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FULL LOAD SHOP TESTING OF 18 000-hp GAS TURBINE DRIVEN CENTRIFUGAL COMPRESSOR FOR OFFSHORE PLATFORM SERVICE: EVALUATION OF ROTOR DYNAMICS PERFORMANCE

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The results for in-plant full load testing of a 13.4 MW (18000 HP) gas turbine driven centrifugal compressor are presented and compared to analytical predictions of compressor rotor stability. Unique problems from both oil seals and labyrinth gas seals were encountered during the testing. The successful resolution of these problems are summarized.

SYMBOLS

Values are given in both SI and U.S. Customary Units. The measurements and calculations were made in U.S. Customary Units.

a, A	real part of eigenvalue, growth factor (sec^{-1})
N	rotor speed, Hz (RPM)
N_{cr}	rotor 1st critical speed (CPM)
N_d	rotor damped critical speed (CPM)
P_2	compressor discharge pressure, N/m^2 (lb/in^2)
P	pressure rise across compressor, N/m^2 (lb/in^2)
Q	aerodynamic cross-coupling, N/m (lb/in)

INTRODUCTION

The design and testing of centrifugal compressors have demanded increased interest and concern as a result of documented cases of compressor vibration instability when actual startup conditions are first encountered (1,2,3,4).

For this reason, two 13.4 MW (18000 HP) aircraft derivative gas turbine driven compressor trains for North Sea platform service were recently purchased with requirements for standard four-hour mechanical shop tests and full load performance testing. Actual contract gas composition was required to satisfy the customer specifications. The facility for this full load testing was constructed in the United Kingdom. Figure 1 shows this test facility with one of the two enclosures and associated hardware.

The contract gas moleweight was 18.3 and consisted mainly of methane. The discharge pressure was 7585 kPa (1100 psig) and the suction pressure was varied from 4137 kPa (600 psig) at 70% speed to 1724 kPa (250 psig) at 100% rated speed to simulate field conditions for year 1 through year 6 operation as the gas field pressure decays. This paper is concerned specifically with the rotor dynamics evaluation of the mechanical and full load tests of the two seven stage series flow centrifugal compressors for the year six, 100% speed condition. The relative position of this design to past experience at the authors' company is shown in Figure 2. The log decrement for the compressor supported on its bearings and without any other destabilizing element was 0.144 which was similar to numerous other units running without vibration problems.

EVALUATION OF OIL SEAL PERFORMANCE

One of the major components known to cause instability in high pressure compressors is the oil ring type seal. The original design of the seal for the subject compressor is shown in Figure 3. The predicted worst case stability parameters for this design are shown in Table 1 as growth factor for various operating speeds considering both soft start and hard start conditions (3). The year six condition is used for stability work since this condition had the highest speed and the highest pressure rise across the compressor. The initial year 6 design speed was 194.2 Hz (11650 RPM). A growth factor greater than +50 has been found to cause unbounded vibration. The authors' company has used a value of +15 as the upper limit of acceptance if seal and aerodynamic instability mechanisms are included in the analysis.

The mechanical testing of the compressors revealed unexpected problems from the oil seals. The outer seal lapped surface that contacts and seals to the housing was fretted after a short running time as illustrated by the typical ring shown in Figure 4. The vibration characteristic for this ring failure mode is shown in Figure 5. The low frequency oil ring whirl instability would typically start when compressor speed was just over 100 Hz (6000 RPM) and track with compressor speed. This subsynchronous vibration would vary in magnitude and frequency with typical frequencies from 0 to 30 Hz.

Re-evaluation of the oil seal ring resulted in reducing the sealing length and width of the lapped face, and increasing the steady holding force of the springs to attempt to both stop the low frequency ring whirl and to improve the locked ring rotor stability. Table 2 summarizes this re-design condition which has much better stability for both soft and hard start conditions. Table 3 indicates that even with 1751 N/mm (10,000 lb/in) cross-coupling at midspan the design had acceptable stability for normal seal ring holding force. Further operation produced unacceptable results due to fretting damage and the outer seal was further modified to the final configuration shown in Figure 6. The predicted stability for three seal designs versus rotor speed are plotted in Figure 7 and compared to the condition without the influence of oil seals. The interaction of the aerodynamic excitation is plotted in Figure 8 for the re-design and final oil seal design as compared to the condition without oil seal ring influence. The final modification was made to the seal assembly procedure to assure that the casing end plate that mates to the oil ring lapped face was not distorting upon assembly. This combination of oil ring seal design and modified assembly procedure produced results that did not experience either fretting damage or oil seal re-excitation of the compressor 1st bending critical.

RESULTS OF FULL LOAD TESTING - AERODYNAMIC EXCITATION

The full load test facility was constructed during the same time period that the mechanical tests were being conducted. The first run up for the full load facility using inert gas with a moleweight of approximately 14 was made on June 24, 1984. The entire test facility was functional and the vibration looked very promising as indicated by all the vibration frequency scans. The thrust end horizontal spectrum plot given in Figure 9 was typical and indicated the absence of nonsynchronous vibration at the oil seal whirl frequency and very small components at or near the first rotor critical speed frequency. The entire project was considered to be satisfactory at that time.

The first run on contract "live" gas with a moleweight of 18.3 occurred on July 3, 1984. As indicated by Figure 10, a compressor speed of only 170 Hz (10,200 CPM) was possible before the vibration grew in excess of the overall acceptance level with re-excitation at the first critical frequency going as high as .038 mm (1.5 mils). Upon disassembly, no damage was found on the rotor. These results prompted the use of a standard balance piston shunt line modification to essentially reverse the flow on the back of the disk of the last stage impeller. A single feed line and circumferential groove was used to distribute the flow into the leading edge of the balance piston. This modification allowed maximum continuous speed to be obtained. The vibration had sub-synchronous pulsations with the steady levels at just over 0.6 mil and occasional spikes to 0.9 mil (see Fig. 11). The vibration was bounded and sustained operation was possible but the contract specification of .01 mm (0.4 mil) sub-synchronous was not satisfied. At that time and without bringing the unit down in speed, the loop gas was reduced in moleweight from 18.4 to 15.5. The compressor vibration reduced to acceptable levels indicating without doubt the aerodynamic origin of the excitation. These results gave increased evidence of the possibility that the balance piston labyrinth swirl excitation was a major contributor to the sub-synchronous instability. A unique modification was designed to produce a uniform swirl in the leading teeth of the balance piston. The orifice jets were directed against rotation to further negate or reduce any forward swirl of the balance piston gas. This modification was installed and on August 8, 1984, the compressor was capable of full speed operation with acceptable nonsynchronous vibration (see Fig. 12). A cross-section of this particular balance piston modification is given in Figure 13. This design is now referred to as the NEGASWIRL Insert (patent pending).

The results for the thrust end probes for year six design point are shown in Figure 14 where the average sub-synchronous levels are indicated to be less than .005 mm (0.2 mil). The compressor aerodynamic performance mandated that a new lower design speed be selected to match the field required flow and pressure condition. The results for the second unit for its new year six design point (N = 10622 CPM = 177 Hz) is shown in Figure 15. The nonsynchronous vibration levels are indicated to be only slightly higher than the unit one levels, while the synchronous component was lower indicating a slightly better dynamic balance condition for unit two. The overall vibration levels were within contract specification.

EVALUATION OF DESIGN STANDARDS FOR CENTRIFUGAL COMPRESSORS

The design standard from reference (5) has been overplotted with the inert gas run, the run limit without NEGASWIRL, and the operation with NEGASWIRL. It is evident that operation less than the unacceptable line can only be achieved by the incorporation of 1) stabilized oil seal designs and 2) balance piston and other labyrinth seal designs that enhance stable operation. The general slope of this standard is believed to be valid and the region to the left of the design limit bandwidth is considered to be fully achievable with standard design features.

Upon completion of the unit one testing on this order, a new computer analysis for labyrinth seals was developed following the theory and general analysis of Iwatsubo (6) and Childs (7). The results of the stability of the compressor using each labyrinth seal stiffness and damping characteristic as calculated from this new program is given in Table 4. The influence of the moleweight reduction from 18.3 to 14 drops the growth factor from 7.3 to 5.5. Removing the balance piston influence results in a growth factor of 2.4 for the 18.3 moleweight condition.

The vibration spectrum from these units and other high pressure gas compressor units operating in the field have common characteristics. The nonsynchronous vibration pulses in a nonsteady fashion with occasional larger spikes that are quickly suppressed. To reject a machine for an occasional rise over the specification which is proven to be bounded and harmless is not realistic. To standardize the acceptance of units in the field and machines on full load test, the following standard has been developed to judge the acceptance of nonsteady sub-synchronous vibration.

The magnitude of any discrete nonsynchronous vibration shall not exceed 40% of the standard overall vibration acceptance limit (with $N_{mcos} = RPM$):

Compressors:

$$R_{non-syn} \text{ (mil peak to peak) } \leq 0.4\sqrt{12000/N_{mcos}}$$

Turbines and Expanders:

$$R_{non-syn} \text{ (mil peak to peak) } \leq 0.4 \times 1.15 \times \sqrt{12000/N_{mcos}}$$

The magnitude of the discrete frequency component shall be measured using eight (8) or more exponential averages. A frequency range of 0-250 Hz shall be used for units with design speeds less than 7500 RPM, 0-500 Hz shall be used for units with design speeds greater than 7,500 but less than 15,000 RPM, and 0-1000 Hz shall be used for units having design speeds greater than 15,000 but less than 40,000 RPM. The frequency spectrum shall be monitored at the maximum continuous operating speed. If the nonsynchronous vibration is consistent to the point that the average level is above 40% of the acceptable overall level, alternate designs should be considered to enhance the rotor stability.

RECOMMENDATIONS

The following recommendations can be given as a result of the analysis and results of the full load testing reported in this paper.

1. It is essential to test under the actual gas moleweight condition to verify compressor stability.

2. The analysis of labyrinth seal characteristics are required to predict the aerodynamic excitation of the impeller labyrinth seals and the balance piston.

3. The stability analysis should consider locked seal excitation for rotor stability. Oil seal holding force must be adequate to prevent seal whirl which can result from low compressor suction pressure or seal element distortion.

4. The upper design limit for growth factor is considered to be +15 (log decrement of -.07) when oil seal and aerodynamic excitations are accounted for. The value of zero growth factor should remain the design limit for new machinery.

5. Acceptance specification for nonsynchronous vibration under field or simulated field conditions should be based on an average level as proposed in this paper. The overall vibration limit, warning limit, and trip level are not altered by these considerations.

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	C-radial (in.)	N(RPM) = 5000	7500	10000	12500
TEST 70 PSI	.003	-45.8	-13.5	1.5	10.2
FIELD 250 PSI	.005	-55	-10	-2.65	12.8

TABLE 1 - STABILITY GROWTH FACTORS WITH SEALS AT PREDICTED OPERATING CLEARANCE FOR TEST AND FIELD CONDITIONS ORIGINAL SEALS: 1.25 IN. OUTER SEALS

	C-radial (in.)	N(RPM) = 5000	7500	10000	12500
TEST 70 PSI	.003	-35.2	-15.3	-4.95	1.17
	.004	-31.9	-15.7	-7.15	-1.83
	.005	-30.2	-15.4	-7.58	-3.2
FIELD 250 PSI	.003	-54.5	-11.6	10.0	22.6
	.004	-46	-15.8	2.07	13.5
	.005	-39.9	-17.9	-3.5	6.4

TABLE 2 - STABILITY GROWTH FACTORS FOR TEST AND FIELD CONDITIONS FOR THREE SEAL CLEARANCE CONDITIONS REDESIGN SEALS: REDUCED SEAL LIP O.D.; 1.0 IN. OUTER SEAL

Q(lb/in)	100	1000	10000	50000
Worst Case	3.38	4.22	12.3	45.5
Normal Conditions for Seals	-.3	.5	8.39	40.8

TABLE 3 - STABILITY GROWTH FACTORS FOR WORST CASE AND NORMAL SEAL LOADING CONDITIONS REDESIGN SEALS: INFLUENCE OF Q - AERODYNAMIC CROSS-COUPLING

----Condition----		Total	W/O Bal. P.	W/O B.P. or Oil Seals
Baseline	Aero=0.0	-2.6	-2.6	-6.5
Labyrinth Analysis	MW=14	5.5	1.59	-2.4
	MW=18.3	7.3	2.37	-1.7

TABLE 4 - RESULTS OF GROWTH FACTOR AT N = 194.17 HZ (11650 RPM) NOTE: 1.0 LB/IN = 0.175 N/mm

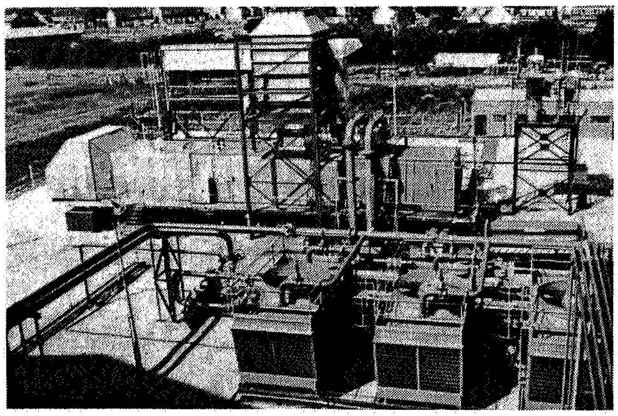

Figure 1. - Full-load test facility.

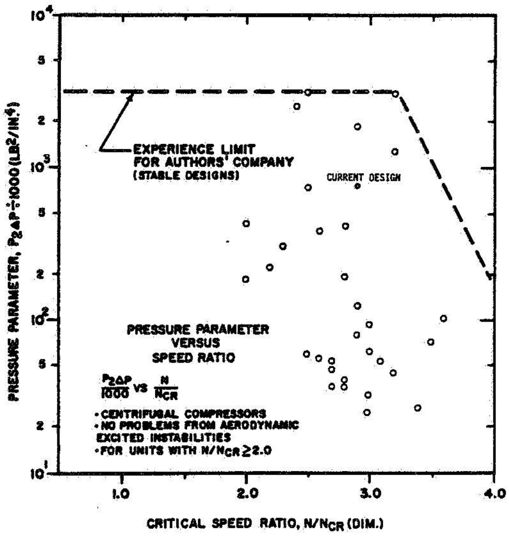

Figure 2. - Current design conditions.

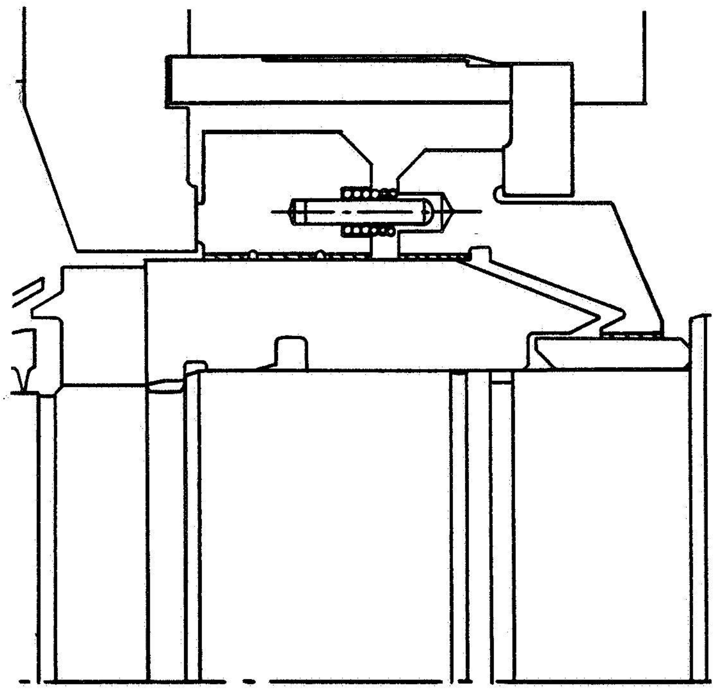

Figure 3. - Typical cone seal.

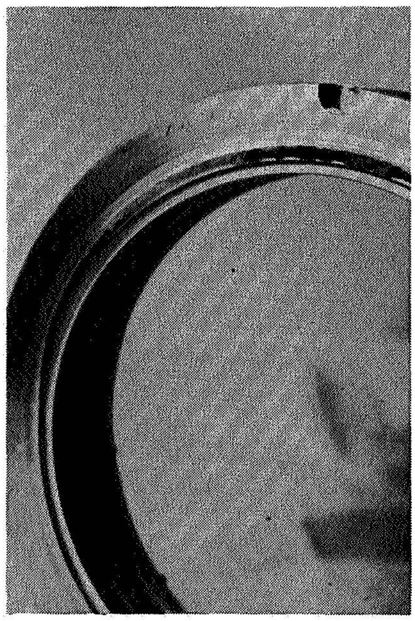

Figure 4. - Damaged seal ring.

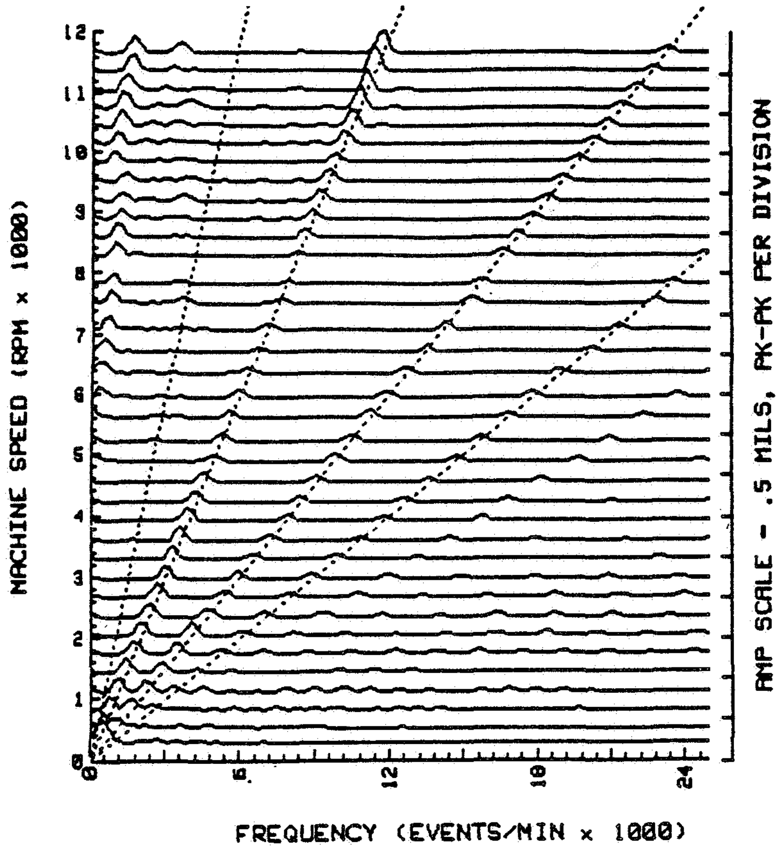

Figure 5. - Vibration resulting from oil ring seal whirl.

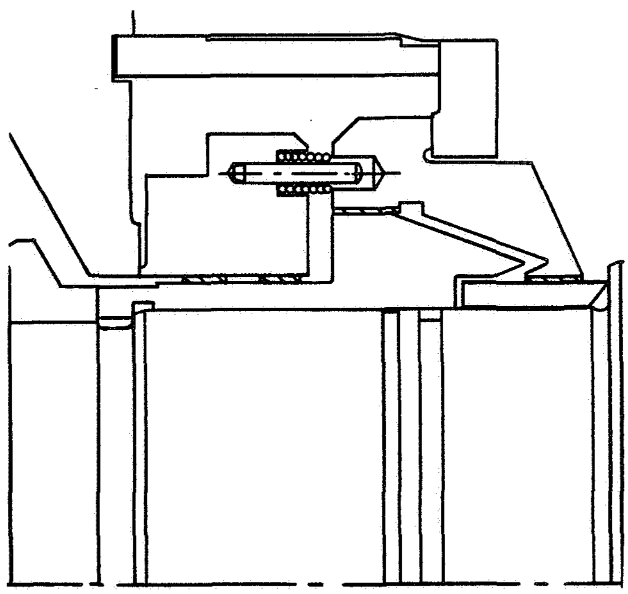

Figure 6. - Pressure balanced seal.

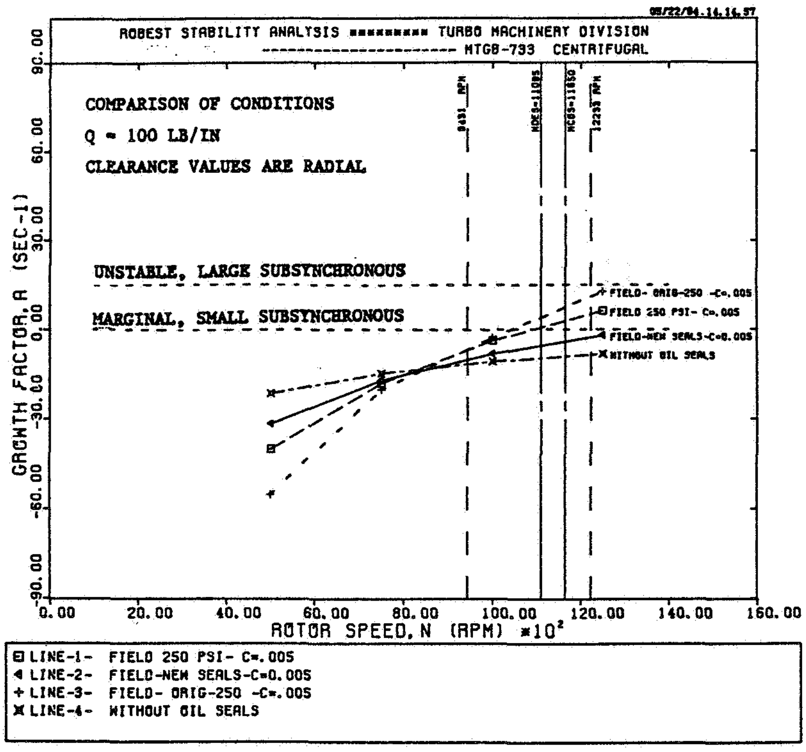

Figure 7. - Comparison of seal type versus rotor speed.

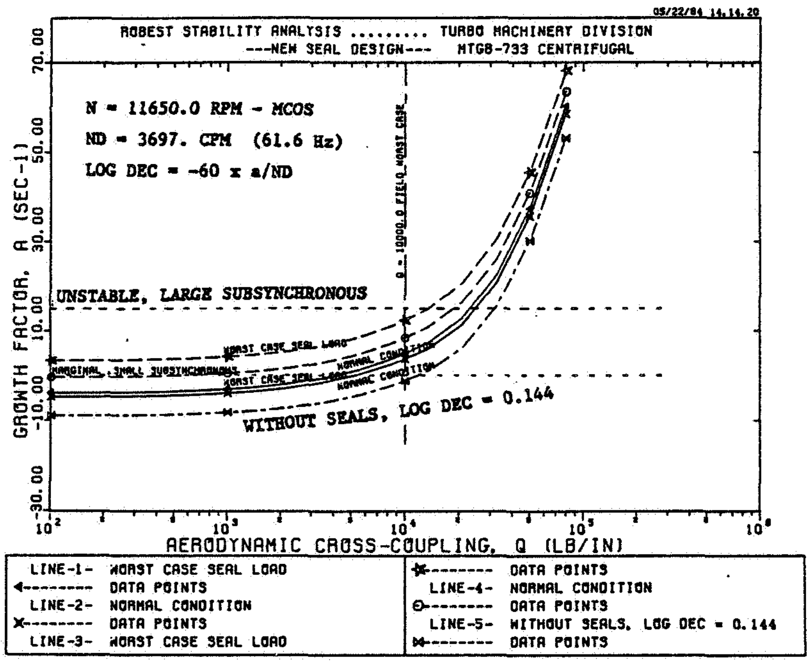

Figure 8. - Influence of aerodynamic excitation at M.C.O.S.

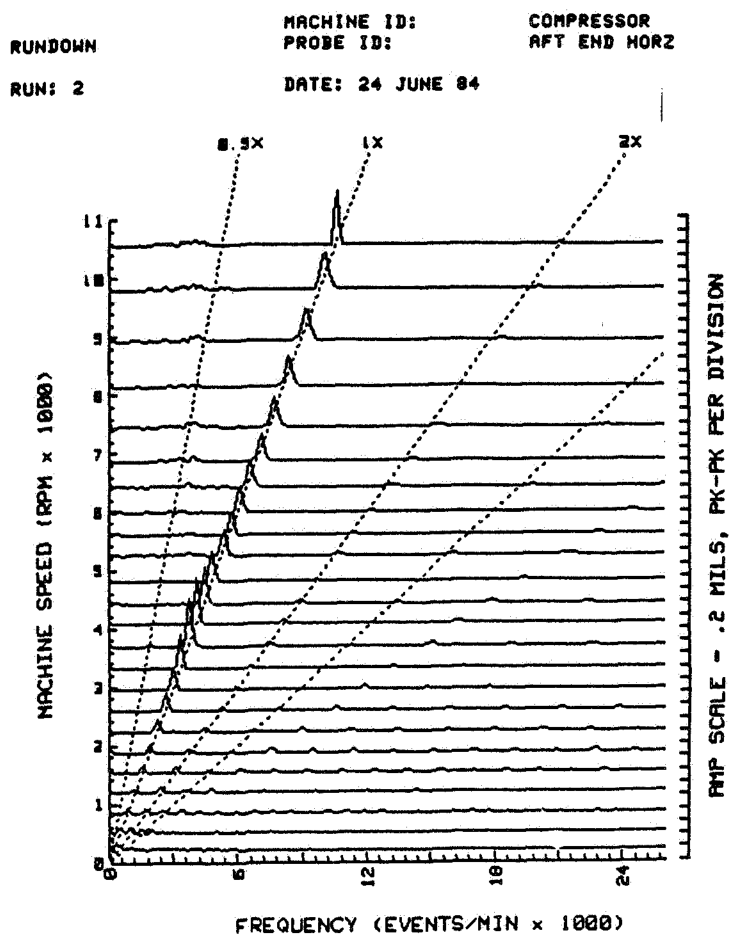

Figure 9. – Rundown with inert gas.

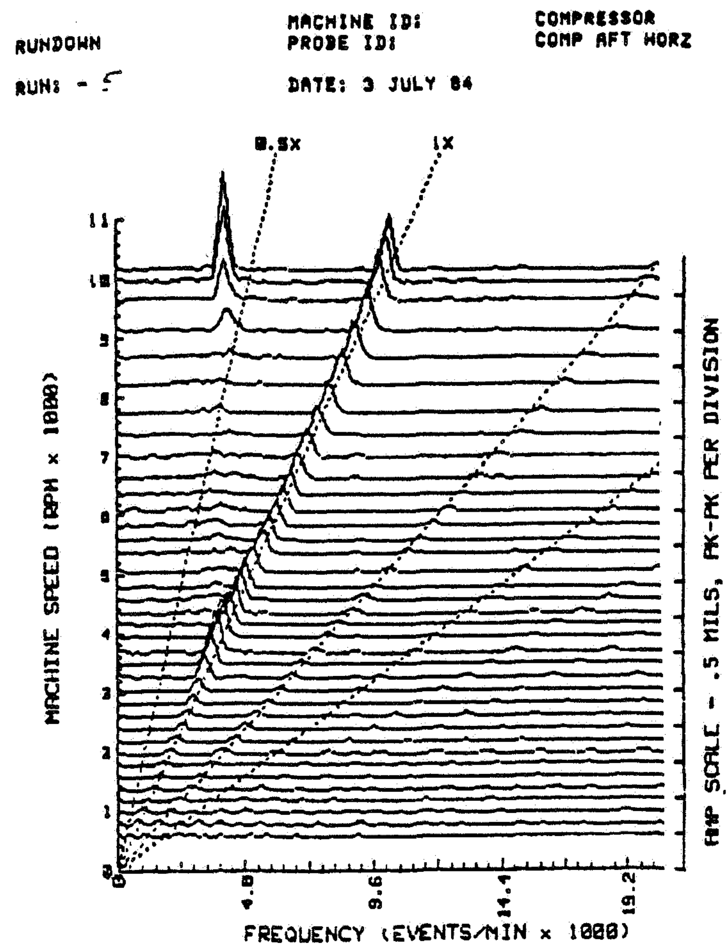

Figure 10. – Rundown with live gas.

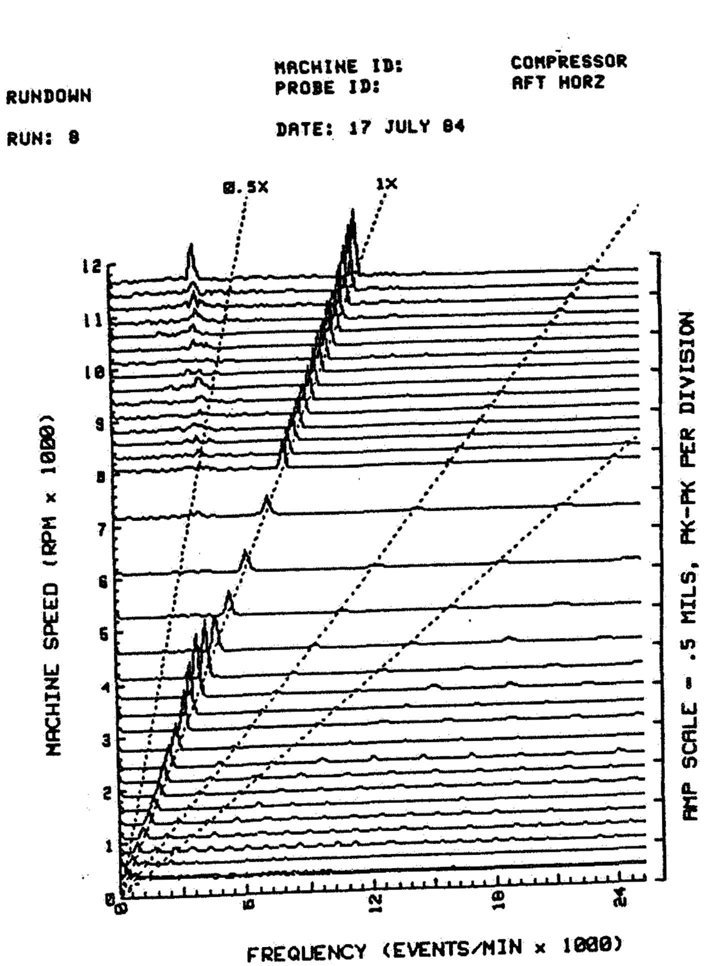

Figure 11. - Rundown with balance piston shunt line.

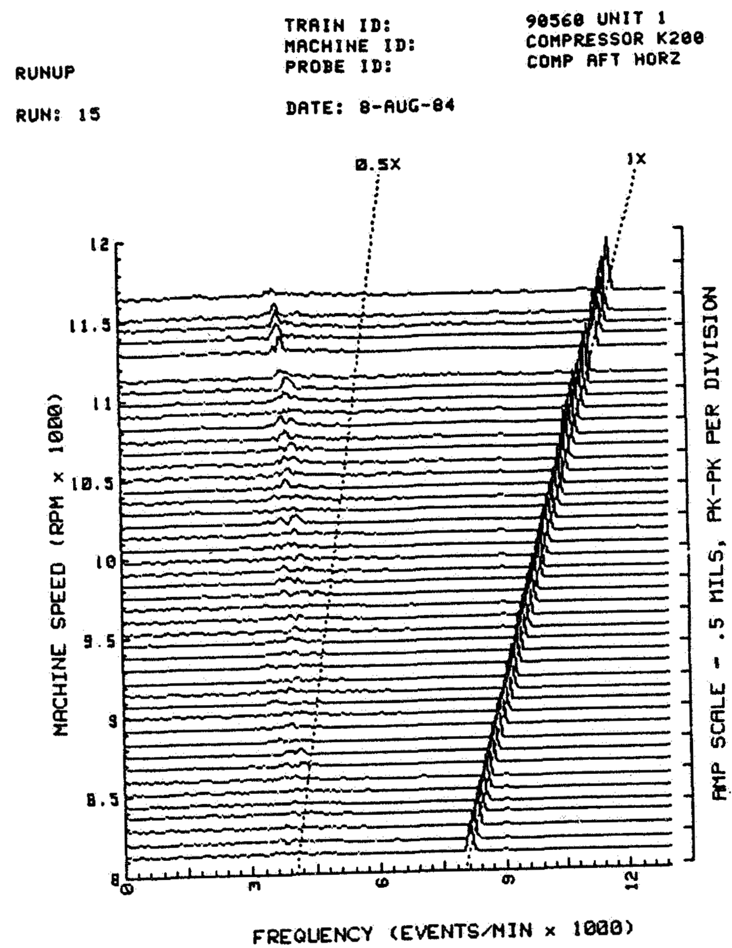

Figure 12. - Runup with negaswirl.

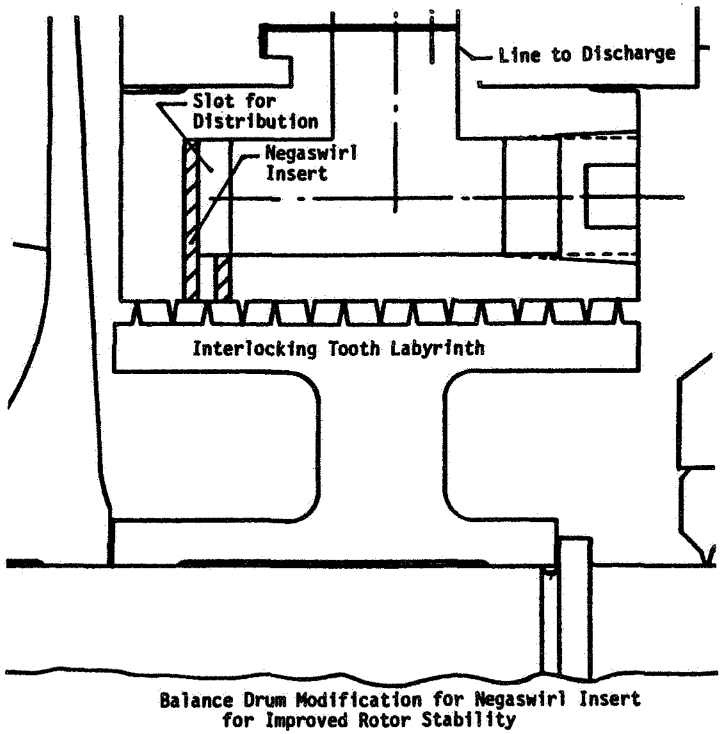

Figure 13. - Typical balance piston.

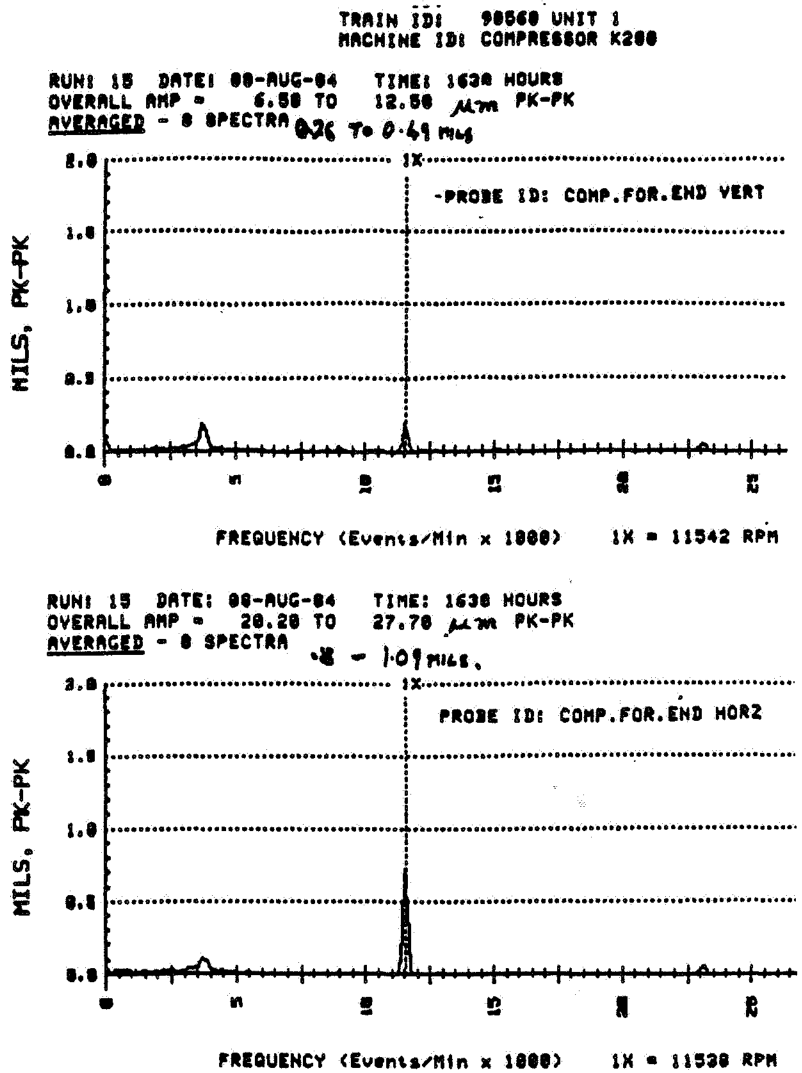

Figure 14. - Unit one.

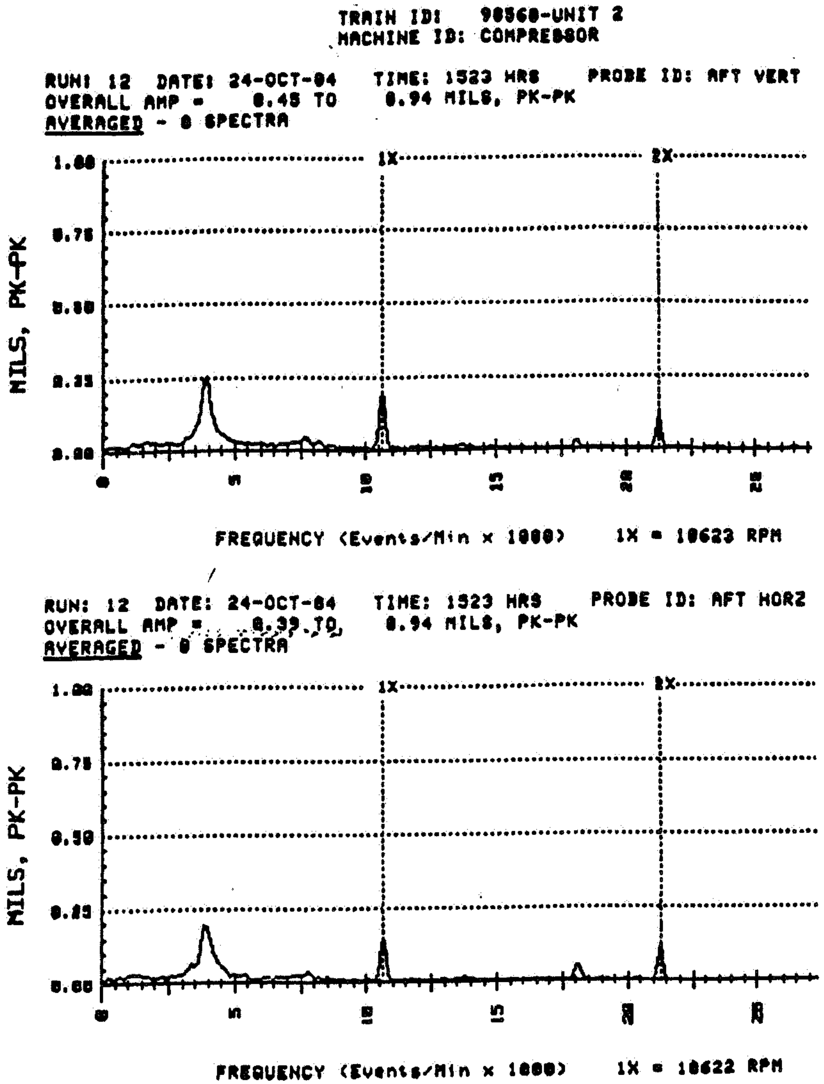

Figure 15. - Unit two.

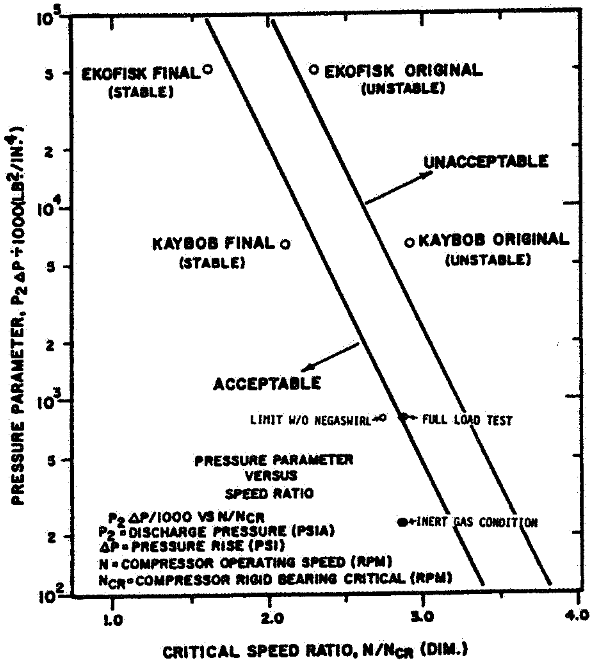

Figure 16. - Compressor design map showing full load test conditions of interest.