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ASME PTC 10 MODIFIED TEST FOR MECHANICAL ASSESSMENT OF CENTRIFUGAL COMPRESSORS

by

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ABSTRACT

The rotordynamics paragraphs of the latest version of the API 617 Standard (2002) have been thoroughly reformulated and the prevention of subsynchronous vibrations and instability phenomena are dealt with in a very comprehensive and detailed form. However, this stability analysis may suffer from the uncertainties of numerical simulations. Some manufacturers have developed evaluation tests based on magnetic actuators but these methods are still in an experimental phase. In order to ensure that no instability problem will be present during operation, the full load tests have been of major importance. Recent experience on the ASME PTC 10 (1997) Type 1 test demonstrates that in many cases the manufacturer is not able to fully comply with its requirements. This study presents results obtained by the application of a full density test in order to use it as a mechanical test. Modifications on the requirements are proposed in order to adapt the ASME tests and allow using them as tests for mechanical and stability assessment. The advantages of using the modified requirements are demonstrated by the presentation of test results of different compressors.

INTRODUCTION

The objectives of this study are to present the requirements to ensure the mechanical behavior of centrifugal compressors and to discuss the application of a *full density* test, adapted from ASME PTC 10 Type 1 test. Besides the thermodynamic performance assurance, other significant advantage of the Type 1 test has been to allow for the early verification and thus prevention of subsynchronous vibrations and instability on high-density gas compressors. Many users have successfully employed this test for instability verification. Otherwise, even synchronous response changes have been reported when operating on full load, due to thermal and/or pressure effects on the rotor, bearing, and seals system. These changes cannot be detected by a standard, unloaded, mechanical running test. A Type 1 test should reproduce as near as possible the design conditions, including the gas composition. The PTC 10 tables 3.1 and 3.2 define the maximum allowable departures from the operation conditions and parameters, respectively. Since the power requirements for this type of test are near to the maximum design value and the test bed drivers are of low power, usually the driver is the very contracted motor. For this reason, the Type 1 test is always performed together with the complete unit test (string test).

The rotordynamics sections of the latest version of the API 617 Standard (2002) have been thoroughly reformulated and the prevention of subsynchronous vibrations and instability phenomena are dealt with in a very comprehensive and detailed form. These sections define a two-step analysis procedure where the first requires only a basic model in order to assess the proximity of the equipment to an instability threshold, based on a modified Alford's equation. For that equipment found to be not well away from the instability threshold, a second step is required, based on a detailed model, which includes seal stiffness and damping coefficients. However, this stability analysis may suffer from the uncertainties of numerical simulations. Some manufacturers have developed evaluation tests based on magnetic actuators but these methods are still in an experimental phase. In order to ensure that no instability problem will be present during operation, the full load tests have been of major importance.

In many instances the manufacturer is not able to fully comply with the requirements of the "pure" Type 1 test, particularly with regard to the gas composition. Either practical or even safety restrictions prevent manufacturers from handling explosive gases in the test bed area. Even if a manufacturer is allowed to handle natural gas, it is sometimes necessary to add carbon dioxide (CO_2). Therefore the mol weight deviates substantially from the ASME requirements.

This study presents modifications on the requirements of table 3.1 of the ASME PTC10 (1997) code. The modified table was developed in order to accommodate the actual limitations faced by the majority of the centrifugal compressor manufacturers.

In order to demonstrate the advantages of the modified requirements some case studies are presented. This paper reports the test results of two offshore, electric driven, three section compressor trains of 6 MW and 9.6 MW, respectively. Other comparisons are also presented, including a 2 MW electric driven high-pressure gas storage compressor and one gas lift, 14 MW variable frequency drive (VFD) driven, three section compressor train. In the later case, thermal induced change of residual unbalance was observed at the test bench.

API STABILITY REQUIREMENTS

As is well known, turbomachinery rotors may present subsynchronous vibrations that are caused by the rotor movement itself, i.e., they are self-excited. In being so, they become unstable when the dissipative forces of the rotorbearing system are not capable of limiting the amplitude of vibration.

It is worth mentioning that, as discussed in API 684 (2005), the research on the causes of subsynchronous vibrations has contributed to the development of the tilting pad journal bearing, the squeeze film damper, the dry gas seal, the swirl brakes, and the damper seals (honeycomb, hole pattern, and pocketed labyrinth). Hand in hand with these was the development of numerical programs for modeling and analysis of the dynamic behavior of rotors and its components. Nevertheless, due to the ever-present concern on subsynchronous vibrations, specific sections on this matter have been introduced in the latest revision of the API 617 Standard (2002).

The rotordynamic stability analysis adopted by API 617 (2002) employs *eigenvalue* evaluation, which is also used in diverse engineering areas. It is important to note that eigenvalue extraction from *general eigenvalue problem* has been traditionally employed to obtain the undamped critical speeds and their plane mode shapes that are, by its turn, a good estimate of the actual critical speeds and their skewed whirling modes. However, for stability analysis, a damped eigenvalue evaluation must be employed. This is achieved by transforming the equations of motion, damping included, which form a set of second order differential equations, into state equations (first order equations). These are used in the analysis of general dynamic systems and lead to a *standard eigenvalue problem*. For general dynamic systems, the eigenvalues may be real or complex. However, if the standard eigenvalue problem was derived from a rotordynamic or other damped vibration system, the eigenvalues appear as n complex conjugate pairs in the form:

$$\lambda_j = -\omega_j \zeta_j \pm \omega_d i \qquad \textbf{(1)}$$

where:

ω_j is the undamped natural frequency of the jth mode,

ζ_j is the modal damping ratio of the same mode,

$\omega_d = \omega_j \sqrt{1-\zeta_j^2}$ is the damped natural frequency of the same mode and i is the imaginary unit.

The natural frequency ω_j is always positive but this may not occur to ζ_j. The contribution of the jth mode to the free vibration response may be written as:

$$y = e^{-\omega_j \zeta_j t} sen(\omega_d t + \phi) z \qquad \textbf{(2)}$$

For the system to be stable, all free vibration responses must decay. Equation (2) shows that for this to happen, since the natural frequencies are already positive, the damping ratios must also be positive. It is interesting to note that this is equivalent to the stability criterion employed in control theory, that all poles (i.e., the eigenvalues of matrix A) should be placed in the left semi-plane of the complex numbers, as expressed in Ogata (2002) and other textbooks on control theory.

In the API 617 Standard (2002), although employing a criterion equivalent to requiring positive damping ratios derived from standard eigenvalue extraction, stability analysis has two peculiarities. First, instead of the damping ratio ζ, the adopted stability parameter is the logarithmic decrement δ, which is the natural logarithm of the ratio of two successive displacement amplitudes. As shown by Inman (1996), δ is related to ζ by:

$$\delta = 2\pi\zeta / \sqrt{1-\zeta^2} \qquad \textbf{(3)}$$

Second, only the first forward mode must be analyzed, except for double overhung machines, where the first two forward modes must be considered.

The standard also defines a two-step analysis process, where the first employs the same mathematical model used for the unbalance response in order to assess the proximity of the equipment to the instability threshold. For that equipment found to be over the instability threshold, a second step is required, also of damped eigenvalue extraction, but based on a more detailed model, which should include seal stiffness and damping coefficients. As noted by the API standard, the level I stability analysis procedure was introduced with two objectives. The first objective is to supply a preliminary screening process. In this way the procedure is conservative, i.e., if a rotor fails to pass, this is not an indication that the rotor is unstable. The second objective is to provide a standardized procedure, equivalent in difficulty to obtaining undamped critical speeds.

A further indication of this standardization concerns the tilting pad bearing coefficients. To model a tilting pad bearing with k pads, 5k + 4 stiffness coefficients and an equivalent amount of damping coefficients are required. This large number of coefficients can be reduced to the traditional eight stiffness and damping bearing coefficients, but a vibration frequency must be assigned to perform the reduction. As observed by API 684 (2005), there is a controversy on with which frequency the reduced coefficients should be calculated, the synchronous frequency or the subsynchronous one. The API 617 (2002) avoids this controversy by requiring that this reduction should be performed using the rotating speed frequency.

For a level I stability analysis, all destabilizing sources on the rotor, i.e., aerodynamic cross coupling, impeller eye and hub seals, and case labyrinth seals, are represented by a cross coupled stiffness coefficient Q. This is introduced in the rotor plus bearings basic model, at the rotor midspan, for between-bearings rotors, or at the impeller center of gravity, for the overhung case. Generally speaking, the Q stiffness coefficient affects all the complex eigenvalues of matrix A, but, considering that it is cross coupled and applied at midspan, it should particularly affect the modal damping ratio ζ of the first forward mode and by extension, the logarithmic decrement δ.

For centrifugal compressors, the API 617 Standard (2002) requires a level II stability analysis if the evaluated level I logarithmic decrement is less than 0.1. Further restrictions are also applied on the values of the cross coupled stiffness coefficient, on the critical speed ratio, and on the gas density.

For a level II stability analysis, all destabilizing sources on the rotor should be considered separately and can be applied at its rotor location, instead of an overall Q cross coupled stiffness applied at the center of the rotor. This should provide a more accurate and hence less conservative δ calculation. Another difference relative to level I is that the analyses should now be carried out for clearance and oil temperatures that lead to the minimum logarithmic decrement. Per API 617 (2002), the stability of the rotor is considered satisfactory if the calculated logarithmic decrement is larger than 0.1, but the authors' company requires at least 0.2.

As destabilizing sources, the API 617 Standard (2002) explicitly lists labyrinth seals, balance piston, impeller flow, shrink fits, and shaft material hysteresis. As discussed in literature, e.g., Gunter (1973), the last two are important sources of rotordynamic instability. However, these sources are nonlinear while the API stability criterion is eigenvalue based and, therefore, is a linear criterion. Due to this dichotomy, the standard recognizes that methods may not be available to model the destabilizing effects from all listed sources. In fact, two recent papers on the application of the new API 617 (2002) stability requirements consider only the first three causes (Nicholas and Kocur, 2005; Li, et al., 2005).

Other stability criteria have been proposed in the past. Kirk and Simpson (1985), for example, proposed an empirical curve based on a parameter defined as the product of the discharge pressure with the pressure difference across the compressor. They considered both pressure rise and discharge pressure as critical issues with regard to stability. The pressure rise P_2-P_1 applies to single section compressors. For back-to-back configurations, pressure drop through the intersection division wall is approximately the difference between the discharge pressures of both sections.

MODIFIED ASME REQUIREMENTS

The previous section shows that the last API 617 (2002) revision has introduced a very detailed and thorough stability analysis procedure. Nevertheless the authors' firm has required a full load test on all newly purchased centrifugal compressors, in order to verify stable rotordynamic behavior. The reasons for this test requirement are:

- Some destabilizing effects present numerical modeling difficulties. This becomes more difficult if the procedure is linear, as adopted by the API 617 Standard (2002).
- Numerical simulations may suffer from wrong assumptions. On this aspect, Li, et al. (2005), present a very illustrative example of a centrifugal compressor that, although being the object of a detailed numerical evaluation whose result was unstable, was commissioned and has been working without any problem. The contrary may occur.
- Due to Galileo's cultural heritage, one may only accept that equipment attends a requisite if it is experimentally verified.
- The existence of significant historical record of cases with unsuccessful design of centrifugal compressors, which have led to substantial production losses caused by instability and high vibrations, even where detailed analysis was applied.

This paper proposes a new type of test in order to assess the mechanical and stability behavior of centrifugal compressors. The test introduces some modifications on the requirements for the ASME Type 1 test, which have been made not only to accommodate the eventual limitations faced by some centrifugal compressor manufacturers, but also in order to reduce costs and increase test safety conditions. The modifications intend to create conditions to submit the compressor to the aerodynamic forces as near as possible to the design conditions, rather than to reproduce the similarity required for performance verification. In some cases both rotordynamics and performance can be verified, but the major objective is to check the former, considering that the default type 2 tests are enough for performance evaluation.

The ASME PTC 10 (1997) tables 3.1 and 3.2 define the maximum allowable departures from the operation conditions and parameters, respectively. Recent experience with the Type 1 test shows that in many cases the manufacturer is not able to fully comply with the requirements of the test. Particularly with regard to the gas composition, it is sometimes difficult to achieve the molecular weight of the design condition. Either practical or even safety restrictions prevent manufacturers from handling explosive gases in the test bed area. Besides, the cost of the test with inert gas is significantly lower than with hydrocarbon mixtures. Even if a manufacturer is allowed to handle natural gas, for high molecular weight gases, it is sometimes necessary to add carbon dioxide instead of propane or butane, and the mol weight may deviate substantially from the Type 1 test requirements.

The main objective of the ASME code is to establish the requirements for thermodynamic tests for dynamic compressors. As the Type 1 test is made at full load and design operating conditions, this type of test is also considered as an excellent mechanical test. This happens not only due to the full load and full speed conditions, but also mainly due to the close to design gas density. The high density itself is the worst villain of the aerodynamic phenomena that can lead to instability. The focus of the test is rather the reproduction of the aerodynamic conditions inside the compressor, including density and the flow path through the labyrinth seals and internal channels. With this new focus in mind this paper analyzes every single requirement of the code in order to assess the impact on stability for each one of them.

ASME table 3.1, shown in Table 1, defines the specific requirements for Type 1 test and is related to the operational conditions and the test gas. The Type 1 test shall also meet the limitations of ASME table 3.2, which defines the allowable deviations for dimensionless parameters. As ASME table 3.2 is related to the similarity control, it accounts more for the thermodynamic performance. Otherwise ASME table 3.1 is of major interest under the rotordynamics point of view.

Table 1. Permissible Deviation from Specified Operating Conditions for Type 1 Tests.

Variable	Permissible Deviation
Inlet pressure	5%
Inlet temperature	8%
Speed	2%
Molecular weight	2 %
Cooling temperature difference	5%
Coolant flow rate	3%
Volumetric capacity	4 %
Inlet gas density	8%

The combined effect of the deviations shall not exceed the limits of ASME table 3.2:

- Specific volume ratio: ± 5 percent
- Flow coefficient: ± 4 percent
- Mach and Reynolds numbers: limited according to ASME charts

Parameters Influence on Stability

Among the operating conditions, the compressor inlet pressure shall be kept under the deviation limits of ASME table 3.1, and the discharge pressure, although not mentioned, is indirectly kept close to design by the combined effect of application of both tables. In order to create conditions that are representative under the stability point of view, it is more relevant to reproduce the density profile instead of limiting the suction pressure and molecular weight. Moreover, the destabilizing effect in the seals is also a function of mass flow, which is dependent on the pressure drop. Therefore, a good strategy to test the stability conditions can be developed considering:

- Specify limits for the compressor pressure rise or the pressure drop through the balance piston/division wall labyrinth
- Tighter limits for the inlet density
- Specify limits for the discharge density

On the other hand:

- Relax or even remove the limits for suction pressure and
- Relax limits for molecular weight.

The new strategy allows gas mixtures different from the specified gas and, in most cases, it is possible to use inert gases only or a mixture of hydrocarbon and inert gases. The disadvantage of inert gas as substitute for hydrocarbons is the higher temperature rise. The adiabatic exponent k is usually higher for inert gases than for hydrocarbons. Therefore it is advisable to include in the new strategy also:

- Specify limits for the discharge temperature, and
- Increase the allowable speed (rpm) departure.

The limit to discharge temperature is due because, otherwise, the temperature gradient would be much different from design, with strong impact on the differential thermal growth and on the static-to-rotating parts clearances. In order to limit the temperature rise, it may be necessary to change the speed. Previous studies and experience shows that, in most cases, the increase of the speed allowed deviation from ±2 percent up to ±4 percent is enough to keep the discharge temperature within a deviation of ±10 percent. All the proposed modified requirements are shown in Table 2.

Table 2. Modified Requirements.

Requirement		original ASME type 1	proposed new req.	Comments
temperature (K)	Inlet	8%	8%	keep requirement of Table 3.1
	Disch	-	10%	New requirement to limit temperature gradient
pressure (kPa a)	Inlet	5%	-	Requirement relaxed to allow density adjustment
	Disch	-	8%	New requirement to obtain full pressure
capacity (m3/h)	Inlet	4%	4%	keep requirement of Table 3.1
	Disch	-	-	
density (kg/m3)	Inlet	8%	5%	Departure narrowed to better verify stability
	Disch	-	5%	New requirement to better verify stability
specific volume ratio vi/vd		5%	5%	keep requirement of Table 3.2
pressure ratio P2/P1		-	-	
Power (KW)		-	-	
Molecular Weight		2%	-	Requirement relaxed to allow density adjustment
pressure rise P2-P1 (kPa)		-	5%	New requirement to control flow thru balance labirinth
P2*(P2-P1)		-	5%	New requiriment to verification of Kirk parameter
Head (m)		-	-	
speed (RPM)		2%	4%	Requirement relaxed to limit discharge temperature
cp/cv	Inlet	-	-	
	Disch	-	-	
flow coeficient φ	Inlet	4%	4%	keep requirement of Table 3.2
	Disch	-	-	

Approaching Surge

The proposed test, although aiming at the stability concerns, can be also applied, in some applications, as a thermodynamic performance test provided that the similarity condition is achieved. That is the reason why the proposed requirements still keep those of ASME table 3.2. The performance test procedure foresees the verification of the curve by testing at least five points. The test point "approaching surge" is a good opportunity to check for subsynchronous vibrations.

EXAMPLE #1—MAIN COMPRESSORS OF AN OFFSHORE PLATFORM

The first example of an offshore platform, 180,000 barrels/day of oil and 6 million cubic meters/day of gas, was witnessed during 2005. The system is designed with a configuration of three fixed speed electric driven, straight-through compressors, for the low-pressure stages (first section). The three high-pressure compressors are also fixed speed driven and they are designed in a back-to-back configuration (second and third sections). During the design phase of the compressors, ASME Type 1 tests were specified for one out of three low-pressure compression strings and one out of three high-pressure strings. Tables 3 and 4 show both design and test condition and parameters for the first and third sections, and the comparison with both original and modified ASME requirements.

Table 3. Process Parameters for Full Load Test of Main Compressor, First Section.

Item	Requirement		design	test	Deviation	original ASME type	proposed new req.
1	temperature (K)	Inlet	308,539	309,15	0,20%	8%	8%
2		Disch	440,848	472,142	7,10%	-	10%
3	pressure (kPa a)	Inlet	899,3	900	0,08%	5%	-
4		Disch	4421,3	4526	2,37%	-	5%
5	capacity (m3/h)	Inlet	8618,229	8610,33	-0,09%	4%	4%
6		Disch	2516,085	2652,421	5,42%	-	-
7	density (kg/m3)	Inlet	7,763	8,224	5,94%	8%	5%
8		Disch	26,592	26,697	0,39%	-	5%
9	specific volume ratio vi/vd		3,4255	3,2462	-5,23%	5%	5%
10	pressure ratio P2/P1		4,9164	5,0289	2,29%	-	-
11	Power (KW)		5311,41	5617,93	5,77%	-	-
12	Molecular Weight		21,568	23,06	6,92%	2%	-
13	pressure rise P2-P1 (kPa)		3522	3626	2,95%	-	5%
14	P2*(P2-P1)		15571819	16411276	5,39%	-	5%
15	Head (m)		28710	28715	0,02%	-	-
16	speed (RPM)		11490	11490	0,00%	2%	4%
17	cp/cv	Inlet	1,2611	1,3124	4,07%	-	-
18		Disch	1,2289	1,2577	2,34%	-	-
19	flow coeficient	Inlet	0,7500634	0,749375979	-0,09%	4%	4%
20		Disch	0,2189804	0,23084604	5,42%	-	-
21	Head coeficient		0,0002175	0,000217505	0,02%	-	-
22	gas type		Natural Gas	Natl. Gas + CO2			

Table 4. Process Parameters for Full Load Test of Main Compressor, Third Section.

Item	Requirement		design	test	Deviation	original ASME type	proposed new req.
1	temperature (K)	Inlet	308,954	309,15	0,06%	8%	8%
2		Disch	352,001	358,516	1,85%	-	10%
3	pressure (kPa a)	Inlet	11843,3	11800	-0,37%	5%	-
4		Disch	19822,3	19550	-1,37%	-	5%
5	capacity (m3/h)	Inlet	588,366	618,479	5,12%	4%	4%
6		Disch	389,077	408,347	4,95%	-	-
7	density (kg/m3)	Inlet	123,785	118,770	-4,05%	8%	5%
8		Disch	157,896	151,988	-3,74%	-	5%
9	specific volume ratio vi/vd		1,2756	1,2797	0,32%	5%	5%
10	pressure ratio P2/P1		1,6737	1,6568	-1,01%	-	-
11	Power (KW)		1593,71	1627,06	2,09%	-	-
12	Molecular Weight		20,479	21,37	4,35%	2%	-
13	pressure rise P2-P1 (kPa)		7979	7750	-2,87%	-	5%
14	P2*(P2-P1)		158162132	151512500	-4,20%	-	5%
15	Head (m)		8457	8556	1,17%	-	-
16	speed (RPM)		11490	11490	0,00%	2%	4%
17	cp/cv	Inlet	1,8561	1,7807	-4,06%	-	-
18		Disch	1,6317	1,6181	-0,83%	-	-
19	flow coeficient	Inlet	0,0512068	0,053827589	5,12%	4%	4%
20		Disch	0,0338622	0,035539339	4,95%	-	-
21	Head coeficient		6,406E-05	6,48083E-05	1,17%	-	-
22	gas type		Natural Gas	Natl. Gas + CO2			

The proposed test was performed with a mixture of light natural gas and 21 percent of CO_2. The tables demonstrate that, although mol weight is not within ASME original tolerance, all other variables are well close to the design ones. Some minor adjustments on the suction pressure were done during the test, in order to match specific volume ratio, flow coefficient, and density.

During the design phase the manufacturer performed stability analysis for both level I and II. For the low-pressure (LP) casing, the analysis predicted a log decrement of 0.034 for level I analysis, as shown in Figure 1. For level II the value of 0.101 was predicted (Figure 123). For the high-pressure (HP) casing the analysis predicted -0.123 in level I and 0.641 for level II. Other parameters and results are presented after the examples. During the shop tests no subsynchronous vibration was observed for both casings. The compressors are currently under commissioning.

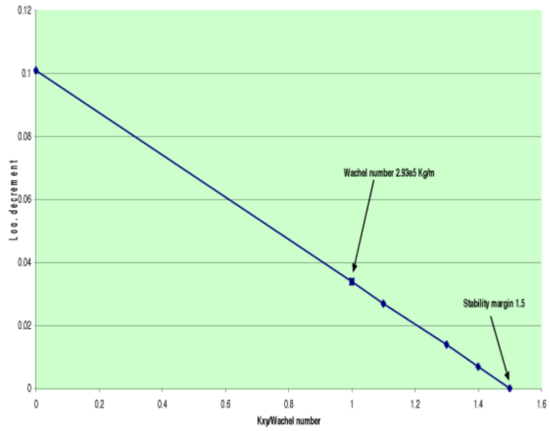

Figure 1. Level I Analysis of LP Rotor, Example #1.

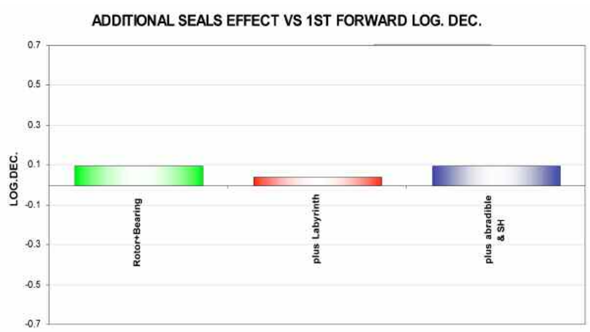

Figure 2. Level II Analysis of LP Rotor, Example #1.

EXAMPLE #2

Another offshore facility for 6 million cubic meters/day is currently under design. The system will be designed with three compression trains. Electric motor and variable speed hydraulic couplings will drive each train with two casings, one with back-to-back LP casing and the other with the HP straight-through. The test specification already includes the new modified requirements. Simulations of the process parameter are presented in Table 5 for the high pressure third section, using nitrogen instead of hydrocarbons. It can be demonstrated that all the proposed requirements can be met.

Table 5. Process Parameters for Full Load Test with Nitrogen, Third Section.

Item	Requirement		design	test	Deviation	original ASME	proposed new req.
1	temperature (K)	Inlet	317.65	298.15	-6.14%	8%	8%
2		Disch	413.7	453.11	9.53%	-	10%
3	pressure (kPa a)	Inlet	7230	6400	-11.48%	5%	-
4		Disch	19781	19100	-3.44%	-	8%
5	capacity (m3/h)	Inlet	1101.16	1058.3	-3.89%	4%	4%
6		Disch	597.4	589.58	-1.31%	-	-
7	density (kg/m3)	Inlet	68.76	71.812	4.44%	8%	5%
8		Disch	126.74	128.905	1.71%	-	5%
9	specific volume ratio vi/vd		1.8432	1.7950	-2.61%	5%	5%
10	pressure ratio P2/P1		2.7360	2.9844	9.08%	-	-
11	Power (KW)		3858	3607	-6.51%	-	-
12	Molecular Weight		21.27	28.01	31.70%	2%	-
13	pressure rise P2-P1 (kPa)		12551	12700	1.19%	-	5%
14	P2*(P2-P1)		248271331	242570000	-2.30%	-	5%
15	Head (m)		18702.26	17416	-6.88%	-	-
16	speed (RPM)		10350	9987.7465	-3.50%	2%	4%
17	cp/cv	Inlet	1.5522	1.5058	-2.99%	-	-
18		Disch	1.4277	1.4536	1.81%	-	-
19	flow coeficient φ	Inlet	0.10639227	0.1059598	-0.41%	4%	4%
20		Disch	0.05771981	0.0590303	2.27%	-	-
21	Head coeficient		0.00017459	0.0001746	0.00%	-	-
22	gas type		Natural gas	Nitrogen			

For the LP casing, the stability analysis predicted a log decrement of 0.074 and 0.065 for the HP casing both for level I analysis. For level II the calculated values were 0.561 and 0.146 for LP and HP compressors.

EXAMPLE #3—GAS REINJECTION COMPRESSOR

For the reinjection and storage gas compressor of an offshore facility the test was originally specified as a Type 1 test. During the bid phase an alternative full density test with nitrogen was agreed to, considering the legal restriction preventing one manufacturer from performing a hydrocarbon test. The specification called for a dry, high-speed direct drive, magnetic bearing motor-compressor configuration. The original specification was relaxed in order to improve bid competition, considering that, at the time of the bid, there were only two manufacturers with sufficient experience with the required technology. The main parameter profiles of the performed test are shown in Figure 3.

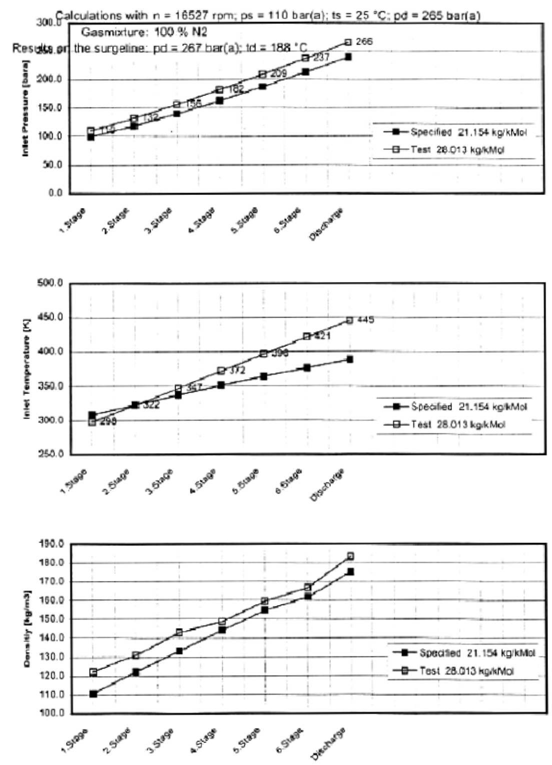

Figure 3. Full Density Test of Reinjection Compressor.

In Table 6 the test and design conditions are compared with both ASME and modified requirements. The actual modified requirements (Table 2) were not yet developed at the time of the test. It is clear that the test did not comply with many of these requirements. Table 7 shows an alternative test procedure that could be used in case of application of the modified requirements, and it can be observed that the new test would be under reasonable compliance. The use of inert gas allowed substantial cost reduction and improved safety conditions on the test bed. The injection compressor is currently running perfectly on the offshore platform.

Table 6. Injection Compressor—Full Density Process Parameters.

Item	Requirement		design	test	Deviation	original ASME	proposed new req.
1	temperature (K)	Inlet	308,95	298,15	-3,50%	8%	8%
2		Disch	389,15	446,14	14,64%	-	10%
3	pressure (kPa a)	Inlet	9950	11000	10,55%	5%	-
4		Disch	23950	26500	10,65%	-	8%
5	capacity (m3/h)	Inlet	295,6	295,53	-0,02%	4%	4%
6		Disch	229,6	208,19	-9,32%	-	-
7	density (kg/m3)	Inlet	111,5	123,160	10,46%	8%	5%
8		Disch	171,5	174,9	1,98%	-	5%
9	specific volume ratio vi/vd		1,5381	1,4201	-7,67%	5%	5%
10	pressure ratio P2/P1		2,4070	2,4091	0,09%	-	-
11	Power (KW)		1727,5	1997	15,60%	-	-
12	Molecular Weight		21,167	28,01	32,34%	2%	-
13	pressure rise P2-P1 (kPa)		14000	15500	10,71%	-	5%
14	P2*(P2-P1)		3,35E+08	410750000	22,50%	-	5%
15	Head (m)		9929,46	10482	5,56%	-	-
16	speed (RPM)		16527	16980,61	2,74%	2%	4%
17	cp/cv	Inlet	1,469	1,582	7,69%	-	-
18		Disch	1,858	1,522	-18,08%	-	-
19	flow coeficient φ	Inlet	0,017886	0,017404	-2,69%	4%	4%
20		Disch	0,013892	0,0122605	-11,75%	-	-

Table 7. Injection Compressor: Process Parameters in Accordance with the New Requirements.

Item	Requirement		design	test	Deviation	original ASME	proposed new req.
1	temperature (K)	Inlet	308,95	293,15	-5,11%	8%	8%
2		Disch	389,15	430,65	10,7%	-	10%
3	pressure (kPa a)	Inlet	9950	10200	2,51%	5%	-
4		Disch	23950	23500	-1,88%	-	8%
5	capacity (m3/h)	Inlet	295,6	307,215	3,93%	4%	4%
6		Disch	229,6	219,345	-4,47%	-	-
7	density (kg/m3)	Inlet	111,5	116,856	4,80%	8%	5%
8		Disch	171,5	163,669	-4,57%	-	5%
9	specific volume ratio vi/vd		1,5381	1,4006	-8,94%	5%	5%
10	pressure ratio P2/P1		2,4070	2,3039	-4,28%	-	-
11	Power (KW)		1727,5	1805,54	4,52%	-	-
12	Molecular Weight		21,167	28,01	32,34%	2%	-
13	pressure rise P2-P1 (kPa)		14000	13300	-5,00%	-	5%
14	P2*(P2-P1)		3,35E+08	312550000	-6,78%	-	5%
15	Head (m)		9929,46	9585	-3,47%	-	-
16	speed (RPM)		16527	16237,803	-1,75%	2%	4%
17	cp/cv	Inlet	1,469	1,582	7,69%	-	-
18		Disch	1,858	1,522	-18,08%	-	-
19	flow coeficient φ	Inlet	0,017886	0,0189197	5,78%	4%	4%
20		Disch	0,013892	0,0135083	-2,77%	-	-

EXAMPLE #4—THE THERMAL INDUCED UNBALANCE CASE

This case has occurred with a gas lift, 14 MW VFD driven, three section compressor train. Although the contracted test was a full Type 1 test, high vibration on the third section (high pressure casing) was detected and corrected during pretests with inert gas at full load and full pressure. With a nominal discharge pressure of 200 bar (2900.7 psi), during the test the vibration exceeded the acceptance level, reached 46 µm and prevented the compressor from going above 160 bar (2320.6 psi). The manufacturer diagnosed the problem as a *thermal induced unbalance vibration*, i.e., double shrink fit between rotor components and shaft causing a bend on the shaft due to differential expansion. There were also concerns regarding insufficient axial gaps between rotor components that could be constraining from free thermal expansion, causing asymmetric expansion, shaft bowing, and vibration.

After modifications on the balance piston design, where two high shrink fits were replaced by one high plus one low shrink fit, the following pretesting did still not go as expected, with the vibration level on the compressor approaching unacceptable limits of 38 µm, at full discharge pressure. Although the modification to the balance piston attachment did greatly improve the thermal sensitivity of the rotor, there was then sensitivity to pressure as the machine approached full discharge pressure. The manufacturer diagnosed excessive dynamic stiffness of the hole pattern damper seal of the balance piston, shown in Figure 4. In fact the rotordynamic simulations have confirmed that the hole pattern seal stiffness was as high as the rotorbearing system, considering the squeeze film damper. The combined effect was an intermediate behavior between double supported and overhang beam. The softer side showed a higher level of vibration than the stiffer balance piston side. To solve this problem the seal design was modified and a hybrid seal was installed. The modified seal has part bladed labyrinth-part hole pattern, as shown in Figure 5.

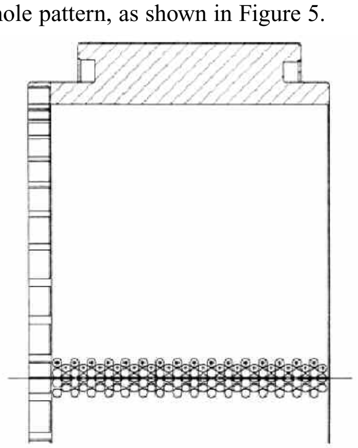

Figure 4. Damper Seal with Hole Pattern.

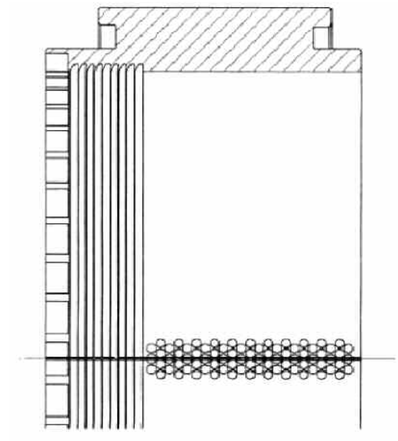

Figure 5. Hybrid Seal Labyrinth plus Hole Pattern.

A new test run was conducted, still with inert gas and the vibration level fell below 33 µm at 193 bar (2799.2 psi). Considering that in the inert gas test the discharge temperature was about 30°C (86°F) above the design temperature, it was expected that the vibration would be even lower if running with hydrocarbons. The final test was conducted with a mixture of hydrocarbons and met all Type 1 requirements. Vibration maximum level was then 27 µm. Figure 6 shows the undamped critical speed map of the high-pressure casing bearings, while Figure 7 presents the level I logarithmic decrement analysis results. Other parameters and results are given in the following section.

This example demonstrates the importance of full load tests to detect dynamic problems that usually do not show up in the standard unloaded mechanical running test. Besides, it demonstrates that the use of inert gas test can be a conservative approach with regard to thermal-pressure combined effects on the rotor bearing system.

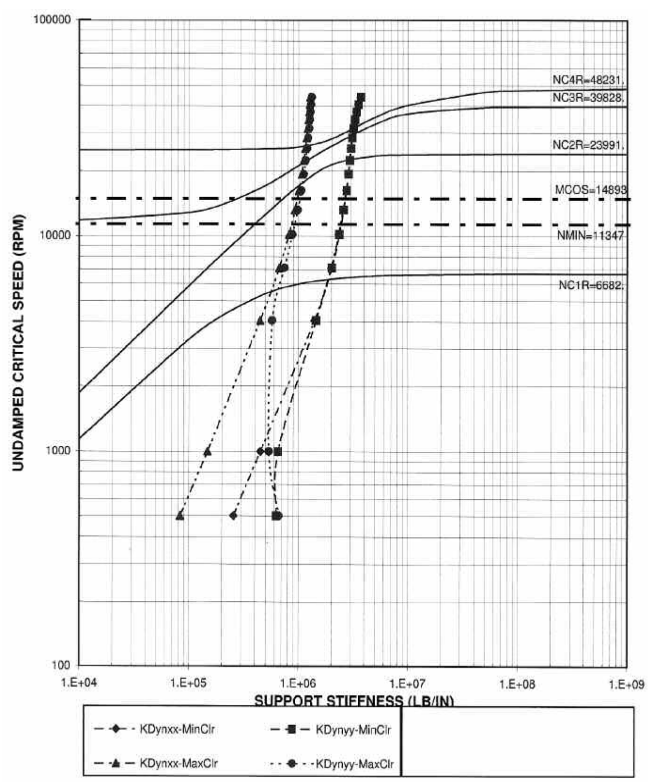

Figure 6. Undamped Critical Speed Map—HP Casing, Example #4.

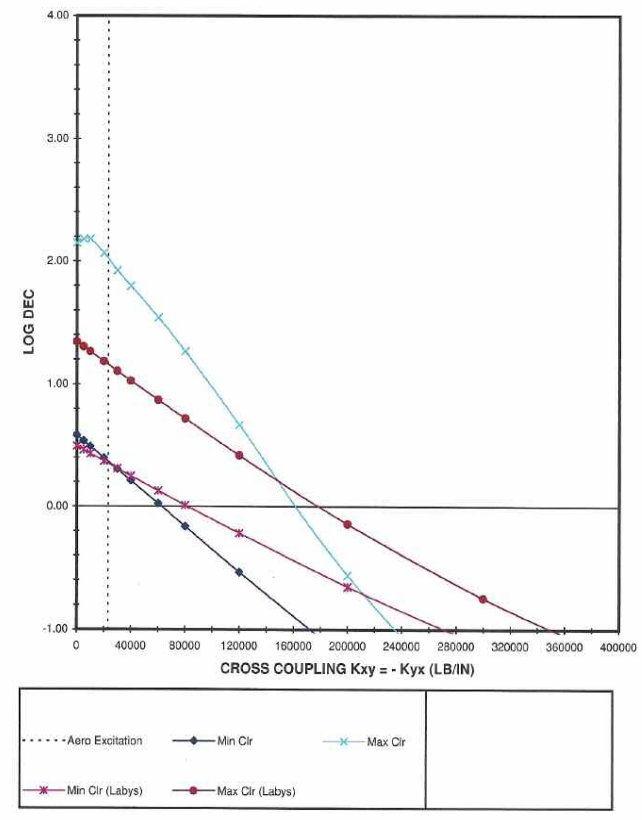

Figure 7. Level I Analysis Plot—HP Casing, Example #4.

SUMMARY OF THE RESULTS

A summary of the results is shown in Table 8, Figure 8, and Figure 9.

Table 8. Data Summary of the Four Cases.

Example	#1		#2		#3	#4	
Rotor	LP	HP	LP	HP	HP	LP	HP
Type	StThg	BtoB	BtoB	StThg	StThg	BtoB	StThg
Bearing Diameter (mm)	120	120	110	100	75	101.5	88.9
Bearing Span (mm)	1526	1618	1730	1514	970	1529.1	1308.6
Hub Diameter (mm)	166	170	175	158	103	153.5	132.3
Bearing span to hub diameter ratio	9.19	9.52	9.89	9.58	9.42	9.96	9.89
Impeller diameter (mm)	460	400	520	450	212	360	309
Exit width at last impeller (mm)	10.65	6.7	12	6	6.7	11.23	5.84
Maximum continuous speed - MCS (rpm)	11490	11490	12494	12494	18800	14893	14893
Rigid support critical speed - NC1R (rpm)	6000	5800	4750	5700	8000	5701	6682
MCS/NC1R	1.92	1.98	2.63	2.16	2.35	2.61	2.23
Av. Dens. (kg/m3) (for API screening plot)	17	76	23	80	141	18	74
Pd / Ps (bar)	43.08/8.57	198.81/43.17	65.34 / 7.52	198.72 / 61.84	240/100	54.25 / 8.61	197.15 / 53.79
Pdx(Pd-Ps) (bar2)(for Kirk's plot)	1487	30942	3778	27200	33500	2476	28260
Level I results - log dec	0.034	0.123 (neg)	0.074	0.065		0.64	0.37
Level II results - complete log dec	0.101	0.641	0.561	0.146		0.57	0.38

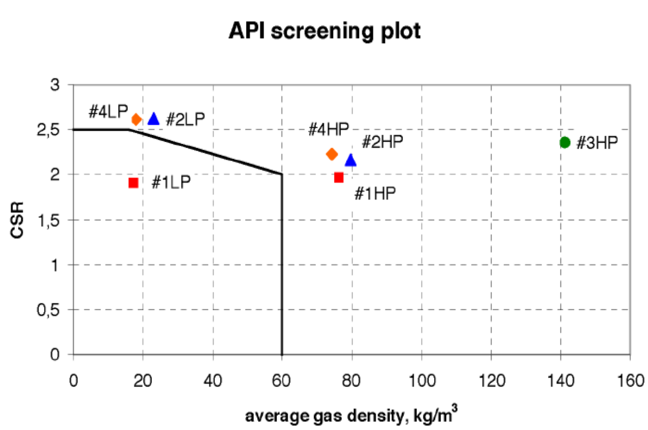

Figure 8. Summary of the Plots for the Four Cases, API Screening.

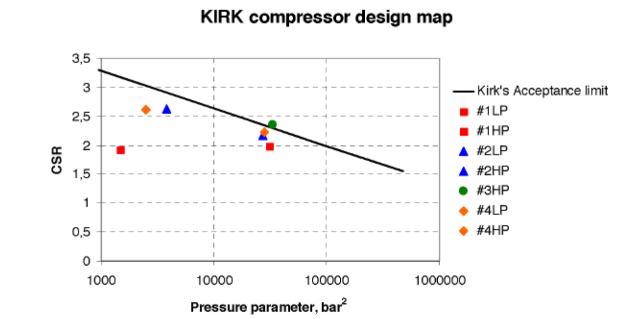

Figure 9. Summary of the Plots for the Four Cases, KIRK Screening.

CONCLUSIONS

In order to prevent subsynchronous vibrations in rotating machines, high-density gas centrifugal compressors included, the latest editions of the API standards have established new design requirements, which include theoretical models for stability analysis. Taking into account the approximate nature of numerical analysis, complementary tests have been used. When directing the focus on rotordynamic instability and other load related phenomena verification, it is possible to introduce some requirements, adapted from the ASME standard and, at the same time, to allow for the utilization, on the test bench, of a gas with a composition different from the design gas, allowing, in some cases, the partial or integral use of inert gases instead of hydrocarbons. Such focalization has, as a consequence, a significant reduction in the test cost, without interfering with the main objective of instability prevention. These modifications have been introduced in the authors' firm's technical specifications and standards.

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