizes have been used. Pneumatically controlled pressure regulators with all the proportional bands and resets just do not cut it in the larger sizes. A governor type direct control head or possibly two smaller regulator types are necessary on *large* oil consoles. The oil pumps here were 250 to 350 HP for mental reference.

Cylindrical oil tanks were used on this job with good success. They have been tried on other projects with no success (for many economic reasons). Cylindrical tanks are available from many sources. They are more rugged, easier to clean and the slope is set by the saddle heights, plus plate coil tracing is simple. Who said the reservoir had to be a rectangular construction? These are difficult to clean, fabricate, maintain and have been ripped open on several compressor seal failures from personal contacts with others. As an aside, the author's Company has installed 8 in. conservation vents with reliefs (weighed at 8 ounces pressure) above the operator's positions in services where a reservoir explosion or overpressure is possible, e.g. hydrogen or synthetic gas.

Another Murphy provision was installed on standby motor start switches. These are generally referred to as "Hand-Off-Auto" switches. However, an operator can manually check a standby pump on Monday morning by placing the switch in "Hand" (manual) position. *If* this operator should go to the "Off" position and not over to the "Auto" position, *then* there is no provision for the standby pump to operate as backup protection.

In a technical discussion (semi-violent argument) over this concern, this project provided an annunciator alarm in the control room to alert the operator that the switch was *"not in automatic"* position. This is still not foolproof and three shutdown causes by this type of Murphy oversight have been found.

The electrical code states there must be a lockable "Off" position within easy access (within 10 ft.) of the motor. The normal oil pressure *fall*, standby pump *start*, oil pressure *restore*, standby pump *off but in readiness*, procedure really doesn't need an "Off" position, but simply "Auto - Manual" and a momentary reset button for manual action to shutdown, on pressure restore, in the "Auto" position.

Figure 29 is offered here as a solution and has been introduced for incorporation in the API 614 Lube Standard, now in CRE vote for update revision.

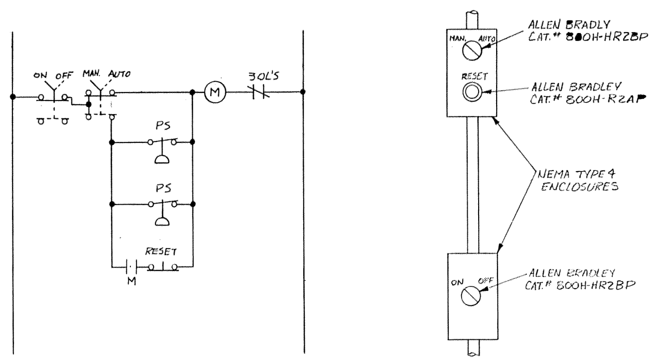

Figure 29. A Logic Diagram for Control of the Motor Driven Standby Oil Pump Removing the Normal "On-Off-Auto" Switch in Favor of "Manual" or "Auto" for Normal Control.

Transfer valves have been a problem with many consoles for more than a decade. The traditional suppliers and new suppliers (some without a tapered plug) are considered on future projects. The greatest pressure drop occurs in the transfer valve as opposed to the filter on API systems. If the holddown on the plug is not released, operation will score the seats in trying to rotate the plug without releasing the holddown device. Further, if the holddown device is left open (a tricky move by operators not prone to constructive criticism), then the oil moves port to port, bypassing the filters. An economical move, regarding transfer valves, is to group the coolers and filters in the same loop, halving the number of transfer valves. If the quality of these valves does not improve, installation of four-line blinds may be required to allow for tight shut-off for maintenance. A new approach is to specify bronze fitted, cast steel bodies, with 400 series stainless plugs.

Finally, a second fresh note was that all the problems were overcome prior to run-in tests and without project delay. Actually, the commissioning of the lube oil console and system should be first on the agenda and the heated oil should be left continuously in circulation. A small pony pump can often be temporarily installed to reduce power costs.

Pipe Header Stress Checks

Steam lines were first blown down with a series of 15 second blasts against target plates and bars to rid the lines of loose scale. Steam at 1200 psi (83 bar), 900°F (427°C), was slowly pressured to the turbine trip throttle valve to note any movement of the uncoupled turbine. Movements were less than 1 mil (0.001 in.) (25.4 μ M). Measurements were made on all four corners with both dial indicators and eddy current probes. On the exhaust shaft, indicators and probes were also installed. The unit had to be roped off to prevent the dial indicators from being knocked off. The eddy current probe is nice to have as a precaution should the trip throttle valve leak when the header is pressurized. Both valves did leak and both turbine throttle valves (TTV's) were overhauled, which is no easy task. The turning gear and vacuum system had to be prechecked and ready for this test.

A start-up screen was tack welded to the standard strainer on the TTV's per purchase specifications, so that the turbine was better protected through trial runs for hot alignment and control check-out. The start-up screen can then be removed without disturbing the steam lines.

The couplings were never installed except for equipment settings; they were removed to prevent any DC currents should improper welding occur. One incident did occur. It was checked by using a gaussmeter and required degaussing at that location only.

The compressor piping had been positioned early in the installation when the casings were put in cold alignment. This down connected piping was supported with cross-bracing (strong back beams) to the correct mezzanine and the compressor flange bolting was removed.

After all spring hanger adjustments were checked along with "anchor stops" and the pipe insulation was 90% completed (heavy weight added), the stress check was made on the compressors. This was similarly measured as in the turbines, but the compressor piping was flanged up, one at a time, and movements noted. All were within ½ mil (0.005 in.) (12.5 μm).

Control and Auxiliary System Checks

Vacuum

The vacuum system worked well once all the leaks were located. Flanges were taped with "tell-tale" holes punched to locate leaks.

The vacuum traps were installed upside down on the condenser and worked okay once corrected. The oversized hogger jet was helpful in commissioning the condenser-vacuum system, but was rarely needed after that, as it should be.

The special provisions for troubleshooting the hotwell pumps *all* paid off. These were:

1) Sight-glasses in the vent lines.
2) Sight-glasses in all of the seal and packing vents.
3) Emergency water provisions to gland seals to establish vacuum and prime pumps.
4) Automatic and manual return discharge recirculation lines to the condenser from the hotwell pump discharge.

People seem to forget that one cannot open the vent on a condenser hotwell pump to determine if it is primed with condensate. The vent doesn't blow — it sucks.

The hotwells were large on these condensers which was good. The level had a difficult time getting quickly out of viewable sight-glass control (which made operators and machinery people nervous).

Valves

An extraction non-return valve cracked its shaft on solo runs which allowed one to become an admission turbine rather than an extraction turbine as purchased. Governor and extraction positioners on the turbine helped in the diagnosis of this problem.

The extraction valves on one turbine did not close properly and realignment of the bar lift mechanism guides and freedom of the stem gland bushings was required.

The control oil regulator valve(s) for the turbine was short on response time by one order (10x too slow). These were replaced with modulating type regulator relief type valves.

The electro/hydraulic governor valves for the two big turbines required two to four weeks in calibration and response checks primarily because of impedance matching for long distances (800 to 1000 ft).

A significant amount of I&E instrument checkout and recalibration was needed on this job and is typical of all projects.

Turning gears were a blessing on this equipment, specifically the hand crank for positioning rotors for inspections, repairs, and alignment. Several adjustments were made to one turning gear to prevent unexpected disengagement.

Bearing Checks, Rotor Float And Cold Alignment

All of the bearings were checked for any damage and the journals coated with STP.

Nitrogen purging had faired well with all of the internal inspections through various openings.

The rotor floats on both of the turbine(s) and the thrust collar to radial bearing measurements showed that the steam inlet bearing standard (pedestal) was out of square with the shaft by 0.020 in. across the thrust collar radius. All construction shipping keys were in the proper position and clearance. Both bearing standards were rotated about their trunnion pins and reshimmed with 30 in. × 30 in. square shim plates to effect a proper axial squareness to the radial bearing shim support area which was also remachined. Eccentricity run-out readings were taken from the shaft to the bearing house ID as the rotor was rotated in its journals. Also, corrected thrust collar to radial bearing face inside micrometer readings were made to assure squareness was achieved. This required one week.

Thrust floats were made for total casing travel, proper nozzle stand-off, and cold active-to-inactive rotor float in the thrust bearings. Solid shim plates approximately 1⅛ in. thick were machined to accommodate these dimensions.

A few damaged thrust shoes were replaced from babbit scraps and damaged dual tip thermocouples corrected, as confirmed by conductivity tests.

Missing EYS thermocouple conduit seals were an excitement as oil filled all of the conduit system on the first solo runs. Temporary rags held until run-up was completed and a proper seal compound was packed.

Precalculated reverse indicator alignment values were obtained within acceptable pre-run-in tolerances. Three sets of charts were maintained. One in the millwright shack on the

Treatment of a broader area is favored, as rotors do exhibit thermal growth and axial floating, i.e., a different surface can be exposed to the probe.

A technique noted earlier by Dale Beebee with Turbodyne has also been used successfully. The probe area can be rolled in a balancing machine with the rollers directly on the probe path. Rolling continues only until an observed haze or matted appearance remains on the rotor. Overworking would be detrimental.

Often, a combination of treatments may be necessary, e.g., an uneven carburization layer may exist from forging which could be removed by machining or grinding. If so, it would be better to machine first, degauss, then burnish, if needed. Do not ever burnish if it is not needed. It will most often introduce electrical run-out that was not present before.

Figures A-1 and A-2 show the results from a proper treatment of a rotor shaft. A center holder (or vee blocks) was used to hold the rotor on a granite table. A two-pen recorder is handy to record run-out. Large degaussing coils (8 in. range) were available to make it an easy and proper manner for inspection.

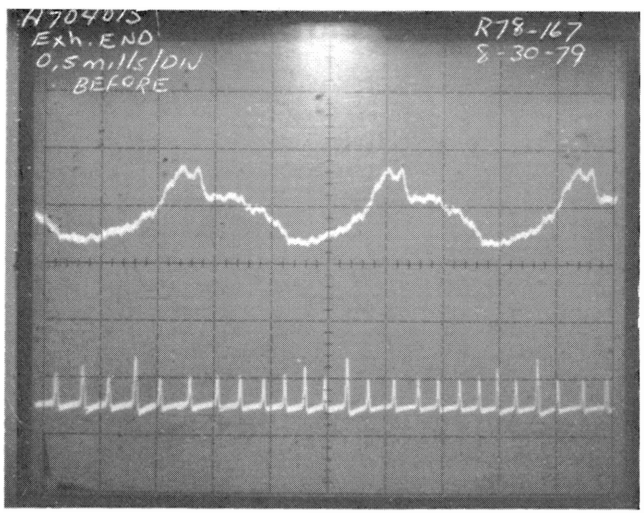

Figure A-1. Shaft Signature Before Run-Out Treatment.

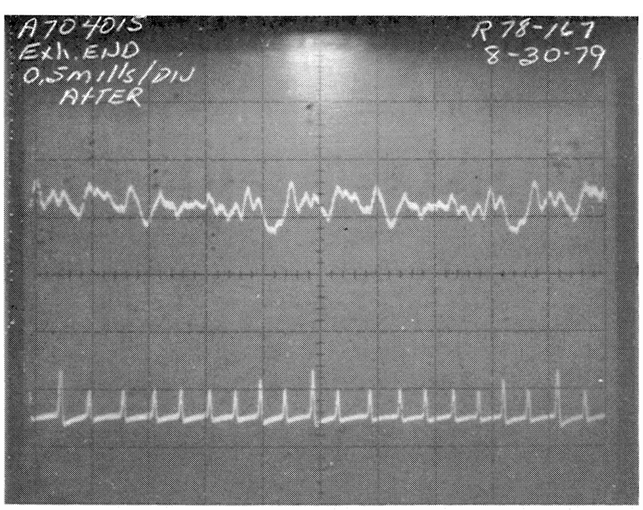

Figure A-2. Signature of Shaft After Treatment to Reduce Run-out.

To conclude this discussion, these observations are made. They are only observations, but seem to hold true with experience. However, little is absolute.

1. Steam turbine manufacturers have less trouble with electrical run-out than compressor manufacturers because the shafts are often double temper stabilized alloy steel shafting.
2. Commercial rolled bar stock shafting seems to offer more susceptability to electrical run-out.
3. Forged steel shafting with minimum upsets and close finish dimensions (for reduced metal waste) can offer problems from carburized or oxide layers near the finished journal OD. Further, either forging jaws or three and four jaw lathe chucks may be causing double run-up, 2x, type electrical run-out patterns.
4. Precipitating grades (15-5, 17-4 pH) are notorious and will remain with little hope of betterment. This alloy is not homogeneous because of other elements which have been placed interstitially in the melt to derive the heat treating properties at low temperatures.
5. Any form of surface coating (chrome plating, nickel plating, metal spraying) will complicate electrical run-out. Further, a DC voltage gain occurs with austenitic type materials. They seem to correlate with thermal growth coefficients. An aggravation to plating, as a cure of electrical run-out, is that the person removing metal prior to plating must receive proper instructions. The machinist taking off metal may not be too interested in centering his work as the plating finish can correct any errors he might have introduced.
6. Burnishing and rolling are more practical, cover a wider area, are more uniform, and have given the best results.