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STABILITY TESTING OF CO_2 COMPRESSORS

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ABSTRACT

This paper presents results from stability analysis and testing on high pressure CO_2 centrifugal compressors, beginning by a preliminary analysis conducted on an academic institution. It includes results of stability tests that were carried out in manufacturer facilities. The tests were done on two different compressor designs, both operating with high content of CO_2 in super critical condition. Due to the similarities on testing and results, only the results of one compressor type are

presented. An important gain from the tests was a clearer understanding of the behavior of this compressor for the stability for different load conditions.

INTRODUCTION

The so called Replicants project was developed for oil and gas production at the Brazilian offshore pre-salt fields. One of main objectives was to develop a single standard project for eight FPSOs to be located at different pre-salt sites, minimizing costs and reaching high productivity. An important design challenge was the compression system, due to the high carbon dioxide content of the associated gas, requiring its removal and reinjection. Considering furthermore that the same project should be capable to handle a high molecular weight range varying among different sites and among different modes of operation in the same site, an elaborate compression plant was devised, comprising twelve compressors in five different services: the Main and Export compressors, respectively upstream and downstream the CO_2 removal system, the Low and High Pressure CO2 compressors, handling the side stream from the membrane separators, and the Injection compressors.

Of concern of this paper are the CO_2 HP and the Injection compressors. The first is a nine stage back-to-back centrifugal compressor while the second is a six stage one, also back-to-back, but of a single section only. The CO_2 compression train was specified to operate on a fixed pressure increase condition, from 4 to 250 bar, but with MW ranging from 26 to 39 (CO_2 content from 33% to 83%). The operating conditions of the Injection compressor are more diverse, with MW varying from 21, in the case the membranes are not in operation, to 39 (CO_2 content from 3% to 83%). The suction pressure is also fixed, of 250 bar, but the discharge pressure depends on the column weight of the injection well and may vary from 300 to 550 bar. Thus, the injection compression will work on more demanding conditions, due to its higher discharge pressure and higher molecular weight range. Nevertheless, despite these differences, both are to operate with high density, super critical fluid.

Considering their highest density condition, the compressor locations in API 617 (2002) Level I screening criteria are depicted in Figure 1. Taking into account the importance of gas density on the stability of centrifugal compressors, stability tests were specified for these two compressors. Similar to API's unbalance response test, it was specified to verify the stability calculations and, as such, the test was done in only one for each type of compressor, i.e. one out of the 16 CO_2 HP compressors for the eight FPSOs and another for the 16 Injection compressors. Thus, the testing procedure had to contemplate different operating conditions and also extrapolation criteria to determine minimum stability margin applicable to all 16 compressors based on test results from only one unit.

There are two sources of instability inherent to a centrifugal compression process, the first of which is the labyrinth seal that separates the different levels of pressure inside the compressor and the second is the actual movement of the gas caused by the rotation of the impellers, generally referred to as "aerodynamic cross-coupling." Published papers and internal records of operator companies show these fluid induced instabilities may lead to high production losses. Thus, stability analysis is part of the dynamic design of the rotor. An excellent reference on these issues is the recent book by Childs (2013) and much is in API 684 (2005).

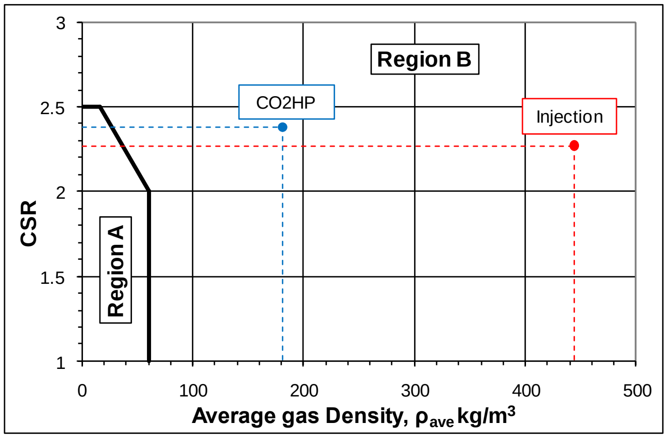


Figure 1. Location of the compressors in Level I screening criteria for the highest density condition

In the current context of dynamic analysis of rotating machines, there are various procedures for stability analysis of these systems, many of them based on numerical modeling and analysis of eigenvalues and eigenvectors. It is known, however, that there are many uncertainties associated with numerical modeling, due to theoretical approximations or uncertainties related to manufacturing and assembly of the rotor and its components. Indeed, a survey on numerical modeling and stability analysis conducted by Kocur et al. (2007) points out to variations on the results that range from 10 to 1. Therefore, techniques were developed for experimental evaluation of the stability of rotors.

The operator has previous experience with another OEM with stability testing with a magnetic bearing exciter on a low pressure, low density downstream compressor, without any device to enhance damping or reduce cross-coupling (Pettinato et al., 2010). Stability was low and testing was important in order to assure it was within the specified limits.

The operator has a freighted platform where this OEM was involved in stability testing with magnetic bearing exciters of high pressure, high density back-to-back CO_2 compressors for injection service (Colby et al, 2012). For those compressors there were devices used to enhance damping - squeeze-film dampers centered by a mechanical spring at the tilt pad journal bearings and hole pattern seals at the division wall and there were devices used to reduce cross-coupling - shunt holes and swirl brakes at the division wall hole pattern seals and swirl brakes at the impeller eye toothed labyrinths. All of those compressors had dry gas casing end seals.

Overall stability is a net result between the forces caused by destabilizing mechanisms and those from any stabilizing devices. The present ones are similar to the high pressure, high density CO_2 compressors mentioned above and have the same stability enhancing devices. They also have tilt pad bearings and dry gas casing end seals. In common with the equipment mentioned above, the present ones share the fact that the guaranteed point is a condition where compressors seldom operate. Testing is required in more than one load condition in order to obtain an experimental curve of log decrement vs. load.

ACTIVITIES AT UNICAMP

In order to provide the operator with a better understanding of stability testing, theoretical and experimental studies were conducted by researchers at Campinas State University (UNICAMP). The most important feature of the test bench, shown in Figure 2, is that it is basically a Jeffcott rotor supported on cylindrical bearings with an electromagnetic actuator on the non-drive end. Therefore, its dynamic behavior is well known and, hence, due to its simplicity either for practical experiments or for theoretical modeling, the test rig can be considered appropriate for preliminary testing of the proposed method.

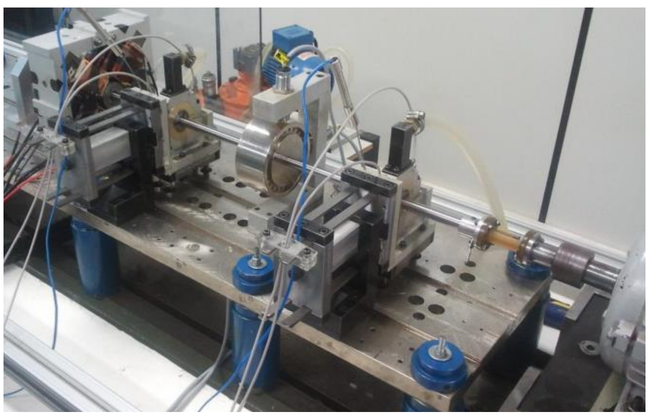
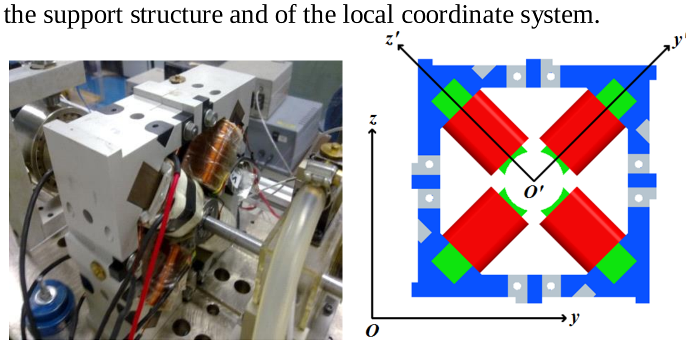


Figure 2. UNICAMP test bench

The electromagnetic exciter

To obtain the frequency response of the rotor and perform the procedures for stability analysis, the test bench is equipped with an electromagnetic actuator as a source of external excitation (Mendes, 2011). Figure 3 provides additional view of the support structure and of the local coordinate system.



(a) *(b)*

Figure 3. (a) Support structure of the magnetic actuator; (b) Coordinate system rotated by 45° (Mendes, 2011).

A finite element simulation showed that the lowest natural frequency of the structure and coils set is around 576 Hz (Figure 4), which is far above the operating range of the test bench (approximately 0 to 100 Hz). To excite all the inertia involved in the test bench, the actuator is able to apply an excitation of 150 N up to 100 Hz, considering an air gap of 2.5 mm and a current of 3A.

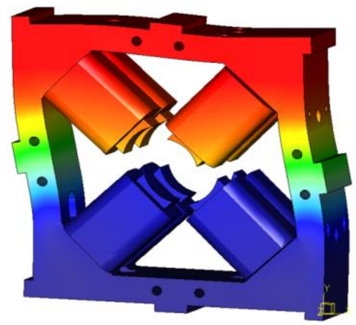


Figure 4. First mode of vibration of the actuator at 576 Hz (Mendes, 2011).

The actuator force control is done by controlling the magnetic field (Chiba et al., 2005), as shown in Figure 5, where B_{ref} is the magnetic field corresponding to the reference force, which is compared with the measurement of the magnetic field made by a Hall type sensor. This comparison is the input to the proportional controller. The output voltage of the controller is converted into current by a PWM amplifier (operating in current control mode) and sent to the actuator coil, generating a magnetic force F_m. Thus, using the differential set with a magnetic bias field (B_b), the resulting linear force can be written as (Schweitzer et al. 2009; Maslen, 2000)

$$F_m = 4 \cdot A_g \cdot B_b \cdot B_{ref} / \mu_g \qquad (1)$$

where A_g is the area of each pole, B_b is the bias magnetic field, B_{ref} is the desired magnetic field, μ_g is the air permeability. The force measurement model was calibrated using a short shaft assembly monitored by load cells and accelerometers in both ends (Castro et al. 2007).

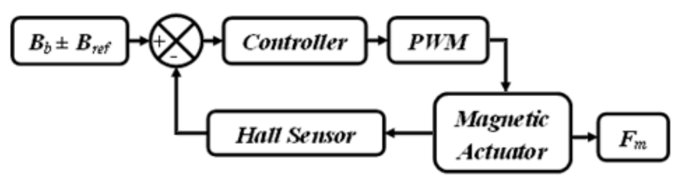


Figure 5. Diagram of the Magnetic Actuator System (Mendes, 2011)

Unlike a magnetic bearing, in the case of a magnetic actuator, control is performed based on the magnetic field because the goal is that the actuator applies a force on the shaft of predetermined external excitation, regardless of the shaft displacement.

Test Methodology

The methodology for the tests aims to establish a stability criterion for rotating systems based on estimation techniques of the logarithmic decrement, using an electromagnetic actuator as a source of external excitation. One of the proposed techniques, termed Blocking Testing (Kanki et al, 1986, Pettinato et al., 2010), is applied in time domain and does not require the measurement of the excitation force, while in classical methods of modal analysis, it is necessary to know the external excitation, as this is part of the experimental model. This procedure is compared with the classical excitation techniques in the frequency domain, called by Pettinato et al. (2010) as Stepped Sine Testing.

During the tests, the latter was applied first in order to establish, through the analysis of the frequency response, the forward and backward natural frequencies, which were then used for the Blocking Testing excitation. Although instability generally appears only through the excitement of a forward precession mode, one must also identify backward modes in order to differentiate them properly and get the log dec strictly associated with the forward mode. After all, even when they are not dominant, both are present in the dynamic response, affecting the modal damping factors and thus the logarithmic decrement (Cloud et al., 2009). Strictly speaking, forward and backward excitation are actually not necessary. Accurate measurements of the forward mode can be obtained using an appropriate ID techniques (MOBAR or PEM) using only forward or directional excitation. The best estimates for a particular mode are provided when the excitation focuses on that mode, but it's not necessary to excite both modes in order to get accurate estimates of either. Nevertheless, in order to get a broader insight of the results, tests were done with forward and backward frequency excitation for each operation condition.

In the test bench, the operating condition was modified changing the rotation speed. Thus, in the tests, the same procedure was applied to a single rotor configuration, whereas with five different levels of stability represented in this case by five different rotation frequencies of the rotor, namely, 50 Hz, 55 Hz, 70 Hz, 75 Hz and 80 Hz, i.e., above and below the critical speed which is about 62 Hz.

Because it is a validation of a testing procedure for the identification of the logarithmic decrement, it becomes essential to evaluate its sensitivity to different levels of damping. The behavior of the real part of the eigenvalues to be estimated at different rotational speeds, gives the tendency to change sign from negative to positive, thus setting the threshold of instability in the frequency domain. In the case of the logarithmic decrement, this tendency goes from positive to negative according to:

$$\delta = \frac{-2\pi\,\mathrm{Re}(\lambda)}{|\mathrm{Im}(\lambda)|} = \frac{2\pi\zeta}{\sqrt{1-\zeta^2}} \tag{2}$$

where δ is the logarithmic decrement, λ the eigenvalue and ζ the damping factor.

For time domain identification of the stability level of a rotating system, the MOBAR method was studied and tested. The MOBAR, Multiple Output Backward Autoregression (Hung and Ko, 2002; Cloud et al., 2009) is an identification method of modal parameters that considers multiple degrees of freedom, which are analyzed in the time domain. An important feature of this method is that it does not require the measurement of the excitation signal.

The recommended test for generating the output response signals is the Blocking Testing, where a rotating excitation at a particular frequency is applied for a period of time and then stopped abruptly. The application of MOBAR on the results of the Blocking Testing provides a dynamic system whose eigenvalues are used to calculate the logarithmic decrement. Moreover, this rotating excitation force can be forward or backward, as shown in Equation 3 (Jang et al., 1996):

$$\begin{aligned} f_{yF} &= F\cos(\omega t + \Phi); \quad f_{zF} = F\sin(\omega t + \Phi) \\ f_{yB} &= F\cos(\omega t - \Phi); \quad f_{zB} = -F\sin(\omega t - \Phi) \end{aligned} \tag{3}$$

Figure 6 shows the response of the system under Blocking Testing. Note the decrease in amplitude after the rotating force has been interrupted. Thus, one can analyze the displacement decrement after excitation cessation (Cloud, 2007).

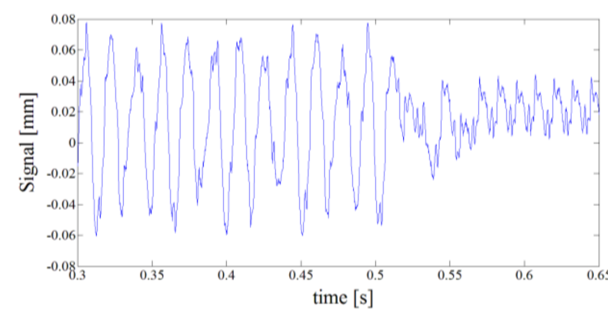


Figure 6. Blocking Testing – Forward excitation at 4800 rpm Horizontal response on bearing 2

Another way to obtain the modal parameters of the system is to perform a sine sweep, the Stepped Sine Testing, where non synchronous excitations are applied to the rotor in a predetermined range of frequency in two orthogonal directions of the vibration plane of the rotor and the frequency response function is obtained in the frequency domain. After the test, one may apply classical techniques of modal analysis in accordance with the literature, such as Ewins (1984), due to the knowledge of the external excitation force. In this paper, however, a model identification technique based on error minimization was applied (PEM - Prediction Error Method). The systems identified are the type SIMO (Single Input Multiple Output). Therefore, the responses of horizontal and vertical displacement sensors were considered. Figure 7 illustrates the application of this method. After obtaining the experimental FRFs (black), the mathematical model was adjusted to the experimental points, which allowed the extraction of modal parameters such as natural frequency, damping factor and vibration modes. For each rotation, two FRFs were identified (y and z directions) in the frequency range of 40 to 80 Hz. The reconstitution at 3900 rpm excited in horizontal direction (y) is represented in Figure 7 (green line).

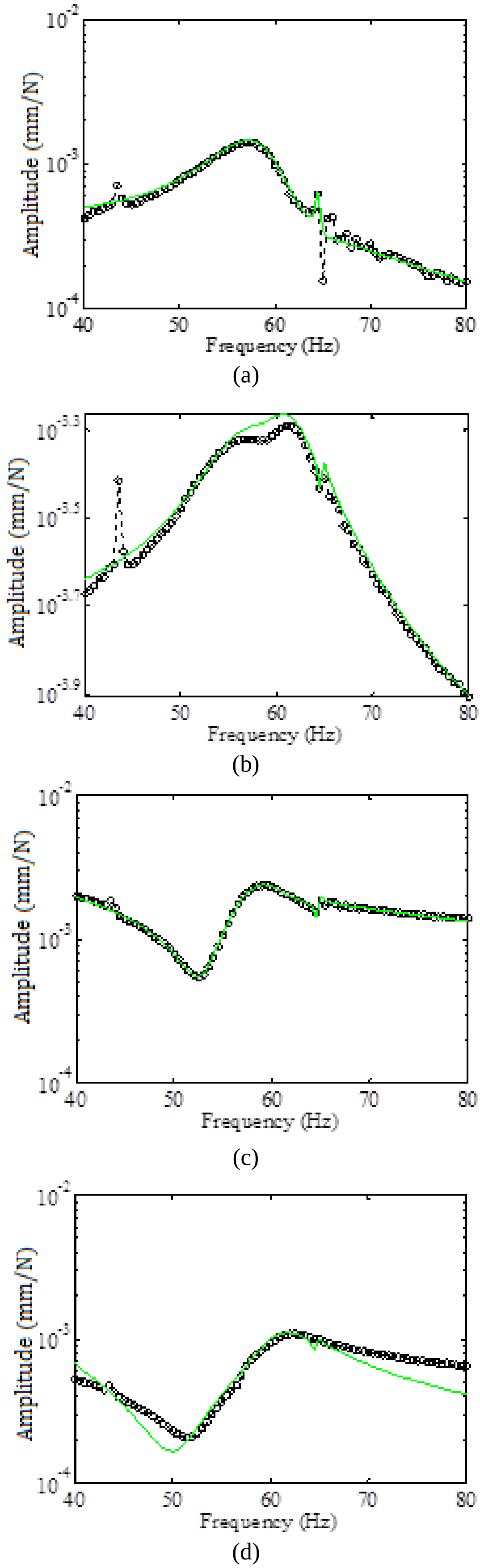


Figure 7. FRFs Reconstitution at 3900 rpm excited in horizontal direction: (a) Bearing 1- horizontal; (b) Bearing 1 - vertical; (c) Bearing 2 - horizontal; (d) Bearing 2 - vertical

Results and discussion

Figure 8 shows the logarithmic decrement vs. rotation for the applied identification methods. The experimental results are compared to the responses from a software developed at UNICAMP for rotor dynamic analysis. It is noticeable that the numeric simulation values are in very good agreement when compared to forward precession experimental results. The logarithmic decrements related to backward natural frequency are more conservative concerning the instability analysis. However, it is important to point out that even for backward the tendency is in good accordance between experiments and numerical simulations.

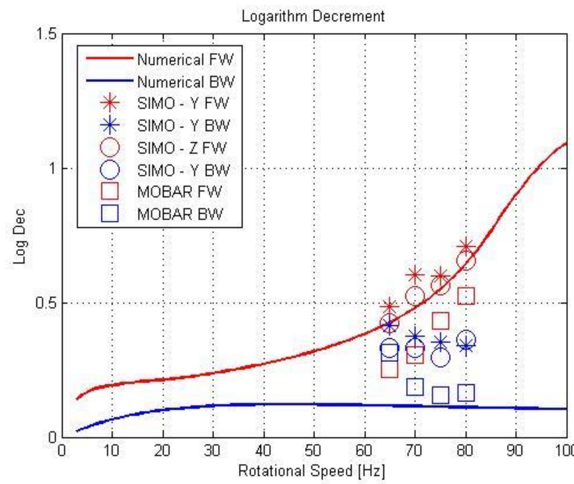


Figure 8. Log Decs vs. Rotation from Simulation, SIMO (Experimental Stepped Sine Testing) and MOBAR (Experimental Blocking Testing) results

The frequency domain log dec estimates agree more closely with the numerical forward mode predictions for the test rig, than the time domain estimates, although for the backward mode, better correlation with the numerical predictions were achieved by the time domain estimates. However, frequency domain tests demand higher time-costs and also the previous knowledge of the excitation force, which measurement can be one more uncertainty source in the entire procedure. The time domain tests are easier and faster to be accomplished, but attention must be given to the measurements and identification techniques to achieve an acceptable accuracy. Moreover, in the tests carried on in the laboratory, the time domain tests leaded to more conservative stability analysis.

Finally, it is important to highlight the main objective of this section. The approach with the academic environment allowed a better understanding of the whole procedure theoretical basis of the stability tests and procedures. Hence, a standard procedure to be regularly adopted to qualify rotating machines to be applied at the operators' plants could be established.

DESIGN OF COMPRESSORS FOR HIGH DENSITY GASES AGAINST FLUID INDUCED INSTABILITY

The early history of the OEM in the usage of stability enhancing devices to insure stable operation with high density gases has been described in a series of papers by Coletti and Crane (1981), Shemeld (1986) and Memmott (1990 and 1992). These papers also describe the application of full load, full pressure tests in the shop on inert gas, starting in the 1970s, and on hydrocarbon gases, starting in the 1980s, to validate stable operation.

During the 1970s, the OEM started using the stability enhancing devices as described as follows and applied them to both original equipment and upgrades. Tilt pad seals were first used by the OEM in 1972 and have been applied to over 600 compressors since then. Tilt pads were inserted into the outer rings of oil-film seals to make then act like tilt pad bearings and not like instability inducing sleeve bearings. Damper bearings (Squeeze-film dampers in series with tilt pad journal bearings) were first used by the OEM in 1973 and have been applied to over 800 compressors since then. These are used to provide optimum damping at the journal bearing locations. Shunt holes were first applied by the OEM in 1974 and have been routinely applied to most medium and high pressure compressors since then. Shunt holes tap the last stage diffuser to send higher pressure gas radially in to the entrance of the division wall seal of back to back compressors or at the balance piston seal of an in-line compressor. This eliminates the destabilizing tangential inlet swirl to those seals that otherwise would come down from the back side of the last impeller. See the papers listed in the previous paragraph for sketches of these parts and the early history of their usage. For a more recent survey of damper bearing applications, see (Memmott, 2010).

Honeycomb seals in centrifugal compressors at the division wall or balance piston were first applied as damping devices in the mid-1990s. These were ideal locations to apply the large amount of direct damping that could be obtained from these seals for high density gases to stabilize the compressor. Honeycomb seals had been used years before that at the balance pistons of some syn gas compressors before rotordynamic benefits were recognized. The experience of three different OEMS showed that the honeycomb seals needed to be deswirled and shunt holes were used to do this. See the papers by Memmott (1994), Gelin et al (1996), and Camatti et al (2003). Analytically hole pattern seals have similar dynamic characteristics as honeycomb seals and a summary of experience with the use of honeycomb and hole pattern seals is given by Memmott (2011). The OEM has applied hole pattern and honeycomb seals to well over 400 compressors, most of them of the hole pattern type.

Papers were written by Nielsen and Myllerup (1998), Nielsen et al (1998), and Moore and Hill (2000) analyzing and describing the application of swirl brakes, which are used to deswirl the gases entering toothed labys, such as at the impeller eyes, and at the hole pattern and honeycomb seals at the division wall or balance piston.

Since the early 2000s magnetic bearing exciters have been used by the OEM in full load full pressure tests to validate the predicted log decs and the design of high pressure high density compressors for stable operation. All of those tests were conducted on compressors with hole pattern seals and shunt holes. Some of the compressors had squeeze-film dampers in series with the journal bearings and some did not. Data was taken at low, intermediate and full pressures. With the hole pattern seals as the pressure and thus the density increased the log dec increased significantly, unlike what would be expected with toothed labyrinth seals. There was good agreement between the testing, which was done with a frequency sweep and a single degree of freedom model for the data collection and the analysis of the data, which analysis is described in the next section of this paper. See the papers by Moore et al (2002), Moore and Soulas (2003), Gupta et al (2007), Soulas, et al (2011), Memmott (2011), Gupta (2011), and Colby et al (2012).

For all of the stability analysis shown in those papers and in this paper, the log decs were calculated with the effects of the toothed labyrinths, the hole pattern seals and with the inclusion of the Modal Predicted Aero Cross-Coupling (MPACC) (Memmott, 2000a) for the effects of the anticipated cross-couplings from the impeller-diffuser interaction. See the discussion in the next section for a description of the MPACC number and the necessity for its inclusion along with the effects of the toothed labyrinths and the hole pattern seals.

Also see (Chochua and Soulas, 2006) where a method was proposed for computations of rotordynamic coefficients of deliberately roughened stator gas annular seals using computational fluid dynamics (CFD). Rotordynamic coefficients predicted by CFD for a hole pattern seal were in good agreement with seal test data and the ISOTSEAL 1-d code prediction.

Because of the experience described in Memmott (1994) the OEM for that compressor began to use honeycomb seals at the division wall or balance piston in all high-pressure high-density applications and also in many medium-pressure medium-density applications. Since 1999 this OEM has used hole pattern seals at those locations. There has been no magnetic bearing exciter testing by this OEM of compressors with toothed labyrinth seals at the division wall or balance piston.

A further description from the 1994 paper will illustrate more in depth why toothed labyrinths are no longer used by this OEM at the division wall or balance piston of high-pressure high-density compressors. This is a back-to-back compressor and has a division wall. There were no problems on the low pressure mechanical test. On the FLFP HC test there were unacceptable levels of SSV (*sub-synchronous vibration)* with a toothed labyrinth at the division wall, even though shunt holes were used. This was not a test with a mag bearing exciter, it was done before the mag bearing exciters were used. The shunt holes had been used before on hundreds of compressors with great success. But toothed labyrinths do not provide much direct damping. When the toothed labyrinth was replaced by a honeycomb seal without shunt holes (they were just covered up) there still were unacceptable levels of SSV, and the frequency was higher. It showed that the large direct damping from the honeycomb was insufficient without deswirling. It also showed that there was a significant direct stiffness term from the honeycomb seals. The shunts were uncovered and it was stable with the honeycomb seal and this proved again that shunt holes work.

The paper by Gelin et al (1996) describes much the same experience. It describes two compressors which each started out with a toothed labyrinth at the balance piston and each of which ended up with a honeycomb seal with a new shunt hole system. This also was before mag bearing exciter testing

STABILITY ANALYSIS BY THE MANUFACTURER

The log decs were calculated using the OEM's rotor dynamic software suite, which is a state-of-the-art program linking the centrifugal compressor aerodynamic selection and compressor modeling tools with the rotor dynamic programs into one cohesive engineering tool (Ramesh, 2002).

The tilt pad bearings were analyzed by the program of Nicholas (Nicholas, et al, 1979). The stability analysis was done by the transfer matrix program of Lund (Lund, 1974, Smalley, et al, 1974). The toothed labyrinths were modeled by the two control-volume bulk flow method program DYNPC28 by Kirk (Kirk, 1985. 1986. 1990), The hole pattern seals were modeled by the ISOTSEAL program of Texas A&M (Kleynhans and Childs, 1996, Holt and Childs, 2002, Soulas and San Andrés 2002, and Childs and Wade, 2003). Usage of the tilt pad bearing and stability programs, especially as related to the requirements of API 617, is described in (Memmott, 2003).

The excitation arising from the centrifugal impellers is estimated using a modified form of the Alford and Wachel numbers (Alford, 1965 and Wachel and von Nimitz 1981). After benchmarking the formulation on numerous test cases operating with different mole weight gases, the OEM adopted the form referred to as the Modal Predicted Aero Cross-Coupling (MPACC) (Memmott, 2000a). Instead of the factor of MW/10 that is used in the Wachel number it is replaced by 3. By taking a modal sum of these excitations at each impeller based on the first forward whirling mode shape, an effective aerodynamic cross-coupling is calculated and applied at the mid-span of the rotor for the API 617 7^{th} Edition Level II analysis along with the dynamic coefficients from the toothed labyrinths and the hole pattern seals. The formula at each impeller is the same as was adopted by API 617 7^{th} edition for the arithmetic sum of the anticipated cross-couplings at each impeller used in the Level I screening criteria without the labyrinths and without the hole patterns.

In the paper (Memmott, 2000a) it was shown that the MPACC number needed to be included in the Level II analysis in addition to the stiffness and damping coefficients for the labyrinths in order to predict instabilities seen in several different types of compressor, a 400 bara (5800 psia) discharge back-to-back gas injection compressor, a large propane compressor with multiple side streams, a 168 bara (2432 psia) discharge in-line CO_2 compressor and a 94 bara (1368 psia) discharge in-line syn gas compressor.

The gas injection compressor, discussed in (Memmott, 2000a), is predicted to be stable from a Level II analysis with all of the labyrinths, but without the MPACC number. It had a toothed division wall labyrinth with shunt holes. If the MPACC number was included along with all of the labyrinths it is predicted to be unstable. It was not stable until the division wall toothed labyrinth was replaced with a honeycomb seal with shunt holes. Then it was predicted to be stable with the inclusion of the MPACC number, the toothed labyrinths, and the honeycomb seal with shunt holes. This experience is also discussed in (Memmott, 1994), which was written before the introduction of the MPACC number.

The propane compressor is predicted to be stable with a toothed balance piston seal and the other toothed labyrinths, but without the MPACC number. If the MPACC number is included along with all of the labyrinths it is predicted to be unstable. It was not stable until application of damper type seals with small pockets and swirl brakes at all of the labyrinths. With the MPACC number and the coefficients for these damper seals then it is predicted to be stable. This compressor experience also showed that the modal sum of the anticipated cross-couplings at each impeller should be applied and not with an overall density ratio of outlet density over inlet density or with a section by section sum based on density ratios at each section, This experience is also discussed in (Memmott, 2000b).

The CO_2 compressor and the syn gas compressor, both discussed in (Memmott, 2000a), are predicted to be stable with a toothed balance piston seal with no shunts and no deswirling and all the other toothed labyrinths and non-damper bearings, but without the MPACC number. If the MPACC number is included along with all of the labyrinths then both are predicted to be unstable. They were not stable until application of damper bearings and shunt holes at the balance piston. Swirl brakes were not used at those locations. With the MPACC number and damper bearings and the coefficients for the toothed labyrinths with shunts at the balance piston both are predicted to be stable. The CO_2 compressor is also discussed in (Memmott, 1990), with no results on modeling of the labyrinths and again in (Memmott, 2010) with the results for modeling the labyrinths. For the syn gas compressor without the 30/MW x MW/10 = 3 modification to the Wachel formula the instability seen with the non-damper bearings and no shunt holes is not predicted. With the modification it is predicted to be unstable. For the syn gas compressor the MW is 9.3 in the make-up section and 11.44 in the recycle impeller. The syn gas compressor experience led to the modification to the Wachel formula. The experience with the syn gas compressor is discussed again in (Memmott, 2010), with the results for modeling the labyrinths.

For the tested CO_2 HP compressor, analytical curves of log dec vs. average gas density were developed to show the striking effect of increasing log dec with increasing density at the hole pattern seals at the division wall and to study the effects of the various parts in the system. See Figure 9. The effects of journal bearing design clearances, squeeze-film damper design clearances and eccentricity, and hole pattern clearance on the stability of the compressor can be studied. Also included are the effects of minimum and maximum oil inlet temperatures. Minimum design oil temperature was paired with minimum design bearing and damper clearance and maximum allowable eccentricity for the damper for the stiffest system. Maximum oil inlet temperature was paired with maximum design bearing and damper clearance and centered damper for the softest system. The log decs include the effect of the MPACC number. The testing was done up to a value of approximately 7 LB/FT^3 (112 kg/m^3), and this was full load and full pressure, but not full density. The last three points on each curve had the same final discharge pressure and approximately the same load, but not the same MW. It is seen that plotting against discharge pressure or against load is not always appropriate.

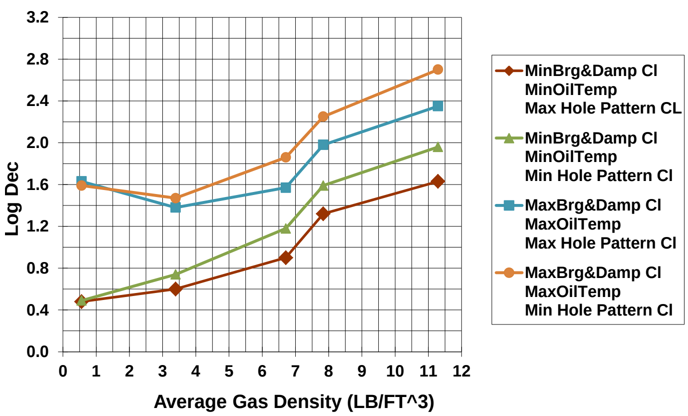


Figure 9. Log dec vs. Average Gas Density for the CO_2 HP compressor

Prior to the tests, the log decs of the tested compressors were calculated for the stipulated operating conditions and the measured bearing and gas annular seal running clearances throughout the compressor by the OEM. Table 1 presents the calculated log decs for the tested CO_2 HP compressor based on the API 617 Level II analysis with the inclusion of the MPACC number.

Table 1: Calculated Log Decs for the tested CO_2 HP compressor

Precession Mode	Low Load	50% Load	100% Load
1st Forward	0.640 @ 5414 cpm	0.759 @ 6549 cpm	1.092 @ 7613 cpm
1st Backward	0.931 @ 5327 cpm	2.041 @ 6585 cpm	2.997 @ 7752 cpm

Curve (b) in Figure 13 shows the 1st forward mode calculated values of log dec in terms of average gas density for the tested CO_2 HP compressor on the three different load conditions. As the average gas density increases, so does the damping of the damper seal in the division wall which leads to the growth of the log dec, as shown in Table 1 and Figure 13.

STABILITY TESTING AT THE MANUFACTURER

Test Description

Stability tests were carried out in the first CO_2 HP and Injection compressors to undergo factory acceptance tests, from a total of 16 each. Hence, they are called by the operator as a type test, that is, they are not for all equipment, but only for one of a type. As discussed in the Introduction, the purpose of the tests is to establish the minimum stability margin of a machine type based on test performed on one of the machines. The operator technical specifications stated that the log dec should be measured on at least two different operating conditions in order to obtain an experimental curve of its dependency to cross coupling stiffness for the tested compressor. The extrapolation of this curve to the worst design conditions, i.e., those that produce the smallest log dec is discussed next.

Figure 10 describes the procedure recommended by the operator to obtain corrected log decs from on a hypothetical compressor. It depicts, in blue, three curves of calculated log dec, of which the center one corresponds to the actual test conditions, such as bearing clearance and preload and the lowest one is the curve of minimum calculated log dec, i.e., the calculated curve for the worst conditions of these parameters, e.g. minimum clearances. Similarly, the three curves in red represent the same curves, but corrected from the results of the test. Of these, what matters is the curve of minimum corrected log dec obtained from the minimum calculated log dec curve, translated by the value of the largest difference between the measured and calculated values of log dec for the different test conditions.

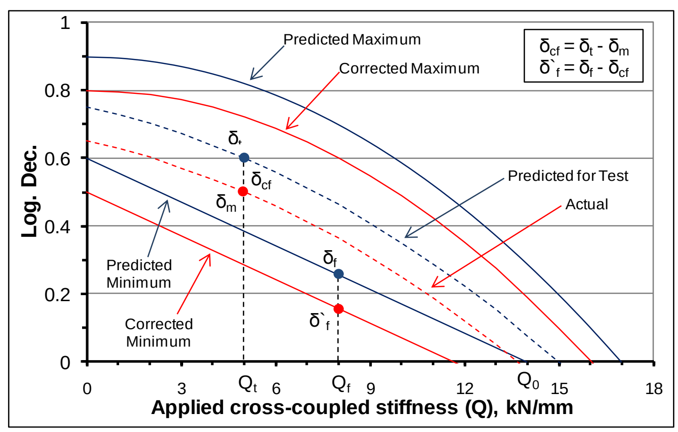


Figure 10. Hypothetical curves of Log Dec vs. cross-coupled stiffness – calculated curves in blue; corrected curves in red

In accordance with the procedure established by the manufacturer for factory acceptance tests, the stability tests were performed in sequence with another type test, the FLFPFS - Full Load, Full Pressure and Full Speed test, using the same test bench in three operating conditions, called 100%, 50% and Low load. The first condition is the same of the FLFPFS test, the second with approximately 50% gas power and the last one with very small power, all three conditions established before the test and with the log decs also previously calculated (see Table 1). The rotation speed during stability testing was the maximum continuous, the same speed of the FLFPFS. That is, while on the UNICAMP bench the stability level was varied by rotation, at the factory it was through load. The excitation applied to the rotor was accomplished by a magnetic bearing coupled to an extension of the rotor on the non-drive end, as represented by the sketch of Figure 11.

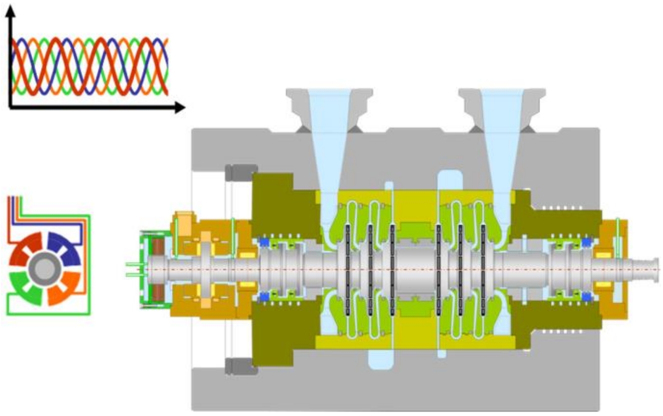


Figure 11. Representation of a compressor with the magnetic bearing attached to the non-drive end.

Test Results

Stability test data was acquired and analyzed by a consulting firm specialized in modal testing. Similar to what has been done on the UNICAMP bench, a frequency domain MDOF method was used. It was based on the PolyMAX modal parameter estimation (Peeters et al., 2004a,b and El-Kafafy et al., 2012) and was applied to determine the natural frequencies and damping of the first forward and backward modes. As represented in Fig. 11, top left, signals that excite the forward or the backward modes were provided to the windings through the magnetic bearing controller. A sweep sine signal over the entire frequency range was used. Figure 12 shows spectra produced by successive sine wave frequency sweeps. Time Domain MDOF parameter estimation was also attempted but did not produce consistent results

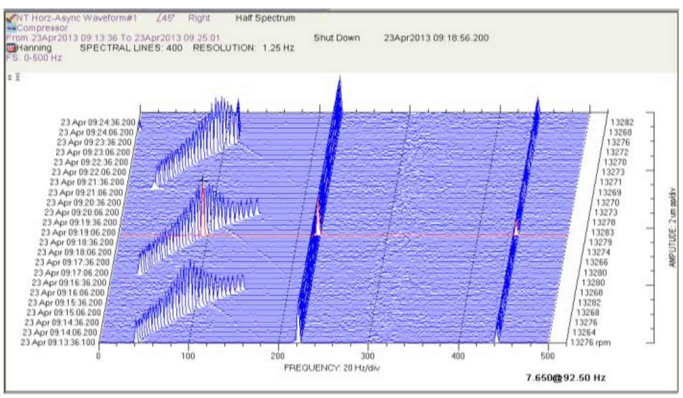


Figure 12. Spectra generated by sine wave frequency sweeps

Table 2 shows the measured log decs. In blue are the results that presented acceptable repeatability, that is, when at least five measurements were within the range of the mean value ± 0.03, in red when repeatability was unacceptable. For forward modes, only the frequency domain results for Low and 50% loads presented acceptable repeatability. All three Forward mode results, good or not, are higher than the respective calculated values presented in table 1. Hence, the stability analysis done by the manufacturer is basically conservative.

Table 2. Measured Log Decs - CO_2 HP compressor

	Frequency Domain MDOF		
	Hz	cpm	Log Dec
Low Load			
Forward	93.4	5606	**1.18**
Backward	91.3	5479	**1.33**
50% Load			
Forward	114.1	6844	**1.73**
Backward	108.7	6524	**1.15**
100% Load			
Forward	109.0	6538	**1.34**
Backward	107.7	6461	**0.87**

Based on the two forward log decs with acceptable repeatability, the required Fig. 10 curves were constructed and presented in Figure 13, where the dashed lines are calculated and the full ones are the corrected ones based on the test results. In line with Figure 10, the corrected curves of Figure 13 were obtained as follows:

- For Low Load, the Corrected Maximum and Minimum values are the predicted ones plus the difference between the measured and predicted values for the tested rotor.
- For 50% Load, an equivalent procedure applies.
- For 100% Load, since there is no measured value for this condition, the three Corrected values, Maximum, Tested Rotor and Minimum are calculated by adding to the predicted values the difference between the measured and predicted values for 50% load.

The abscissa was changed to average gas density, since, as discussed in the previous section, it is more straightforward and more appropriate for high density compressors than cross-coupled stiffness. Once again, the figure shows that the calculated values are very conservative, e.g., while the minimum predicted log dec is around 0.5, the test result for low load shows that this value should be above 1.0. Comparing Figure 13 with Figure 1 it can be seen that the average gas density for the shop load tests did not reach that which can be seen in the field. Nevertheless, as presented next, the full load stability response already presented a flat response.

For comparison purposes, Figures 14 and 15 present frequency response functions for Low load and 100% load. While for the first FRFs, the natural frequency is clearly defined by the peak amplitude and phase shift on the sensor signals, for the second set of FRFs, identifying the natural frequency is very difficult since the curves present a flat aspect. It is noteworthy that another compressor with very similar specification but from another manufacturer also presented this flat response when tested at full load.

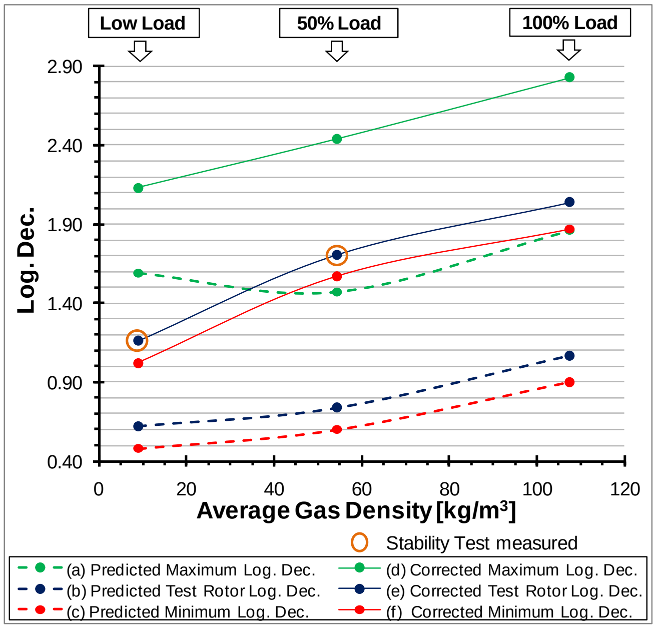


Figure 13. Curves of Log Dec vs. average gas density for CO_2 HP compressors – dashed lines are calculated; solid lines are corrected from test results

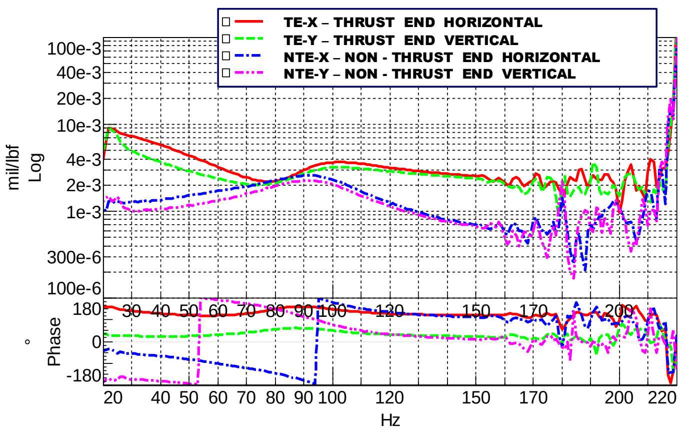


Figure 14. FRFs for the four compressor probes – forward precession and 0% load

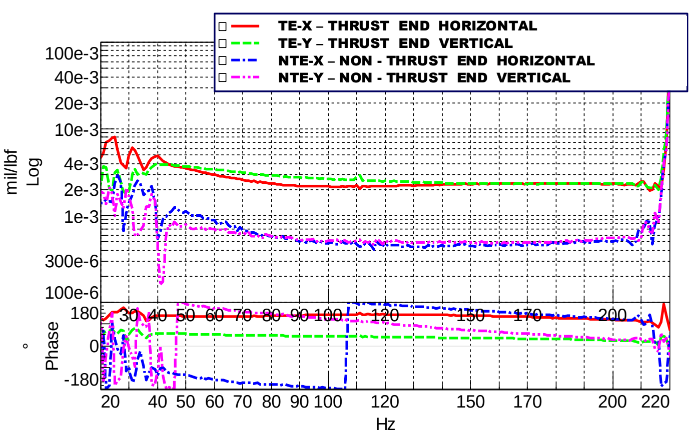


Figure 15. FRFs for the four compressor probes – forward precession and 100% load

RESULTS DISCUSSION

There is a saying that the design condition is a point where it is sure that a machine will not operate. Taking this into account and also noting that since the stability test is a Factory Acceptance Test and not a test done on a test bench rotor, where good agreement between experimental and numerical model results are important, it should be pointed out that the results of the test, showing that the measured stability level is higher than the predicted by the OEM is considered desirable by the Operator, leading to a more robust compressor, in the sense that it will also be stable in off design conditions. This desirability is further enhanced by the survey conducted by Kocur et al. (2007), where a compressor designed and tested as unstable had its stability analyzed by many specialists, most of them with positive log dec results.

Nevertheless, a discussion on the results is an opportunity that should not be looked down since it provides an occasion to try to get a better insight on the contribution of the rotor components to its dynamic behavior. To begin with, there is the tilting pad and squeeze film damper set. The Unbalance Response Test, where, in addition to the rotor itself, these two components are the only ones to provide stiffness and damping, gave a good agreement with the calculated results (and also showed well damped behavior). However, on an unbalance response test there is only synchronous vibration. During the magnetic bearing actuated stability test, the tilting pad bearings are simultaneously subjected to synchronous and sub-synchronous vibration while API 617 requires that the bearing dynamic coefficients should be synchronously reduced. So, good agreement between calculated and measured behavior during an unbalance response test does not mean that the contribution of the bearing and squeeze film damper sets will be well predicted during a stability test, although it does give some confidence that the dynamics of the bearings and SQFD are predicted somewhat accurately. Then all stability contributions should be considered together: the destabilizing effects of the labyrinth seals and aerodynamic cross couplings and the extra damping provided by the bearings, squeeze film dampers and hole pattern seal, discussed next.

In previous sections an historical retrospective of compressor design and modelling at the OEM is given, with examples on why the OEM has stopped using labyrinth seals at the division wall or balance piston of high pressure and high density compressors and, more important to this discussion, how the MPACC method of calculating the aerodynamic cross-coupling forces has evolved, describing four examples where only after this calculation methodology was applied that it was possible to obtain stable behavior of the compressors. In other words, besides the annular gas seal coefficients, the MPACC destabilizing forces needed to be considered and to achieve stability then damper seals, honeycomb or hole pattern, were applied for the first two examples, and squeeze-film dampers and shunt holes were applied for the last two examples. Additionally, the tendency, for the present compressor, of the calculated stability increasing with density, i.e., with pressure and MW, shows that it is the hole pattern seal that produces a dominant effect on the stability of the present compressor.

The testing procedure was specified based on a previous experience of the operator with stability testing, but on low pressure, low density gas compressors without squeeze film dampers and damper seals (Pettinato et al., 2010). So the main results of the present test, of higher density leading to higher stability level, full load flat FRFs and tested log decs higher than calculated ones did produce some (pleasant) surprise onto the operator staff. Nevertheless, the results are consistent and in line with previous experience of the manufacturer and also consistent with witnessed tests by the operator at another manufacturer.

CONCLUSION

Compressors for high density gas are susceptible to fluid-induced instability. Events that have produced high production losses are found in literature and also in internal records of operators. In order to avoid this problem, hole pattern or honeycomb seals with deswirling devices have become standard practice for many OEMs. For the present compressors hole pattern seals are used at the division wall and the deswirling devices are shunt holes and swirl brakes at the division wall and swirl brakes at the impeller eyes. Squeeze-film dampers are also used at the tilt pad bearings.

Numerical modeling methods have been developed in order to assure stable behavior of these machines. Although a detailed procedure for the numerical evaluation of the stability level of centrifugal compressors has been added in the API 617 standard, experimental evaluation of the stability level of centrifugal compressors are becoming more common.

Stability tests were done on two different type of compressor, both designed to operate with high content of CO2 in super critical condition. Due to the similarities on testing and results, only the results of one compressor type are presented. Test data were acquired and analyzed taking into account forward and backward modes of vibration and employing a frequency domain MDOF acquisition method, obtaining results with good repeatability in two different operating conditions. The final result was a curve of experimentally corrected minimum log dec versus average gas density applicable for that type of compressor.

The results also showed that the manufacturers´ log dec calculation procedure is very conservative for these particular compressors in the sense that the experimental results were greater than the calculated ones. The operator regards that conservatism in this case, in the design of the hole pattern seal, is desirable since it gives higher endurance against seal degradation and robustness to off design operation conditions.

Despite the hardware limitations, the support provided by the academic institution was clearly advantageous in the aspect of allowing the preparation of the operators´ professionals across theoretical basis related to the implementation of the stability tests.

Finally, it is considered as an important gain from the tests a clearer understanding of the behavior of this compressor concerning its stability for different load and density conditions.

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