SHOP FULL-LOAD TESTING OF CENTRIFUGAL COMPRESSORS

by

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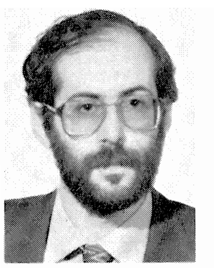

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ABSTRACT

The high reliability traditionally required of turbocompressors has become even more imperative in recent years, due to increased unit power and the extreme environmental conditions of many applications. These machines are now often used in highly demanding compression service, such as reinjection and sea pipelines. In these applications there have been, unfortunately, some examples of malfunctioning. As a result, machines are often requested to undergo all possible testing before leaving the factory. Full-load tests are now being requested in addition to the more traditional no-load mechanical and performance tests.

In the field of natural gas machinery, full-load tests were initially performed mainly for the purpose of verifying the rotor-dynamic stability of compressors at high pressure, as well as their overall performance. Subsequently, full-load testing of the complete compression system (driver, coupling and gear, lube and seal oil systems, control system, coolers, etc.) was performed. Finally, testing was carried out on the entire module.

INTRODUCTION

The main features of a full-load test are described, both for a single compressor and for the whole system. The bases for evaluation of these tests, as obtained from past experience, are presented. Lastly, a detailed description of shop test facilities and of test design and performance is given.

FULL-LOAD TESTS

Full-load testing of single units

These tests are performed chiefly to check and/or correct compressor functioning as regards:

- vibration caused by aerodynamic excitation;
- actual as compared to predicted thermodynamic performance, taking into consideration the imperfection of the compressed gas;
- axial thrust caused by secondary effects occurring in off-design conditions;
- end seal performance under full pressure.

The best method of testing compressor soundness consists in scanning the whole compressor characteristic curve from high flow to surge over the whole speed range. The verification that the machine can withstand full surge conditions, guarantees that failures will not take place even if surge occurs owing to incorrect operation. To be sure of reproducing both the pressure and the velocity profile of design conditions, as regards aerodynamic excitation and secondary effects on thrust, flows and specific volumes must be kept close to design values throughout the compression path. The temperature level should also be similar to maintain consistent end seal operation and to obtain the same thermal effects.

With regard to thermodynamic performance, if the contract gas is fairly imperfect (high pressure natural gas, carbon dioxide, etc.), it is advisable to use a gas of similar composition in testing to minimize uncertainty in adjusting tests results to design conditions. A slight modification of the ASME PTC-10 Class I requirements may provide more representative test results, as described below. The permissible departures from specified operating conditions for Class I tests (Table 1), together with no limitations on Zi and K (= Cp/Cv) values, lead to deviations of

$$\frac{N}{\sqrt{Z_i\,R\,T_i}} = \frac{N}{\sqrt{p_i\,v_i}}$$

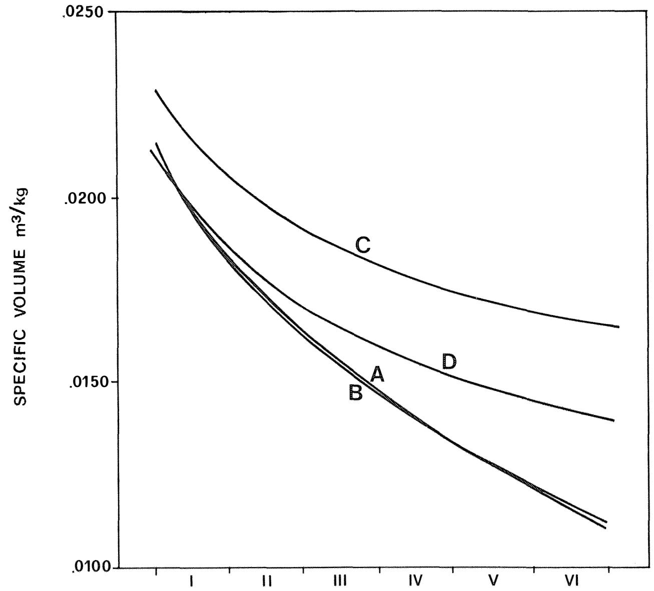
up to 8% (effecting volume ratio), and of Mach number up to 10% or more. Therefore it is possible, within the limits of Table I, to run tests beyond the 5% tolerances on the volume ratio and Mach number allowed (permissible departure from specified design parameters for Class II and Class III tests). These tolerances must instead be respected in order to reproduce the design velocity profiles.

The insertion of the term $Z_i\,R\,T_i$ in Table 1, in place of specific gravity G and the addition of the 5% requirements, would permit Class I tests to be performed as originally intended. From the conditions described above, it can be deduced that, for machines compressing natural gas under high and medium pressures, it is sometimes necessary to use natural gas with the addition of inert gas in testing [1]. To give an example, Figure 1 shows the shape of compression path specific volumes for a compressor with the following design data:

- natural gas — molecular weight 18.15
- suction/discharge pressure (bar) 59.5/176.2
- suction/discharge temperature (°C) 38/167.6

Curve A shows the design specific volume shape; curve B the shape obtained at the same speed and pressure level with a mixture of natural gas and carbon dioxide, chosen in accordance with the criteria proposed above; curve C, the shape obtained with a nitrogen-helium mixture strictly conforming to ASME PTC-10 Class I; curve D, the shape obtained with a mixture of nitrogen and helium, attempting to apply the criteria proposed above.

From this example it appears that, with an inert gas mixture, above a certain compression ratio, it is impossible to respect the design specific volumes along the whole compression path, even exceeding the limit on specific gravity set by ASME PTC-10 Class I. In this case, for example, the last stage discharge specific volume is about 26% higher than the design value. In these conditions, aerodynamic excitation caused by rotating stalls may not occur in the compressor during testing. It can also be seen that the discharge temperature with inert gas is higher than that of normal operation.



CURVE	GAS COMPOSITION	MOLECULAR WEIGHT	INLET CONDITIONS PRESSURE bar abs	TEMPERAT. °C	SPECIFIC VOLUME m³/kg	DISCHARGE CONDITIONS PRESSURE bar abs	TEMPERAT. °C	SPECIFIC VOLUME m³/kg
A	CH_4 = 88.9 % C_2H_6 = 5.3 % +others	18.153	59.5	38	.0213	176.2	167.6	.0112
B	CH_4 = 91 % CO_2 = 9 %	18.56	59.5	38	.0213	175.2	174.3	.0113
C	N_2 = 60.5% He = 39.5%	18.53	62.5	30	.0226	135.4	192	.0167
D	N_2 = 72 % He = 28 %	21.29	59.5	38	.0210	144.8	217.1	.0141

Figure 1. Example of Compression Path Specific Volume Shape.

Full-load testing of complete compression systems

Early experience in off-shore installations has shown that dramatic costs and delays can be caused by apparently insignificant problems. Thus the concept of testing not only main machines but also their auxiliary or connected systems has been developed.

The main system components and the problems most likely to arise are the following:

- drivers and couplings (interaction vibrations, abnormal axial thrusts);
- control panels and motor control centers (incorrect cabling, instability of regulation);
- lube and seal oil consoles (improperly sized components, control problems);
- main skid (misalignment owing to insuffucient stiffness, resonance, inappropriate supports);
- coolers (marginal sizing, excessive pressure losses, bundle tube breakage caused by vibration).

It is advisable, when possible, to test complete single-lift modules, limiting also the disassembly for shipping purpose. In this way, site problems involving errors or breakage during disassembly and consequent reassembly are minimized.

Furthermore, by testing complete systems, all the control instrumentation is set-up and its operation is checked. The criterion on which the test is based is that of duplicating, as far as possible, the field set-up, start-up and operation sequences of the system. The checks to be made should be listed in card

TABLE 1. SUMMARY OF SHOP FULL LOAD TESTS WITH CENTRIFUGAL COMPRESSORS

Type	Service	Rated speed (RPM)	Delivery pressure (Kg/cm^2)	Driver used for the test and relevant horse power	Main job auxiliaries system components tested	Test period
BCL 306-1/b	Ammonia synthesis	16000	350	Shop electrical motor 1500 HP	—	Summer 1965
2BCL 306/b	Ammonia synthesis and recycle	14923	291 (320 recycle stage)	Shop gas turbine N.P. MS3002 and 7000HP (ISO)	—	Summer 1966
BCL 156/b	Urea synth.	26000	400	Shop steam turbine N.P. SSG25-2000HP	—	Winter 1968-69
BCL 205/a	Urea synth.	13000	220	Shop steam turbine N.P. G250-2530HP	—	Winter 1968-69
PCL 501	Pipeline	10290	70	Shop gas turbine N.P. MS1002A—5000HP (ISO)	—	Summer 1971
2BCL 306/a	Urea synth.	13943	160	Job steam turbine N.P. EK 1100-2- 14070 HP	—	Spring 1975
BCL 405/c + BCL 405/d	Nat. gas reinjection	8426	700	Shop gas turbine N.P. MS5001 25000 HP(ISO)	—	Summer 1975
PCL 502	Pipeline	10290	70	Shop gas turbine N.P. MS 1002B- 5200 HP (ISO)	—	Spring 1976
PCL 802	Pipeline	6500	81	Job gas turbine N.P. MS 3002-14600HP (ISO)	Gas turbine control panel Seal and lube oil system Gas cooler system	Summer 1979
BCL 406/a	Natural gas reinjection	6500	196	Shop gas turbine N.P. MS 3002-14600HP (ISO)	Seal oil system	Summer 1979
BCL 406/b	Natural gas reinjection	6500	280	Shop gas turbine N.P. MS 3002-14600HP (ISO)	Seal oil system	Autumn 1979
BCL 405/c	Natural gas reinjection	10905	433	Shop gas turbine N.P. MS 5002-35000HP(ISO)	Lube and seal oil system	Winter 1979-80
BCL 406/b	Ethane injection	10290	70	Shop gas turbine N.P. MS 3002-14600HP (ISO)	—	Spring 1980
BCL 406/b	Natural gas	11840	350	Job gas turbine C.V. RB211 30550HP (ISO)	Gas turbine and compressor control panel. Gas and oil cooler systems, lube and seal oil system.	Winter 1981-82
BCL 607-6/a	Pipeline	4000	150	Job gas turbine U.T.I. FT4C-3F 45694HP (ISO)	Control system house including gas turbine and compressor control panel M.C.C., lube and seal oil systems, gas and oil cooler systems. Antisurge valve.	Spring 1982
BCL 405/a + BCL 305/b + BCL 305/c	Natural gas injection	9740	527	Job gas turbine G.E. LM 2500-27500HP (ISO)	Single lift module with lube and seal oil systems. Gas turbine and compressors control panel.	Autumn 1982
PCL 804-2/36	Pipeline	4670	80	Job gas turbine N.P. MS 5002-35000HP (ISO)	Gas turbine and compr. control panel, lube and seal oil systems. Gas and oil cooler systems.	Autumn 1982
PCL 603-2/24	Pipeline	6500	71	Job gas turbine N.P. PGT 25-27500HP (ISO)	Gas turbine and compr. control panel — Lube and seal oil systems.	Autumn 1982
PCL 502	Pipeline	10290	70	Job gas turbine N.P. MS1002D-6400HP (ISO)	—	Autumn 1982

form and the operators required to fill in the spaces with the actual values measured or adjusted and add their signature and the date as confirmation that the procedure has been followed. If a part of the system proves to be critical in the design phase or has not been previously experimented, additional tests should be planned. For example, the three-point skid support first verified in this type of testing as shown in Figure 2.

Figure 2. Three-Point Skid Support on the Test Stand.

Test Experience

Experience in full-load testing from 1965 to the present, initially on compressors only and subsequently on complete compression systems (see summary in Table 1), has mainly involved the following:

- Compressor rotor dynamics and performance
- system set-up and problems encountered

Compressor rotor dynamics and performance

The first full-load tests were done on compressors for ammonia synthesis service [2]. The machines for this service have quite flexible rotors and relatively high speeds. The tests made it possible to check and set up the methods for calculating critical speeds and the criteria for determining stability of bearings and oil end seals.

Later tests on carbon dioxide and high pressure natural gas compressors drew attention to vibration of aerodynamic origin caused by the high density of the gas [3].

The full-load test performed in 1975 at 700 Kg/cm^2 on a compressor train for reinjection of natural gas made it possible to analyze aerodynamic excitation, correlating it with pressure pulsations caused by rotating stalls. The resulting vibrations were recognized as being of the forced type as shown in Figure 3 [4-5]. An example of the forced vibrations in the various zones of the characteristic curve is shown by Figure 4. Thus the need to develop a program of research on the individual stages, using a special test stand suitably equipped to study the influence of stator components with different geometries, became evident [6].

The machines designed on the basis of the experience

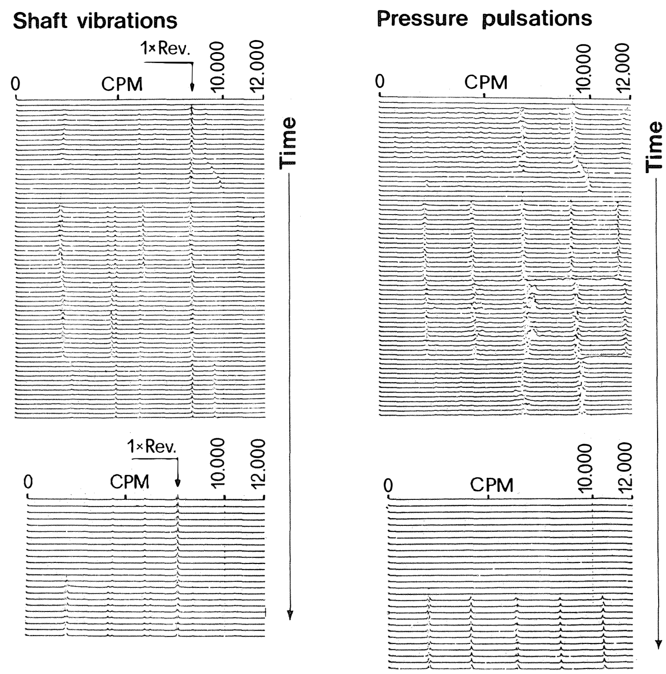


Figure 3. High Pressure Compressor Vibrations and Pressure Pulsation Spectra.

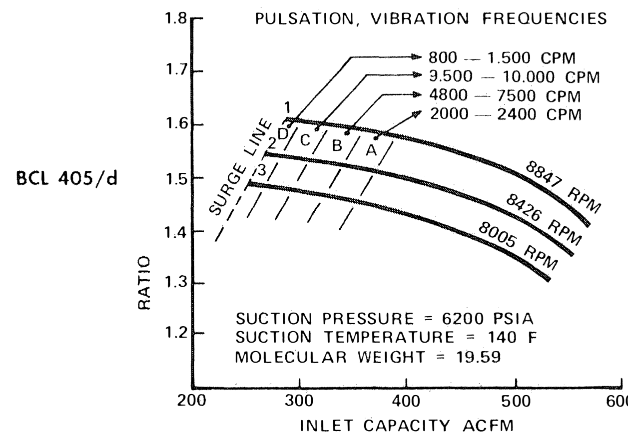


Figure 4. High Pressure Compressor Aerodynamic Force Frequency Versus Capacity.

acquired in single stage tests demonstrated, under full-load testing, that vibrations were fully synchronous, and only on reducing the flow towards surge did a stable vibration invariably appear at a frequency of 10-15% of running speed as shown in Figures 5-6 [7].

Another type of vibration of aerodynamic origin sometimes arises on back-to-back compressors. Destabilizing forces on the central labyrinth seal produce low frequency vibrations [8-9]. A spectra of vibrations of this type is shown in Figure 7. With a special central labyrinth seal configuration, it has been possible to limit this phenomenon and to transfer it outside of the normal operating range.

Correlation of phenomena with their causes has been possible by shop full-load testing due to the ease with which minor variations can be applied. For example, oil end seals can be depressurized or overpressurized to determine whether a

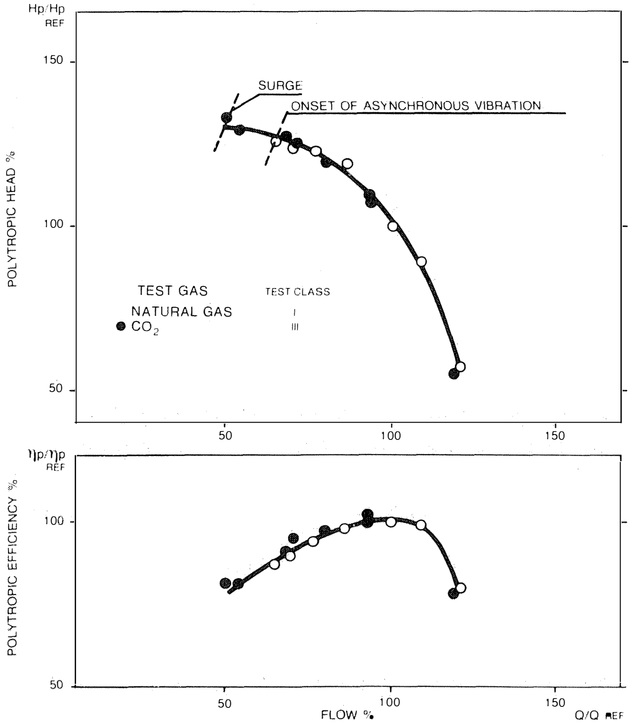


Figure 5. Ethane Compressor Performance Curves.

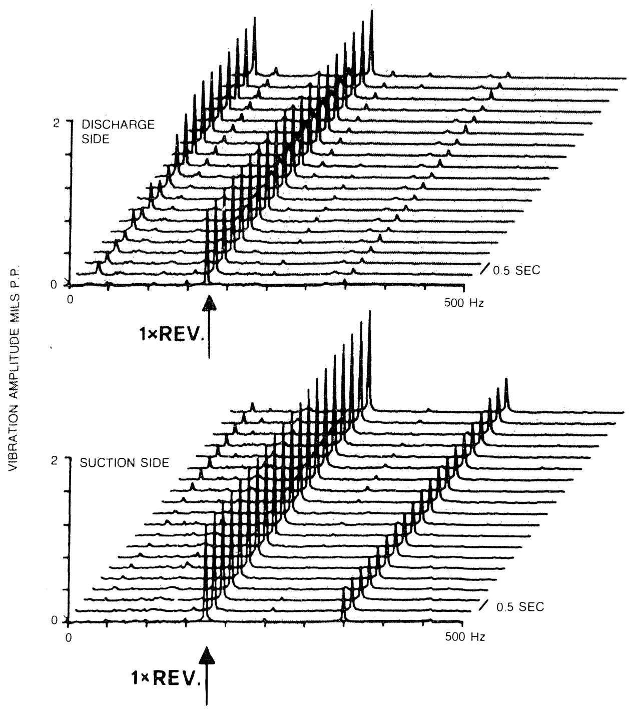


Figure 6. Ethane Compressor Vibrations Spectra.

certain type of vibration originates from aerodynamic excitation or from seal oil whirl.

With regard to thermodynamic performance, it can be noted that the stages used in centrifugal compressors come from families of optimized standard stages. Each stage is generally air tested for different values of impeller tip Mach

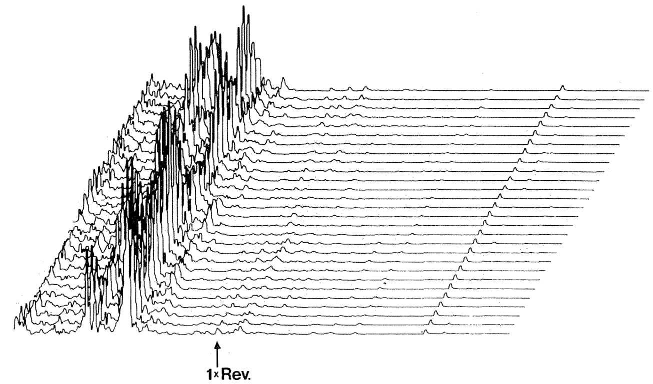


Figure 7. Vibration Spectra of a Back-to-Back Natural Gas Reinjection Compressor.

number, and its performance is stored in a computer. A computer program corrects these performances for different Reynolds numbers and gas specific heat ratios.

The validity of the above correction methods has been confirmed during full-load tests with different gases as shown in Figure 8 [10-11].

System set-up and problems encountered

Full-load testing on complete systems has involved various problems arising from accidental causes or unexpected difficulties.

A major problem was the breakage of a gas generator blade in a job aeroderivative gas turbine. This entailed disassembling the gas generator and power turbine to permit inspection and replacement of damaged parts. Another severe problem, which arose during testing of a turbo-alternator with all job auxiliaries and control panels, demonstrated that even elements not considered theoretically critical can cause major trouble. In this case, the auxiliaries motor control center suffered serious failure due to short circuit. It was necessary to modify the design, increasing the distance between two adjacent conductors.

Two examples of setting-up work are given below.

A separate lube and seal oil system required some adjustment.

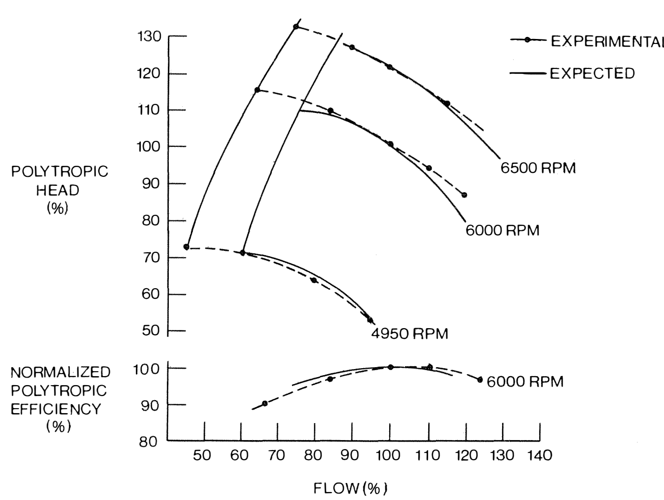


Figure 8. Natural Gas Reinjection Compressor: Full-Load Experimental and Expected Curves.

Clearance of seal rings was reduced to match correctly the seal oil pump capacity. Modification of the lube and seal oil separation device was also necessary.

Problems were encountered in the accelerometers on a speed increasing gear and subsequent modifications made on the monitor electronic cards.

TEST FACILITIES

Initially, full-load tests were performed only sporadically; thus, the investment required for permanent installations appeared unjustifiable. As a result, the first tests requiring flammable gases were performed at nearby petrochemical plants (Figure 9) and those with inert gas on internal test stands. The first outdoor test facility was built at the Florence factory in 1975 (Figure 10); subsequent expansion increased the number of outdoor stands to four, as shown in Figure 11.

With these facilities, it was possible to carry out two tests simultaneously with a total power of 80,000 HP, developed by two different type aeroderivative gas turbines. The need to test single-lift modules led to construction of another stand at the Massa plant, near the port of Marina di Carrara as shown in Figure 12.

A description of the Massa test stand, detailing the various elements of the layout, follows (Figure 13):

A) Test stands: two in number, which can be extended to three. Each stand is 34 x 11 m in size and can support a maximum load weight of 800,000 Kg.

   Stands have been designed so that all skid configurations can be installed.

B) Gas coolers: the test facility is equipped with three variable effect air coolers which can dissipate approximately 22×10^6 Kcal/h with pressures up to 400 bars, corresponding to the suction pressure of the last compressor section. Should the need arise, job air coolers can be installed nearby.

C) Gas feeding system: a remote control system can inject the following gases into closed loops:

   - natural gas; straight from gas network or bottle packages or through reciprocating compressors.

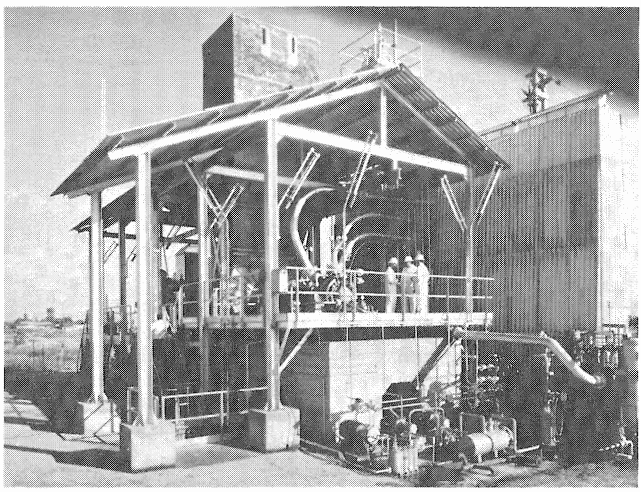


Figure 9. Ravenna 1966 — Full-Load Test on the First Ammonia Synthesis High Pressure Compressor Carried Out at a Petrochemical Plant.

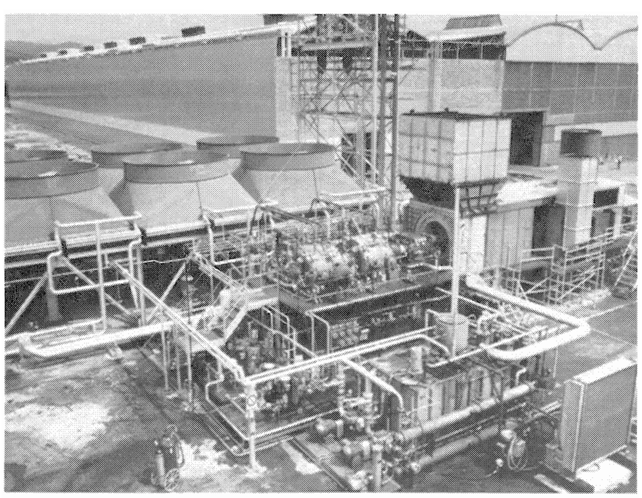


Figure 10. Florence 1975 — Full-Load Test on a High Pressure Reinjection Compressor Under Actual Service Conditions — The Maximum Recorded Delivery Pressure was 700 Kg/cm^2.

Figure 11. Florence 1981 — Outdoor Test Facilities with Four Test Stands — Simultaneous Testing of Two Turbocompressors with a Total Power of 80,000 HP.

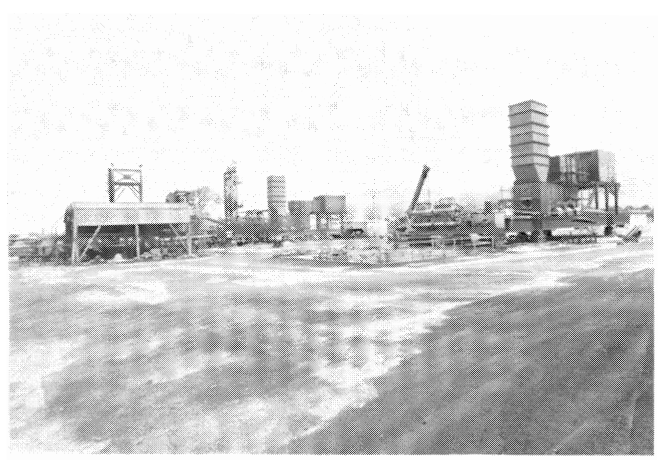


Figure 12. Massa 1982 — Outdoor Test Facilities with Two Stands — Preparation of a Single-Life Module for Testing.

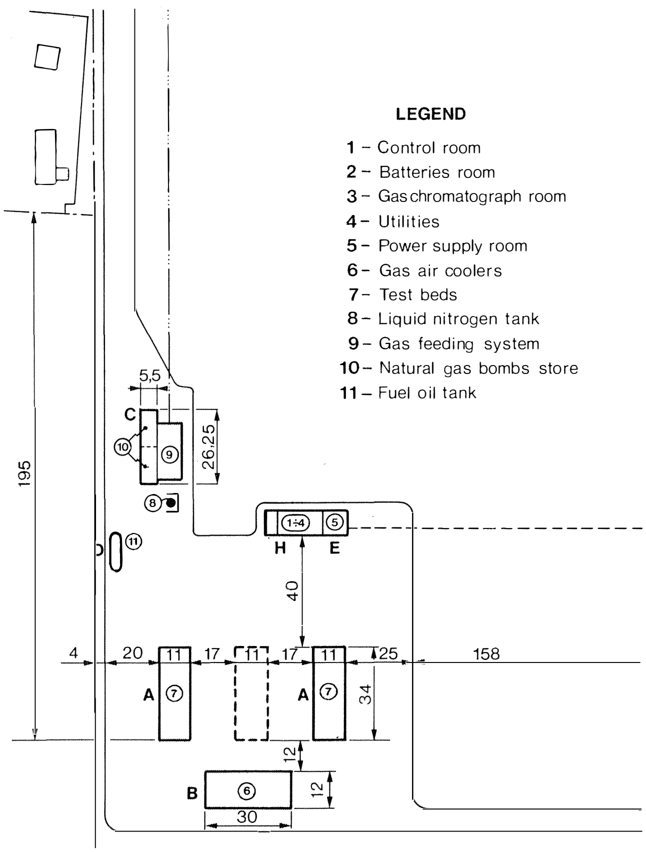


Figure 13. Massa Outdoor Test Facilities Layout.

Natural gas is also used to feed the aeroderivative gas turbine starter motor,

- nitrogen; straight from a tank with a vaporizer or through reciprocating compressors,
- helium; straight from bottle packages or through reciprocating compressors,
- carbon dioxide; from a tank through pumps and a vaporizer.

D) Diesel fuel feeding system: feeds gas turbines at a maximum rate of 400 lit/min., corresponding to the requirements of two turbines of approximately 50,000 HP each.

E) Electric power supply: up to 3200 KVA at 400 V, 50 Hz. This is used both for the test facilities and the job systems, and can be controlled either by test or job motor control center.

If necessary, a 500 KVA, 60 Hz conversion unit is available to feed 60 Hz systems.

The facility is also equipped with a battery system to supply power to emergency devices.

F) Auxiliary facilities: service air and water supply required for any lube and seal oil system are available.

G) Instrumentation: a computerized data acquisition system (DAS) gathers data in conformity with ASME PTC-10 requirements and processes them according to the BWRS equation. The composition of the test gas is continuously analyzed by a gas chromograph. Vibrations are picked up by the DAS and are simultaneously recorded and analyzed by tape recorders, fast fourier transformers, digital vector filters, etc. as shown in Figure 14.

The gas pressure pulsations can be analyzed in similar fashion.

The closed loop valves are all remote controlled and the main pressure and temperature parameters are continuously monitored and recorded as in an industrial plant.

The alarm and shutdown systems can be either those of the job or the test stand.

H) Control room: a building near the test stands houses computer, control panels, gas chromatograph room, battery room, power center, and a small storage deposit and workmen's locker-room.

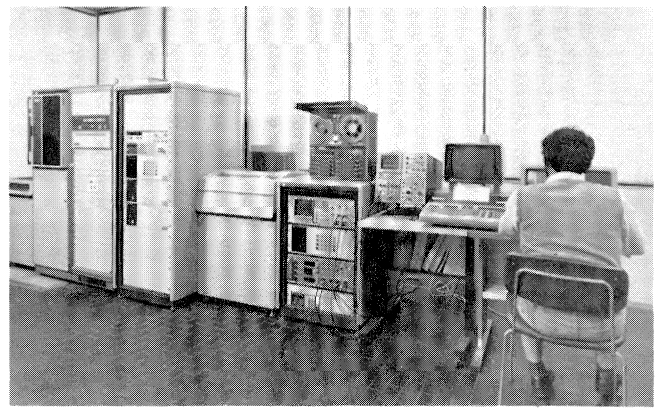


Figure 14. Data Acquisition System and Auxiliary Instrumentation (Tape Recorder, FFT, DVF) Used for Measuring Full-Load Test Parameters.

Test Design and Performance

Test preparation initiates together with the designing of the machines and system. Procuring and constructing some test equipment and instruments requires as much time as manufacturing job equipment. Obviously, it is necessary to determine the critical parts in advance, on the basis of previous experience in similar tests. Test design must be done in close collaboration with the designers of the machines and system. The client also must collaborate in organizing a really meaningful test by providing all available information on the expected conditions of use.

The actual testing period is the most critical moment. Performance and analysis of the tests must proceed simultaneously, since analysis after conclusion of the tests could show that modifications or additional tests are required. This would have a negative effect on delivery schedules.

The cost of testing is generally high. A testing team consists of about ten people (two engineers, four technicians and four operators); the greatest cost, however, is that of fuel (consumption is approximately 170 lit/min of diesel fuel for a 50,000 HP gas turbine).

Therefore, an experienced team that works well together, making technically valid decisions rapidly, is a must from both technical and financial point of view.

CONCLUSIONS

The purpose of full-load testing and its design philosophy have been described as concerns individual machines, complete systems and modules. Mention has been made of the fact that a slight modification of ASME PTC-10 Class I may provide more representative test results. Some experiences with full-load tests involving compressor rotordynamics, compressor performance and system set-up and problems have been illustrated.

Available test facilities and the organization necessary to prepare and perform such testing have been described.

Whether full-load testing is worthwhile or not can only be decided case by case, in close collaboration between client and manufacturer, weighing up the cost of the test against the risks involved in setting up on site.

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NOMENCLATURE

N = Speed, RPM

Z = Compressibility factor

R = Gas constant

T = Absolute temperature, °K

p = Absolute pressure, ata

v = Specific volume, $m^3\ Kg^{-1}$

Subscripts

i = inlet conditions