RETROFIT, UPGRADE, AND DEMOTHBALL OF TWO CENTRIFUGAL COMPRESSOR TRAINS

by

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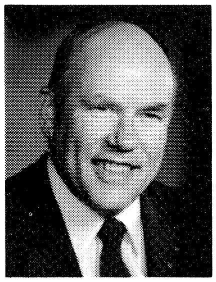

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He served on the API Mechanical Equipment Subcommittee for 22 years. He has a BSME degree from Texas A&M University and an AAS degree in Electronics Technology from the College of the Mainland. He is a member of Pi Tau Sigma and Tau Beta Pi. He has published over 50 technical papers and one book, The Practical Vibration Primer.

ABSTRACT

The design premise in upgrading two operating plants for more compression capability is discussed.

The first is a syn gas plant designed for 1000 tons/day of methanol but converted to syn gas only for a primary acetic acid market. The upgrade concentrates on a carbon monoxide (CO) compressor train which upstaged an original pair of compounded centrifugal compressor casings (15 impellers) to a larger casing (first body), roughly four sizes larger at 14 impellers compound compressor casings(two). The new train uses an available larger horsepower steam turbine available from the previous process conversion and it s concrete mezzanine and intercoolers located just south of the present CO train. The timetable called for conversion in one year, plus shutdown and switchover at 30 to 45 days.

The second upgrade required adding a third steam turbine driven centrifugal compressor air train, which had been "mothballed" some 13 years prior. The turbine and compressor rotors were vertically immersed in plant fabricated steel containers holding VSI inhibited ISO 43 turbine oil. The empty casings on a 20 foot mezzanine were coated with a tacky rust preventive and dead head blanketed with nitrogen. The time frame was 18 months.

The intent of presenting these two conversions is to focus on the design considerations to be addressed for such a project to be successful and also upgrade the state of the art after 13 to 18 years.

INTRODUCTION

The two cases are addressed with the conversion of the CO compressor handled first. This presentation charts the upgrade of two rotors, one casing, and the coupling drives for the location to a new nearby site and the upgrade of the drive from a larger steam turbine in prior service for 18 years and rated in the same 10,000 to 11,000 rpm speed range. The horsepower was increased from 3160 to 5000+ at 10,700 rpm on a nearby foundation modified to receive the two compounded compressor casings in this train and reuse an existing rerated steam turbine. The oil system was converted from lube and seal to lube oil only. The train was commissioned on December 23, 1988.

The second case, which takes a large four impeller, 48 in diameter, by 172 in long centrifugal air compressor rotor at 12,000 lb weight on seven inch diameter journals out of "mothballs" will then be discussed. The steam turbine, with one velocity-compounded curtis stage, plus five rateau stages rated at 10,000 hp and 4,000 rpm, is also taken from a submerged-in-oil container and refitted, reset, aligned, and recommissioned in October 1989.

RETROFIT/UPGRADE OF STEAM TURBINE DRIVEN TWO CASING CO COMPRESSOR

The existing train is described as a back pressure steam turbine rated at 3,160 hp and 10,500 rpm driving two compounded centrifugal compressors, i.e., four sections of compression and intercooling within two compressor casings. The compressor casings were horizontally split with seven impellers in a casing capable of nine stages for the first body, along with eight impellers in a casing capable of holding nine impellers. The train had been in successful service for 18 years, taking primarily extremely clean CO from a cold box having operated internally in the $-325°F$ range.

The steam turbine had been upgraded once to the current 3,160 hp which was limiting with plant expanded operating conditions. In 18 years, the compressor was opened once on an overspeed, due to a TTVs failure to close. One case had been opened to change a rotor out on a rotor shift using the spare rotors. Turbine and compressor rotors were spared in rotor cans of inhibited turbine oil. Operation wise, the biggest handicap, in operating well over the original design capacity, was that one section could be in "stonewall" while another section was near surge.

The compressors take CO gas from a cold box near zero pressure and deliver it to a reactor at 600 psig. If the pressure from the cold box was a few pounds higher, e.g., winter conditions, then the steam turbine became overloaded.

The entire cold box had been recently superseded by a totally new larger unit which greatly improved the compressor's output by furnishing gas at the suction at five to eight psig from a former zero to one psig condition. When delivering gas to a reactor in another plant at the 600 psig regulation pressure, this is a tremendous improvement alone. One must remember that a centrifugal compressor is a volumetric machine; it can only handle what you can get into the suction in inlet flow, icfm. If the density of that volume is higher . . . good; it can meet discharge conditions easier but can draw a higher horsepower. The polytropic head is really ft-lb of work per pound of gas handled. It is easier to compress a higher weight (mass) flow and inlet pressure, but it is more sensitive to volume conditions rather than pressure conditions. A recip it is not.

To set the scene, it was desired to utilize the new/larger cold box to the fullest, upgrade the centrifugal compressors to 54,000 lb/hr capacity, and the turbine driver to at least 5,000 hp and reduce pressure drop to/from the compressors, e.g., suction from cold box and the discharge run to the reactors in another operating block. The piping changes were accomplished during scheduled outage times. A steam turbine previously driving two compressors was available rated at 6,150 hp and at the same speed range within two percent.

PROJECT MECHANICAL DESIGN PREMISE

• Since the turbine was available, the original compressor design gave satisfactory performance, the impellers can be staged forward (push in wider ones at the inlet and narrower ones out the discharge), a single source supplier would be used in bidding.

• A Section 1 Premise would be written to inform the contractor and the compressor vendor of the complete intent of the user, i.e. stamp out any misunderstandings of the goals.

• The compressors would be built to API 617 Standards with the owner calling out all bulleted (•) items. The mechanical running tests would be required on both contract and spare rotors, each rotor would be mechanically fitted in the new casings. Only one performance run will be required and performed on the "spare" rotor which fits in the casing first. (Cost advantage against risk (spare *vs* contract) on any needed design changes by an unsuccessful performance run.)

• Rotors to be sequentially stack balanced, with the contract coupling(s) a separate step in balancing, and calibration verification (not sensitivity test) required.

• The couplings were to be upgraded to hydraulic fitted dry type with both torsional and lateral analysis included. Reduced moment coupling were stipulated as the existing lubricated flexible gear type (lighter) were to be replaced. An 18 in spacer was to be maintained to allow for more misalignment.

NOTE: Only moderate "sludging" had occurred in the past using 5.0 micron followed by ½ micron filters along with three gpm minimum oil spray over the 18 year span; but why not eliminate this threat.

A two inch minimum clearance by the coupling enclosure was to be maintained.

• Labyrinth seals were to be continued with CO service with staging eductors and nitrogen buffering (*safety*).

• Heat rise and axial growth data to be calculated by the compressor vendor who was also to be the *train responsible vendor*.

• The scheduled project time and the 45 days conversion time from "old" train to "new" and the speed/weight considerations, dictated a heavier "I" section over channel section (*owner decision*).

The fabricated baseplate would be built to rest on adequate (10) level plates using greased "jack-up-leveling" and "pull-down-anchor" bolting (Figure 1).

• The fabricated baseplate was to be immediately set from the flat bed delivery truck. The truck would drive straight through with two drivers (Figure 2). Prearranged "mirror-optics" would allow proper alignment to the existing steam turbine and grouted with low shrink epoxy grout after removing the catalyzed primer from the baseplates (Figure 3).

Fill holes and vents provided in the base plate would allow the baseplate to be totally grout filled (cheaper grout), after proper alignment has been confirmed (Figure 4). Full stainless

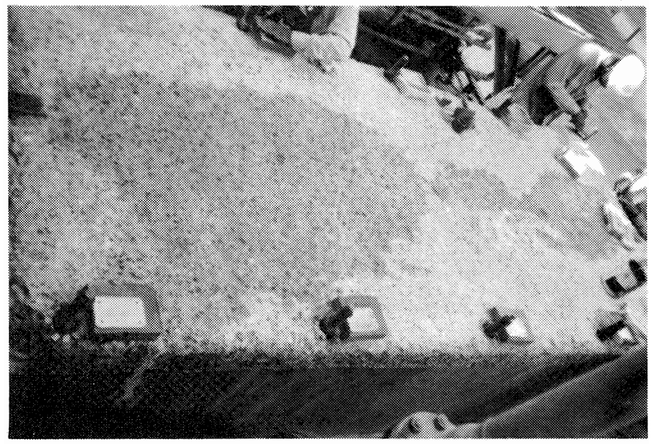

Figure 1. Concrete Mezzanine Base Prepared for Baseplate.

shims at ¼ in total thickness (⅛ in + 1/16 in + 1/16 in) would be provided by the compressor vendor during witnessed shop assembly (Figure 5).

• The owner would optically shoot the existing/repaired steam turbine rotor shaft for centerline projection and elevation. Owner to provide minimum ½ in thick plates at each anchor bolt/jack-up bolt location for the lifting bolts to seat against (Figure 6).

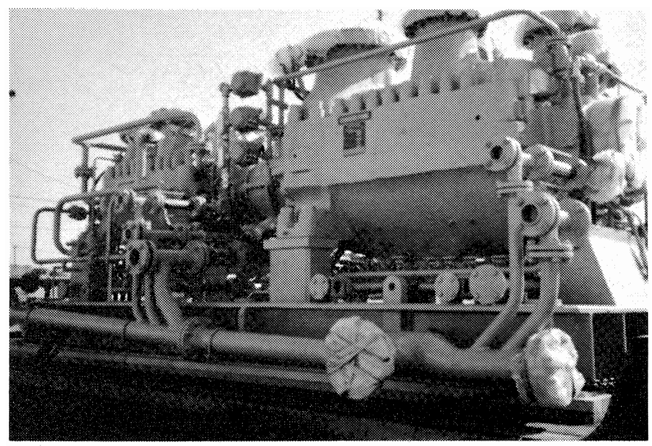

Figure 2. Two Compound Compressors on Baseplate Arriving on Site.

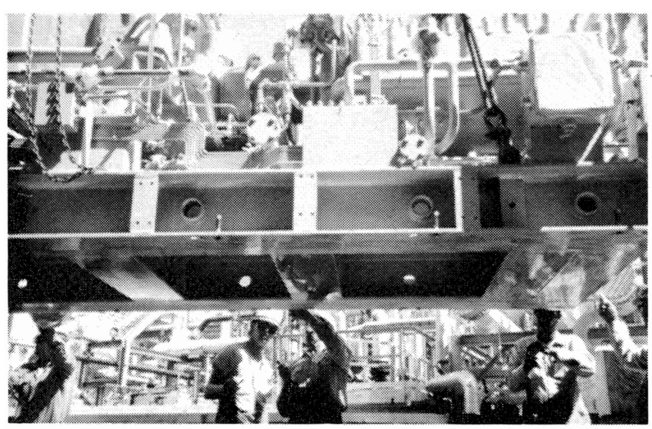

Figure 3. Baseplate Underside Being Cleaned before Setting.

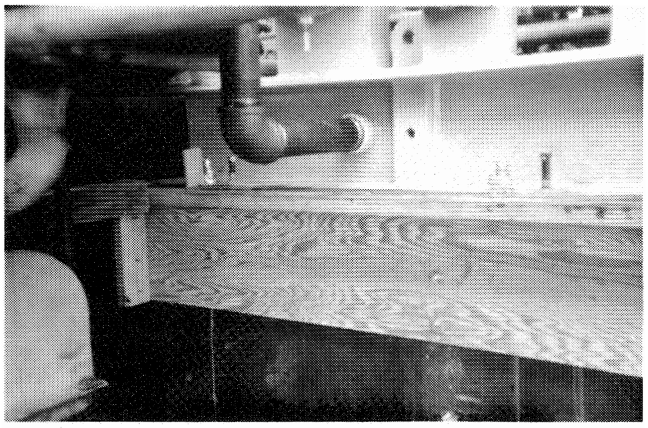

Figure 4. Baseplate Set and Formed for Grouting to Main Frame.

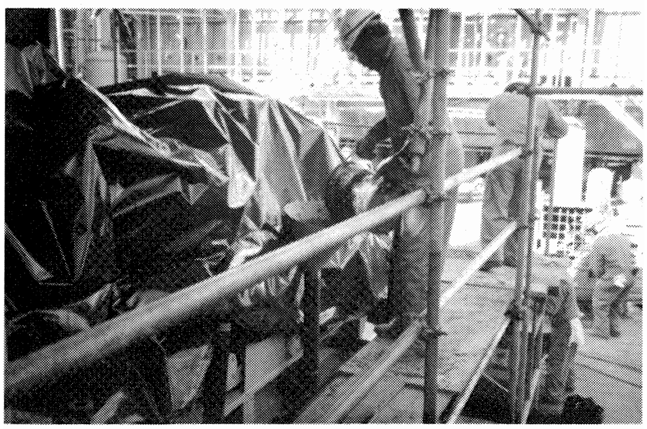

Figure 5. Baseplate Being Poured in Frame Full with M.B. Grout.

Figure 6. Optical Alignment with Mirror at Turbine Exhaust Shaft End.

• Alignment brackets to be provided, by owner, for "reverse indicator" alignment prior to *first* grout of the baseplate and for *final* alignment after all positioning of the compressors is complete with discharge/suction spools in place for new piping make pieces on "shutdown/switchover" period (Figure 7).

• Cold Alignment to be based on previous hot-growth data taken with Jackson eddy current probe/water stands over the previous winter/summer 18 year operating conditions (see Alignment Map in Figure 8).

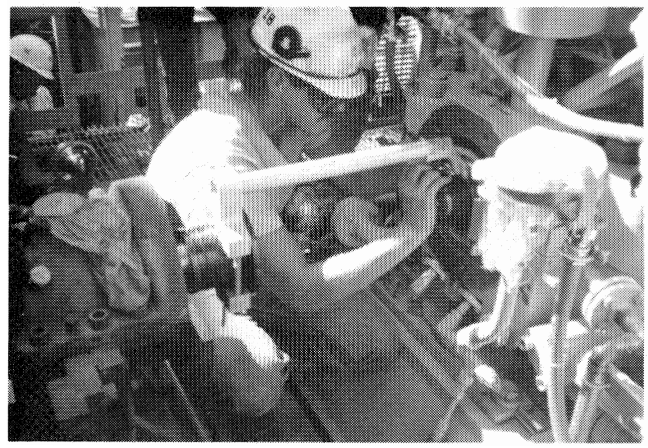

Figure 7. Reverse Indicator Alignment Across Coupling Span.

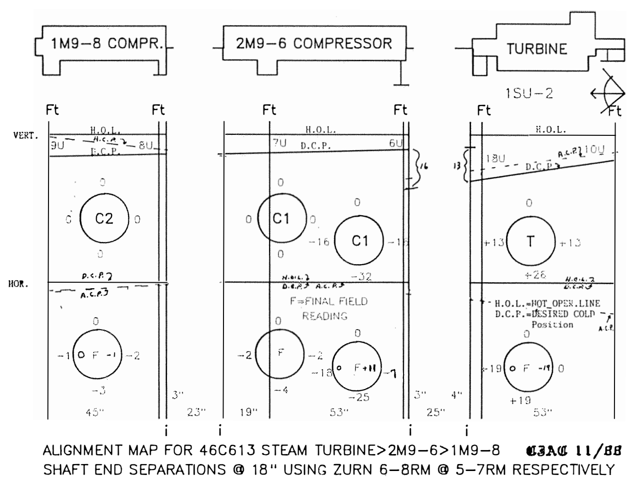

Figure 8. Alignment Map for Turbine and Two Compressors Based on Thermal Heat Rise in the Field on a Minimum of Three Measures.

• The steam turbine to be removed and the turbine rotor shipped to the vendors shop for full inspection. The rotor turned out to require only light shaft and rotor dressing and rebalancing without any blade changes. The diaphragms, however, required extensive rebuilding and remachining.

• It was decided to keep the governor since there were no good reasons to change this system.

• The inlet steam valve position indicators were provided in the first design (1968-69) and would be retained (Figure 9).

• There was a flurry of design tic-tac-toe correspondence on whether the inlet valve sequencing, sizing, lifts, etc., should be changed and what percentage improvement would be obtained. Any changes were thrown out, based on the fact that seven inlet valves were already incorporated and 80 percent of the original design horsepower was being used and no agreement could be obtained indicating even two percent improvement by redesigning. (A good decision also based on:)

• The steam turbine had gone into partial admission "pure oil whirl" in 1970 on startup with pressure dam bearings after about two to three steam valves were opened (Figure 10). These bearings were changed out by surprisingly fast vendor response at

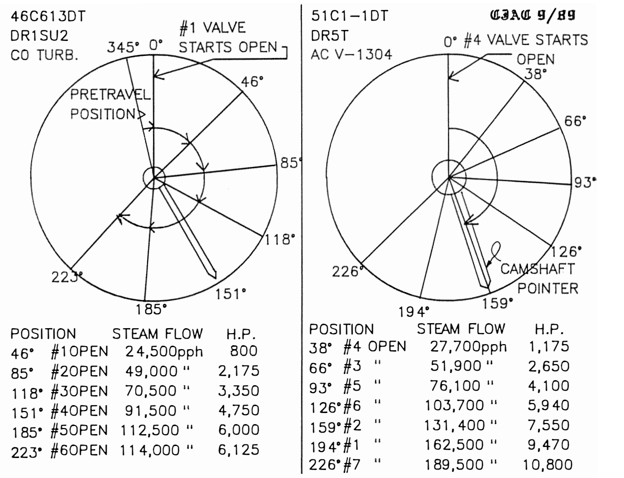

Figure 9. Steam Turbine Inlet Valve Position Indicator on Camshaft. One is for this train's turbine. The other for the second train.

that time to a four lobe design which has been successful for 17½ years and would be retained (Figure 11).

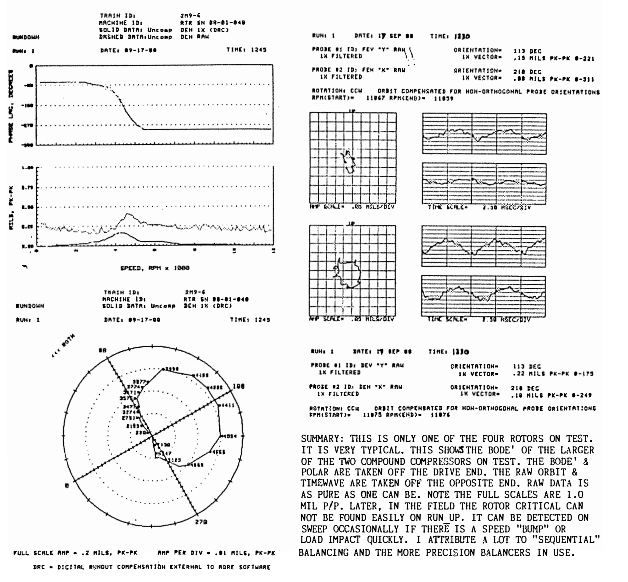

Figure 10. Startup Data Taken at the Factory on Mechanical Run. This is typical data for one of the four rotors tested.

This incident was reported in a 1973 ASME paper. (It was also 1 of 13 case histories for a paper in the *Proceedings of the Fourteenth Turbomachinery Symposium*, 1985). Incidently, it was learned via "orbit analysis," at that time (hint by Albert Kingsbury), that the pressure dam bearing could be rotated in the housing to affect a "limit cycle whirl" giving two weeks at that time to obtain and install the corrective four lobe design. The oil whirl by partial admission was proven, at the time, by gagging the inlet valves totally open (full admission) and bringing the turbine and compressor up to full speed and load *without oil whirl*.

• A fitting mandrel would be made (Figure 12) to fit the diaphragms into place in a quicker and accurate manner.

• The vibration probes and the monitors would be advanced two designs to -24 volt systems with liquid crystal display for *all*

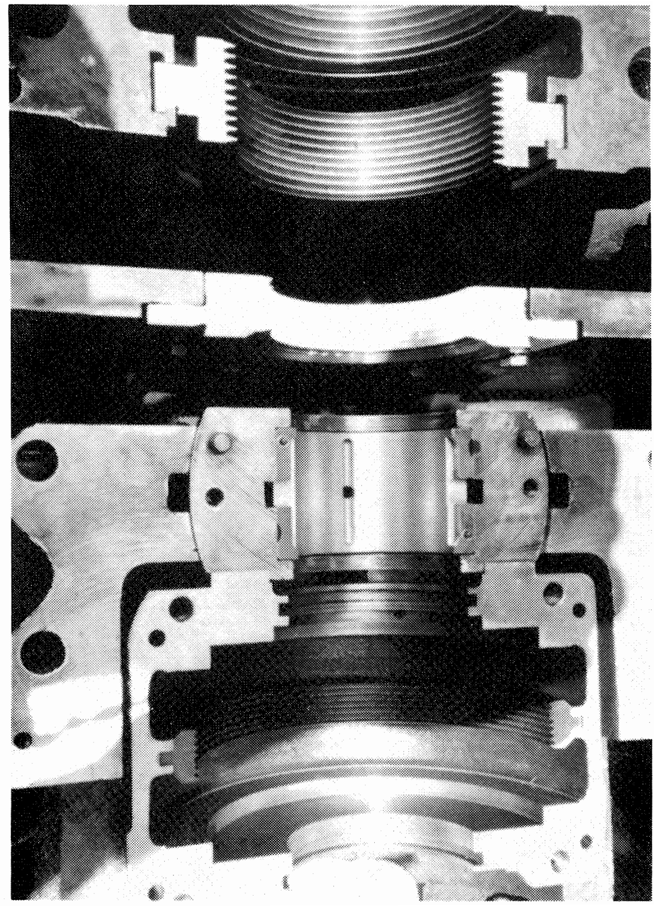

Figure 11. Four Lobe Bearing Installed to Prevent Whirl (in 1970).

probes installed. Dual voting logic on thrust would be provided, as before with alarms at 15 mils and *automatic* shutdown at 25 mils, with three seconds delay. The radial vibration alarms would be continued at 2.2 mils p/p for alarm and 4.2 mils p/p for *operator committed manual shutdown*.

• Thrust and radial bearing temperatures were installed with 200°F. degrees for alarm and 225°F for *NON*-automatic decision shutdown.

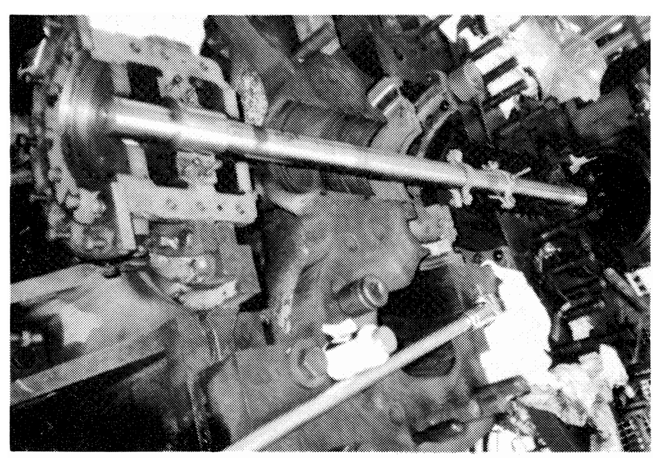

Figure 12. Fitting Turbine Diaphragms with Fitting Mandrel in Field.

• Great care would be exercised in assuring that the prestretch dimensions (28 and 30 mils) were achieved at the shaft end separations (BSE) for each of the two multiconvolution dry couplings; with the three rotors in the normal operating position, i.e., on each "active" thrust bearing.

The train layout is shown herein to explain where the casing anchors (dowels) are located and to show how the thermal growths, casing and rotor, are accommodated to appreciate "why" a prestretch (shaft gap greater than coupling free length) is so important (Figure 13).

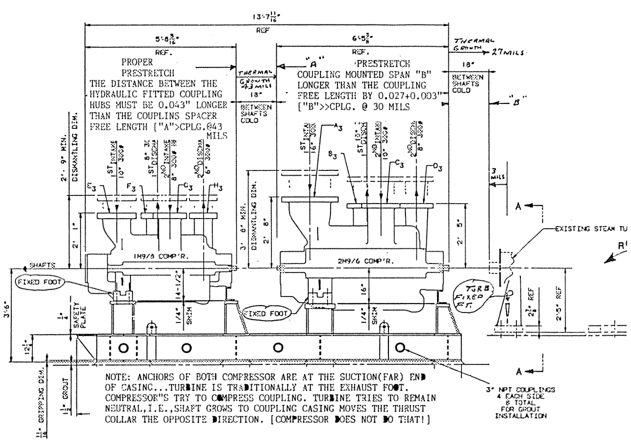

Figure 13. Compressor/Turbine Layout to Affect Proper Prestretch of Couplings.

• Three dimension alignment bolting is required to properly move the compressors (Figure 14). The steam turbine remains fixed and assumed to be in the "desired cold position."

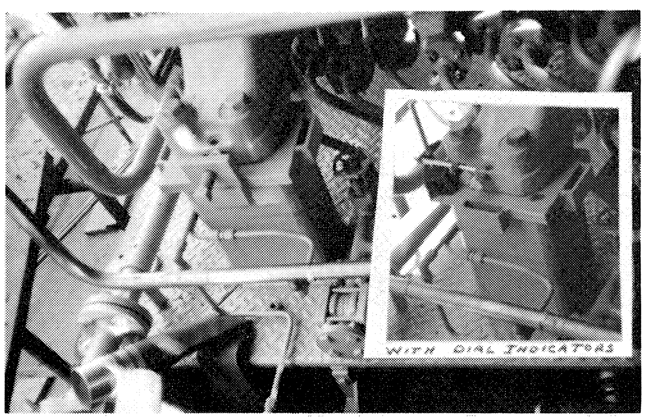

Figure 14. Alignment Bolts, X-Y-Z, for Compressor Aligning.

• The owner added an eight minute minimum rundown tank in the upper pipe rack as a backup against the turbine main oil pump and the motor "standby" API 614 lube scheme.

NOTE: The previous compressors in this "slot" used floating oil bushing seals with overhead tanks. This CO Compressor uses labrinth seals (safety).

CONCLUSIONS

The two API 617 performance tests (spare rotors only) and the four rotor mechanical fitting and the four mechanical (four hour) run test were performed. See Figure 15 for Bodé on runup of one typical rotor.

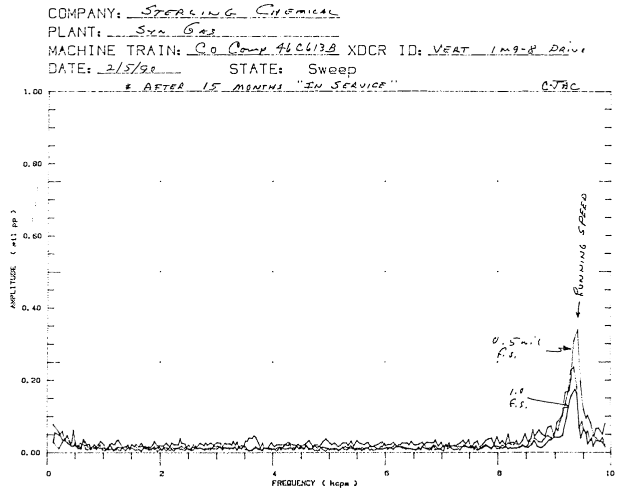

Figure 15. "Sweep" Plot of That Same Point to Confirm the Vibration Is Only at Running Speed (1×).

The delivery was within one month of schedule. The conversion went as planned. The commissioning went off at 0400 on December 23, 1988, beating the end of year commitment for customer product. The maximum vibration down the three piece train was less than ½ mil p/p vibration on the x-y proximity probes for the three bearings. Data taken 15 months later can be seen in Figures 15, 16, and 17.

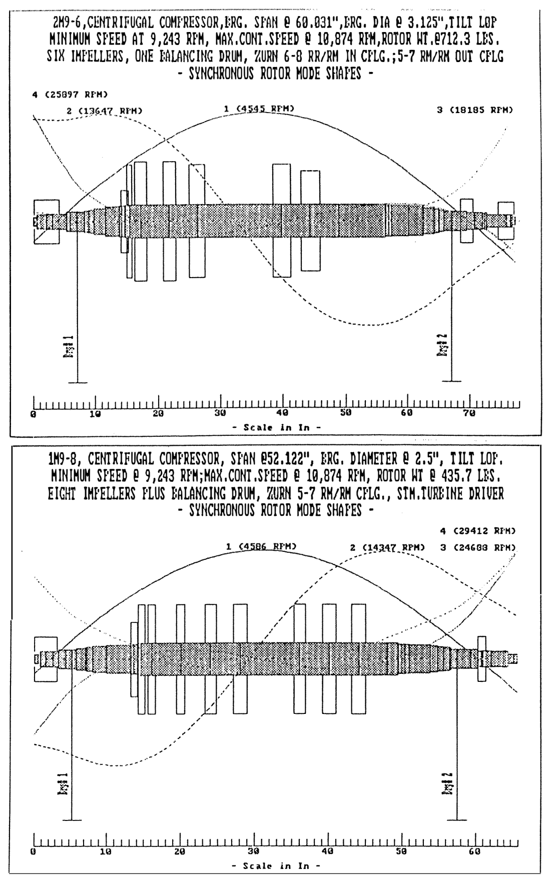

*Figure 16. Calculated Critical Speed Mode Diagrams for **the** Two New Compressors. One for each.*

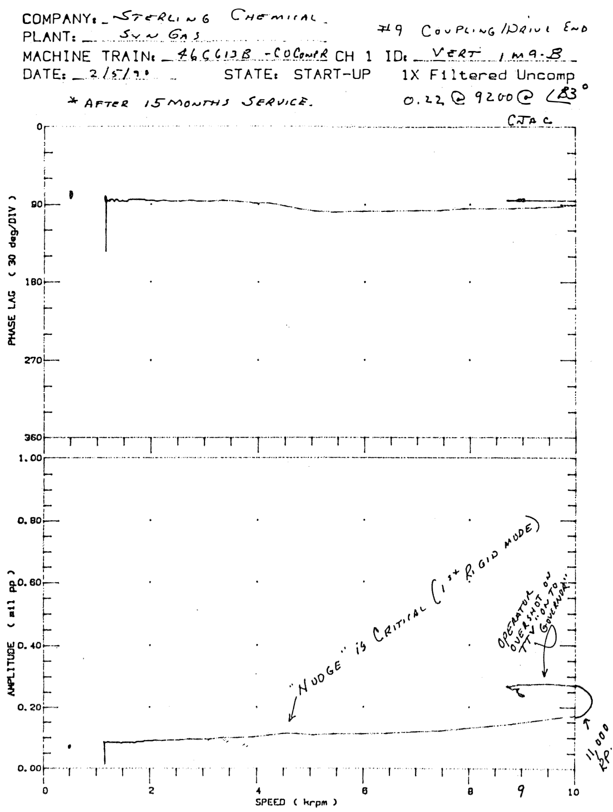

Figure 17. Bodé Plot on One of Eight Data Points Taken After 15 Months in Service. Full scale is 1 mil p/p (should have been ½ mil p/p). The critical is difficult to detect.

The thrust of this presentation was the conversion premise for those not familiar. It is hoped that the premises may be food for thought, should one consider a similar upgrade conversion. Close inspection in all three shops involved was performed by professional people knowing what was necessary, carried out in selected, responding and responsible shops.

DEMOTHBALL OF STEAM TURBINE DRIVEN CENTRIFUGAL COMPRESSOR

Introduction

Two of three steam turbine driven centrifugal air compressors were installed and commissioned in 1978. The third train was placed on the mezzanine, but the rotors were removed since the third train was withheld pending higher process product marketing demands (Figure 18).

The steam turbine rotors (two, including spare) and the two air compressor rotors (contract + spare) were immersed into higher inhibited ISO 32 turbine oil, suspended by the lids of the shop fabricated, false bottom canisters (Figure 19).

The casings were left coated with a heavy nonrusting sprayed on inhibitor, typical of "long-term" storage. The ends of the casing were closed with special machined aluminum plates with gaskets. The two casing were left with a slight positive pressure, eight ounces, by dead heading nitrogen via tubing from regulated bottles, and special warning signs—*Under Nitrogen Blanket*.

Projected process forecasts in 1988 initiated a large project to complete the third reactor train and compressors for a startup in the third quarter of 1989. Therefore, this report covers the premise to be carried out by the owner, his consultant, and the selected repair shops and subcontract field people.

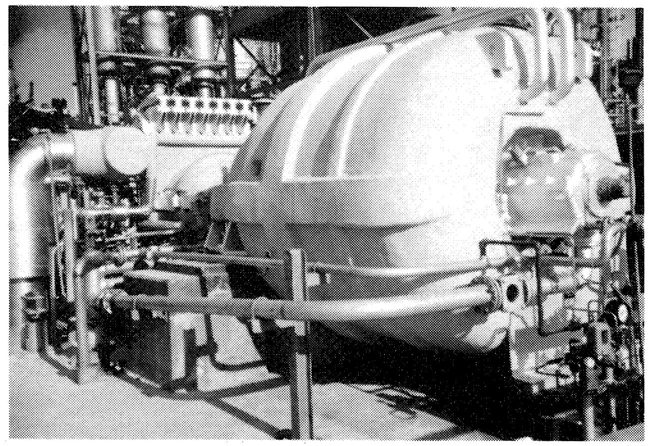

Figure 18. Steam Turbine Driven Centrifugal Air Compressor.

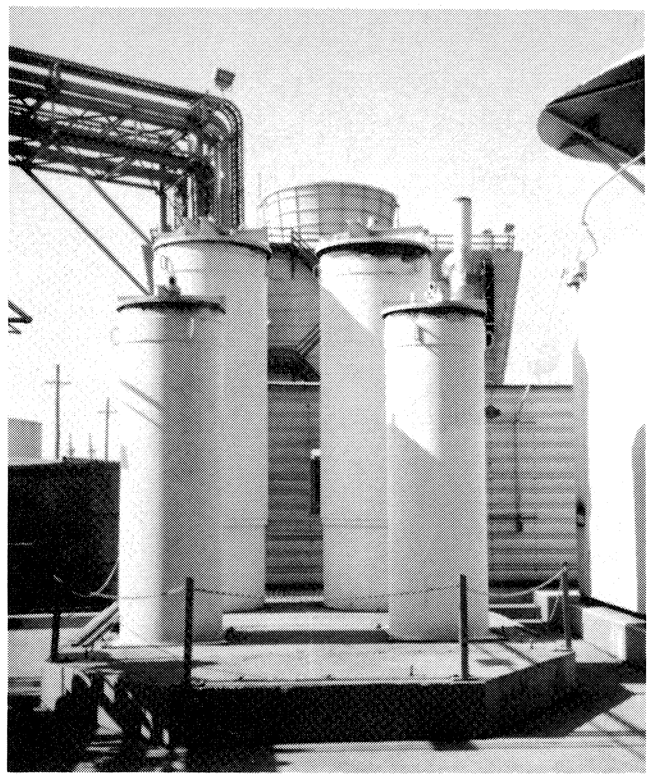

Figure 19. Oil Filled Canisters for Turbine and Compressor Storage.

Again, it is the planned steps which are being presented as they seem always to be so important to successful startups; i.e., it's no real difference from the initial commissioning in planning. However, after a 13 year hiatus, it does allow upgrading to the latest concepts.

PROJECT MECHANICAL DESIGN PREMISE

- Remove turbine and compressor rotors from "submerged-in-oil" storage canisters and truck to repair shop in LaPorte, Texas, for truth checks, inspection, shaft taper checks, bearing journal inspections, and check balance with "trim" from 1/10 "g" values to 4W/N (oz-in) of residual unbalance with 6 point data plot (See Figures 20, 21, 22, 23, and 24).

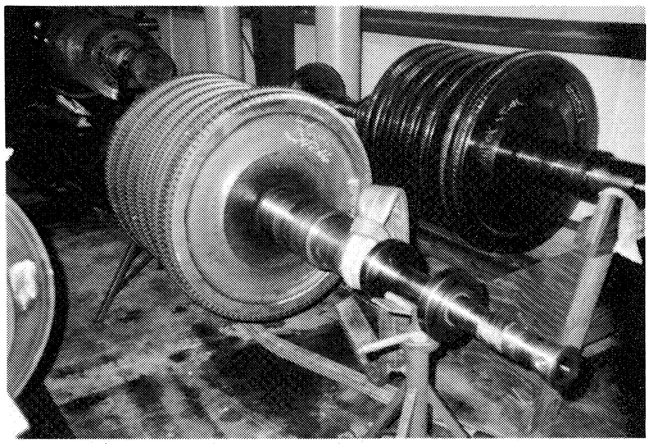

Figure 20. Steam Turbine Rotor (2300 lb) in Local Repair Shop.

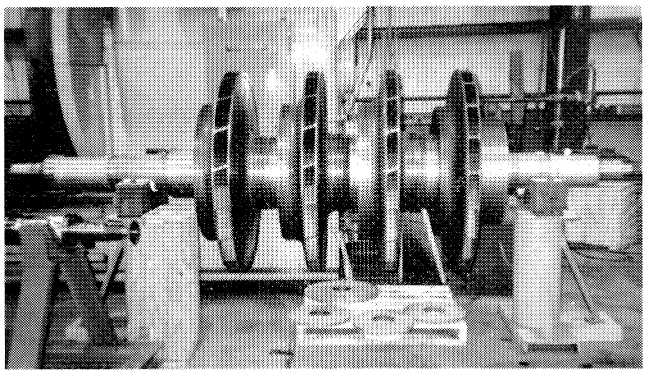

Figure 21. Air Compressor Rotor (11,000 lb) in Repair Shop.

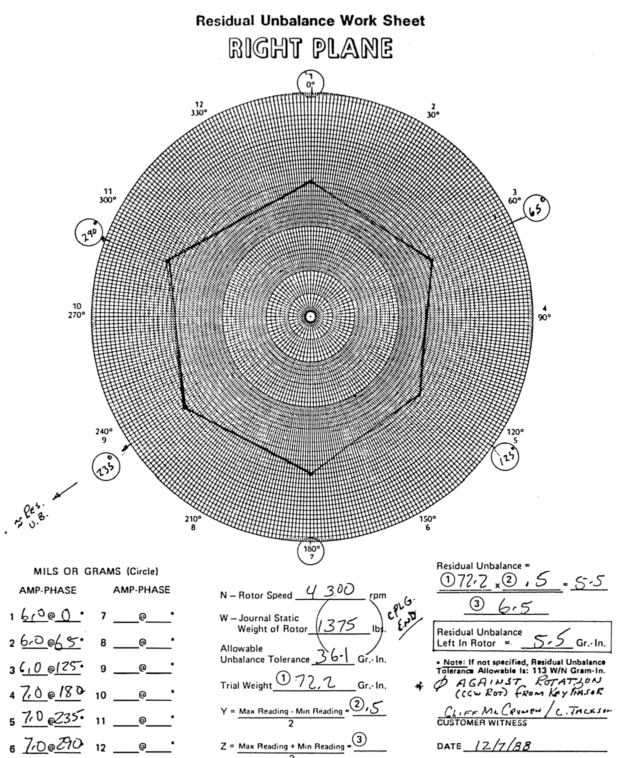

Figure 22. Six Point Residual Unbalance Data 4W/N Balance.

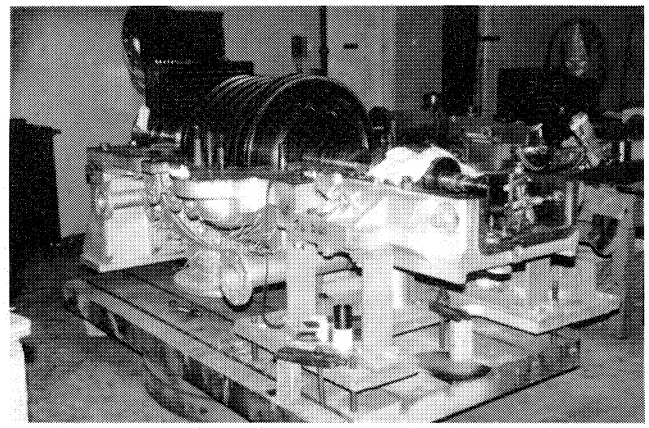

Figure 23. Steam Turbine Rotor (10,000 HP) Being Fitted in Shop.

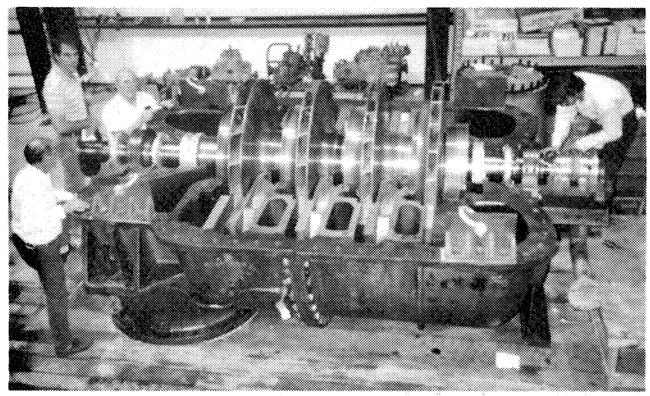

Figure 24. Four Stage Compressor Rotor Being Fitted in Shop.

- Open turbine and compressor casings in the field (mounted on 20 ft concrete monolithic mezzanine) for visual inspection of "rust-ban" type coating followed by nitrogen blanket for the storage period of 13 years. Transport casings to "same" repair shop for inspection, remachining the connecting flanges, installing new "back welded" connections *from* the casings.
- Prepare lift (slings, attachments, shackles, and angles) diagrams for safe "rigging" to be approved by the owner's Superintendent of Maintenance and Rigging Consultant. Heaviest lift was at 81 tons (Figure 25).
- Prepare inspections and repair reports with the repair shop on all rework. Inlet valves to steam turbine along with the governor linkages and servo piston were known to need rework and replacement. The hydraulic servo was to become one piece machined hot rolled bar and to provide two piston rings *rather* than one.
- All control oil and lube oil piping to be replaced with stainless steel.
- All proximity probes to be upgraded to -24 vdc systems maintaining the reverse mounted bodies in new holders and condulets. The monitors to be upgraded to the latest liquid crystal displays using the previous "alarm" and "shutdown" values, i.e., x-y radial alarm 2.4 mils p/p and shutdown (manual committed) at 4.5 mils p/p. Thrust or axial position (using the redundant Figure 14 API 670 R/2 mountings) was to be at 15 mils from "bump" or "commissioning position" (both directions) for alarm and "*AUTOMATIC*" *Shutdown* at 25 mils from the same reference, using three second delays. The active thrust position will be against the active thrust bearing, i.e., toward the exhaust on

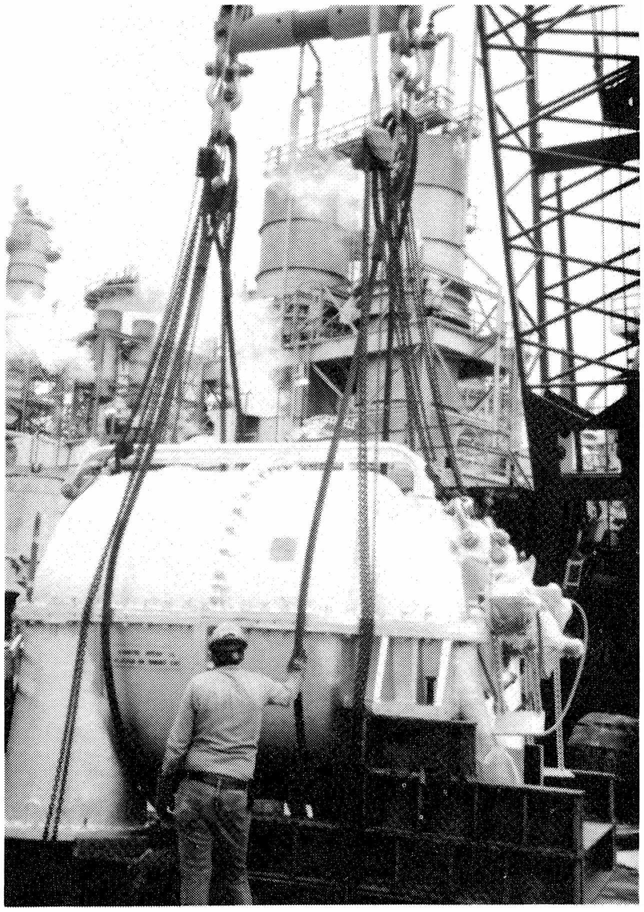

Figure 25. Compressor at 81 Tons Being Lifted from Trucks.

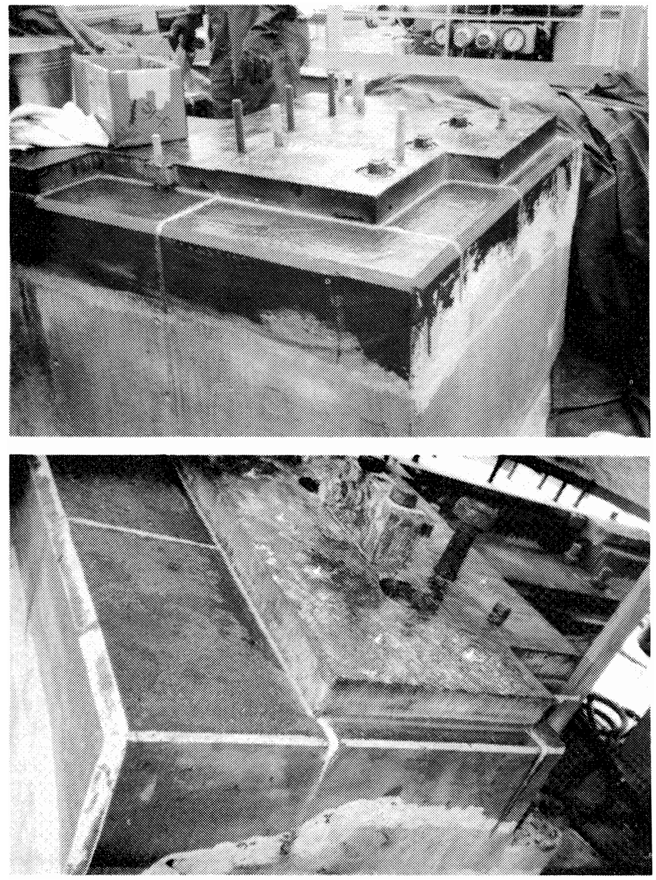

Figure 26. Regrouted Sole Plates in Field-Turbine and Compressor.

the turbine and towards the suction on the compressor. The axial position probe voltage will be set to correspond to a + 7.00 mils (normal) monitor reading. The voltage would approximate the calibration curve for that position.

• Upgrade all metal temperature dual tip sensors to a new liquid crystal panel with six points per train (three trains) and all spare backup thrust temperatures on the fourth panel to keep operator logic simple. Bearing metal temperature alarms at 200°F and shutdown at 225°F (emoving temperature recorders/alarms instruments-space).

• Remove the pressure dam steam turbine bearings and install new four tiltpad radial bearings, LBP, in the steam turbine (earlier conversion by Malcolm Leader for the two operating units). Install the newer design self leveling thrust bearings also per past conversions.

• Replace antisurge system with the newest (high response) systems successfully proven on the past two syn gas conversions.

• Regrout the existing soles plates (4.0 in thick) for the compressor without removing or relocating the sole plate anchor bolts. Chip out regrout, recoat plates against rust, and provide expansion joints in the grout (rounded corners on all grouted plates) and pressure grout for any "voids" after pouring (Figure 26). Replace the anchor bolts and the *large* x-y-z alignment bolts with new ones. Precut and properly filed stainless steel shims for the full compressor feet to be at ¼" thickness (⅛ in + 1/16 in + 1/16 in).

• Drill "fixed end" suction feet anchors pins (1⅛ in diameter) parallel to the "earth" after alignment is achieved. Simplify and "use four body bound bolt-down" rather than welded construction for the 500 lb plate forming the gib key holders for the two to four mils clearance "keel block."

• Remove the turbine pedestal base off its soleplate. Reinstall with two, only, (8 in × 30 in) stainless steel shim plates. Drill tapped holes (vents and supply) under the turbine pedestal plate for pressure grout to be "poured" after final alignment for a no "drum head" final set under the front (steam end) bearing.

• Reinstall new inlet valve positioner plate on the turbines inlet lift camshaft from stainless with Lucite™ cover "view" plate.

• Replace all instrumentation for the turbine and compressor.

• Remove the oil reservoir, exchangers, filters, and pumps for total cleaning and replacement of all instruments. This work was done in a local repair shop in LaMarque, Texas. A running test with response test on all instruments, relays, etc., was performed as on the original API 614 tests (Figure 27). All oil lines in questionable condition were replaced with stainless steel. The main oil pump/standby pump control switching was converted from "hand-off—auto" to "manual-auto" per past good success in reliability. A *second* standby "start" switch on low supply oil pressure was added to the steam turbine driven main oil pump between the pump discharge and its discharge check valve per CJ conversion in 1978, with a promise from the API Subcommittee that it will be incorporated in the next API 614 draft for lube console control.

Note 1. This switch is in parallel with the standby pump start switch, traditionally placed downstream of the accumulator's discharge check valve, i.e., "either" switch can start the standby pump. Logic: on a loss of the main oil pump, the standby pump will come on about four seconds quicker to prevent shutdown of the train to a drop in control oil/lube oil pressure.

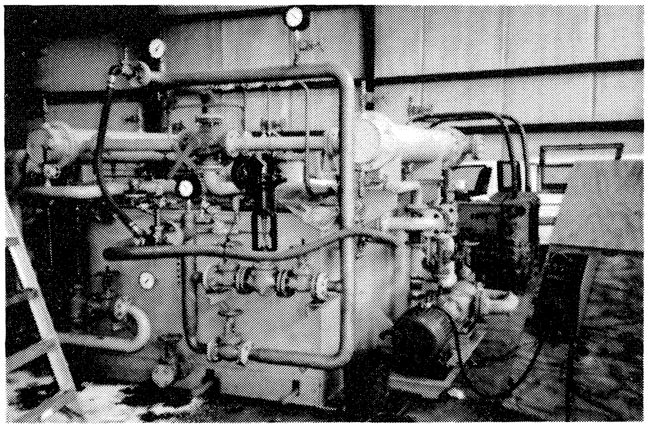

Figure 27. Lube Console on Test in Local Repair Shop.

Note 2. On this original design, the vendor left out the check valve ahead of the accumulator. This accumulator is normally specified by this author at four seconds (rather than the often supplied one second), but this one was reused based on good success for 12 years with the added start switch and check valve.

Note 3. The four second accumulator value is based on several trip-out "tests" where the standby pump *does not come-to-speed "delivering" oil past its check valve* in sufficient time. Further proven by multipoint strip chart recordings of (1) the pressure at the main pump discharge; (2) the pressure at the standby pump discharge; (3) the speed of the main pump; (4) the speed of the standby pump; and (5) the trip point relay initiation, etc. Improper location of the pump discharge relief valves relative to the oil reservoir level, have caused 10 seconds delay in delivering oil forward. *INTERESTING?*

- Prepare startup manuals for each phase of the commissioning.
  - Part 1: Acidizing the new lube oil piping.
  - Part 2: Flushing lube oil lines to/from turbine/compressor: a) stainless screens @ 100 mesh, location, acceptance; b) sequence of systems to be exercised; c) line connections and precharge of L.O. accumulator.
  - Part 3: Commissioning of the lube oil system: 1) startup of the lube oil system with standby pump; b) check out pump discharge RVs and instrumentation; c) soft start main steam turbine oil pump system; d) check out of main oil pump RVs and instrumentation; e) check oil control PCVs single and dual oil pumps; f) hand tryout of main oil pump; g) restart of main pump. Proper standby setup.
  - Part 4: Blowing the 600 psig steam header: a) steam flow at targets. Duration/number of blasts.
  - Part 5: Inspection of inlet air filters and compressor inlet.
  - Part 6: Instrument checkout, warmup, and overspeed tests.
  - Part 7: Coupling fit up, prestretch, alignment.
  - Part 8: Piping stress check, dial indicators in place.
  - Part 9: Startup and run to minimum speed and load-venting.
  - Part 10: Incipient surge at minimum, medium, and full speed.
  - Part 11: Attempt to surge with controllers active.
  - Part 12: Tripping out main oil pump. Trip out main turbine.
  - Part 13: Shut down, slow roll and secure requirements.
  - Data taking during commissioning. Data reduction. This train operated at load for about eight hours, for various tests and valve checkouts. The vibration level on the train was less than ½ p/p (Figure 28). These trains had passed the API 612, 614, and 617 tests at the vendor's shop 13 years before. The spare rotors had also been tested and each rotor had been mechanically fitted in each casing. The coupling was purchased at 18 in spacer length to API 671, hydraulic dilated to 1.75 mils interference/inch of diameter. *NOTE:* Hydraulic dilation for coupling fitting is through the coupling hub. This is the only unit bought this way out of many trains, and will be the only one in the future; however, with special care, this has caused only minor problems.

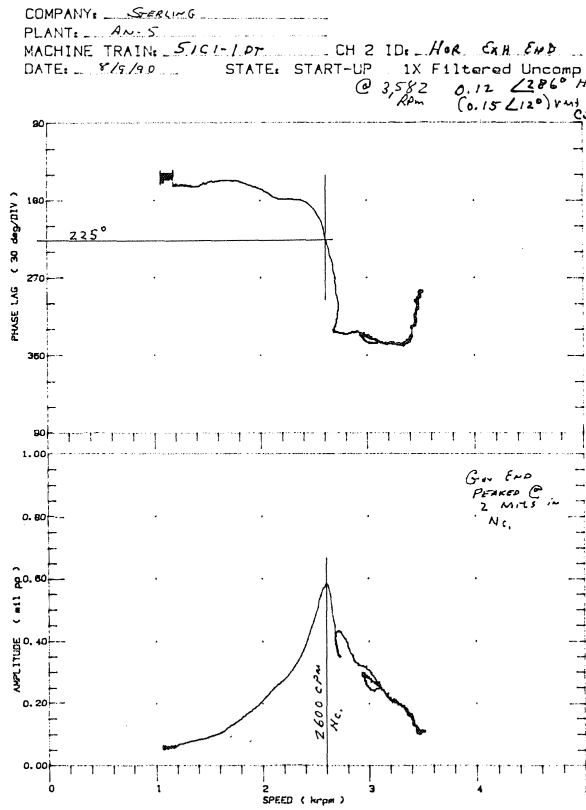

Figure 28. Bodé Plot During Run up at Commissioning from the Exhaust (Coupling) End. Please note that full scale is only ½ mil p/p. The governor end peaked (2 mils) critical.

- Two sets of special coupling bolts, match-weighed, were bought for this job. The first set was installed, with proper torque after all the initial alignment and first commissioning runs were completed.

CONCLUSION

This project was finished on schedule and precommissioned in August 1989. It was commissioned to its reactor in October 1989. The shaft relative vibration data (0.2-0.4 mils p/p) was taken from the "patch boards" behind the control room and recorded on FM tape recorders. Process data flows, pressures, steam flow were recorded. The bearing metal temperatures were below 160°F and recorded along with other mechanical data.

The compressor critical was 1,900 cpm and well damped with A.F. approximately 2.0 Figure 29. The steam turbine critical speed was about 2600 cpm with a higher amplification factor, e.g., A.F. = 12. The compressor was operated at all speed ranges, 3,700 rpm to 4,430 rpm.

The keel blocks were placed on both turbine and compressor. The compressor was fixed at its suction feet with 1⅛ in diameter 400SS solid dowels in the shim pack joint parallel to the "earth."

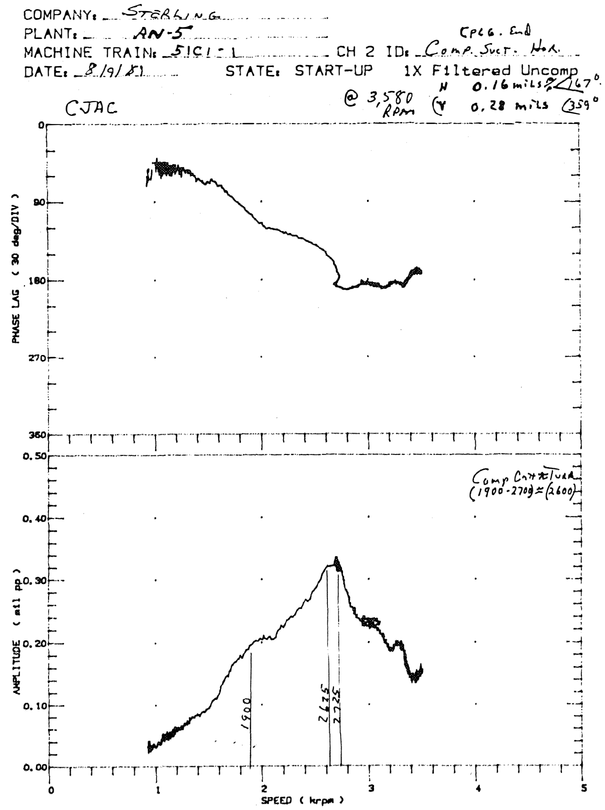

Figure 29. *Bodé Plot of the Centrifugal Compressor Coupling End, Horizontal Sensor, also on a ½ mil Full Scale. Split criticals are at 1900 and 2700 CPM. These compressors have been plagued with 2× electrical runout, since supplied, which exceeds the shaft absolute vibration.*

The final alignment was based on a 23 mils rise on the compressor discharge end wobble plate, a "zero" rise at the compressor suction (ambient air inlet), an 11 mil rise on the turbine exhaust end pedestal, and a four mils rise on the governor end wobble plate center (Figure 30 Alignment Map). the cold alignment was by *reverse indicator* measure using a two inch diameter aluminum bar with a sag of one mil in 22 in bar span (Figures 31 and 32) are shown:

0 0

−21 C −24 +23 T +18

[−38] −44 +43 [+42] [Design]

(Design based on water stands/eddy current probes for 18 hours to reach equilibrium temperatures (Figure 33).

Peelable (full pack @ 125 mils) stainless shims were used to set the single membrane dry coupling at zero prestretch. About ⅛ in of single cut shims had to be placed in the inactive stored position, because the vendors data on long bolts *vs* short bolts was incorrect. The coupling guard was perforated and a good three inches from the coupling OD which was 16 in.

The thrust of this discussion is directed to those who might be faced with a similar situation and offered as a guide to things needing attention or at least consideration. The original commissioning of two of these three trains was in an ASME paper (78-PET-48) written in 1978 (1977 commissioning). However, this coverage is following a theme of retrofit, upgrade, and demothballing.

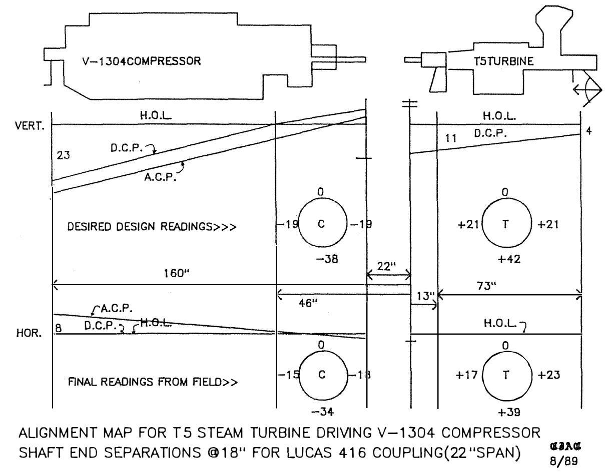

Figure 30. Graphical Plotted Alignment Map—Turbine Compressor.

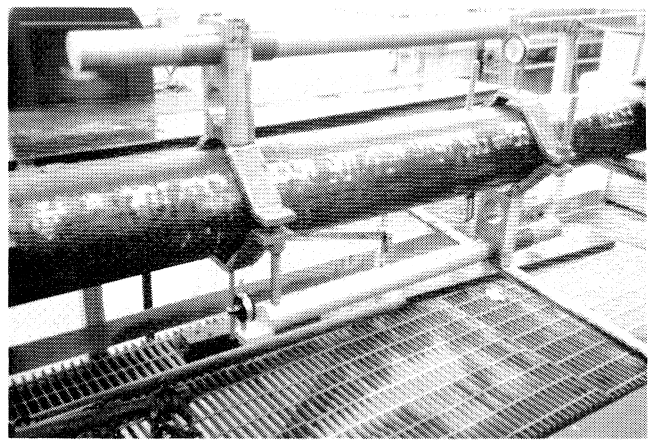

Figure 31. Sag Check on Alignment Reverse Indicator Bars.

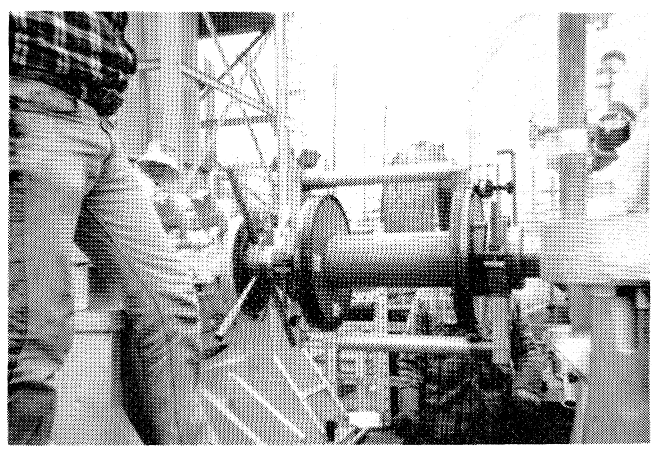

Figure 32. Reverse Indicator Bars in Place over Coupling.

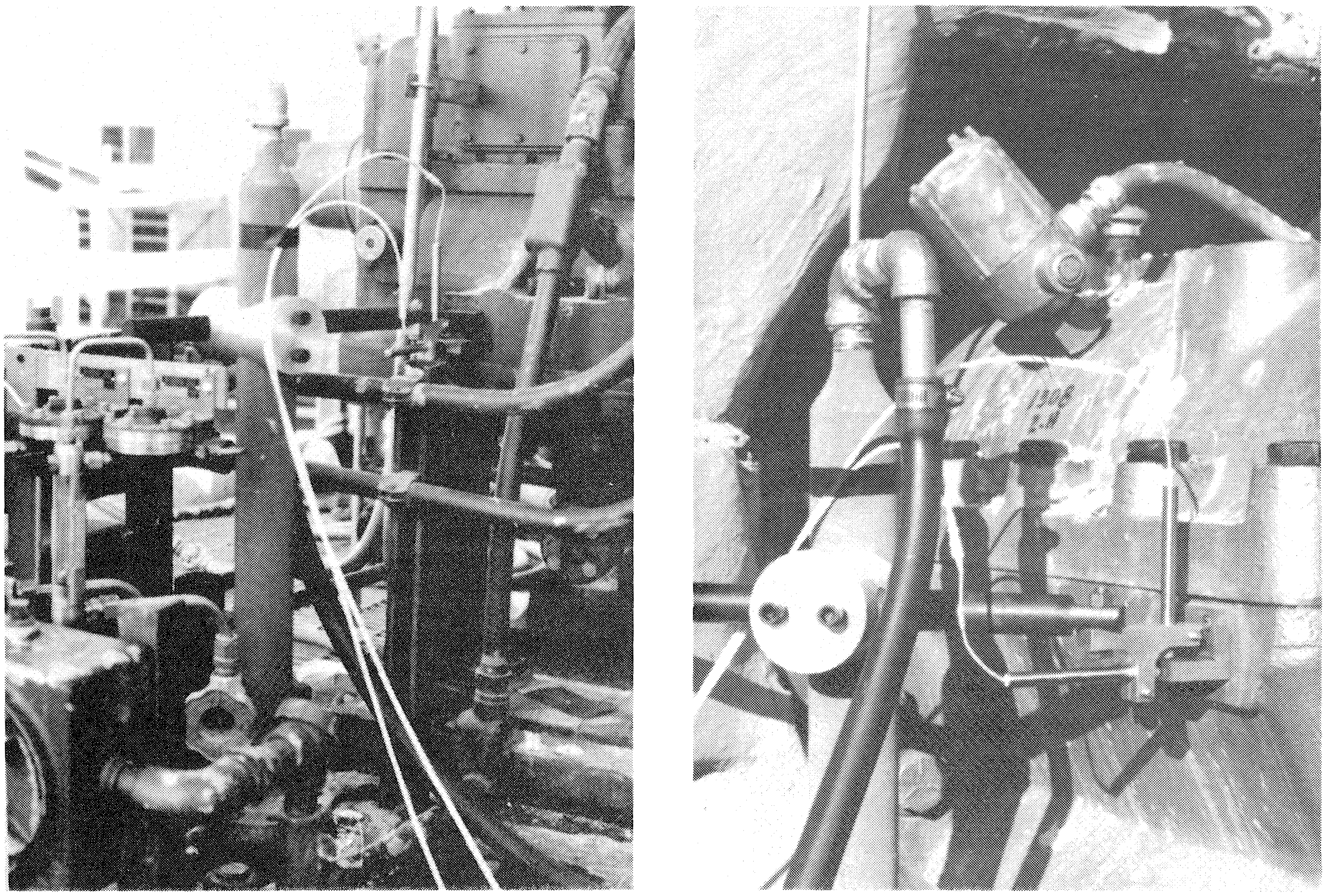

Figure 33. Thermal Rise Data Taken in 1977 Using Jackson Stands.

SUMMARY OF BOTH PROJECTS

Successful projects are a result of a lot of special work on specific details. Rotors have to be fitted correctly and balanced correctly. Bearings should be correct and fitted correctly. Attention to detail yields straight lines on the vibration data and excellent performance elsewhere. People have rejected high speed equipment as long as I can recall, but I seem to have more problems with the slow, heavy, lumbering along equipment. Did here!

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