# STARTUP OF A LARGE COMPRESSOR TRAIN—TESTING VERIFIES DESIGN

by

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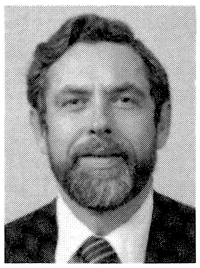

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## ABSTRACT

The startup of a reliable compressor train involves not only the design and design audit, but also startup testing to verify the functionality of the design. Startup testing usually includes mechanical and electrical system checks, solo runs, alignment measurements, and extensive vibration monitoring.

There are other test programs conducted at startup that can also play an important role in the successful startup of a critical process train. Recently, a large compressor train was installed and brought on stream in an air oxidation process. Described briefly herein are five tests that helped verify the design and that are contributing to the reliability of the train. These tests were:

- shaker testing a low tuned concrete foundation,
- transient torsional torque measurement during startup,
- measurement of stator vane stresses during surge,
- compressor surge tests, and
- measuring rated point performance.

## INTRODUCTION

The reported compressor train is shown in Figure 1 and is described from the upper right to the lower left corner of the picture. The train is driven by a 22,000 hp synchronous motor through a speed increaser. A two case compressor train provides 65,000 scfm of air to the process. Power is recovered from the waste gas by a 13,000 hp expander. The compressor train is mounted on a low tuned spring supported foundation.

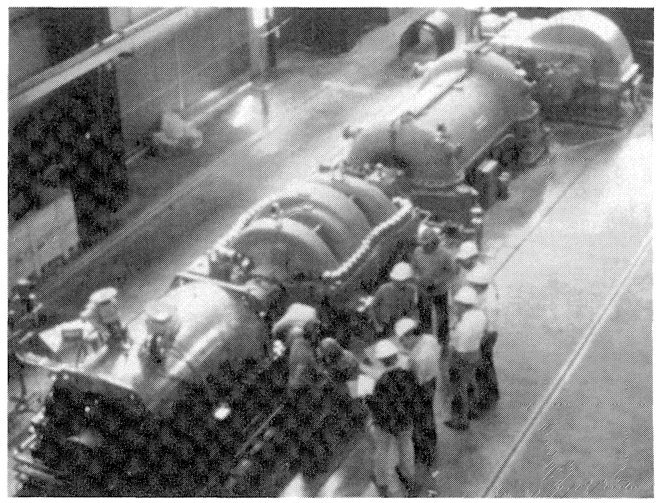

*Figure 1. Compressor Train. Front to rear: expander, high pressure case, low pressure case, speed increaser, and synchronous motor.*

The process schematic diagram is shown in Figure 2. The low pressure compressor (LPC) is an axial flow machine with variable stator vanes. The high pressure compressor (HPC) has seven impellers and is divided into two stages for intercooling. Air exiting the compressor may go either to the process or spill back to the expander.

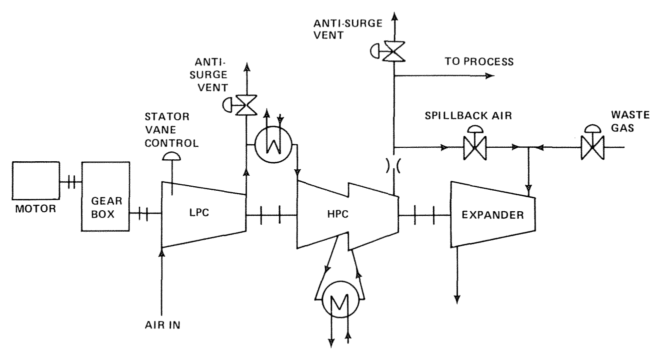


*Figure 2. Compressor Train Schematic Diagram.*

Energy recovery from the spill back air reduces energy usage during the time that the process is being brought online. At reduced rates, the spillback air may be mixed with waste gas from the process. Compressor train energy usage is an important part of the product cost, and compressor turndown is important to minimize the cost during periods of reduced production.

The following design audits were carried out parallel to the vendor's analyses:

- Motor shaft stress analysis
- Lateral rotor response
- Torsional rotor response
- Synchronous motor transient torsional excitation
- Foundation analysis and modeling
- Blade vibration modal analysis for the axial compressor and the expander
- Computer simulation of the interaction between the process controls

Several design changes were made in the compressor train prior to fabrication as a result of the design audit.

Startup testing was extensive and included the following:

- Mechanical and electrical system checks
- Solo runs for motors, compressors, etc.
- Hot and cold shaft alignment measurements
- Extensive vibration monitoring that included continuous spectrum analysis with tape and computer data gathering equipment

There were five additional tests that were conducted during the startup. These tests were:

- shaker testing a low tuned concrete foundation,
- transient torsional torque measurement during startup,
- measurement of stator vane stresses during surge,
- compressor surge tests, and
- measuring rated point performance.

The purpose of these tests were to both verify the component designs and to improve system reliability.

## SHAKER TESTING A LOW TUNED CONCRETE FOUNDATION

The compressor train is mounted on a monolithic reinforced concrete foundation that rests on spring supports as shown in Figure 3. The 24 ft wide by 79 ft long by 5 ft thick slab weighs 625 tons and was cast in place in a continuous pour. The machinery and piping weighs an additional 175 tons for a total weight of 800 tons.

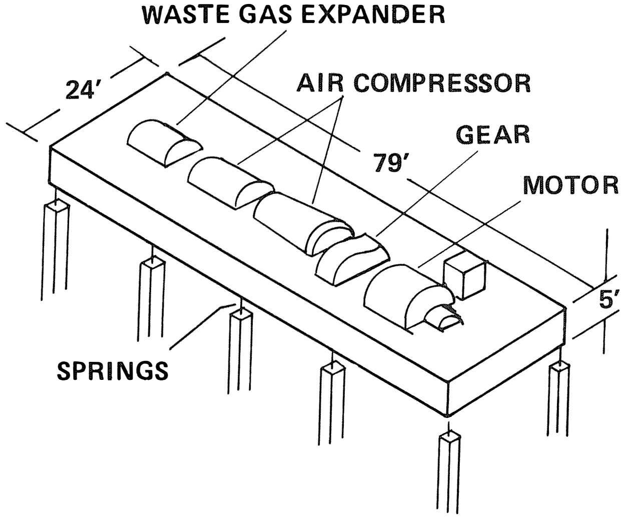


*Figure 3. Isometric View of Compressor Foundation.*

This compressor train is located in an area with soft soil and a high water table. The primary purpose of a low tuned foundation is to isolate the subfoundation and supporting soil from the machinery's vibratory forces which can slowly drive friction piles down into soft soil. The lower dynamic load transmitted to the columns and pile caps reduces their size and provides more room under the foundation for pipes, vessels, etc. A foundation can be low tuned without using springs, but springs are easy to change in the field. The disk spring assemblies used on this foundation can be removed and inspected in about 15 minutes.

Low tuned foundations are uncommon in the United States. The main concern was that the design would fail to properly isolate the system. The primary shaking force was expected to be unbalance at the motor speed. For effective isolation, the ratio of motor frequency divided by the first natural frequency of the foundation should be greater than the square root of two ($\omega/\omega_n > (2)^{0.5}$). Higher frequency vibration modes also should not fall on the motor or compressor speeds. Analytical and experimental models of the foundation [1] predicted that the 1/1 torsional mode would be close to the 1200 cpm motor speed. The 1/1 torsional mode has the two ends rotating out of phase, as shown in Figure 4.

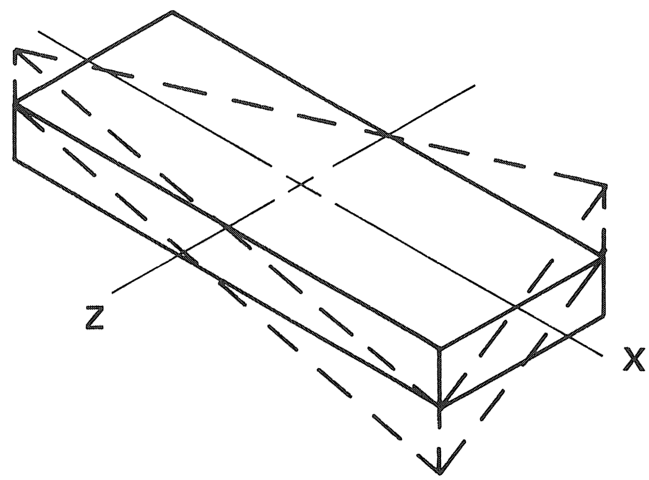


*Figure 4. Isometric Depiction of 1/1 Torsional Mode of Foundation.*

In Figure 5, the foundation is shown sitting on the springs, shortly after the forms were removed. A hydraulically driven shaker (1000 lb capacity) was mounted on the foundation at three different locations and shaker tests were conducted to determine the natural frequencies and mode shapes of the foundation. A two-channel spectrum analyzer was used

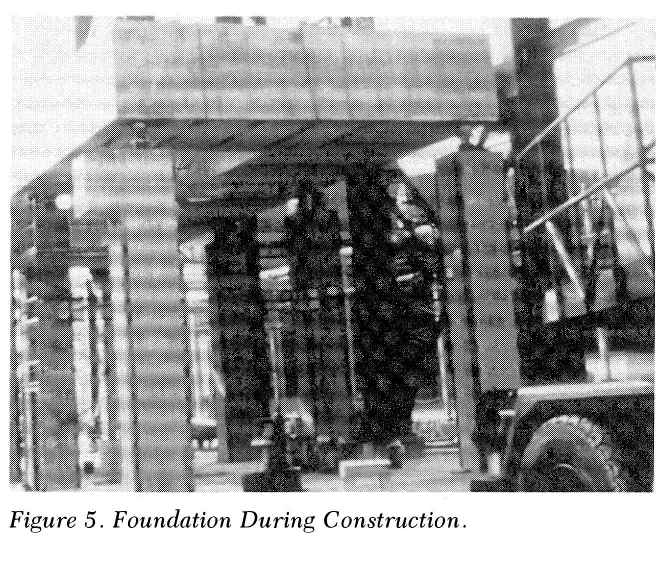

*Figure 5. Foundation During Construction.*

for transfer-function analysis and for spring transmissibility measurements. The shaker tests were repeated after the machinery was in place. The foundation vibration modes and predicted and measured frequencies are shown in Table 1. The frequency of the 1/1 torsional vibration mode was measured to be 1260 cpm or within five percent of the motor speed.

*Table 1 . Predicted and Measured Frequencies for the First Six Foundation Vibration Modes.*

| VIBRATION MODE | PREDICTED | MEASURED |
|---|---|---|
| TRANSLATION | 332 (CPM) | – |
| ROCKING, Z-AXIS | 370 | 420 (CPM) |
| ROCKING, X-AXIS | 578 | 608 |
| 1ST BENDING | 708 | 780 |
| X, Z, - 1/1 (TORSIONAL) | 1125 | 1260 |
| 2ND BENDING | 1512 | 1658 |

The 0.2 mils O-peak vibration amplitude measured at the corners of the foundation with the motor running met the acceptability criteria (Figure 6), but only because the motor rotor balance was excellent. Since it was so easy to change springs, the stiffness of the corner springs was doubled. The 1/1 torsional vibration frequency was increased to 1350 cpm, ten percent away from the operating speed, without shifting the other frequencies enough to cause problems. The vibration levels at the corners of the foundation were reduced by a factor of eight. The stiffer springs were left in service.

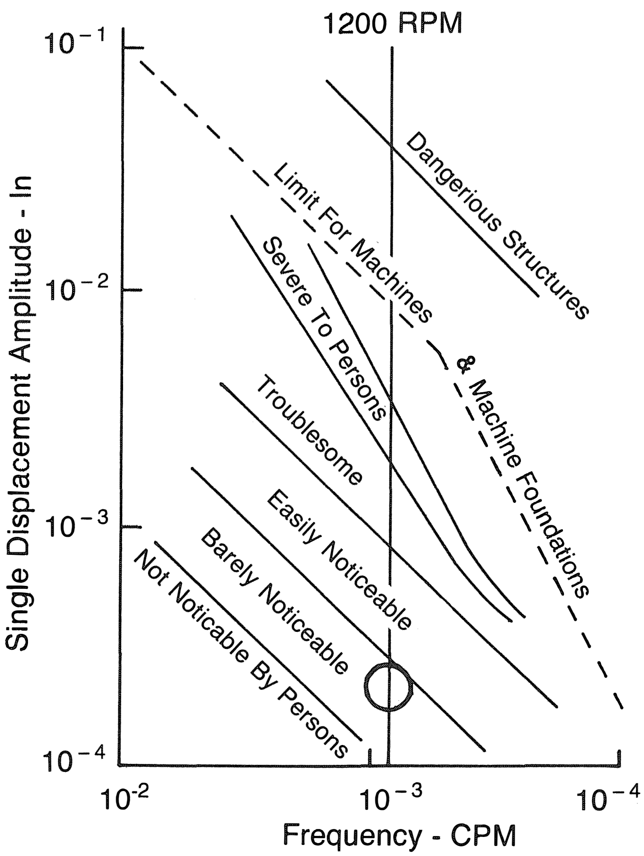


*Figure 6. Foundation Vibration Acceptance for Torque Measurement [2].*

Analytical and experimental modelling gave good results, but field testing the foundation provided the final data that led to the decision to change the stiffness of the corner springs.

## TRANSIENT TORSIONAL TORQUE MEASUREMENT DURING STARTUP

During startup, synchronous motors produce a pulsating torque at twice the slip frequency that excites the shaft system at its torsional natural frequencies that fall between 0 and 120 Hz [3]. The transient torsional response calculations for the train forecast a peak amplitude of 390,000 ft-lb at the first torsional natural frequency of 1039 cpm. This was an amplification of 4:1 over the 96,000 ft-lb full load torque at the motor coupling. As a result of the calculations, the motor shaft, which is most highly stressed at the keyways adjacent to the coupling hub, had been improved to a forged ASTM A-668 that is similar to AISI 4340. The design life of the shaft, due to low cycle fatigue, was calculated to be as low as 4300 starts, based on calculated motor starting characteristics and using a new transient torsional analysis program. Some compressor vendors were recommending resilient couplings for large synchronous motor drivers. It was decided that it would be prudent to measure the torque amplitudes during startup in the field.

The motor coupling, a size ten oil filled gear tooth type, was converted to a torque transducer by mounting a full bridge strain gauge circuit on the coupling sleeve. The coupling assembly is shown during testing in Figure 7. Batteries to power the bridge circuit and miniature strain transmitters were contained in a ring clamped to the sleeve. The strain output was transmitted via an FM carrier signal to a staionary antenna leading to a signal conditioner and a high speed stripchart recorder.

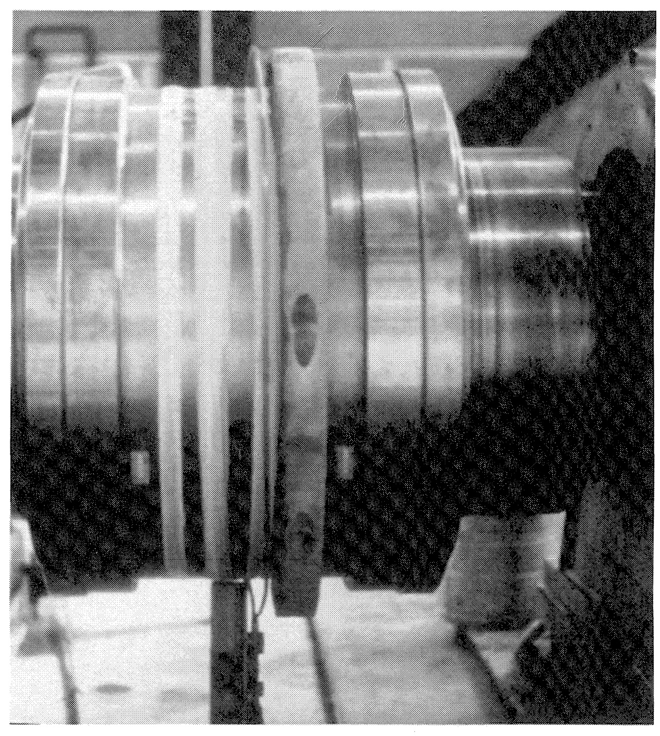

*Figure 7. Motor Coupling Instrumented for Torque Measurement.*

The data shown in Figure 8 are typical of the recorded startups. The first three torsional natural frequencies are evident; they appear in reverse order because the slip frequency of the motor pulsating torque that excites them decreases as the motor speed increases. The peak torque at the first torsional natural frequency was measured to be 230,000 ft-lb, which is considerably lower than the 390,000 ft-lb calculated value. Fortunately, the analysis program and the motor manufacturer's data were conservative. Available motor starting torque was higher than predicted, so the dwell time near the torsional critical frequencies was shorter than expected. The actual system damping at the first torsional critical also appeared to be higher than the value used for analysis. Major problems may have developed if the measured torque had been that much larger than the calculated value.

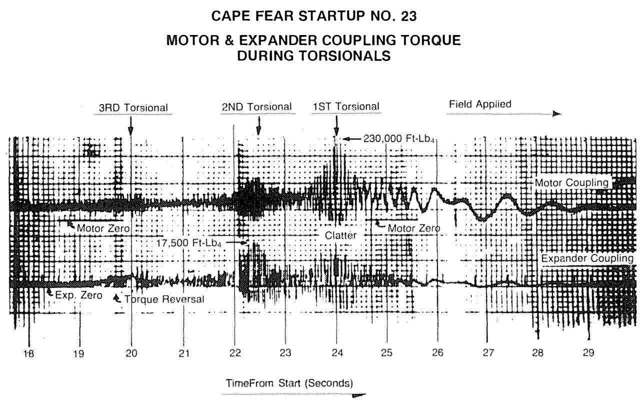


*Figure 8. High Speed Chart Recording of Motor T•rque During Startup.*

## MEASUREMENT OF STATOR VANE STRESSES DURING SURGE

The low pressure compressor (LPC) has 14 axial flow stages with variable stator vanes that are used to extend the compressor operating range. The stator blades are constrained by linkages to rotate together (Figure 9). The first or 0-row blades vary from 54 degrees (minimum) to 97 degrees (maximum) opening. The opening range of each succeeding row is smaller than the last. The vendor's standard policy was to instrument axial flow compressors to shut down during the first surge cycle. One reason given was to protect the blades from high stress levels encountered during surge. Since the devastating effects of a 0-row stator blade failure had been experienced previously in a similar axial flow machine, it was decided to field test the three most highly stressed blade rows in the stator during startup. The vendor was very cooperative and agreed with the details of the proposed test.

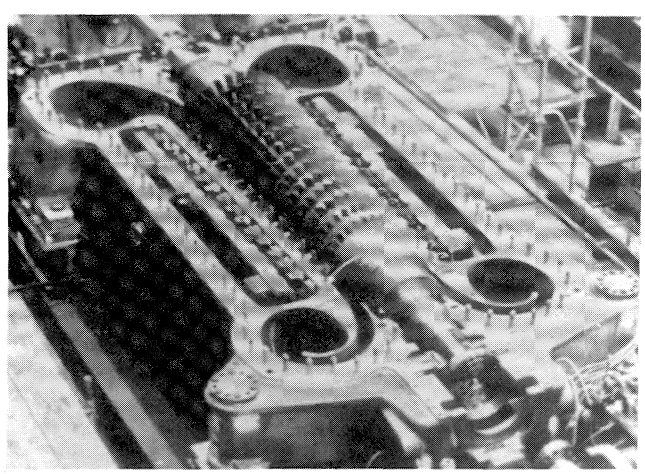

*Figure 9. Low Pressure Compressor with Top Half Removed.*

It was proposed to mount strain gauges on four blades from the 0-row, two blades from the second stage row, and two blades from the fifth stage row. The strain gauges were mounted on stator blades from the three blade rows (Figure 10). A hole was drilled in the shaft of the vane and an intersecting hole was drilled in the platform. Two strain gauges were mounted on each blade (Figure 11); one located in the root radius area, and the other mounted on the platform for temperature compensation. Calibration curves were run for the stator blade and strain gauge assemblies prior to installation. The stator blades were installed during the routine field internal clearance checks prior to startup. The wires exited the casing through a port provided by the vendor. Strain data were recorded throughout initial test runs and surge testing. The modified stator blades were removed when the train was given a routine inspection after one year of service.

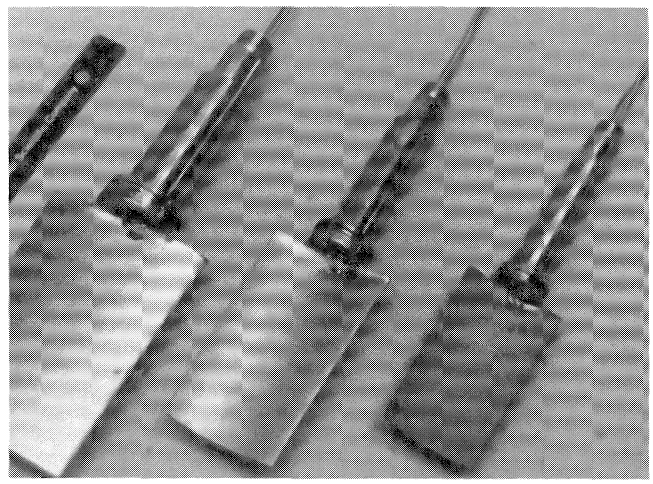

*Figure 10. O-row, Second and Fifth Stage Stator Vanes Showing Strain Gauge Installation. Six inch scale for size comparison.*

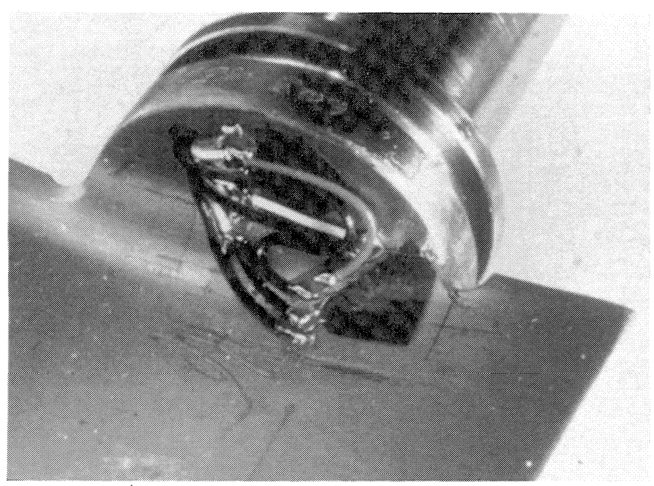

*Figure 11. Close View of Strain Gauges on Stator Vane.*

The modified Goodman diagram (Figure 12) shows the effect of surge on the fifth stage blade stresses. During normal operation, the mean blade stresses are a low 2500 psi. The mean blade stress during surge is higher and increases with the opening of the stator vanes. A mean blade stress of 25,000 psi (an amplification of 10:1) was measured with a 74 degree stator

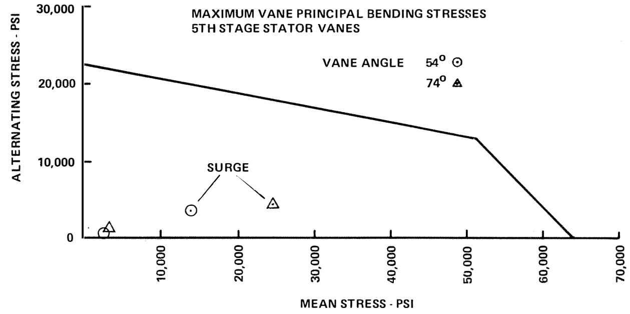


*Figure 12. Modified Goodman Diagram Showing Effect of Vane Angle and Surge on Fifth Stage Stator Vane Stresses.*

blade opening. Similar results were found on the zero and second stage blade rows. A peak stress of 39,000 psi (60 percent of yield) was noted on a 0-row blade during surge.

This test confirmed that the LPC should be instrumented to shut down on the first surge cycle. The shutdown sequence was also changed to close the inlet guide vanes immediately following a surge interlock to reduce the severity of the stresses due to the decaying surge cycles that occur during coastdown.

## COMPRESSOR SURGE TESTS

The compressor train schematic (Figure 2) indicated a surge valve for both the LPC and the HPC. The surge control philosophy for this train showing the control lines on a compressor head-flow diagram is illustrated graphically in Figure 13. The rated point occurs at 100 percent of the flow and head. A system upper pressure limit is established by relief valve settings. The LPC surge valve control line is to the right of the LPC surge line; the valve opens if decreasing flow crosses the control line from the right. The train shuts down when the flow decreases to the LPC surge line. The HPC surge control line is to the right of the LPC line and contains a high pressure override to prevent the operating pressure from reaching the relief valve setting. The HPC surge valve opens if the compressor operating point approaches the control line from below or from the right. The spillback control line is set slightly to the right and below the HPC control line. The response of the spillback valve is slow and designed not to interact with the fast responding HPC surge control system. The operating range of the compressor is the flow along the 100 percent head line between the maximum stator vane opening and the spillback line. At very low production rates, the flow is reduced back to

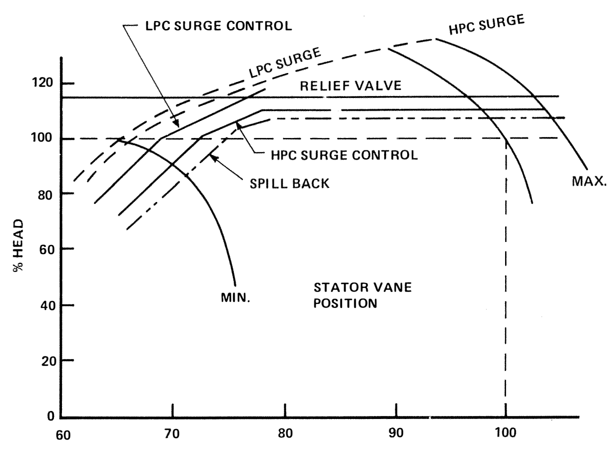


*Figure 13. Generalized Performance Curve Showing Compressor Surge Control Lines.*

the HPC control line and may be reduced even further by lowering the head to the intersection of the minimum stator vane opening line with the HPC control line.

The surge control instruments are electronic and are very fast. During normal process upsets, the HPC surge control system is fast enough to keep the LPC out of surge, so the compressor stays on line [4].

Since the position of the surge control line establishes the lower end of the compressor operating range, the actual surge line is very important to reduced rate operation. The vendor normally establishes the actual surge line in the field and sets the surge controls to it. Although the end user operated the compressor and participated in the testing, the vendor was responsible for the surge testing.

The inlet of the LPC is used as an orifice to measure the flowrate (Figure 14). The pressure drop measurement is used for proportional control of the LPC surge valve and to trigger the LPC surge interlock. Inlet calibration data plotted in Figure 14 are the results of pitot tube traverse data taken in the inlet duct.

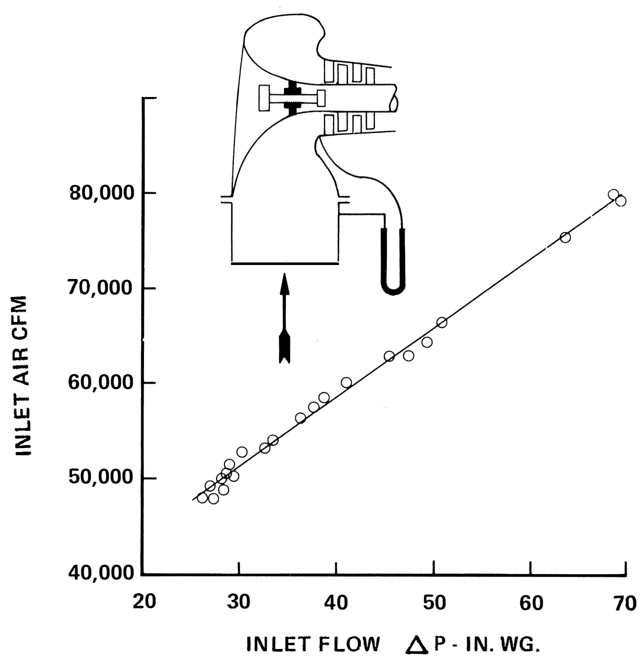


*Figure 14. Low Pressure Compressor Inlet Flow Calibration Data.*

The LPC was run solo and tested to establish its actual surge line. The stator vane angle was held constant and the flowrate was decreased by closing the surge valve in steps. When equilibrium was reached, inlet and discharge temperatures and pressures and flow were recorded. Non-equilibrium data for the actual surge point was recorded, since the compressor shut down at the first cycle. Data for questionable surge points were taken again.

Surge was measured for stator vane angles of 54, 64 and 74 degrees. Discharge pressure and flow data are plotted in Figure 15. Tests were run on two days with widely varying inlet conditions. The data were corrected to a 95°F inlet temperature and the surge control line was offset seven percent from that line. The surge control system for the LPC was exercised thoroughly to verify that it worked properly.

Actual surge data and the vendor calculated surge line and performance curves are presented for clarification in Figure 16. Actual surge for the LPC occurred at a lower flowrate than calculated.

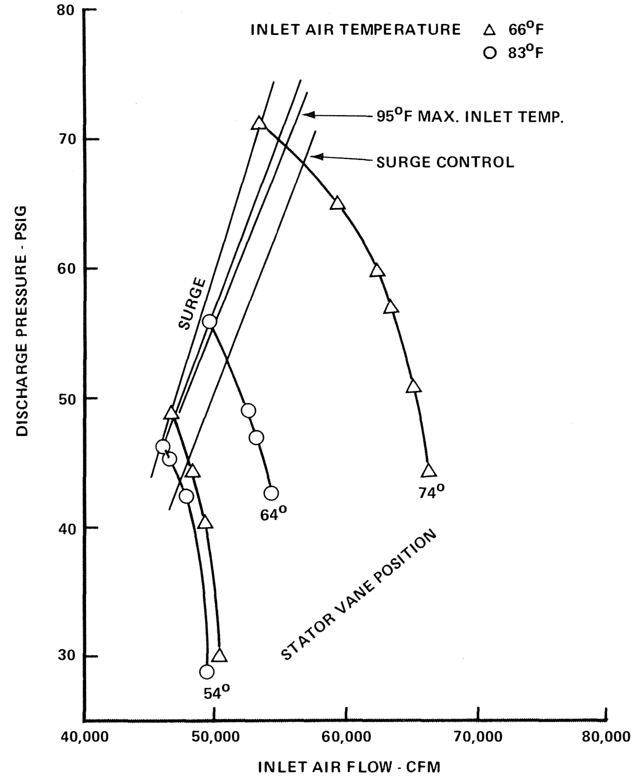


*Figure 15. Low Pressure Compressor Surge Data to Establish Surge Valve Control Line.*

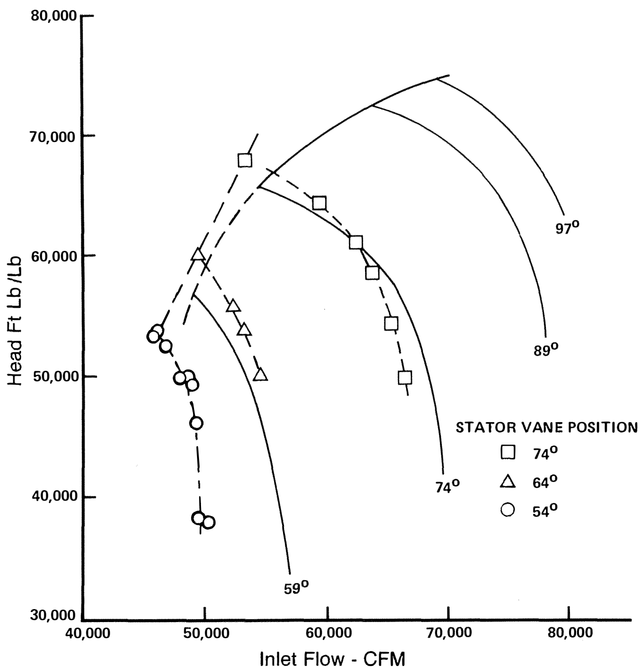


*Figure 16. Comparison of Actual to Calculated Surge Line for Low Pressure Compressor.*

The HPC was connected to the train, and it was tested for surge in a manner similar to the LPC. Surge data for two inlet guide vane angles (Figure 17) were used to set the HPC surge control line and the spillback control lines at seven percent and ten percent offsets, respectively.

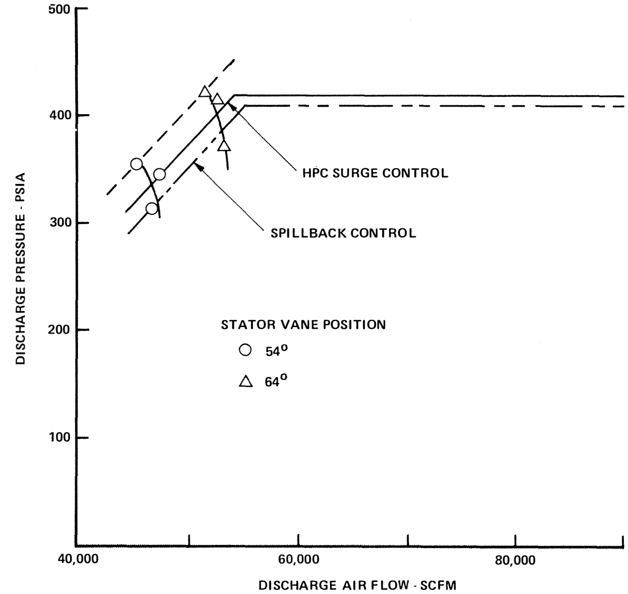


*Figure 17. High Pressure Compressor Surge Data to Establish the HPC Surge System Control Line.*

When the HPC surge data were corrected to design conditions and plotted on the combined performance curve for both compressors (Figure 18), it was obvious that surge occurred at a much higher flow for a given head than calculated by the vendor. This meant that the compressor operating range was less than expected. The loss of operating range needed to be documented with performance data.

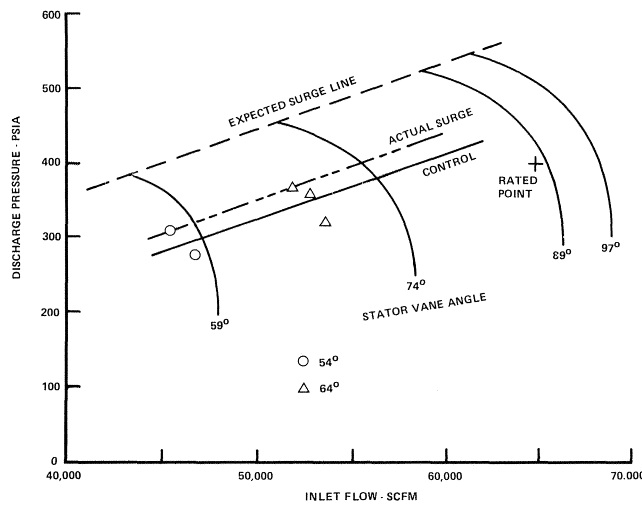


*Figure 18. Comparison of Actual Surge Line with Calculated Surge Line for the Compressor Train.*

## MEASURING RATED POINT PERFORMANCE

A field performance test to compare the required power at rated conditions to the vendor guarantee had been planned into the installation from conception, and the intent had been incorporated into the job specifications. A performance test similar to that defined by ASME PTC-10, Class 1 [5] would be conducted, and the vendor would be invited to participate.

Conducting a field performance test as defined by ASME PTC-10 is not very practical because of the instrument redundancy and the number of long straight pipe runs that would be required. Through meetings with the vendor, well in advance of the startup, data gathering and reduction procedures were agreed upon that would produce results accurate within one to two percent.

A schematic diagram of the instrumentation is shown in Figure 19. Motor power was measured with a precision watt-meter. Voltage, current and phase angle data were also recorded. Since the motor has its own substation, the utility power meter provided redundancy. Dry and wet bulb temperatures were taken at the filter house. A weather station within ten miles provided barometric pressure and redundant temperature and humidity levels. Most temperatures and pressures throughout the train were taken at two adjacent locations. The pressure drop through the intercoolers was measured with differential pressure transmitters. The condensate drained from the intercoolers was collected, measured, and averaged for the duration of each data run. Power from the expander was measured by a permanently installed torque meter that had been calibrated to within one percent accuracy.

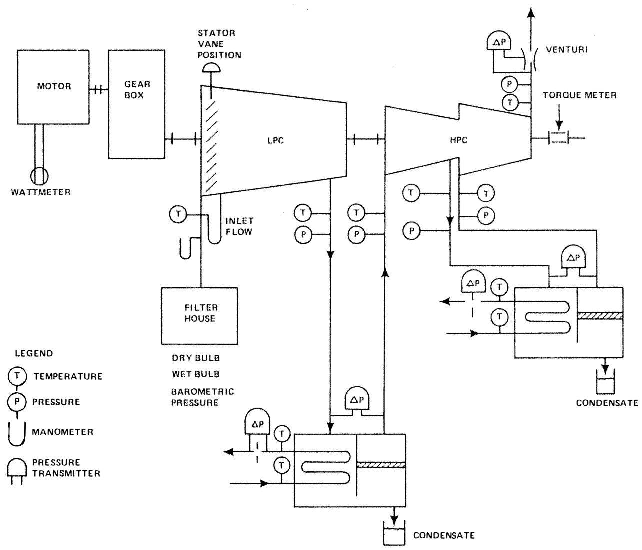


*Figure 19. Compressor Train Schematic Diagram Showing Instrument Locations for Taking Performance Data.*

Precision pressure gauges were acquired and bench tested. All pressure and temperature indicators were tested over the expected operating range immediately prior to the performance test. All errors were either corrected or noted. All critical temperature and pressure indicators were also checked immediately after the test was completed.

Since nature usually will not cooperate and provide design inlet temperature, pressure and humidity conditions, test conditions equivalent to rated conditions were estimated just prior to collecting data. Once the desired test conditions were achieved, the compressor train was allowed to reach equilibrium. Sets of data were taken at five minute intervals until five consecutive sets were taken with all data within these predetermined scatterbands:

| | |
|---|---|
| • Motor power | 1% |
| • LPC inlet pressure | 0.29 psi |
| • LPC inlet temperature | 2.8°F |
| • HPC discharge pressure | 8 psig |
| • Expander power | 1% |
| • Cooling water temperature | 3°F |
| • Cooling water flow | 2% |

The five data sets were averaged and became the data for one test run. Then data for two more test runs were taken at the same conditions. Flow conditions were changed slightly and data for three more test runs were taken on the same stator vane angle curve to bracket the rated point for the train. Test run data were also taken at the HPC surge control line to document inadequate compressor turndown. All test run data points were corrected for known calibration deviations, and power and efficiency calculations were made. The data for equivalent test runs were then averaged. Motor, bearing, gear, and seal losses were estimated using vendor provided data. The measured shaft power was two percent greater than the calculated shaft power. The results were corrected for the design conditions and then plotted.

The performance data for the LPC are shown in Figure 20. The compressor required seven percent to nine percent more power than expected at the rated point; its isentropic efficiency appeared to be low. The low efficiency was possibly caused by blade tip clearances that were at the high tolerance limit and a large balance piston seal clearance that allowed an increased flow of hot air to recirculate to the inlet through a balance line. The primary emphasis was on the combined performance of both compressors because the guarantee was for total power.

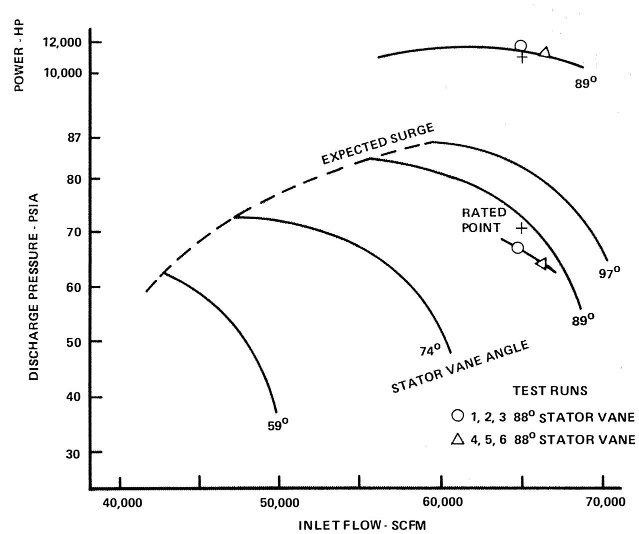


*Figure 20. Comparison of Performance Test Data with Rated Point for the Low Pressure Compressor.*

Corrected performance data for the train are shown in Figure 21. The HPC was more efficient than expected at the rated point so that the power for the train was only one percent to three percent higher than the rated power, which was within the −0 to +4 percent guarantee. The surge control line data point indeed confirmed the limited turndown of the train.

Although turndown was not part of the performance guarantee, the vendor recalculated the aerodynamic performance for the train. It appeared that the surge line for the HPC could be moved to its originally expected position on the head curve without a loss of efficiency at the rated point by changing the diffuser return channel inlet vane angles for the last four impellers in the HPC. The new diffusers have been installed and appear to be performing as expected. Additional high quality performance test data have not been taken, due to production demands.

## SUMMARY

Shaker testing the low tuned concrete foundation in the field identified actual foundation natural frequencies and provided compelling data to change the stiffness of the end springs.

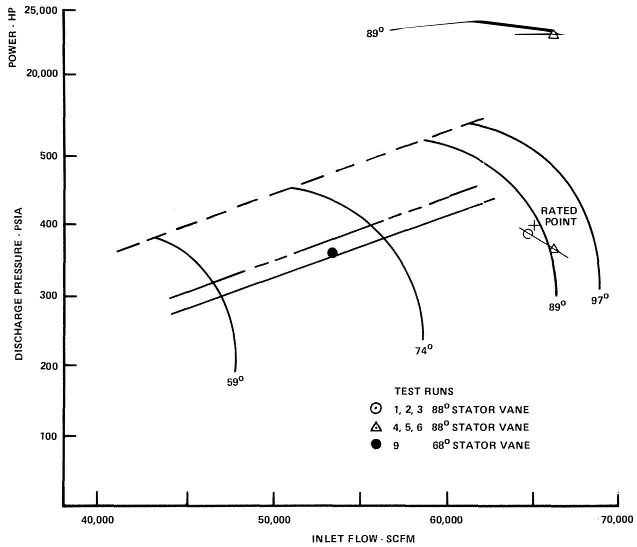


*Figure 21 . Comparison of Performance Test Data with Rated Point for the High Pressure Compressor.*

Measuring the transient torsional torque during startup verified the adequacy of the motor shaft design and confirmed that a resilient coupling was not needed for this installation.

Measurement of stator vane stresses during surge verified the need to interlock the compressor and also provided insight to a compressor operating strategy.

Establishing the actual surge line in the field assured that the maximum attainable compressor turndown was achieved. In this case it also showed that the turndown was less than expected.

The field performance test achieved several things:

• It demonstrated that the vendor met overall capacity and efficiency objectives.

• It documented the inadequate turndown found during surge testing, and economics provided the driving force to revise the diffusers.

• It provided a baseline for the future gauging the internal condition of the compressors and intercoolers, etc.

Several other problems with potentially serious consequences were identified and corrected during the course of the startup testings. They included:

• Differential thermal growth problems in two 20 in surge valves that caused the valves to stick.

• Air flow channeling in the intercoolers that prevented separation of the condensate which was carried into the HPC impellers and would have caused impeller erosion.

• "Worm tracking" in the geartooth coupling between the HPC and the expander that was caused by a coupling design problem.

## CONCLUSION

Each performed test provided valuable insight into the compressor train and either caused a change to improve reliability and operability, or it increased confidence in the existing equipment.

Good vendor design with a parallel user design audit is required for critical process compressor trains. Judicious field testing prior to and during startup is necessary to prove the design and assure reliability.

## ACKNOWLEDGEMENTS

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