

DESIGN AND FULL LOAD TESTING OF A HIGH PRESSURE CENTRIFUGAL NATURAL GAS INJECTION COMPRESSOR

by

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ABSTRACT

This paper discusses the design of a high pressure reinjection centrifugal compressor. This compressor was tested in the manufacturer's shop under full load to simulate the field operating conditions in terms of inlet/discharge pressures, flows, horsepower, gas density, etc. The test data indicates that with the proper design analysis, stable rotor/bearing systems can be designed for high pressure applications. The pressure transducer data verifies that the pressure pulsations at the inlet and discharge of a centrifugal compressor are very small even at high operating pressures. Reliable centrifugal compressors can be manufactured with the use of state of the art technology.

INTRODUCTION

During the past decade, the demand for centrifugal compressors to compress natural gas up to discharge pressures in

the range of 3000-10,000 psig (207-690 bars) has increased rapidly. The centrifugal compressors have been reinjecting natural gas at the Ekofisk Platform, North Sea, at 7000-9000 psig (483-620 bars) since December 1974 [1]. Based on the knowledge and experience gained on these compressors and other high pressure installations, design criteria have been developed for future high pressure applications. The most critical area in the design of these high pressure compressors is the rotor. Extensive analysis is performed during the design phase to minimize downstream problems. Full load testing in the manufacturer's shop further reduces the likelihood of any problems during startup.

During 1977-78, this manufacturer designed and built five low pressure and five high pressure compressors for a reinjection installation in the Middle East to handle 1735 MM SCFD ($2.04 \times 10^6 \text{ m}^3/\text{h}$) of natural gas. The total horsepower requirements of this installation are 125,000 HP (93,200 kW). Each compressor body is directly driven by a GEC Avon gas turbine. The design of these compressors was based on the advanced knowledge of high pressure technology. The scope of supply also included the design of a surge control system. Extensive simulation studies were conducted to insure adequacy of the controls to handle normal as well as upset conditions. Figure 1 shows the equipment arrangement for this installation. The operating conditions for the low and high pressure systems are listed in Table 1.

COMPRESSOR DESIGN

Both the low and high pressure compressors utilized this manufacturer's standard design concepts. Figure 2 shows the cross section of the high pressure compressor with a casing design pressure of 5500 psi (379.2 bars).

Some of the unique design features of these compressors are:

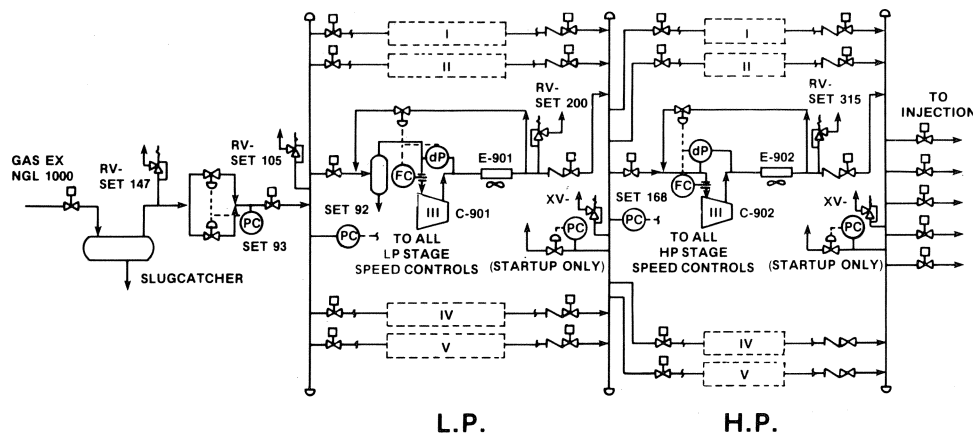


Figure 1. Equipment Arrangement Schematic — Five Low Pressure Compressors and Five High Pressure Compressors.

TABLE 1. OPERATING CONDITIONS FOR THE LOW AND HIGH PRESSURE SYSTEMS.

	L. P.	H. P.
Inlet Pressure — psia (bars)	1326 (91.4)	2391 (164.8)
Inlet Temp. — °F (°C)	99 (37.2)	140 (60.0)
Discharge Pressure — psia (bars)	2411 (166.2)	4038 (278.4)
Discharge Temp. — °F (°C)	195 (90.5)	221 (105.0)
Speed — RPM	6190	6190
Gas	Natural Gas	Natural Gas
Mol. Wt.	19.64	19.64

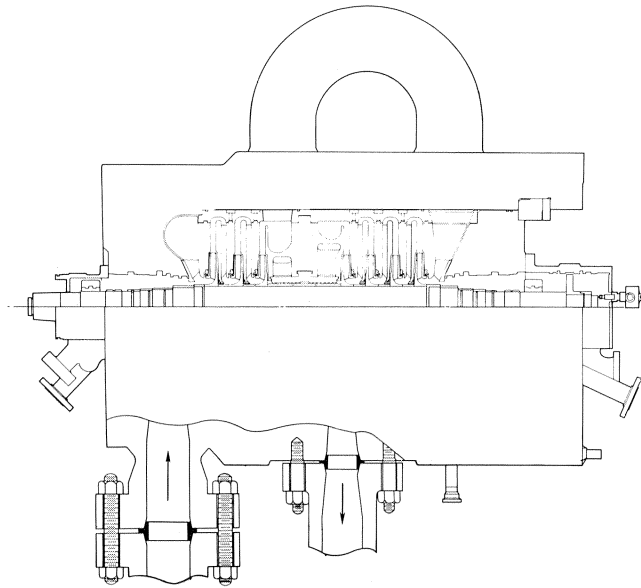


Figure 2. Cross Section of the High Pressure Compressor.

1. Outer Casing and Endwalls

The outer casing is a low alloy carbon steel forging with an integral endwall on one end and a removable endwall design at the opposite end. The removable endwall is retained by an Elliott patented shear ring closure key design. The shear ring keys are retained in an arrangement to balance the forces and moments to minimize the rotation of the key under pressure.

The integral endwall design eliminates the need of an O-ring or an elastomer seal on one end as required in casing designs with shear ring keys on both ends. The solid endwall design thus eliminates the need to remove the endwall to replace the O-ring seal in the field. The solid endwall design is more reliable and provides easier access to the coupling for assembly/disassembly as compared to the closure key design.

The main piping connections utilize a Grayloc type of joint.

2. Inner Casing Assembly

The inner casing and diaphragms are horizontally split for ease of assembly and disassembly. Also, the use of an inner casing eliminates the radial and axial tolerance stack up problem and thus provides more uniform labyrinth clearance along the rotor.

3. Rotor Construction

All impellers utilize a welded design. The impellers are shrunk and keyed on the shaft with sleeves under the labyrinths. All shaft sleeves are shrunk on the shaft. The thrust collar is removable and interchangeable. The thrust collar can be replaced without the need to remove the rotor from the casing.

4. Back-to-Back Impeller Arrangement

Due to axial thrust considerations, the back-to-back arrangement is designed with three impellers in the first section and four impellers in the second section. The back-to-back arrangement is ideal for high pressure applications because it offers the following advantages as compared to the inline arrangement.

- The change in speed and flow through the compressor has an appreciably smaller effect on unbalance thrust. Figure 3 shows that the axial thrust bearing loading variations in the inline arrangement are five times as large as those encountered in the back-to-back arrangement.
- Any increase in labyrinth clearance results in an increase of impeller axial thrust and some decrease of the balancing thrust provided by the balance piston or center seal. The net unbalance thrust caused by labyrinth clearance increase is substantially lower in the back-to-back arrangement than the inline arrangement. Figure 4 shows that 100% increase in labyrinth clearance will result in axial thrust bearing loading changes at 100% flow as follows:

	Thrust Bearing Loading — psi (N/mm ²)	
	In-Line	Back-to-Back
Design Labyrinth Clearance	110 (0.76)	40 (0.275)
100% Larger Labyrinth Clearance	710 (4.89)	110 (0.760)

- The back-to-back arrangement is more efficient than the inline arrangement. The leakage flow across the balance piston labyrinth is considerably

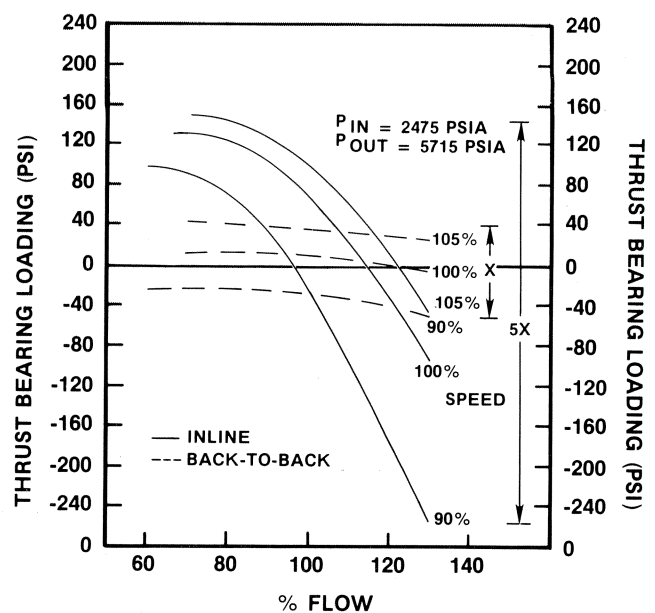


Figure 3. Axial Thrust Loading Comparison of Series and Back-to-Back Impeller Arrangement.

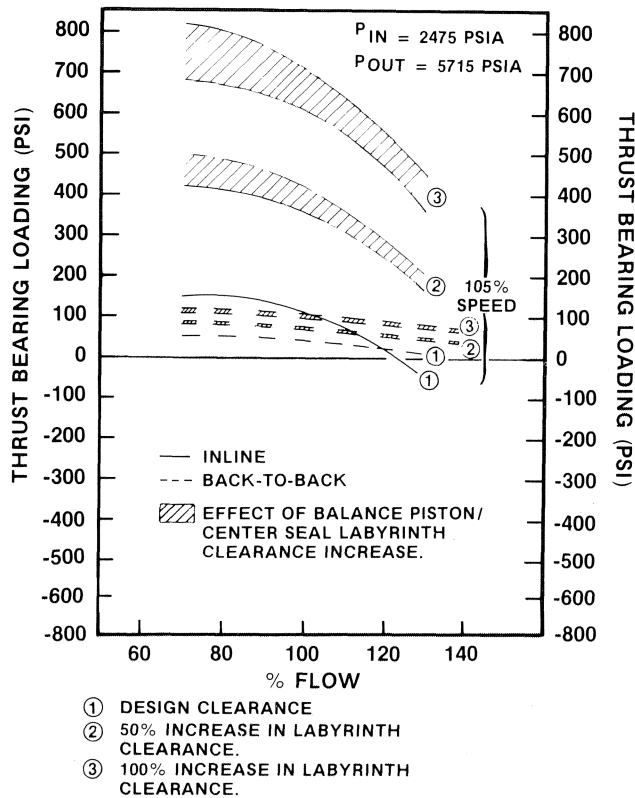


Figure 4. Effect of Labyrinth Clearance Increase on Axial Thrust for Series and Back-to-Back Impeller Arrangement.

higher than the leakage across the center seal. On high pressure applications, the back-to-back arrangement can result in 4-6% lower horsepower than the in-line arrangement.

5. Bearings and Seals

Five shoe tilting pad bearings are used with floating bushing sleeve seals used at each end of the shaft. A pressure balanced breakdown seal design is utilized to minimize seal hangup and its adverse effects on rotor stability. The breakdown seals and the base ring are designed using the finite element analysis techniques.

ROTOR DESIGN ANALYSIS

The most critical area in the design of high pressure compressors is the rotor. Consequently, an in-depth analysis is performed on these rotors to avoid any vibration problems during the test or in the field. The rotors are analyzed for both synchronous and subsynchronous vibration behavior.

Synchronous Vibration Analysis

A detailed rotor response analysis [2] is performed to check the sensitivity of the rotor to static and dynamic unbalances. This analysis includes modelling of the rotor, calculation of bearing characteristics in terms of damping and stiffness, review of the critical speed map, rotor mode shapes and responses due to rotor unbalance. Figures 5 and 6 show that the response is well damped for both static and dynamic unbalances for the entire operating speed. For the static unbalance run, the 0.5g unbalance is located in phase at the middle two impellers. For the dynamic unbalance runs, the 0.5g unbalance was located out of phase at the two outer impellers.

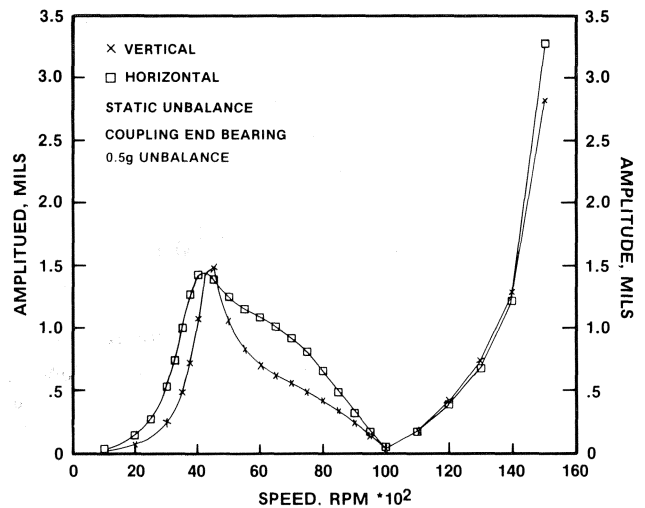


Figure 5. Rotor Response Curve for Static Unbalance. 0.5 g Unbalance was Located at the Middle Two Impellers.

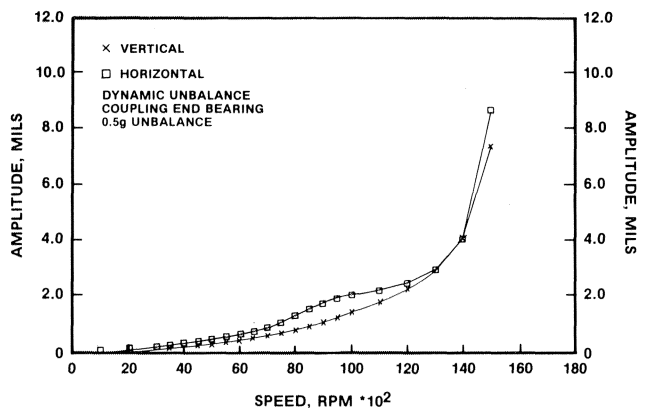


Figure 6. Rotor Response Curve for Dynamic Unbalance. 0.5 g Unbalance was Located at the Outer Two Impellers.

Subsynchronous Vibration Analysis

On all high pressure applications, a detailed analysis of various sources of excitations is performed. Also, a rotor stability analysis is conducted to insure that the rotor will operate satisfactorily in the high gas density atmosphere. A more detailed look at some of these analyses performed on high pressure applications follows.

Aero-Excitation Analysis

Regardless of the aerodynamic stage design, any rotating impeller is bound to create a certain amount of aerodynamic excitation. Of course, this does not imply that an improperly designed stage will not have higher aero excitation. Various mechanisms regarding interaction of aero excitation forces on the rotor dynamics have been theorized. Some of these concepts include: phenomena such as diffuser stall, uneven pressure or work distribution around the outside diameter of the impeller, viscous shear drag of gas, flow through the impeller and interstage shaft labyrinth seals, etc. At high gas density levels, these pressure fluctuations can become severe enough to cause subsynchronous rotor vibrations [3]. The most dominant factor in the magnitude of aero-excitation effects is not the pressure per se, but the gas density which includes parameters

of molecular weight, temperature and compressibility as well as pressure.

The performance calculations of all high pressure applications are reviewed as follows to insure that the aero-excitation sources are kept to a minimum:

- a. The impeller selections are made to provide a sufficient surge margin and a rising head characteristic to insure that there is a minimum possibility of any internal flow instability.
- b. The stage selection is made to satisfy the pressure ratio/stage and diffuser stall criteria. These criteria are primarily gas density dependent as shown in Figure 7.

It must be emphasized that the above criteria are followed to keep the aero-excitation to a minimum to insure maximum rotor stability. As discussed later, the rotor stability is further evaluated with the use of log decrement techniques and rotor flexibility criteria which include both the rotor/bearing characteristics and aerodynamic excitation aspects.

Seal Design Analysis

High pressure applications can have sealing pressure requirements as high as 2000-4000 psi. The breakdown seals and the mating base rings have to be capable of withstanding these high pressures without excessive deflections; otherwise, the seal can hang up radially and will not follow the shaft. A minimum axial loading is maintained on the seal to insure stability of the seal and rotor.

The high pressure seal designs have been analyzed using the finite element method (FEM) to insure satisfactory seal operation. Figure 8 shows the FEM model of the breakdown seal, along with the pressure loading. For a 1750 psi pressure breakdown across the seal, the calculated deflection values are shown in Figure 9.

The seal/rotor interaction effects are calculated for minimum/maximum clearances and eccentricities. This information is incorporated into the rotor stability analysis for further evaluation as is discussed later.

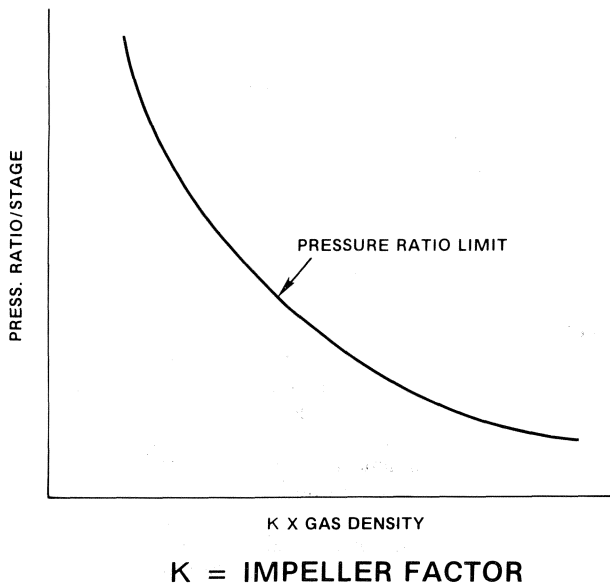


Figure 7. Pressure Ratio/Stage Curve to Minimize Aero-Excitation.

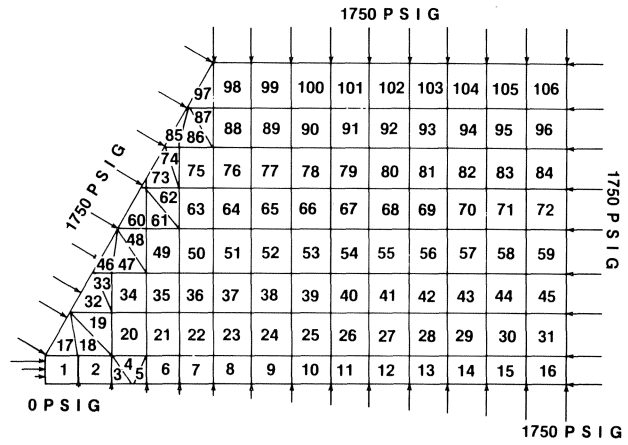


Figure 8. Finite Element Method (FEM) Model of the Breakdown Seal under 1750 psig Pressure.

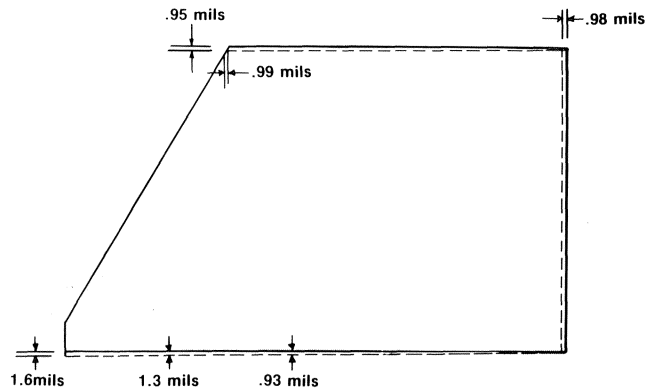


Figure 9. FEM Calculated Deflection Values of the Breakdown Seal under 1750 psig Pressure.

Rotor Mode Shape Analysis

In rotor instability problems, the subsynchronous vibrations most commonly occur at a frequency in the proximity of first bending critical speed [1, 5]. The rotor mode shapes are reviewed to insure that the rotor node points are not located close to the bearing locations. Figure 10 shows that the rotor in the first bending mode has adequate displacement at the bearings, thus indicating that the rotor bearing system will

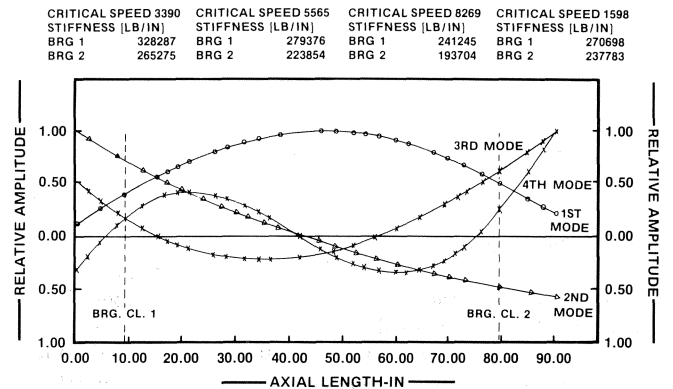


Figure 10. Undamped Rotor Mode Shape Plots of the High Pressure Rotor.

provide the necessary effective damping to control any tendency of the rotor to vibrate in the first bending mode.

Log Decrement Analysis

The next step in the analysis is to check the stability of the rotor bearing system. This analysis involves a detailed calculation of damped natural frequencies of a rotor supported in fluid film bearings. Also, this analysis can provide the stability margin of the rotor in terms of log decrements for various damped natural frequencies. The program has the capabilities of incorporating the destabilizing effects of such items as aero-excitation, seals and internal friction. The subject analysis is called "Log Decrement" analysis [4].

A detailed "Log Decrement" analysis was performed on these compressors. The log decrement associated with the first damped natural frequency of 3922 cpm evaluated at the operating speed of 6500 rpm was 0.443. The rotor was found to be insensitive to the effects of seal interaction. Based on the expected value of aerodynamic cross coupling stiffness value of 6570 lb/in. applied at each impeller location, the log decrement associated with the first damped natural frequency of 3704 cpm was 0.19. These log decrement values were deemed to be more than adequate as has been substantiated by the successful running of the compressor during the full load test.

General Design Criteria

In addition to the detailed Log Decrement Analysis, some general design criteria have been developed for the rotor stability evaluation of high pressure rotors.

The rotor flexibility ratio is defined as the ratio of maximum continuous speed/the first critical speed on stiff supports. A low flexibility ratio means a more rigid rotor. Experience has shown that rotors with a low flexibility ratio are less susceptible to subsynchronous vibration in high density applications. It should be noted that in low pressure applications, the rotors can be operated satisfactorily with a high flexibility ratio. Based on our experience of many high and low pressure installations, a rotor flexibility ratio curve has been developed for various gas density levels as shown in Figure 11.

It must be noted that the state of art does not provide any single criteria to be used in the evaluation of rotor stability. Any rotor stability investigation must include a design review to insure that the destabilizing forces are kept to a minimum with the proper selection of compressor hardware. An evaluation of the system stability can be made with the use of rotor flexibility criteria and a log decrement analysis.

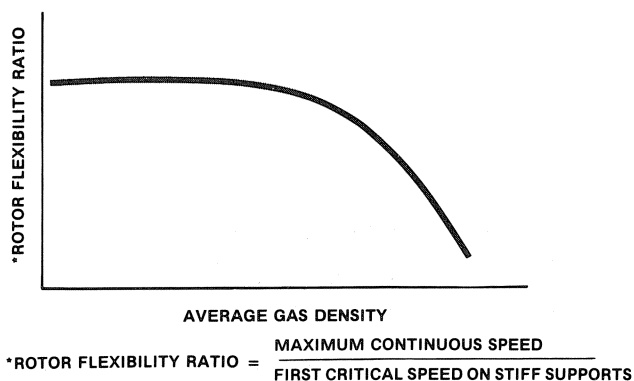


Figure 11. General Rotor Stability Criteria of Rotor Flexibility Ratio vs. Average Gas Density.

TESTING

The five low pressure and five high pressure bodies were tested in accordance with API 617, Paragraph 7.3, Mechanical Test requirements. One low and one high pressure body received an ASME PTC 10 code performance test. In order to minimize any problems at the job site, one low and one high pressure system were full load tested. In this section, the data taken on the high pressure system is discussed in detail.

Full Load Testing

The full load test conditions were selected to simulate the field operating conditions. During the full load testing, important operating parameters such as gas density, pressure, horsepower, etc., exceeded the field operating conditions as indicated in Table 2. These parameters mainly affect the stability of the rotor bearing system.

TABLE 2. OPERATING PARAMETERS DURING FULL LOAD TESTING.

	Design	Test
Inlet Flow — CFM (m ³ /h)	1410 (2395)	1262 (2144)
Inlet Pressure — psia (bars)	2391 (164.9)	2350 (162.0)
Inlet Temp. — °F (°C)	140 (60.0)	120 (48.9)
Discharge Pressure — psia (bars)	4038 (278.5)	4515 (311.3)
Discharge Temp. — °F	221 (105)	280 (137.8)
Avg. Gas Density — #/c.ft. (kg/m ³)	10.0 (160.1)	11.94 (191.1)
Power — HP (kW)	12150 (9059)	13482 (10052)
Speed — RPM	6190	6190

The job coupling, seals and seal oil systems were used during the full load test. Figure 12 shows the high pressure compressor coupled to the shop turbine on the test stand.

Instrumentation

The compressor was equipped with two radial proximity pickups at each bearing location to measure vibration. One axial probe to monitor the shaft position and one keyphasor probe to monitor running speed and phase angle were also provided. Pressure transducers were installed near the inlet

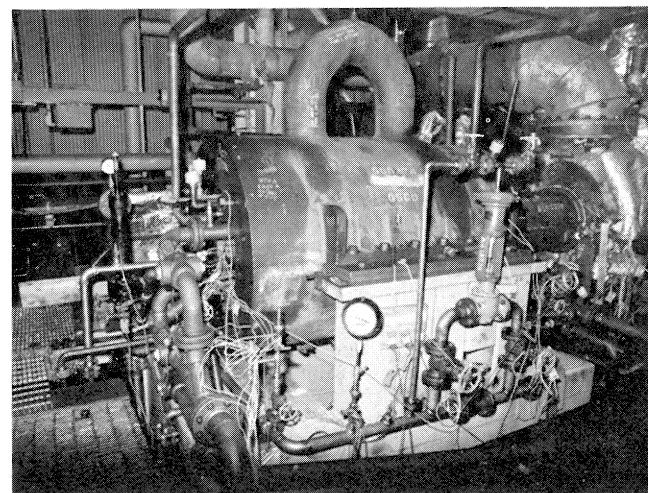


Figure 12. Full Load Testing of H.P. Compressor.

and discharge to measure pressure pulsations. A real time analyzer was used to analyze the vibration and pressure pulsation signals. For accurate analysis of low frequency transient signal during surge, a high speed visicorder was used in conjunction with the real time analyzer. All proximity probe and pressure transducer data was tape recorded for in-depth analysis at a later date.

Rotor Vibration and Pressure Pulsation Data

During the entire full load test period, the vibration levels on each end of the compressor were less than 1 mil (Figure 13). The frequency analysis of the vibration signal indicates that the major component of vibration was at the running speed frequency and with small amplitudes at the multiples of running speed frequencies. Figure 14 shows the vibration signal at the thrust end of this unit taken at the 4500 psia (311.3 bars) discharge pressure condition. The vibration amplitudes shown on this curve include mechanical and electrical runouts. The vibration signal indicates the absence of any subsynchronous vibration. As expected, the pressure pulsation data showed a small peak at the blade passing frequency, $21 \times$ speed (Figure 15). The low frequency data 0-250 Hz is shown on a more expanded scale.

During the full load test, the compressor was run over the complete flow range — surge, design and overload. The coupling end vibrations were below 0.3 mils and the pressure pulsations at the inlet were significantly lower than at the discharge as discussed in detail later. Figure 16 shows the thrust end vibration and pressure pulsations at the discharge for surge, design and overload conditions.

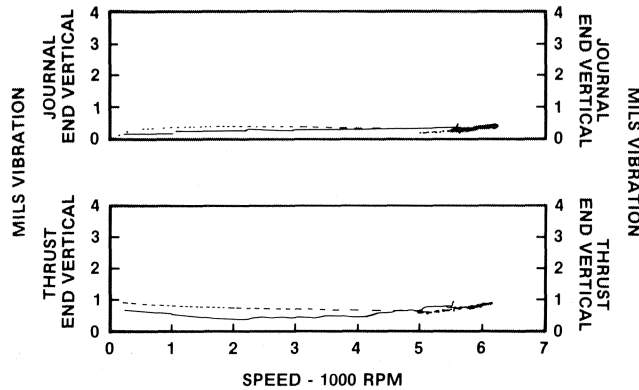


Figure 13. Vibration Amplitudes Peak-to-Peak During Full Load Test.

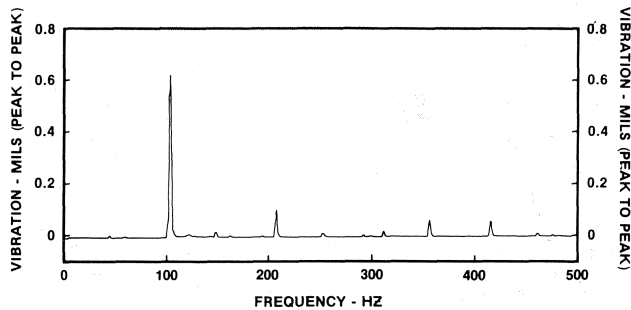


Figure 14. Thrust End Vibration Signal Analysis at 4500 psi Pressure.

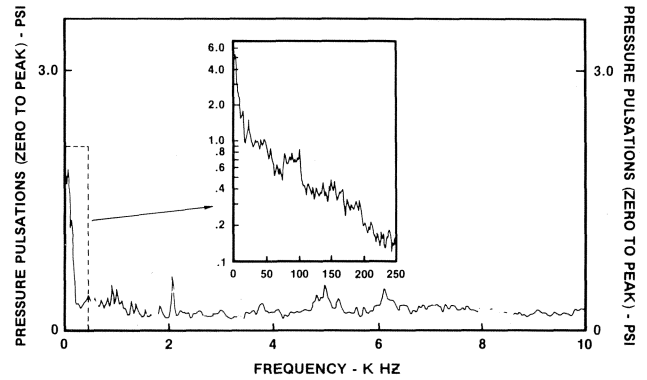


Figure 15. Pressure Pulsations at the Discharge at 4500 psi Pressure.

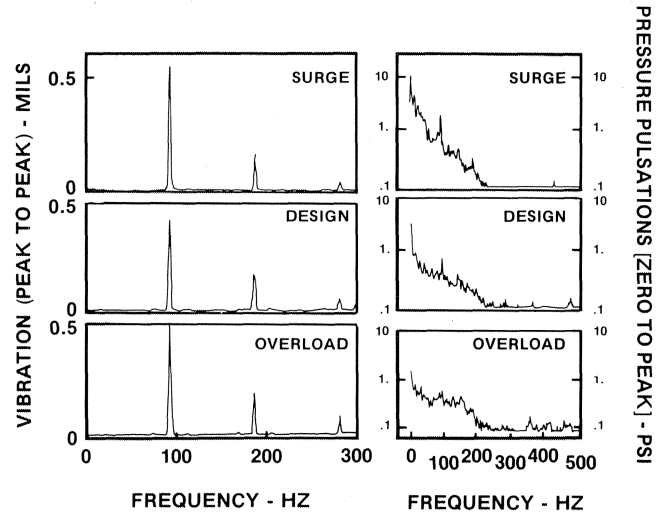


Figure 16. Thrust End Vibration and Pressure Pulsations at the Discharge During Surge, Design and Overload.

The pressure pulsation data is shown for 0-500 Hz range for better resolution of the low frequency data. As the compressor flow is reduced towards surge, the pressure pulsations increase in the low frequency region 0-50 Hz. As mentioned earlier, a high speed visicorder was used for transient signal analysis during actual surge. Figure 17 shows the pressure pulsations at design and during surge. The maximum pressure pulsations (peak-to-peak) were:

	Inlet	Discharge
	psi (bars)	psi (bars)
At Design	3.5 (0.24)	25 (1.72)
At Surge	70 (4.82)	250 (17.24)

The surge cycle appears to occur at a frequency of 0.7-0.8 Hz.

The analysis of high frequency pressure pulsation data indicated that the pressure pulsations at the blade passing frequency were the lowest at the overload conditions and they gradually increased towards surge.

During the full load test, the unit was exposed to many startups, shutdowns and surging to simulate the field operation. It should be noted that the rotor bearing system exhibited excellent stability under all operating conditions, even in the area of surge.

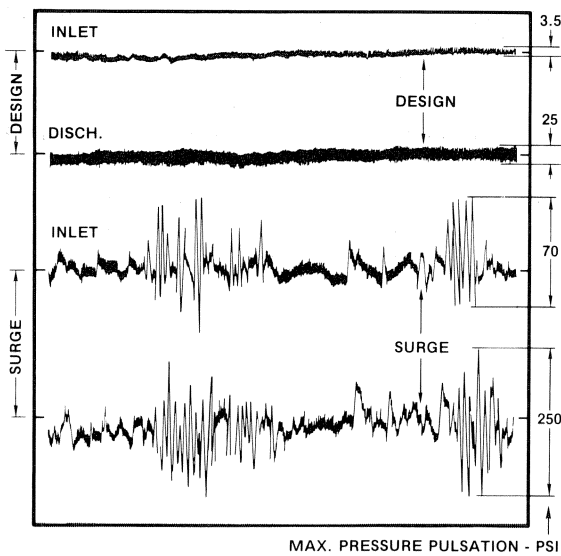


Figure 17. Comparison of Pressure Pulsations at Design and Surge.

Extensive noise data was taken around the compressor and piping using special instrumentation as well as standard octave band analyzers. Airborne noise level generated by the compressor casing was 80-84 dBA at one meter distance.

AERO PERFORMANCE

The aero hardware used on these compressors had been developed in the late sixties with the aid of extensive analytical calculations and single stage testing. Over the years, the performance ratings of these stages have been verified on many applications. The recent high pressure testing provided an excellent opportunity to test the performance of these stages at high pressures under controlled test conditions. Figure 18 shows that the actual performance came very close to the pre-

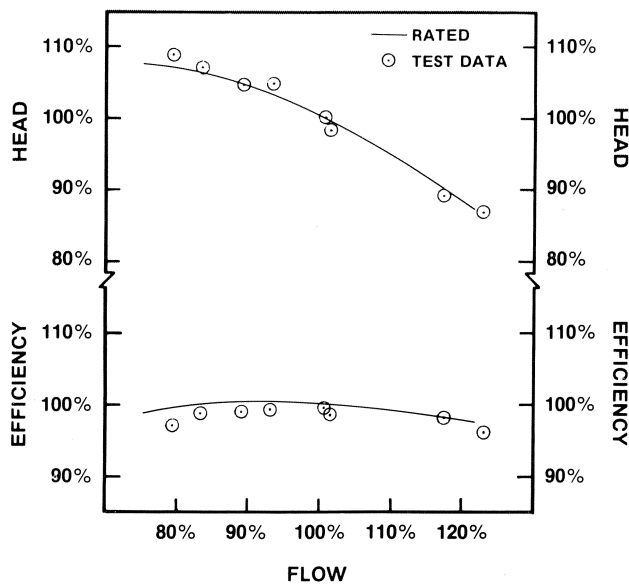


Figure 18. Comparison of Predicted and Actual Performance Based on Full Load Test.

dicted performance. This excellent correlation provides confirmation in the following areas.

1. Base stage performance ratings.
2. Benedict Webb Rubin (BWR) method of gas property calculations [5].
3. Prediction of stage performance at high pressures.

BEARING AND SEAL PERFORMANCE

During the full load tests, the bearing and seal performance was checked. The measured oil flows and horsepower losses were within expected tolerances of the predicted values. The thrust bearing embedded thermocouple data confirmed that the axial thrust in the back-to-back arrangement is quite insensitive to changes in flow and speed (Figure 19). The thrust bearing temperature rise data is based on the average of three embedded thermocouple readings. At 4500 psi discharge pressure condition, the temperature rise across the three thermocouples was 44°F, 46°F and 51°F or 47°F average.

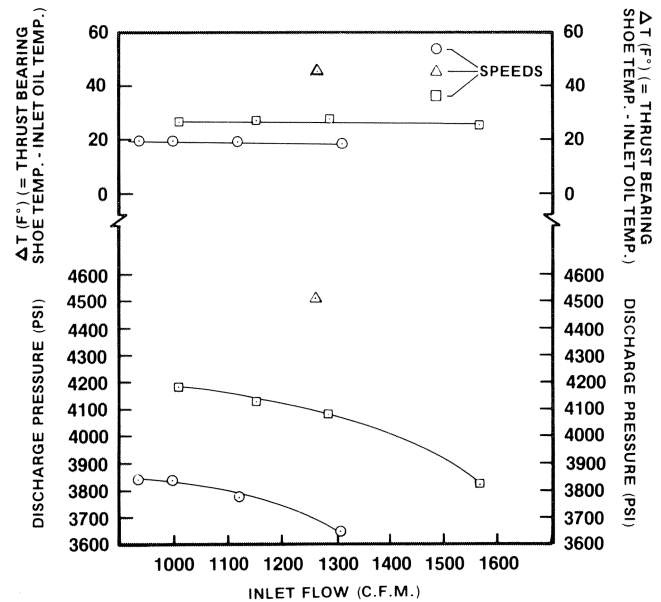


Figure 19. Thrust Bearing Embedded Thermocouple Data Taken During Full Load Test.

CONCLUSIONS

The successful testing of these compressors has indicated that reliable high pressure centrifugal compressors can be designed and manufactured with the use of "state of the art" technology. The full load testing has provided the necessary confirmation of the present analytical techniques for the design of high pressure compressors. The aerodynamic hardware of high pressure compressors can be designed and selected to result in minimal aero-excitation. The rotor/bearing systems can be designed with high effective damping to control the rotor vibrations. The inherently better axial thrust balancing feature of the back-to-back arrangement makes this design ideal for high pressure applications with large pressure rise across the compressor.

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