DESIGN, TESTING AND COMMISSIONING
OF A SYNCHRONOUS MOTOR-GEAR-AXIAL COMPRESSOR
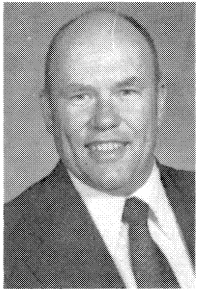
by

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He earned Bachelor of Science and Master of Science degrees in 1977 and 1978 from the University of Virginia, where he worked extensively on theoretical and experimental rotor dynamics. He is an associate member of ASME and a member of Sigma Xi.

ABSTRACT

This paper presents the design features involved in providing two air compressor trains at high efficiency. Testing and commissioning data are also presented, including specific electrical requirements; foundation and assembly features; grade maintained air filter designs at 0.5 in. water drop; torsional features *not* using soft elastomeric couplings; bearing designs for motor, gear and compressors; and test and commissioning data. The test and commissioning data include torsional data, alignment heat rise data, torsional-lateral interaction on run-up, quadrant inlet vane surge tests with flow and vibration responses, eccentricity plots of journals in bearing, transient responses through criticals, and correlations of predicted vs. actual rotor dynamics responses.

INTRODUCTION

This paper is presented because it represents a type of compressor train which is becoming more prevalent but which has been rather lightly covered: axial compressors, specifically those with synchronous motor-gear drivers.

A 5500 hp axial compressor was chosen to improve efficiency at higher flows (<60,000 icfm) and lower head (20 psi, or 1.4 bar) than those of centrifugal compressors. The motor was required to be synchronous, since the plant, a textile plant, would be harmfully affected should the electrical load balance be upset.

The electrical survey put forth several constraints, which eliminated about 70% of the machine vendors:

1. Current in-rush at 500%.

2. Bus voltage drop limited to 10%.

3. Terminal bus voltage drop limited to 20%.

4. Inertial loads limited to 50,000 lb-ft^2.

5. Induction motors not allowed.

6. Six pole synchronous motor specifically required.

The axial compressor was efficiency effective, and variable inlet guide vanes could reduce air flows from 100% to 50% of those designed. A gear increaser was needed for speed matching the axial compressor to 1200 rpm motor speed. Based on these project, process, and electrical requirements, the following mechanical design premises were made:

1. API specifications for compressor (617), gears (613), lube consoles (614), and couplings (671) to be applied to machinery designs and testing.

2. API 670 and 678 to be used for vibration monitoring.

3. Motor to have separate bearing pedestals in order to control rotor stiffness (radial and axial), bearing damping, and stator air gap clearances.

4. Sole plate mounting with epoxy grout and three-piece stainless steel (SS) shims at ¼ in. thickness (⅛" + ¹⁄₁₆" + ¹⁄₁₆") to be used. Gear to use one single spacer, SS shim at ¼ in. thickness. Sole plates to be shimmed at chock blocks for alignment prior to grouting.

5. Down connected piping to be used on the compressor.

6. Inlet air filters to be designed for 0.5 in. H$_2$0 drop of filter and dampers (actually 0.6 in. H$_2$0 obtained). Filter elements to use pre- and final cube cell elements replaceable as a unit (to minimize labor). Filters to be held by non-corrosive frames, holddowns,

etc., and to be free from loose hardware such as nuts, screws, and washers. Filter inlet and outlet from intake rain shield to compressor inlet flange to be no less than white blast inorganic zinc coated (hot dip galvanized allowed) inside and outside, including piping and filter damper housings.

Filters to be at grade elevation and capable of element changeout while in operation, if necessary. Filter DP alarms with direct element pressure indicators and draft gauge to alarm at 2 in. H_2O differential. Intake pipe (free standing) to rise fifty feet above grade with full coned rain hood (Chinese hat) and bird guard. A perforated plate trash preventer to be at grade prior to coarse filter elements. A vaned elbow to be considered prior to calibrated inlet flow nozzle to reduce compressor mezzanine height without risk to flow drops and stratified or turbulent flow.

7. A flexible element to prevent inlet pipe strain to be attached to the compressor inlet flange "Dutchman" (spacer), and support inlet pipe to allow flange bolt access without restraint by concrete mezzanine.

8. Reinforced concrete mezzanine to be monolithic pour.

9. Sole plates to be white blasted and coated with epoxy primer to be removed on bonding surfaces just prior to leveling and grouting.

10. Dry element metallic coupling projected in design based on several years of complaints by soft elastomeric coupling users. Torsional design to be completed and reviewed before motor, compressor gear rotors, material and heat treat, shaft ends, and coupling(s) to be released for manufacture (five months allowed for this single design phase). Complaints against elastomeric torsional damping couplings:
    a. High cost.
    b. High weight rotor overhang moment.
    c. Difficult to balance.
    d. Deterioration of elastomeric stiffness and damping—hardening of elements with time and service.
    e. Often more torsionals passed in transient starting, due to softer stiffnesses.

11. No keyways to be incorporated, due to expected torsional stresses. An early contract feature called for integral shaft-to-coupling flanges motor-to-gear and hydraulic dilation at gear-to-compressor coupling shaft ends.

12. Accelerometers to be used integrated for velocity on motor and L.S. gear bearing horizontal position only. Proximity probes in x-y configurations to be used on H.S. gear shaft and compressor. Dual element redundant axial probes to be used on gear and compressor.

13. Probe type TC, dual element, to be used on motor and gear at load point location (minimum film). Motor bearings expected as plain sleeve hydrodynamic bearings. Gear bearings expected as pressure dam bearings. Compressor bearing to be a stabilized bearing for review. Embedded dual element iron-constantan TC's to not contact babbitt, but to remain 30 mils (0.030 in., 0.662 mm) minimum separation.

14. Surge controls by compressor vendor expected to be high response analog (no digital) instrumentation with hydraulic (no pneumatic) power stroke cylinders.

15. Self leveling thrust bearing (articulating level links) to be used on compressor.

16. Balance chamber to be measured by static pressure gauge.

17. A minimum of *two* parallel vent valves (anti-surge) to be used to result in 1 second response. Design by Monsanto/Contractor. Check valve design to provide minimum fouling features and to be located as close as physically possible to compressor discharge flange while providing vent valve control blowoff between check valve and compressor discharge flange.

18. Oil system to have stainless steel reservoir and filter-to-bearing piping, with remaining piping to be bid both ways. Oil filters to be pleated paper at 10 micron with downstream guard filter of stainless steel mesh at 50 micron and overall pressure drop <5 psid at inlet flow temperatures and new oil cleanliness. Transfer valve to be bronze fitted cast steel bodies. A bladder type accumulator sized for 4 seconds minimum to be used regardless of design transient times, based on measurements taken on many lube consoles over the past thirteen years. Pump discharge relief valve relieving-to-reservoir to be located close to pump (not at reservoir). Modulating relief valves to be used, as opposed to pop blowdown reset-type safety valves. Stainless steel, multi-convolution, flexible connectors to be used in addition to "y" strainers on pump suction only.

19. Spare rotor packages—rotor, seals, bearings, labyrinths—to be purchased for compressor and gear. Spare couplings (one each) to be purchased. Spare compressor rotor to be stored vertically in an inert gas container doubling as a shipping container.

20. Vibration, axial position, bearing and stator temperature monitor systems for the compressor, gear and motor to be installed and protected per API 670, 678 and Monsanto standards. All sensor conditioning to be contained within free standing junction boxes supported to the mezzanine and not attached to machinery (contractor mounted).

21. Compressor trains to be within ventilated roof, building with part wall against driving rain, with bridge crane for heaviest component and drop bay area.

22. Plugged taps at 1 in. NPT to be provided in gearbox, drilled transverse to gear and pinion shafting to allow for modulating type probes to be installed for torsional resonance measurements during commissioning. Dynamometer load to be applied to gear during shop testing.

23. Dual element embedded TC/RTD's to be placed in each motor stator phase. Motor rotor and bearing details to be provided along with compressor and gear for Monsanto design audit.

24. Monsanto alignment "L" or "block" targets to be provided on the four corners of each casing, plus 6 in. × 6 in. square free column space provided by oversized sole plates to accommodate Monsanto 2 in. pipe water column alignment brackets (Jackson stands).

25. Compressor builder to have total train responsibility. Dynamic designs including lateral and torsional (transient and steady state) to be reviewed at compressor location prior to release of final engineering details to manufacturing.

26. Both performance and mechanical tests to be witnessed by shop tests. Energy penalties in bid comparison set at $1400/kW.

RESULTS AFTER VENDOR SELECTION AND ENGINEERING REVIEWS, AND PRIOR TO SHOP TESTING

The arrangement for the synchronous motor gear-driven axial compressor with protection monitoring is shown in Figure 1. The following items should be noted:

1. a. A lemon bore bearing design, along with exceptions to API vibration limits for a 1.8 multiplier, was rejected; i.e.,

$$1.8 \times \left[\sqrt{\frac{12000}{\text{rpm}}} \times 0.25 \sqrt{\frac{12000}{\text{rpm}}} \text{ (runout)} \right]$$

= vibration limit (mils peak-to-peak, unfiltered)

b. A five shoe load on pad (0 to 0.25 preload radial bearing) was accepted as an alternate bearing meeting API 617 vibration limits and electrical runout requirements.
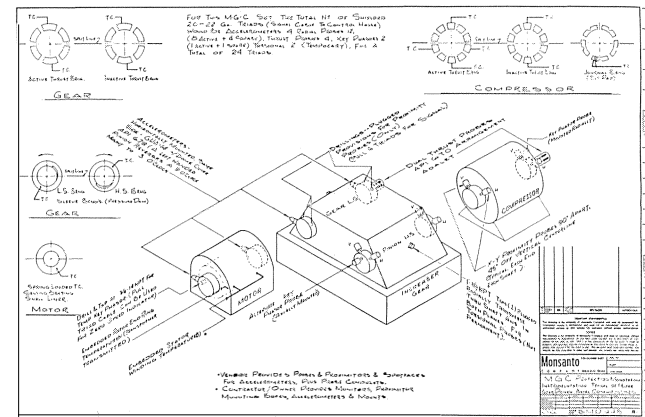
Figure 1. Arrangement for Synchronous Motor-Gear Driven Axial Compressor with Protection Monitoring.

2. The vendor bid a direct lube, thrust temperature instrumented bearing and rejected the need for self leveling links. Monsanto paid the additional costs for self leveling link, directed lube thrust bearings with six metal temperature elements distributed to provide four in active and two in inactive shoes.

3. The gear bearings were plain on low speed and pressure dam in high speed. Upmesh pinion was agreed upon, since the low speed shaft would be lifted at less than design torque. The pressure dam was located in the lower half of the pinion bearing. Pinion and compressor rotation was clockwise, as viewed from the drive motor.

4. A 22 in. diameter, integral flange, stainless steel, multi-convolution coupling was incorporated between motor and gear. A 12 in. hydraulic dilation coupling was incorporated between gear and com-

pressor. Motor and gear rotor materials were of forged electric furnace grade E4340 vacuum remelt quality.

5. The electrical group requested or accepted a motor rotor temperature patented instrument for transformer transmission from gun drilled leads through the shaft to the exciter area.

6. Provisions for oil spraying were incorporated on the compressor coupling housing region, since the coupling is within the bearing housing and close (¾ in.) to the bearing housing bore.

7. Our dynamic analysis for actual criticals (both lateral and torsionals) agreed very closely with that of the compressor manufacturer. Our interference diagram, including coupling axial resonances, is presented in Figure 2.
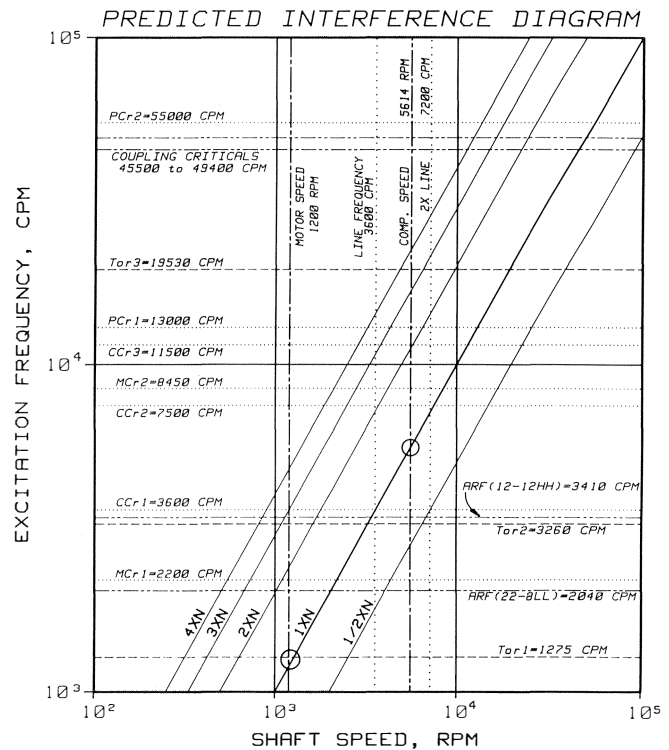

Figure 2. Interference Diagrams of Lateral, Torsional and Coupling Axial Resonances (Criticals).

8. Two torsionals were to be excited on each startup, since a synchronous motor excites all frequencies from 120 Hz (7200 cpm) to zero. The placement of the first torsional, which is excited last, was near peak torque, 82% of full speed. This was felt to be at a point where the alternating torque to steady load torque ratio is low, which is good. The gear teeth were selected at 3 DP, which is quite large and is based on cycle fatigue life. A tooth contact ratio greater than two was obtained (Figure 3).

9. Terminal reduced voltage was to be 24% in starting. Start times were predicted to be 20-24 seconds by most curves, except for one at 15 seconds, which was correct.
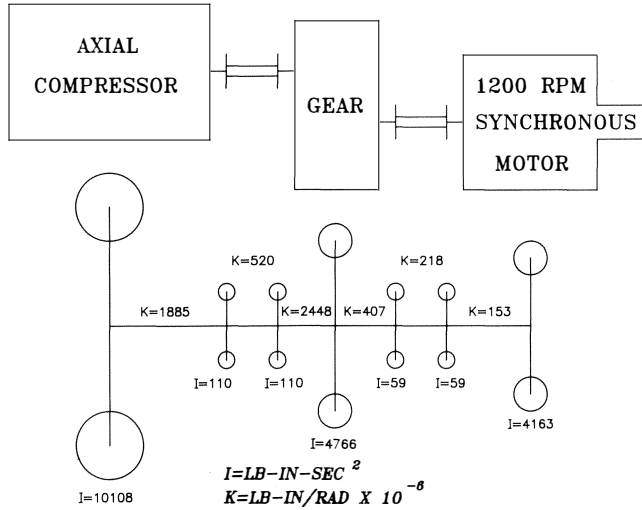
Figure 3. Torsional Model for Steady State Analysis—Transient Analysis Used Three Mass Model.

10. Motor bearings were 7 in., with a 7500 lb rotor. L.S. gear bearings were 9¼ in., with a 5960 lb rotor. H.S. gear bearings were 6¼ in., with a 555 lb rotor. Compressor bearings were 5½ in., with a 4030 lb rotor.

SHOP TESTING—
COMPRESSORS AND GEAR

1. The expected split critical (camel hump) from asymmetric stiffness, as seen on the stiffness map, was observed. A horizontal first rigid translational mode peaked at 2800 rpm. A vertical first rigid translational mode peaked at 3600 rpm (Figures 4a-4b, 5a-5g, and 6).

2. The electrical runout was within 0.375 mils and was very low and synchronous, as can be seen on Figure 7a, but there was a 2 × electrical runout on the suction end of the contract rotor installed in the north compressor. Note the effect on the run-up Bode plot later in commissioning. Electrical runout discrepancy can be seen on Figure 7b. A check used on shop testing at
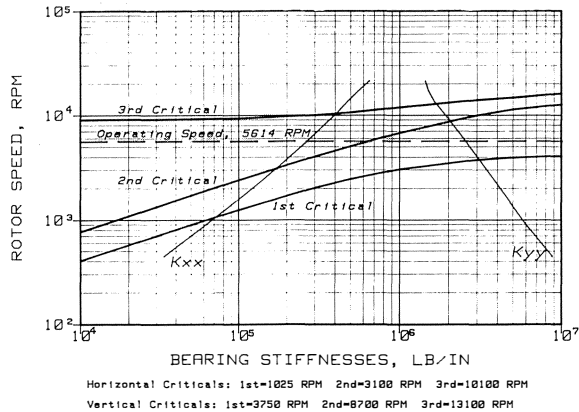

Figure 4a. Undamped Critical Speed Map for Axial Compressor—Two First Mode Criticals Expected; i.e. Horizontal and Vertical Due to Asymmetric Stiffnesses.

DAMPED (IN RPM) *UNDAMPED*	MINIMUM PRELOAD $\overline{m} = 0.10$	DESIGN PRELOAD $\overline{m} = 0.25$	MAXIMUM PRELOAD $\overline{m} = 0.35$
MAXIMUM CLEARANCE c = 0.011"	3667 *3659*	3694 *3632*	3681 *3635*
DESIGN CLEARANCE c = 0.010"	3735 *3727*	**3776** *3701*	3762 *3703*
MINIMUM CLEARANCE c = 0.007"	3968 *3958*	4088 *3936*	4051 *3935*

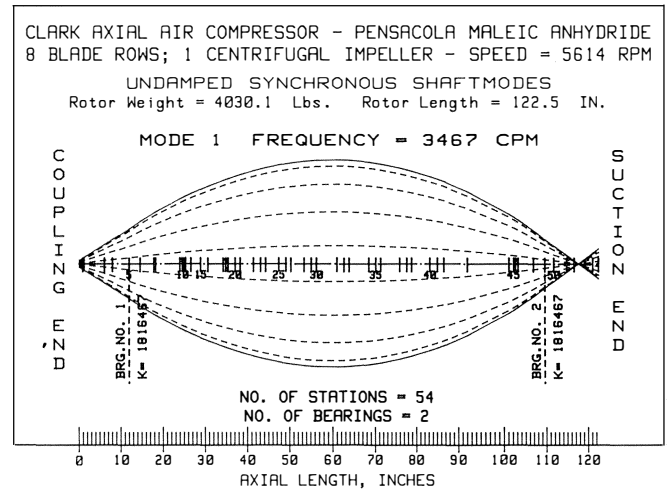Figure 4b. Matrix of First Critical Based on Variable Clearance and Preload.

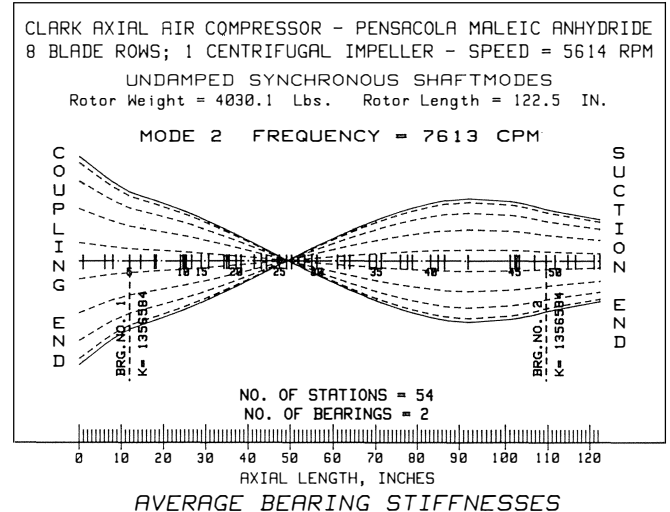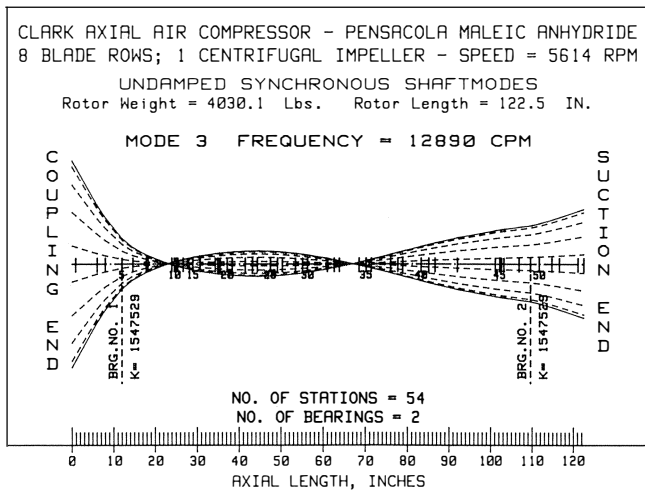Figure 5a. Compressor Undamped Shaft Mode, Translational (Vertical) First Mode at 3467 cpm.

Figure 5b. Compressor Undamped Shaft Mode, Pivotal Second Mode at 7613 cpm.

CLARK AXIAL AIR COMPRESSOR - PENSACOLA MALEIC ANHYDRIDE
8 BLADE ROWS; 1 CENTRIFUGAL IMPELLER - SPEED = 5614 RPM
UNDAMPED SYNCHRONOUS SHAFTMODES
Rotor Weight = 4030.1 Lbs.   Rotor Length = 122.5 IN.

MODE 3   FREQUENCY = 12890 CPM

NO. OF STATIONS = 54
NO. OF BEARINGS = 2

AVERAGE BEARING STIFFNESSES

Figure 5c. Compressor Undamped Shaft Mode, Bending, Third Mode at 12,890 cpm.

CLARK AXIAL AIR COMPRESSOR - PENSACOLA MALEIC ANHYDRIDE
8 BLADE ROWS; 1 CENTRIFUGAL IMPELLER - SPEED = 5614 RPM
UNDAMPED SYNCHRONOUS SHAFTMODES
Rotor Weight = 4030.1 Lbs.   Rotor Length = 122.5 IN.

NO. OF STATIONS = 54
NO. OF BEARINGS = 2
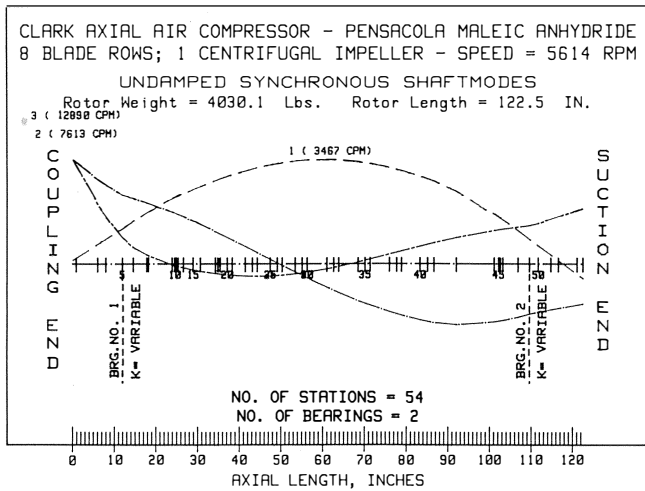
AVERAGE BEARING STIFFNESSES

Figure 5d. Summary Of First, Second, and Third Modes; Stiffnesses Used Are Average Stiffnesses.

"slow roll" and "at speed" is to superimpose the "blanked" filtered signal from two probes over the "blanked" raw, unfiltered trace of the same two probes. A four channel scope is needed, plus an accurate synchronous tracking filter. The blanking and triggering utilizes the AC coupled "Z" axis scope feature for blanking and triggering of the scope and a one-pulse-per-revolution event (phazor probe). The vectorial comparison during one cycle/revolution can then be compared both before and after resonance phase shifts.

As a rather shocking truth from this procedure, one can see that the raw, unfiltered signal is significantly less than the synchronous 1× component. Several rounds of drinks can be won by proving that this can actually happen. Further, it supports the need for minimum electrical runout and runout compensation. The actual run-up of this rotor will plot a negative dip at the critical rather than a peak without a compensated plot.

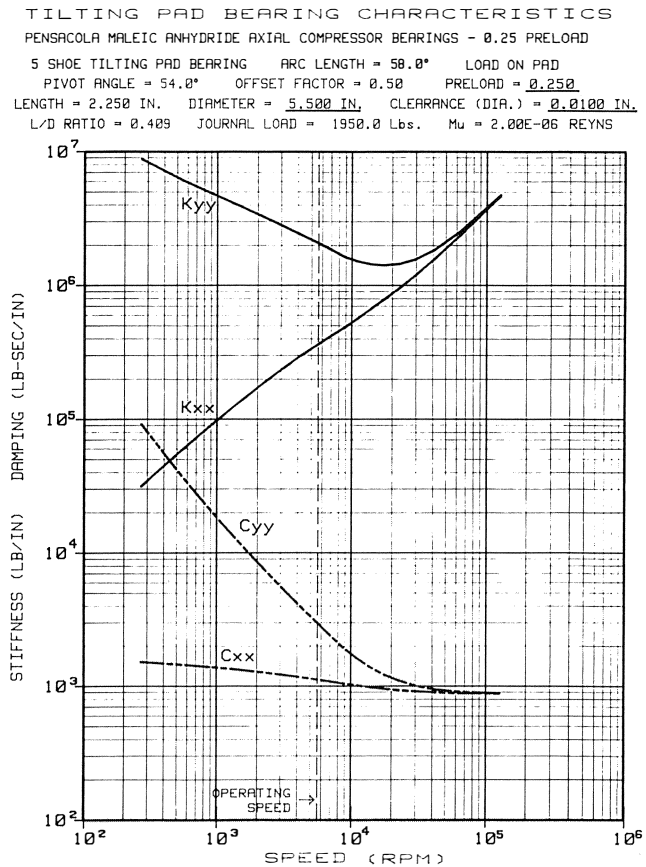3. The bearing metal temperatures ran 200°F on shop test

Figure 5e. Stiffness and Damping Factors for the Compressor Rotor Support.
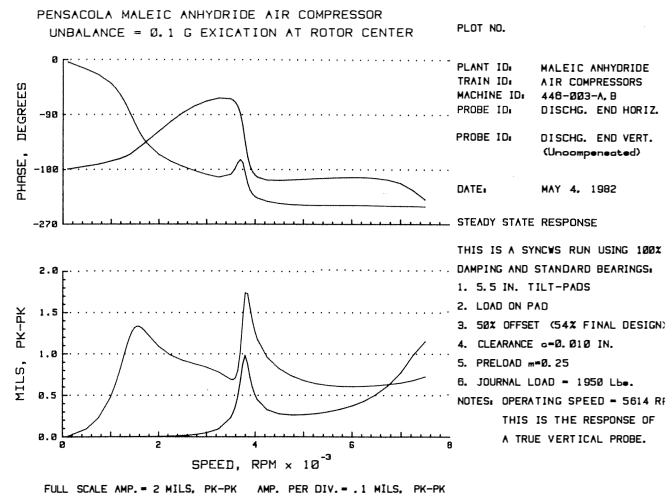

Figure 5f. Unbalance Response Compressor Bode Plot with 0.1 G at Rotor Center—Discharge End Horizontal.

on the free end of the compressor, as opposed to the coupling end of the compressor. After confirming lube hole sizes and measuring oil flow, it was decided to grind 3/16 in. off the trailing ends of the radial bearing pads; i.e., to shift the offset factor from 0.5 to 0.53. This aided the oil in getting across the pad and dropped the metal temperature by 20-30°F. Reverse rotation of these compressors cannot happen on these two trains, as they are not valved or piped to allow even operator error.

PENSACOLA MALEIC ANHYDRIDE AIR COMPRESSOR
UNBALANCE = 0.1 G EXICATION AT ROTOR CENTER
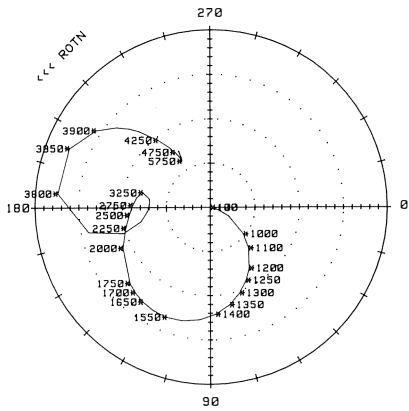
PLOT NO.

PLANT ID: MALEIC ANHYDRIDE
TRAIN ID: AIR COMPRESSORS
MACHINE ID: 446-003-A,B
PROBE ID: DISCHG. END HORIZ.

DATE: MAY 4, 1982

STEADY STATE

THIS IS A SYNCWS RUN USING 100%
DAMPING AND STANDARD BEARINGS:

1. 5.5 INCH DIAMETER TILT-PADS
2. LOAD ON PAD
3. 50% OFFSET (54% FINAL DESIGN)
4. CLEARANCE c=0.010 IN.
5. PRELOAD m=0.25
6. JOURNAL LOAD W=1950 Lbs.

Note: OPERATING SPEED = 5614 RPM
THIS PLOT OF TRUE HORIZONTAL
PROBE.

FULL SCALE AMP.= 2 MILS. PK-PK   AMP. PER DIV. = .1 MILS. PK-PK

Figure 5g. Unbalance Response Compressor, Polar Plot for Figure 5f.

TEST #1   2-8-82
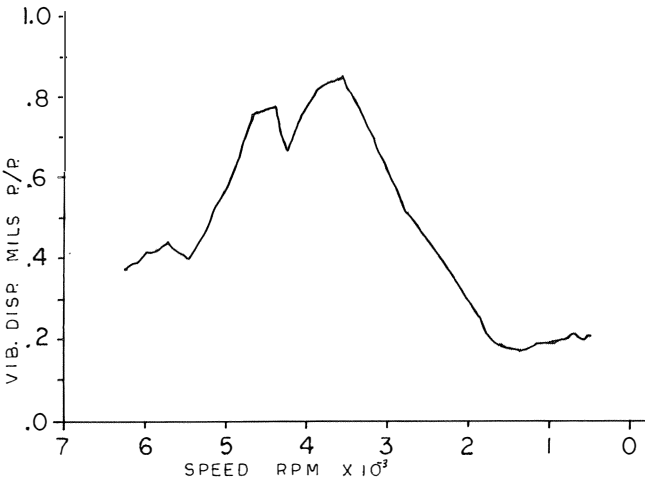CT 211648 (8) AGR 6/8 FLUOR/MONSANT
FREE END HORIZ. DECL. PLOT

Figure 6. Shop Test Bode Plot of Compressor Showing First Mode Split Criticals—Horizontal and Vertical.

4. The gear was tested at full speed, with a step down to the dynamometer to allow 375 hp at full speed, 5614 rpm. The pinion did not lift to the top half, as was predicted by the gear manufacturer. It did go precisely where our pressure dam bearing analysis predicted it would go: 30° below the horizontal split line (Figure 8). The lift of the pinion, as created by the pressure dam (in lower half) during the shop run-up, can be seen using a storage scope and laying out the bearing clearance on the scope plot. Probes are 90° apart, separated at 1 o'clock (vertical) and 4 o'clock (horizontal).

5. The motor test was unloaded and not a responsibility of the mechanical group. Current, temperature and high potential tests were made.

SPECIAL TOOLING PROVIDED

1. Plug and ring gauges for ½ in/ft taper for high speed coupling were provided. A premium was paid to the gear manufacturer to convert his ¾ in/ft taper to ½ in/ft
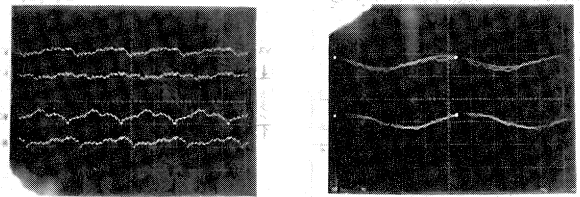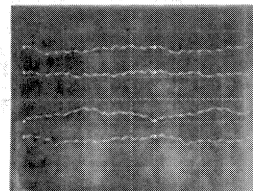

Figure 7a. Slow Roll and "At Speed" Shop Data Taken on Rotor 106 Overlaying Synchronous Plot (1×) over Raw Plot (with Electrical Runout, Also 1×) to Note Vectorial Agreement Per Rev.
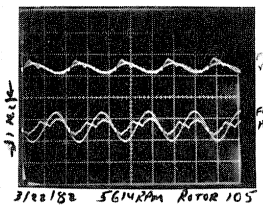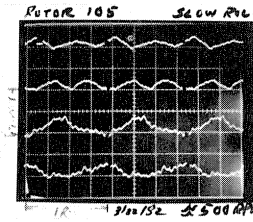

Figure 7b. Same as Figure 7a but on Rotor 105, Which Has an Electrical Runout Anomaly at 2× and When Added to the Synchronous 1×, Signal Disagrees Vectorially Though Within Accepted Limits. Note that the Raw Signal Is Less than the Filtered Synchronous (1×).

DRIVER-END VIEW OF PINION PRESSURE DAM BEARING
NONSTANDARD INVERTED DAM DUE TO UPMESH PINION
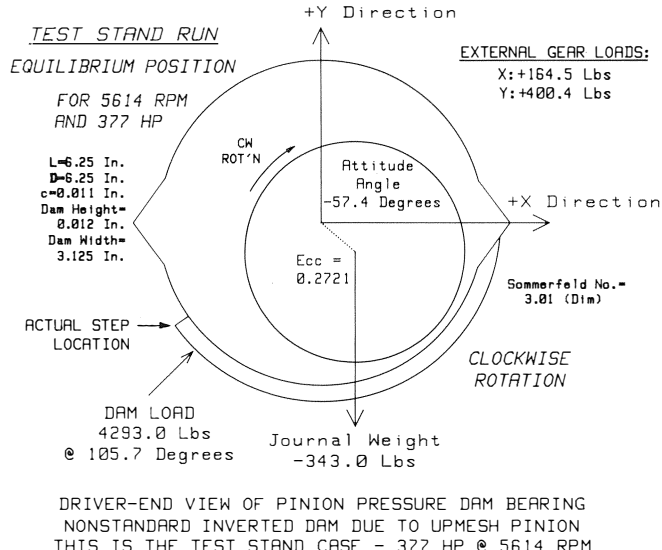THIS IS THE TEST STAND CASE - 377 HP @ 5614 RPM

Figure 8. Pinion Equilibrium Analysis for Shop Test Dynamometer at 5614 rpm and 377 Horsepower. This Is an Upmesh Pinion with Pressure Dam in Lower Half.

for best retention uniformity (both ends) and for prevention of Murphy error.

2. Aluminum 4 in. and 5 in. concentric tube alignment

sets were provided for aligning each shaft end set. Sag was at ⅓ mil, along with 500 mil center zero, 1 mil accuracy, dual scale, revolution counter indicators supported by large shafts.

3. Go-no go, three step plug gauges were transmitted to the operating plant as an option to the 5½ in. diameter mandrel checker for gauging the tilt pad bearing in its retainer. The mandrel checker is preferred, as each pad is gauged.

4. Water stand details were outlined for twelve water pipe stands, in order to gauge twenty-four measuring points on heat rise dynamic alignment vs. 87°F reference to sole plates as a bench mark. Eight points are measured on each casing: horizontal and vertical at each casing (bearing housing) corner. A graphic plot was provided for reverse indicator alignment. Gear horizontal movement was from two #2 precision dowels at each end of the pinion shaft only, never cross dowel (Figure 9).
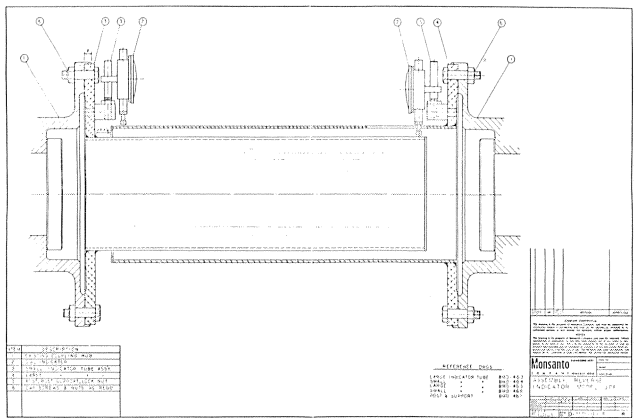

Figure 9. Alignment Reverse Indicator Bracket Design. Compressor Coupling End Is Within The Split Bearing Housing (See Figures 33 and 34 for Alignment Results).

5. API 678 and API 670 spin cap condulets with base cutouts were detailed to enclose axial probes and accelerometers, preventing any strain to each from "big heavy feet."

6. Alarm and shutdown details for vibration and bearing temperatures were transmitted in a manner understandable to three groups—machinists, instrument men, and operators.

COMMISSIONING

The objectives of the commissioning run were to prove the mechanical integrity of the equipment and to record lateral and torsional vibration data for further processing for permanent documentation. Five types of data were obtained from the commissioning:

1. Runup and rundown vibration data, to verify lateral critical speeds and unbalance sensitivity;

2. Torsional FM modulating data from gears, to verify torsional resonances during the runup;

3. Hot alignment data verifying predicted coupling offset, to accommodate differential heat rises;

4. Shaft eccentricity measurements;

5. Machinery response to various compressor surge conditions.

Each of these items will now be described and discussed.

Runup and Rundown of Compressor

The runup and rundown data were obtained from the orthogonal proximity probes on the compressor and from a once-per-turn phasor probe. These data were recorded on an eight-track FM tape recorder at 15 ips. Later, the tape speed could be slowed on playback to optimize the responses of the analysis equipment and also of the operator.

Figure 10 shows a plot of the compressor speed vs. time during the runup. Synchronous speed is achieved in 14.7 seconds, but overshoot occurs, and it takes 19 seconds for the speed to become stable. These speed fluctuations occurring before synchronization are due to the inertias of the rotors carrying the compressor past synchronous speed while the motor is trying to bring the train speed back to synchronous. This causes undershoot, followed by ever-decreasing overshoots and undershoots until the system is stable. Other speed fluctuations occur due to the first torsional resonance, which will be shown later.
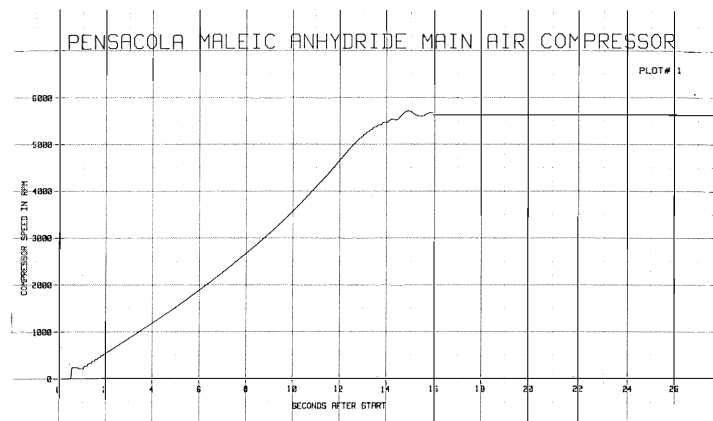

Figure 10. Run-up Plot of Compressor Speed Vs. Time, in Seconds.

The compressor displayed the split first critical that had been predicted and shown on the test previously. Figure 11 shows uncompensated raw and synchronous vibration data and phase data for the runup of the discharge (coupling end) in Bode form. Figure 12 is the vector runout compensated Bode plot, which clearly shows a horizontal first critical at approximately 2800 rpm and a vertical first critical at 3650 rpm. The split critical is even more dramatically seen in the polar (Nyquist) plot of these data (Figure 13). The inside loop is the horizontal, and the outer loop the vertical critical. Without total analysis and close review of the damping shown, one might misinterpret the inside loop to be due to a structural resonance. Figure 14 is the cascade diagram of this runup, showing mainly the 1× synchronous peak.

The rundown data are similar to the runup data, but the data are "smoother" due to the longer time of rundown. Figures 15 through 18 are the corresponding plots to those discussed above for the runup. Again the split critical is very pronounced, although the peak response frequencies have shifted down by about 100 rpm. Throughout the speed range during runup and rundown, the highest total vibration attained was 1.2 mils, with all other points under 1.0 mil. This machine was very sound mechanically and continues to run well.
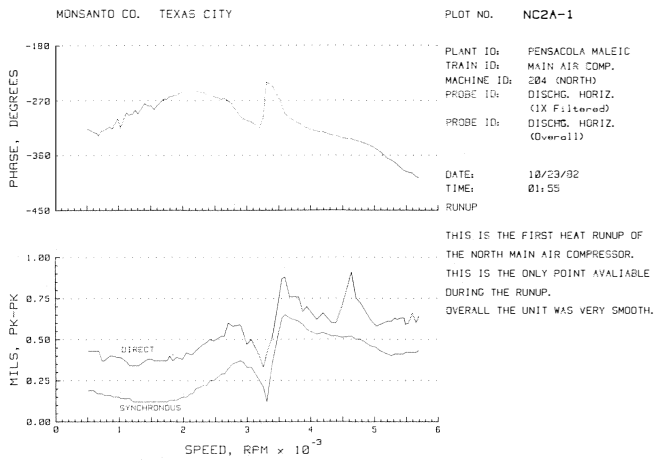
Figure 11. *Run-up Bode Plot of Raw (Uncompensated) and Synchronous (1×) Vibration and Phase Data Taken at Discharge (Coupling End) Horizontal.*
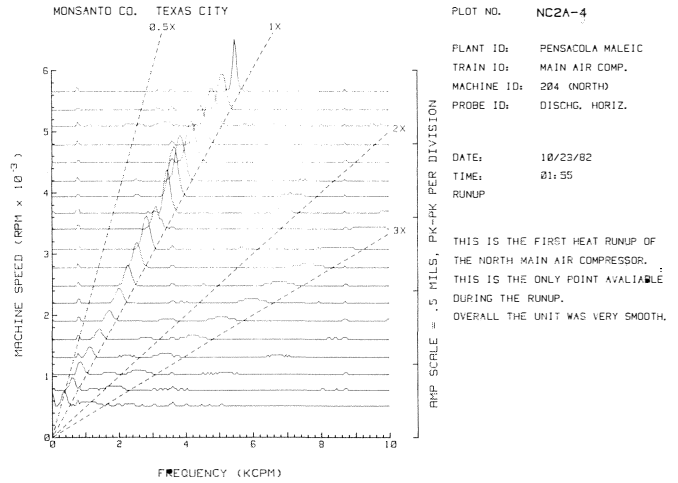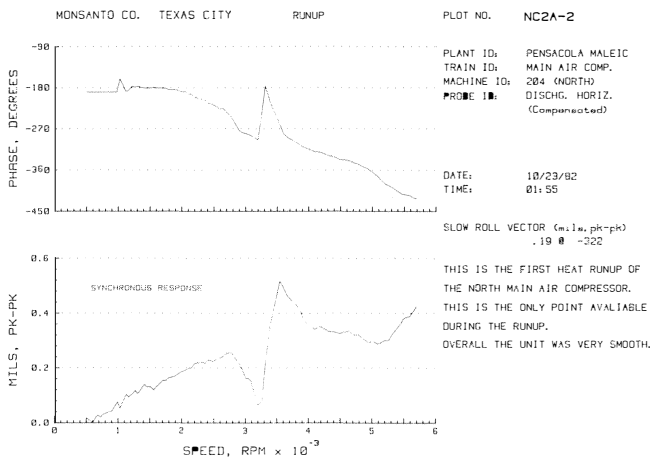

Figure 12. *Run-up Bode Plot of Compensated (Run-Out Compensated) of the Compressor (Same Point as Figure 11). Note Split First Mode Criticals—Horizontal at 2800 rpm, Vertical at 3650 rpm.*

Figure 13. *Run-up Polar Compensated Plot of Same Point.*

Figure 14. *Run-up Cascade Raw Data Plot of Same Compressor Point.*
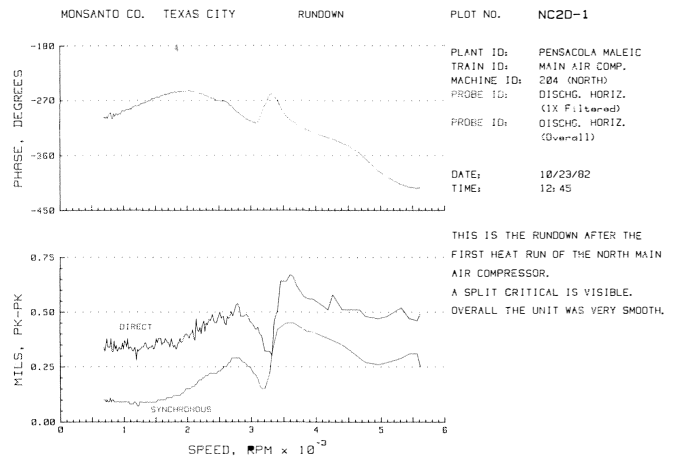
Figure 15. *Rundown Bode Plot of Raw and Synchronous Raw Data—Compressor Much Clearer than Run-up Plot.*
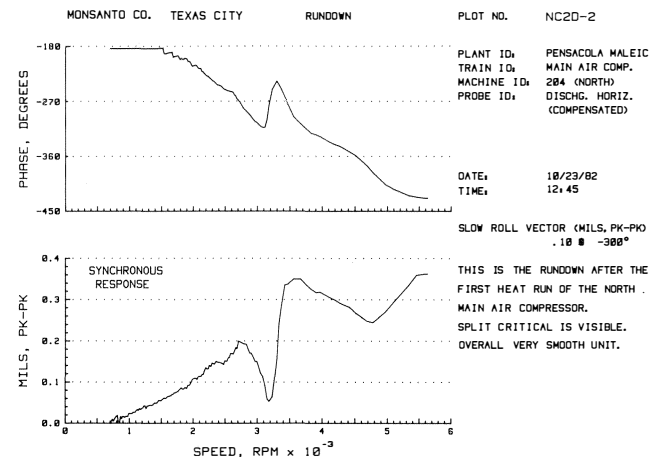
Figure 16. *Rundown Compensated Bode Plot of Compressor.*
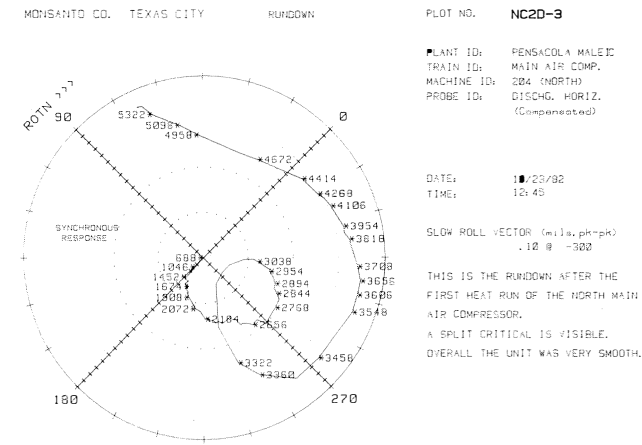
Figure 17. Rundown Polar Plot of Compressor Compares Well with Analytical Data, Figure 5g.
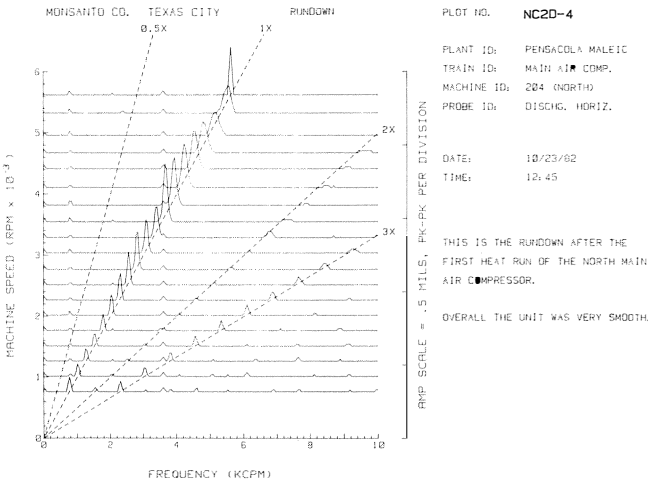
Figure 18. Rundown Cascade Plot of Compressor.

Runup and Rundown of Gears

The data from the gearbox during the runup were unique and significant. A "clatter" was heard from the gearbox during the runup, due to the first torsional resonance. Unexpectedly, the best evidence of the torsional critical speeds came from the lateral pinion proximity probes. Figure 19 shows the raw and synchronous vibration and phase data from the blind end vertical pinion proximity probe. There is a small peak at approximately 3250 rpm, due to the second torsional speed, and there is a very large peak at 4700 rpm, which reaches 5.5 mils. Running speed vibration is 0.5 mils. The synchronous response plot, Figure 20, shows some small peaks, but incoherent phase, indicating that these peaks are not a rotor lateral critical speed. Perhaps the most interesting is the cascade plot of this probe during the runup. Figure 21 shows the spectrums taken during the runup from 1000 rpm to full speed. The synchronous 1× peaks are very low, as are the 2×. The backward sloping line is all possible torsional excitation frequencies going from 7200 cpm (at 0 rpm) to 0 cpm (at 5614 rpm). At 3260 cpm (which corresponds to 3072 rpm and explains the "synchronous" peak near this speed) the *second* torsional critical speed is encountered *first*. Then, as the pinion speed increases to 4620 rpm, the torsional excitation frequency is 1275 cpm and "rings" the system's first torsional resonance. Even before this speed is reached, the 1275 cpm
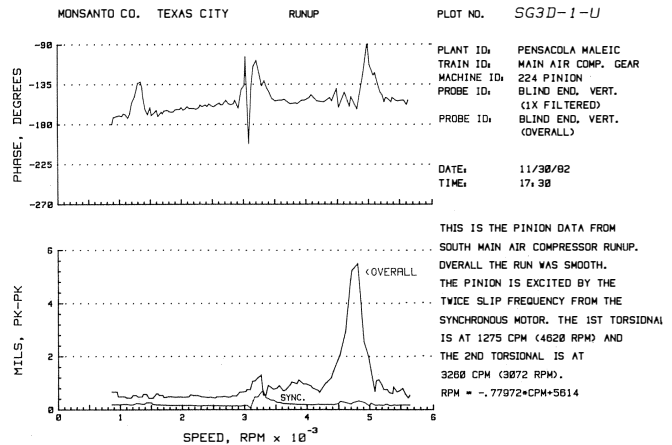
Figure 19. Run-up Bode Plot of Pinion Using Proximity Probe. The Small Peak at 3250 rpm Is Due to the Second Torsional Critical, and the Large Peak (5.5 Mils) at 4700 rpm Is Due to the First Torsional Critical.
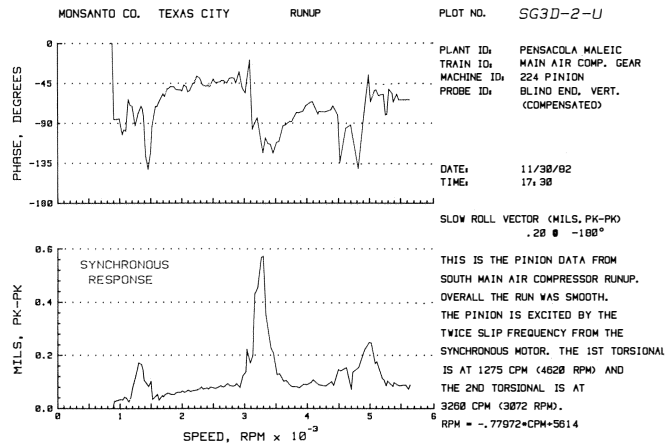
Figure 20. Run-up Compensated Bode Plot of Pinion Blind End, as in Figure 19.
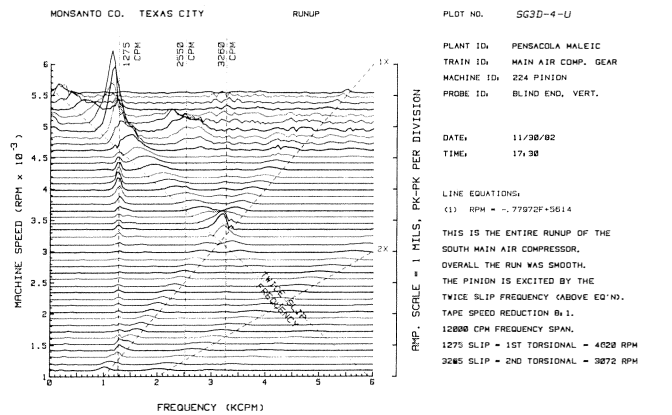
Figure 21. A Vivid Cascade Displacement Plot of the Pinion's Response as the Second and First Torsionals Are Encountered, Respectively, as Excited by the Synchronous Motor Twice Slip Frequency; i.e., 7200 cpm Decaying to 0 During Acceleration.

resonance can be seen on the cascade plot, responding weakly to the torsional pulsing of the synchronous motor. The lateral response of the pinion peaks is at a slightly higher speed (lower excitation frequency), since some finite time is needed for the vibration to build up. The time trace of this probe on a strip chart recorder is very interesting and can be seen in Figures

22a and 22b. This plot shows that at about 11.5 seconds into the runup, the pinion vibration starts to build and then settles out within 1 second. The onset speed is 4600 rpm, and the torsional response is gone by 5000 rpm. Thus, without even looking at the torsional modulating pickups, we have clearly defined the torsional resonance frequencies that are encountered during the runup. The magnitude (in degrees) of the first torsional resonance will be discussed later.

PINION, COUPLING
END VERTICAL (60°)

12 MILS

COMPRESSOR, DISCHARGE
END, HORIZONTAL (45°)

4 MILS

0   TIME (SECONDS)   1   2

0   SPEED (RPM)   250   550
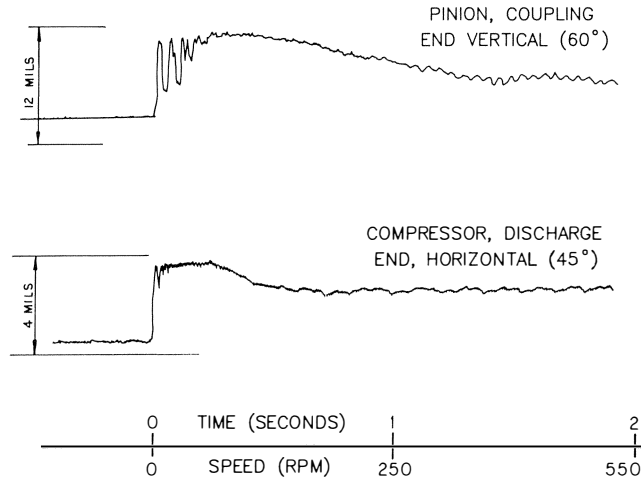
Lateral Response of Pinion
and Compressor During
First Two Seconds of Startup

Figure 22a. Strip Chart Recording of the Compressor and Pinion in Displacement (Mils p/p) During First Two Seconds of Synchronous Motor Startup.

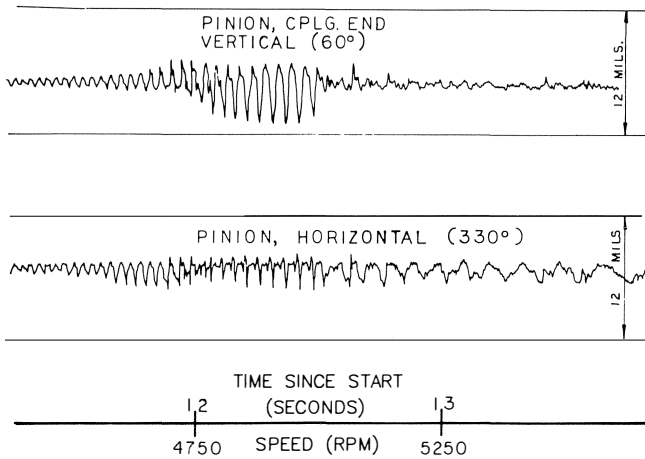PINION, CPLG. END
VERTICAL (60°)

12 MILS

PINION, HORIZONTAL (330°)

12 MILS

TIME SINCE START
1,2   (SECONDS)   1,3

4750   SPEED (RPM)   5250

Lateral Response of Pinion
in Mils Peak—to—Peak
Accelerating Through the
First Torsional Resonance

Figure 22b. Lateral Response of Pinion While Accelerating Through the First Torsional Excitation During Run-up.

The rundown proximity probe data were much cleaner than the runup data, since there was no torsional excitation during coastdown. Figures 23 through 26 are the vibration and phase data from the blind end horizontal proximity probe on the pinion. There is a peak at 3600 rpm, due to the pinion's being sympathetic to the compressor's vertical first critical speed. The solid multi-convolution coupling is able to transmit this to the pinion, due to the pinion's relatively low mass. This effect can be seen as a small loop on the polar plot, Figure 25. The cascade plot shows the vibration to be primarily synchronous, but with some subsynchronous, due to the speed of the bullgear.

The seismic probes on the gearbox are primarily to monitor the health of the gear mesh. During the runup, the gear mesh frequencies and the torsional excitation were observed. Figure 27 shows the cascade diagram of the accelerometer mounted horizontally on the bullgear side of the gearbox. The primary peaks track the 28× (number of pinion teeth) line, with some side bands (mainly upper) showing. There are some speeds where the 28× signal is higher than at others, due to structural resonances in the system. From 4600 rpm to 5000 rpm there is a general broadbend excitation from the first torsional resonance. Note that these data alone cannot identify the torsional frequencies. The highest peak response measured during the runup was 4.0 G's, with the running speed level under 1.0 G.

The rundown data from the accelerometers are shown in the cascade plot of the gear side spectrums. As during the runup, the primary vibration is at gearmesh (28×), with primarily upper side bands. Some twice gearmesh (56×) is also visible. A unique frequency that was encountered during the rundown at approximately 41,520 cpm was thought to be a case resonance. This was confirmed by "rap" testing the gear casing and using a "peak-hold" on the spectrum analyzer. Figures 28 and 29 show the results of one of these tests, giving a broad casing response around 43,000 cpm.

Motor Vibration Data

The 1200 rpm synchronous motors used on this machine train were extremely smooth machines. The runup vibration data are shown in Figures 30a and 30b. The two minor peaks are due, once again, to the system's two torsional critical speeds, but the maximum level is under 2 G's. The running speed vibration is 0.3 G; i.e., cap peak velocity of 0.00 ips. The cascade plot shows the primary frequencies of motor vibration multiples of gearmesh from the gearbox, with one peak at 19,000 cpm during the first torsional resonance. The rundown data from the motor were of such low levels that they are omitted here.

Torsional FM Modulation Data

The proximity probes that viewed the gear teeth passing during runup should "see" an oscillation of the teeth if a torsional resonance is encountered. If the running speed signal is thought of as the carrier wave, it should be possible to demodulate this signal to produce the torsional signal that is riding "on top" of this carrier. The problems encountered during this process are that the torsional signal is low compared to the carrier, and that the carrier frequency is not constant, but is ever increasing with time. Using a torsional signal conditioner and a very sensitive FFT spectrum analyzer, Figure 31 was produced. This is the time trace of the demodulated signal, and from this it was calculated that the bullgear underwent a maximum of 0.10 degree torsional displacement at 1275 cpm and also that the pinion saw 0.19 degree torsional displacement, also at 1275 cpm. This raw signal was also processed in another way: the 28× signal was fed into the digital tracking filter as the speed signal. The output of the tracking filter is a DC analog of this signal and can be plotted versus time. Figure 32 shows the speed range of 4000 to 5800 rpm, or from 10.3 seconds into the runup to 17.2 seconds. At
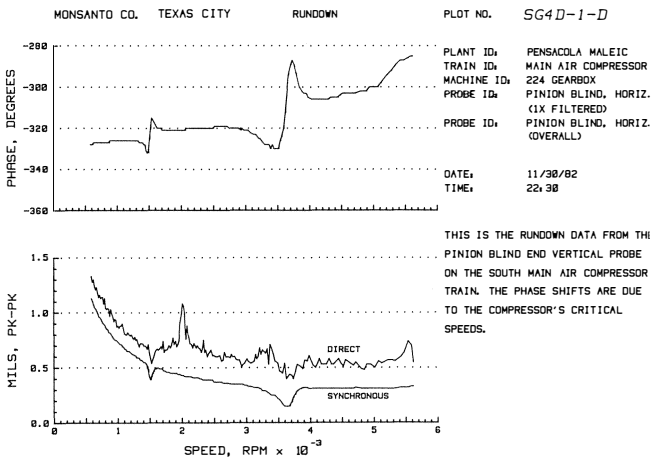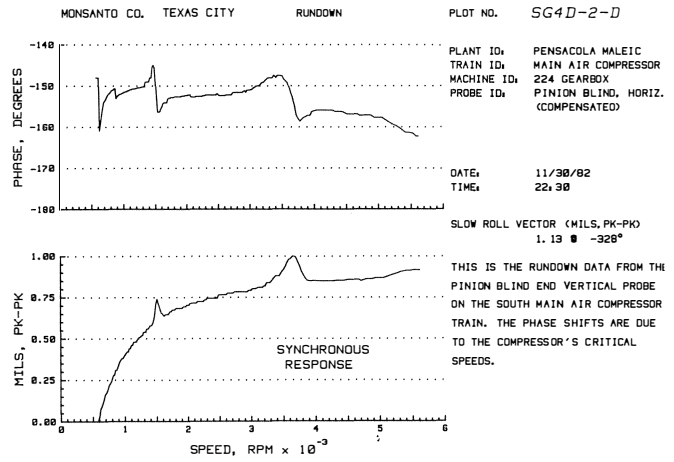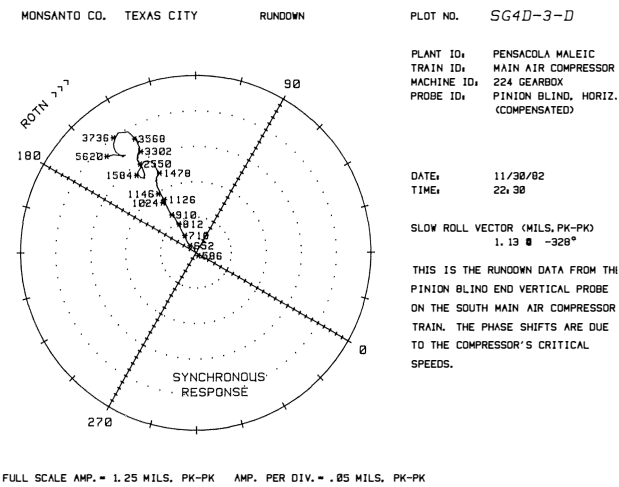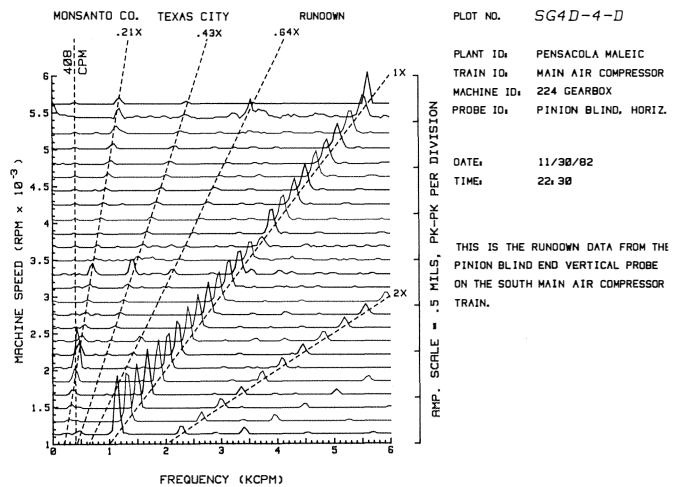
Figure 23

Figure 24

Figure 25

Figure 26

Figure 23-26. These Four Rundown Plots of the Pinion—Bode Raw and Synchronous, Bode Compensated, Polar Compensated and Cascade—Are Free of Starting Torque. They Show the Response of the Compressor's Lateral Critical at 3650 rpm and 1550 rpm. The Compressor Is Able to Transmit Its Critical Response to the Pinion as an Appendage Coupled by a Relatively Stiff Coupling Between Two Distinctively Different Masses (6:1).

about 4600 rpm there appears a fluctuation in the speed that peaks at 4800 rpm to a level of plus and minus 50 rpm. This is again clear evidence of the first torsional resonance, but actual magnitude and frequency are difficult to determine from this plot. The torsional signal from the second torsional critical speed was so low that repeated attempts to measure it were not successful.

Hot Alignment Data

Hot alignment data were gathered using Jackson water stands supplied with constant temperature cooling tower water. Each corner of each bearing housing had a nickel plated target that was viewed vertically and horizontally by proximity probes. The DC gap readings taken from these probes during a twelve hour heat run *and* a complete cool down were converted to displacement measurements and plotted, as shown in Figure 33. This figure shows the heatup values as solid arrows,

and the cooldown as dashed arrows. The net movements, both horizontally and vertically, are shown below diagrams of the compressor train. As can be seen, the figures were repeatable, primarily because the system had been heat cycled once before, allowing any hysteresis to dissipate. Finally, these data were plotted on the reverse indicator alignment plot to show the desired cold alignment readings so that the entire train would be in precise alignment when hot. Figure 34 shows the working alignment plot and the desired and actual machine positions. The predicted offset values were well within the alignment tolerance, so no additional machine shifting was required.

Shaft Eccentricity Measurements

The vibration data recorded from the proximity probes contain additional useful data in the DC bias component. It is possible to monitor the position of the centerline of the shaft
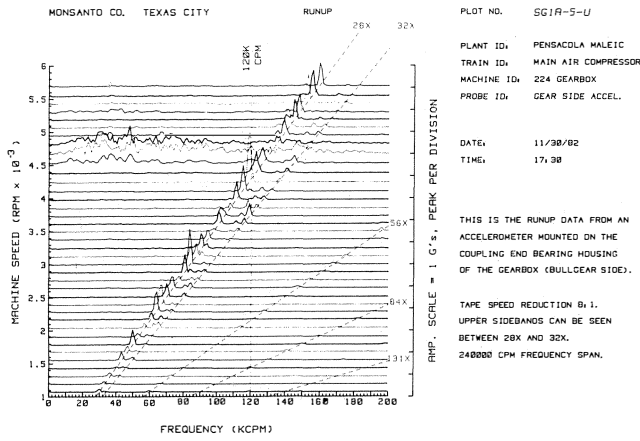
Figure 27. Cascade Diagram Taken From the Bullgear Bearing Accelerometer Tracking the Pinion Gear Mesh Frequency. The Gear Bode Excites All Resonances Seen at the Gearbox; i.e., Four Major Peaks.

Figure 28. Peak Hold Plot of The "Rap" Impact Test on the Gear at Zero Speed. A Resonance of 41,520 cpm on Rundown Is Confirmed, We Believe, by this Rap Measurement in the Broad Range of 43,000 cpm.
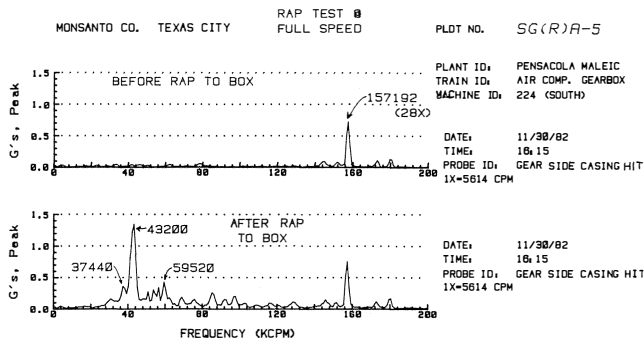
Figure 29. This Is Similar to Figure 28, but a Box Resonance "Rap" Test While Running at Full Speed.

Figure 30a. Motor Bode Plot from Coupling End Accelerometer on Run-up.
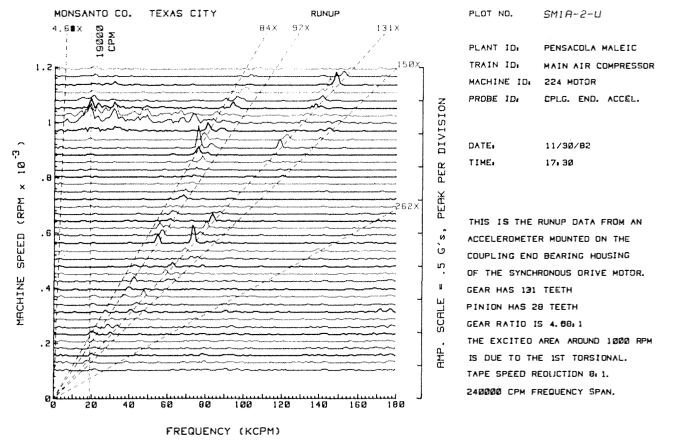
Figure 30b. Motor Cascade Plot from Coupling End Accelerometer on Run-up.
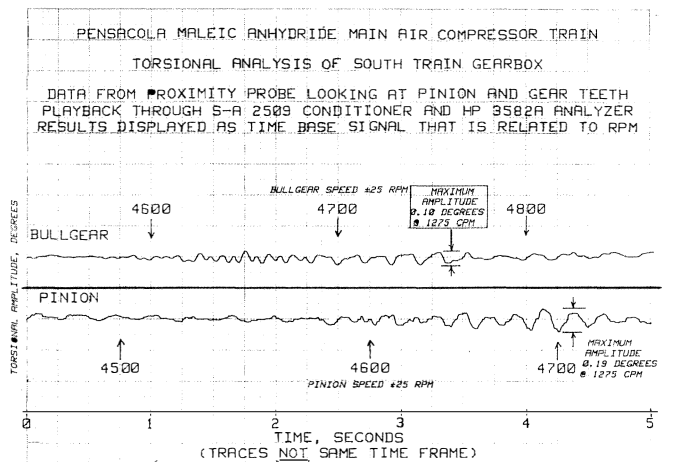
Figure 31. Transient Torsional Peaks, First Torsional, on Gear and Pinion Showing Angular Displacement of 0.1 Degree and 0.19 Degree, Respectively. Duration Is Within the One Second Time Frame.

within the bearing very precisely, in order to determine eccentricity ratio and attitude angle. This is done by filtering out all but the DC component of the signal. These data can then be plotted on a clearance circle to show the path of the shaft centerline either during runup, as load is applied, or
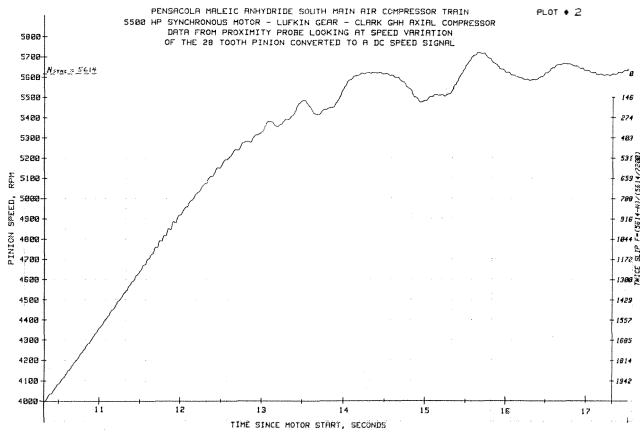
Figure 32. Speed Run-up Tracking the Pinion 28 Tooth Signal from a Torsional Measurement Probe; i.e., Tooth Gap Variations. The First Torsional at Near 4600 rpm Appears to Make a Variation, "Ruffle," in this Plot.
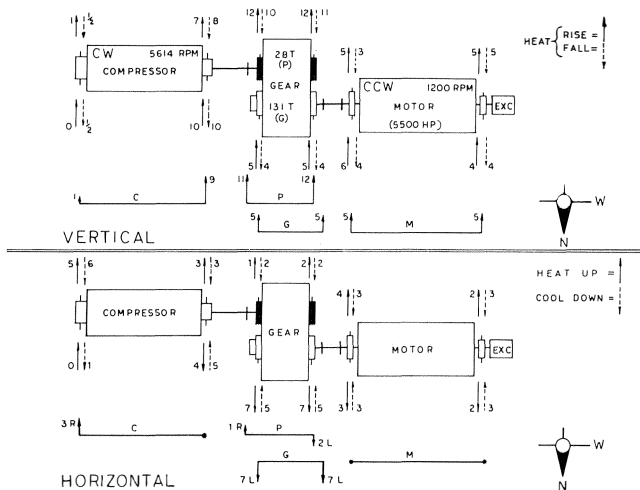
Figure 33. This Hot Alignment Plot Portrays The Heat Up/Cool Down Plots Taken from Continuous Data at 24 Measurement Points by Jackson Water Stands with Probes. (See Figure 9b for Plot).
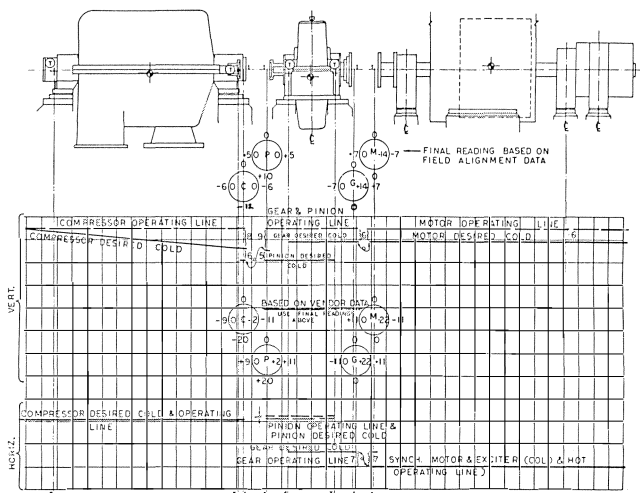
Figure 34. Alignment Graphic Plot; "Before" and "After." See Figure 33 for Movements Measured. Final Cold Reverse Indicator Valves Are Shown Above Grid.

during rundown. The only uncertainty encountered in this process is knowing the precise location before the startup. This is particularly troublesome with the pinion shaft, due to its low mass compared to that of the bullgear, which may be holding it above the journal bottom.

Figure 35 shows the results of the shaft eccentricity measurements from the coupling end of the compressor during the runup and as load was applied. The eccentricity path that the shaft takes is a classic parabola-shaped curve, starting at rest on the center of the bottom pad and rising rapidly within the first 1000 rpm, then rising more slowly up to full speed. An additional decrease in eccentricity ratio occurs as load is applied. The predicted eccentricity ratio for this bearing from the
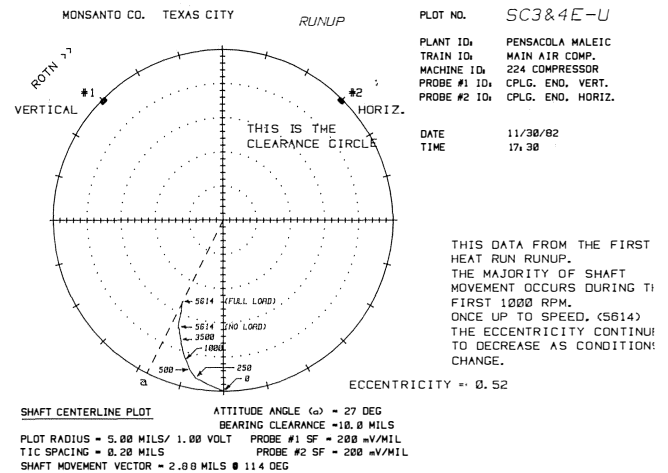
Figure 35. Eccentricity Plot Also Showing Attitude Angle (27°) and Amount (0.52) for the Compressor Coupling End on Pad 5 Shoe Bearing.

finite element-pad assembly computer program was 0.54, and the measured ratio was 0.52. The final attitude angle was 27° in the direction of rotation from the gravity load vector. The rundown of this unit resulted in a retracing of the curve down to zero speed.

Figure 36a is a plot of the runup of the pinion coupling end eccentricity. Due to the large forces imparted on the pinion from the torsional resonances, this was a much more difficult measurement than the compressor's eccentricity. The curve shown is the average position of the shaft centerline at various speeds. The bullgear position is to the left of this plot and is rotating counterclockwise, causing the pinion shaft to lift up in the bearing immediately upon startup to about the two o'clock position. This motion is aided by the presence of a pressure dam pocket located in the lower half of the journal at the 7:30 position, which imparts significant lifting force on the journal (2500 pounds of force at full speed). As speed increases, the pinion "homes in" to the three o'clock position, which is the location of the splitline of the bearing and the 30° oil inlet groove. Note that, as full load is applied, the gearmesh forces push the pinion an additional 0.7 mil toward the oil inlet groove. The initial position of this shaft was not at the exact bottom of the journal. This was determined by producing the rundown eccentricity plot, Figure 36b, which shows that the pinion moves toward the bullgear as speed is reduced and the gear separating forces decline. During the last 100 rpm, the shaft then settles down to the bottom of the bearing. This plot was much "cleaner" than the runup plot, due to the lack of torsional interaction.

The pinion eccentricity data caused us some concern, namely "what is supporting the pinion shaft, since it is running

MONSANTO CO. TEXAS CITY

RUNUP

#1 (VERTICAL)

PLOT NO.   SG3&4E-U

PLANT ID: PENSACOLA MALEIC
TRAIN ID: MAIN AIR COMP.
MACHINE ID: 224 PINION
PROBE #1 ID: CPLG. END. VERT.
PROBE #2 ID: CPLG. END. HORIZ.

DATE   11/30/82
TIME   17:30

ECCENTRICITY RATIO
0.627

THIS PLOT REPRESENTS THE RUNUP
SHAFT ECCENTRICITY OF THE COUPLING
END OF THE SOUTH TRAIN PINION.
THE INITIAL 25 RPM CAUSES A LARGE
"JUMP" AND THE TORSIONALS REALLY
"BOUNCE" THE PINION AROUND DURING
THE RUNUP. THIS PLOT IS THE BEST
AVERAGE CENTERLINE POSITION ABLE
TO BE DETERMINED.
NOTE THAT THE JOURNAL SEEKS OUT
THE HORIZONTAL OIL FEED SLOT AT
EQUALIBRIUM.

SHAFT CENTERLINE PLOT   ATTITUDE ANGLE = -89 DEGREES
     BEARING CLEARANCE =11.0 MILS
PLOT RADIUS = 5.50 MILS/ 1.10 VOLT   PROBE #1 SF = 200 mV/MIL
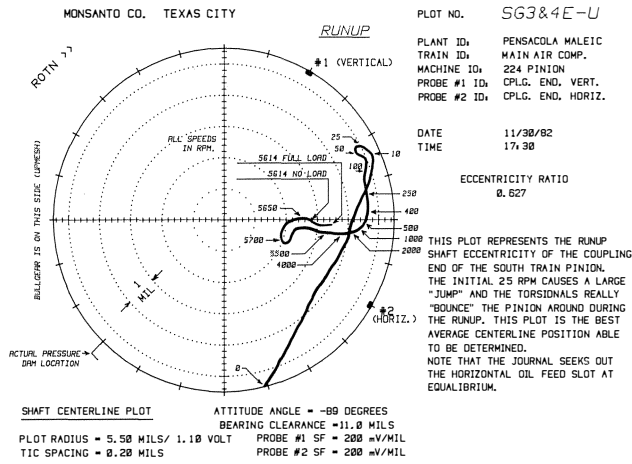TIC SPACING = 0.20 MILS   PROBE #2 SF = 200 mV/MIL

Figure 36a. Eccentricity Plot of the Upmesh Pinion During Run-up. This Journal Resides in Feed Slot (30°), Yet Ran Smoothly. Bearing Is To Be Rolled 30° CW, Maintaining the Same Pressure Dam Position.
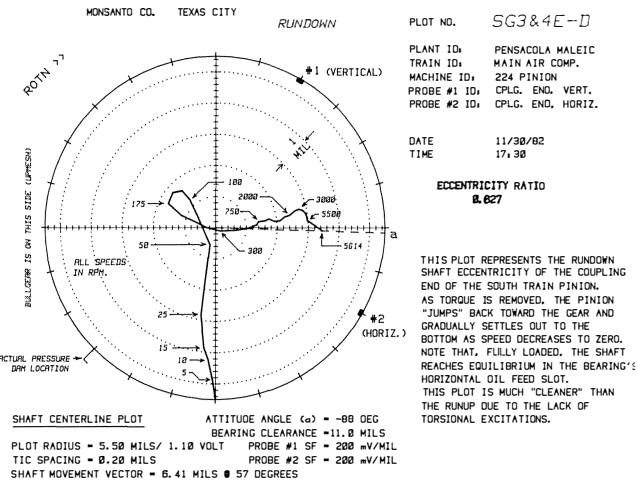
MONSANTO CO. TEXAS CITY

RUNDOWN

#1 (VERTICAL)

PLOT NO.   SG3&4E-D

PLANT ID: PENSACOLA MALEIC
TRAIN ID: MAIN AIR COMP.
MACHINE ID: 224 PINION
PROBE #1 ID: CPLG. END. VERT.
PROBE #2 ID: CPLG. END. HORIZ.

DATE   11/30/82
TIME   17:30

ECCENTRICITY RATIO
0.627

THIS PLOT REPRESENTS THE RUNDOWN
SHAFT ECCENTRICITY OF THE COUPLING
END OF THE SOUTH TRAIN PINION.
AS TORQUE IS REMOVED, THE PINION
"JUMPS" BACK TOWARD THE GEAR AND
GRADUALLY SETTLES OUT TO THE
BOTTOM AS SPEED DECREASES TO ZERO.
NOTE THAT, FULLY LOADED, THE SHAFT
REACHES EQUILIBRIUM IN THE BEARING'S
HORIZONTAL OIL FEED SLOT.
THIS PLOT IS MUCH "CLEANER" THAN
THE RUNUP DUE TO THE LACK OF
TORSIONAL EXCITATIONS.

SHAFT CENTERLINE PLOT   ATTITUDE ANGLE (α) = -89 DEG
     BEARING CLEARANCE =11.0 MILS
PLOT RADIUS = 5.50 MILS/ 1.10 VOLT   PROBE #1 SF = 200 mV/MIL
TIC SPACING = 0.20 MILS   PROBE #2 SF = 200 mV/MIL
SHAFT MOVEMENT VECTOR = 6.41 MILS @ 57 DEGREES

Figure 36b. This Is a Rundown Eccentricity Plot of the Pinion and Gives a More Easily Controlled Measurement, Being Removed from Gear Vector Forces.

in the oil feed slot?" While no bearing stress was detected and the at-speed vibration was very low, there was enough concern to cause a redesign of the pinion bearings. In concert with the gearbox manufacturer, it was decided to rotate the bearing 30° clockwise in the direction of rotation and to maintain the pressure dam position at 7:30. This should encourage the pinion to run in an area where there is full babbitt. In addition, the thermocouples embedded in the bearing will be moved to the minimum oil film thickness region in order to more accurately sense maximum oil temperature. These changes will be made only if bearing distress is noted in the future, or in the event of a major train overhaul.

Mechanical Response To Surge

The principal reason for purposely surging this compressor was to define the surge line at various operating conditions with the actual system resistance and volume, which cannot be duplicated on a test stand. Operations can then have an accurate margin of safety, and the surge controls can be set to protect the machine adequately. The vibration of all points was measured during the surge tests, which were made with various inlet guide vane settings. As might be expected, the

surge was most violent at the 100% flow setting and mildest at the minimum, or 25%, inlet guide vane setting. It is inaccurate to use the word "mild" when describing the surge response of this machine. There was virtually no warning in flow or pressure perturbations or vibrational increases, until there was suddenly a violent bang, lasting only about one second and sounding like a cannon. The compressor rotor was quite excited by this and responded by "ringing" the first critical speed around 3175 cpm, which is about the average of the horizontal and vertical critical speeds. Figure 37 shows the time trace of the 100% vane position vibration signals from all four compressor radial displacement probes. There was an initial burst, followed by the major flow reversal about 0.6 second later. The experimental logarithmic decrement can be accurately calculated from these data and is shown as 0.35. This compares favorably with the predicted logarithmic decrement of 0.36. Figure 38 shows a peak hold spectrum made from the discharge vertical probe during the surge. Note that, while the actual peak-to-peak vibration signal reached 6 mils as shown in Figure 39, the FFT is a windowing device and cannot accurately detect the peak values of transients. Therefore, the 3175 cpm peak comes out at less than 1 mil. This shows the value of being able to slow down the tape recorder speed by as much as 32:1, to produce the strip chart recording for accurate analysis. The large amplitude encountered when surging this machine
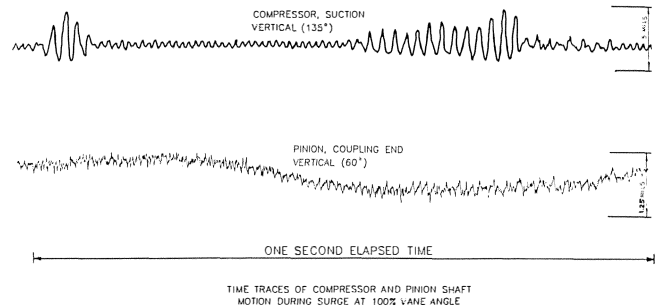
COMPRESSOR, SUCTION
VERTICAL (135°)

PINION, COUPLING END
VERTICAL (60°)

ONE SECOND ELAPSED TIME

TIME TRACES OF COMPRESSOR AND PINION SHAFT
MOTION DURING SURGE AT 100% VANE ANGLE

Figure 37. Time Traces of Compressor and Pinion Shaft Motion During Surge at 100% Vane Angle.

MONSANTO CO. TEXAS CITY

PLOT NO.   NC1R-1

PLANT ID: PENSACOLA MALEIC
TRAIN ID: MAIN AIR COMPRESSOR
MACHINE ID: 204 (NORTH)

DATE: OCT. 23, 1982
TIME: 18:00
PROBE ID: DISCHARGE, VERTICAL
1X=5615 CPM

1X

3175 CPM

0.75
0.50
0.25
0.00

MILS, PK-PK

1.2   2.4   3.6   4.8

FREQUENCY (KCPM)

THIS DATA WAS TAKEN DURING A SURGE TEST AT 100% VANE SETTING.
THE SURGE, WHICH OCCURED IN TWO BURSTS, EXCITED THE COMPRESSOR'S
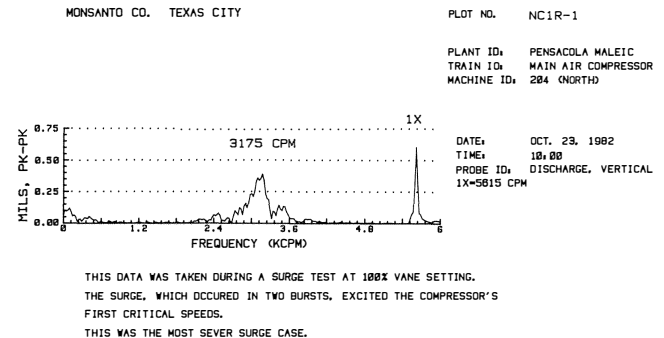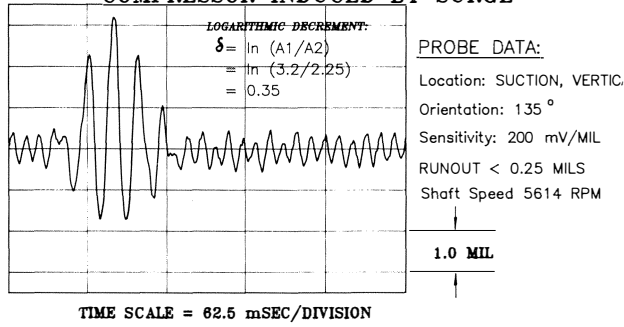FIRST CRITICAL SPEEDS.
THIS WAS THE MOST SEVER SURGE CASE.

Figure 38. Spectrum Capture of Rotor's Criticals (2800 and 3175 cpm) Being Excited During Surge While Running at Full Speed and Load.

graphically shows how damaging continued surge conditions could be to the machine parts and how important adequate testing and surge protection are to producing a reliable and safe unit. The gearbox and motor were virtually unaffected by the surge of the compressor.

CONCLUSIONS

With careful attention to design at the early project level, a compressor system can be conceived which is a culmination of principles based on good and bad experiences over the

RESONATE EXCITATION OF AXIAL AIR COMPRESSOR INDUCED BY SURGE

LOGARITHMIC DECREMENT:

$\delta = \ln (A1/A2)$
$= \ln (3.2/2.25)$
$= 0.35$

PROBE DATA:

Location: SUCTION, VERTIC,

Orientation: 135 °

Sensitivity: 200 mV/MIL

RUNOUT < 0.25 MILS

Shaft Speed 5614 RPM

1.0 MIL

TIME SCALE = 62.5 mSEC/DIVISION

THIS DATA WAS CAPTURED DURING A SURGE TEST
WITH VARIABLE INLET GUIDE VANES 100% OPEN.
FLOW OF 75,000 SCFM @ 30 PSIG DISCHARGE.

Figure 39. Log Decrement of Rotor's Resonant Excitation at 0.35. Confirms Damping in Analytical Model.

years. These conclusions best summarize our desires and final opinions on this project.

The plant requirements forcing a system driver to be a synchronous motor were addressed early. The sole plate mounting of the rotor on separate and sturdy pedestals proved to offer nearly undetectable vibration (≈ 0.004 ips). Further, alignment, electrical magnetic centers, and rotor-stator air gap were in good control. Damping of the larger 7 in. journals was considered to be a good factor as well.

The nonconventional approach to investigating an alternate coupling to the soft elastomeric designs, which are more conventional in torsional susceptible designs, seemed successful. Five months were set aside to complete this work. Emphasis against stress susceptibility was established early; e.g., no keyways, proper metallurgy, inertial balance, and geartooth pitches.

Time constraints to reduce surgings appear to be as low as could be achieved; e.g., minimum volume to dump, two valves operating in parallel for one second action of anti-surge valves, and analog hydraulic actuators for vane control via boosted pneumatics for control.

Air entrance filtration provided not only clean air but also low horsepower losses.

Attention to detail in bearing selection on the compressor and gear, quality control, and shop testing proved to be successful. Attention to proper installation, alignment, and commissioning, along with projection monitors, including temperature, checked out well.

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