Proceedings of ASME Turbo Expo 2011
GT2011
June 6-10, 2011, Vancouver, British Columbia, Canada

GT2011-45634

STABILITY OF CENTRIFUGAL COMPRESSORS BY APPLICATIONS OF DAMPER SEALS

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ABSTRACT

This paper surveys the applications of damper seals to provide the stable operation with respect to rotordynamics of centrifugal compressors. Damper seals are applied as sealing devices at the division wall of back-to-back compressors and at the balance piston of in-line compressors. They consist of a roughened surface on the stator that is typically created by a pattern of holes. Rotordynamically stable operation is shown by a lack of or a small bounded amount of SSV (subsynchronous vibration) at the first fundamental lateral critical frequency. Experience plots showing the use of damper seals will be presented. Case histories of the use of damper seals will be given. Stability analysis and full load full pressure test results will be reviewed. The test results show no SSV at the first critical frequency with the damper seals.

INTRODUCTION

Damper seals as described in this paper are used at the division wall of back-to-back compressors and at the balance piston of in-line compressors. They consist of a roughened surface on the stator that is typically created by a pattern of holes or honeycomb cells. The rotor surface is smooth. A hole-pattern damper seal is shown in Figure 1, as was given in [1]. Damper seals serve as sealing devices to control leakage.

Damper seals have much more direct damping than toothed labyrinth seals, increase the logarithmic decrement of the first critical frequency and the stability margin, and provide stable operation with respect to rotor dynamics. Even with this large amount of direct damping there is a need for deswirling at the inlet of the seal [2], which in that case was done through a shunt hole system as shown in Figure 2, as was given in [2]. Swirl brakes, as shown in Figure 1 from [1] and also in [3], are also used for deswirling.

Experience plots will be given that are used for the preliminary evaluation of the rotor stability of centrifugal compressors. The plots are for compressors with gas or labyrinth casing end seals. Plots similar to these were shown in [4-8]. Distinctions will be made on these plots whether damper bearings or damper seals are used. A sketch of a damper bearing is shown in Figure 3, as was given in [5-8].

The plots will show that damper seals have extended the experience envelope of compressors to higher densities and gas pressures, and that this can be done without the use of damper bearings. The plots will show that there is extensive experience with the simultaneous application of damper bearings and damper seals in the same compressor.

Then case histories of the applications of damper seals will be given. The case histories will show how damper seals have actually raised the log dec as pressure goes up, both analytically and from magnetic bearing exciter testing during full load full pressure tests. The log dec can be so high that the first natural frequency is critically damped. This does not happen when toothed labyrinths are used. The good agreement between analysis and test will be described.

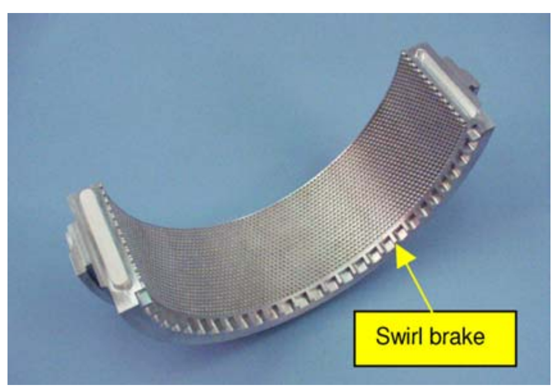

Figure 1. HOLE PATTERN SEAL WITH SWIRL BRAKES ON THE HIGH PRESSURE SIDE

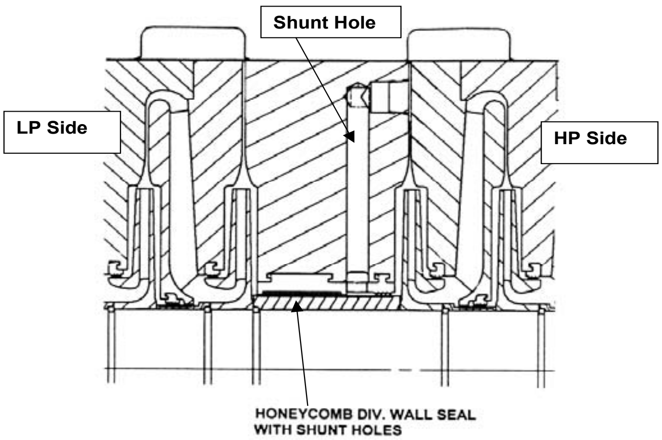

Figure 2. SHUNT HOLE SYSTEM

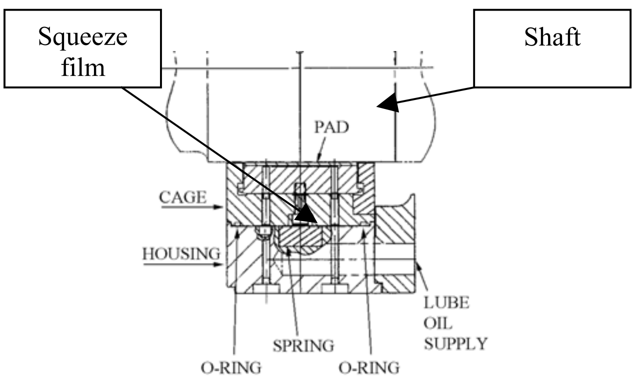

Figure 3. DAMPER BEARING

DAMPER SEALS - HISTORY

Honeycomb seals are a type of damper seal. They were initially used in centrifugal compressors for strength reasons, where differential pressures were in excess of what could be handled by conventional knife-edge seals. They had been used since the late 1960s for that purpose, mainly at the balance piston of high-pressure synthesis gas compressors. There were no deswirling devices used for those seals. Then in the 1990s there were a series of applications for improved rotor stability characteristics for higher pressure and higher density compressors. Some examples are discussed in the papers [2, 9, 10 and 11]. A survey of application of honeycomb seals to high-pressure compressors from 1994 to 1997 is given in [11].

The application of a damper seal at the balance piston during the full-load shop test of a large compressor for propane service is described in [4, 12 and 13]. This particular damper seal was made by machining a pattern of small pockets in aluminum.

It was found that some kind of deswirling device should be used with the honeycomb seal (and certainly also with a hole pattern seal). See [2, 10, and 14], where there were unacceptable levels of SSV without a shunt system and one was eventually used. Some manufacturers use a swirl brake system with no shunt hole system, see [15, 16], where hole pattern seals are used. The author's company uses both [17], except for some in-line synthesis gas compressors it uses just swirl brakes. That paper also contains a short history of compressors with damper seals, both honeycomb and hole pattern types, through 2003.

Although the original honeycomb seals built by the author's company were straight, these new applications of honeycomb seals were built as tapered so that if the stainless steel material of the honeycomb made contact it would be in a small area. For rotordynamic reasons the clearance was smaller at the low-pressure side of the seal. This is called a convergent seal.

Around 1999 the author's company began to make damper seals of aluminum, with a pattern of holes. The production time was much shorter than with the honeycomb and the aluminum material could withstand rubs better than the honeycomb type. A typical hole pattern damper seal with swirl brakes is shown in Figure 1, as was shown in [1]. Also see [15-17] for pictures or sketches of hole pattern seals with swirl brakes.

The author's company has built more than 400 compressors that use damper seals, of which about 60 use the honeycomb seal and the rest use the hole pattern seal. About half of the compressors with damper seals also use damper bearings. These statistics do not include a large number of in-line synthesis gas compressors.

DAMPER SEALS – LOW FREQUENCY INTSABILITY ISSUE

A low frequency instability has been seen when the damper seals are nominally straight. This problem had not been seen when the convergent honeycomb design was used. Analysis and testing has shown if there is a deviation of the straight design to a divergent condition then that may result in a negative direct stiffness. See the experimental vs. analytical results in [18] and the discussion of case histories of this occurrence as given below. If this direct stiffness is high enough in the absolute sense it may overcome the positive direct stiffness from the journal bearings and produce a low frequency vibration with a negative log dec. This frequency is typically described to be the first lateral critical frequency.

The effective cross-coupled stiffness is given by the equation:

$$Qeff = k - \omega C \quad (1)$$

Where k is the cross-coupled stiffness, ω is the first lateral critical frequency, and C is the direct damping.

If there is a large (in the absolute sense) negative K (direct stiffness) introduced then the ω may be lowered and then Qeff increases.

The effective direct damping is given by the equation:

$$C_{eff} = C - (k/\omega) \quad (2)$$

If ω decreases then Ceff decreases.

The stability problems associated with the introduction of a negative direct stiffness has been described in a series of papers.

The paper [14] describes the testing of a back-to-back compressor. This compressor has non-damper bearings and gas seals. A 40 HZ frequency was seen with unacceptably high amplitudes. The first critical frequency was expected to be at 65 HZ. Then the honeycomb seal was made to be convergent, but still the vibration was unacceptable. Shunt holes were added and there was no SSV.

The paper [19] discusses a high vibration in the field at approximately 6 percent of running speed of an in-line compressor. This compressor has damper bearings and contact seals. It was suspected that a fouling on the inlet side of a recently introduced honeycomb seal at the balance piston had produced divergence. The seal was changed back to a toothed labyrinth.

The paper [20] discusses a problem with unacceptable amounts of vibration at a low frequency, approximately 16 percent of running speed, of a back-to-back compressor. This manifested itself on the test stand under load and pressure. The compressor has damper bearings and gas seals. There was a hole pattern seal at the division wall which has shunt holes and swirl brakes and a hole pattern seal at the second section gas balance with swirl brakes. It was likely that there was divergence at the division wall and gas balance seals. The division wall and gas balance seals were redesigned to ensure no divergence and there was no SSV.

The paper [21] discusses high SSV at approximately 7 percent of running speed of an in-line compressor. It has damper bearings and gas seals. There was a hole pattern seal at the balance piston with shunt holes and swirl brakes. Convergence was introduced at the balance piston and there was no SSV.

The papers [14-16 and 20-21] have shown that there are analytical and design procedures that will ensure that this low frequency instability problem is not encountered. A stability analysis should be done to see how much convergence, if any is needed.

Conversely, convergent damper seals can provide a large amount of direct stiffness and this can push the first natural frequency up into the operating speed range, as is shown in [2]. But they also can push the log dec up so that the first natural frequency is critically damped and thus not detrimental to the operation of the compressor.

DAMPER SEAL PROGRAM USED

The analytical basis of the computer code used to model the damper seals and predict their performance is described in the paper [22] by Kleynhans and Childs. The code solves the turbulent bulk flow equations using the Blasius friction factor model but adds an extra degree of freedom to represent the cells [23]. The solution of the turbulent bulk flow equations results in frequency-dependent stiffness and damping coefficients.

Testing done at the Turbomachinery Laboratory at Texas A&M University provided experimental measurements of rotor dynamic stiffness, damping and leakage for both hole pattern and honeycomb seals. Tests were performed at up to 69 bara (1000 psia) supply pressure and at up to speeds of 20,000 rpm. Some of these results were shown in the paper [24] and good correlation was shown between the modeling and the experimental results. The code has also been validated during extensive use in modeling damper seals in high-pressure compressors. Also see [25], 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 test data and the bulk-flow code prediction. This paper also has a picture of a hole pattern seal, but the swirl brakes that are used are not evident.

DAMPER BEARINGS

Damper bearings, as shown in Figure 3, consist of placing a squeeze-film damper and support spring in series with the journal bearing, which in these cases is a tilting pad type. The application of a squeeze-film damper in series with a tilting pad journal bearing is used to aid in providing stable operation, by increasing the logarithmic decrement and stability margin. They provide decreased response to unbalance, by decreasing the amplification factor of the first critical speed and the vibration amplitude at the midspan. The rotor-support system is softened by the squeeze-film damper resulting in less shaft bending at the first natural frequency. Descriptions of damper bearings and modeling and analytical procedures for using damper bearings to aid in the optimization of the rotor dynamic stability and response characteristics have been given in [4-8, 12, 13, 26, and 27]

ROTOR STABILITY EXPERIENCE PLOTS

Four types of rotor stability experience plots will be given that are used in conjunction with an API 617 [28] Level I stability analysis for the preliminary assessment of whether stability enhancements or a more in-depth analysis are needed for centrifugal compressors. These plots were given in [8], but without the discussion of the eight case histories to be given later in this paper. All of the compressors on these plots have dry gas or toothed labyrinth casing end seals (mostly dry gas). The compressors shown in Figures 4A, 5A, 6A, and 7A all use damper bearings and the compressors shown in Figures 4B, 5B, 6B, and 7B all use non-damper bearings. A distinction is made on each plot whether or not the compressor uses damper seals. Solid symbols on these plots mean toothed labyrinths are used at the division wall or balance piston and hollow symbols mean damper seals are used there. Circular symbols mean damper bearings are used and square symbols mean non-damper bearings are used.

All of the compressors given in the case histories to be discussed later in this paper use damper seals. The case histories are indicated with a diamond in the middle of the hollow symbol and are numbered with the arrows pointing to their location. Most of those case histories are on the edge of the experience envelope. Solid triangles are shown for the case histories given in the preceding paper [8], where all those examples did not use damper seals.

Compressors in Region B of these plots are subject to more strict analytical stability criteria than those in Region A. Each plot has a representative sample of the compressors from this author's company. The plots include compressors with ship dates from as far back as 1973.

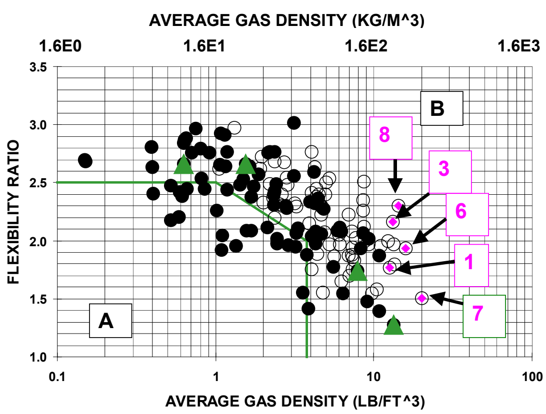

Figure 4A. FLEXIBILITY RATIO *VS.* AVERAGE GAS DENSITY - WITH DAMPER BEARINGS

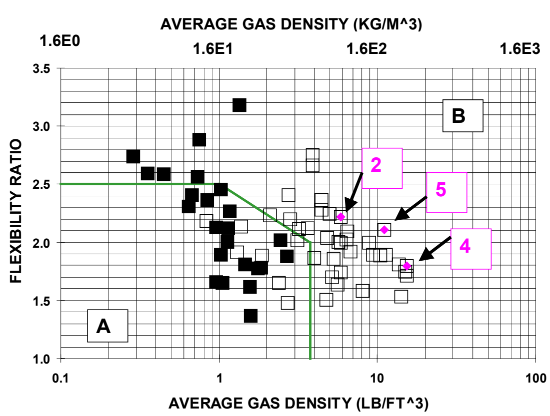

Figure 4B. FLEXIBILITY RATIO *VS.* AVERAGE GAS DENSITY - WITH NON-DAMPER BEARINGS

Most of the high pressure and high-density experience with dry gas or labyrinth casing end seals to this time is included. The usage of these types of plots has been discussed in [1, 4-8]. The compressors with oil-film seals have been given in separate plots in earlier papers [2, 11, 17, 29 and 30], with emphasis on the use of tilting pad oil-film seals.

Figures 4A (damper bearings) and 4B (non-damper bearings) are plots of flexibility ratio, the ratio of the maximum continuous speed divided by the rigid bearing first critical speed, vs. the average gas density, as is used in Specification 2.6.5 of Chapter 1 of API 617 [28]. The average gas density is the average of the inlet density and the discharge density of the gas. These densities are typically calculated for one point, the normal operating (certified) point as is required by API [28]. They are not from tests or experiments. In Region A the API analytical stability screening criteria is not as severe as in Region B. Per API and [4 and 8] there should be a cut-off average gas density above which more stringent stability criteria should apply no matter what the flexibility ratio.

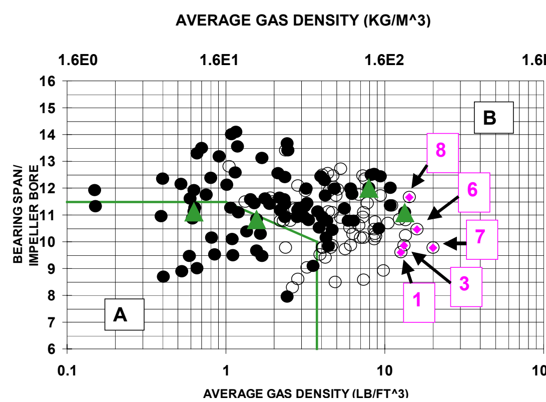

Figure 5A. BEARING SPAN/IMPELLER BORE *VS.* AVERAGE GAS DENSITY - WITH DAMPER BEARINGS

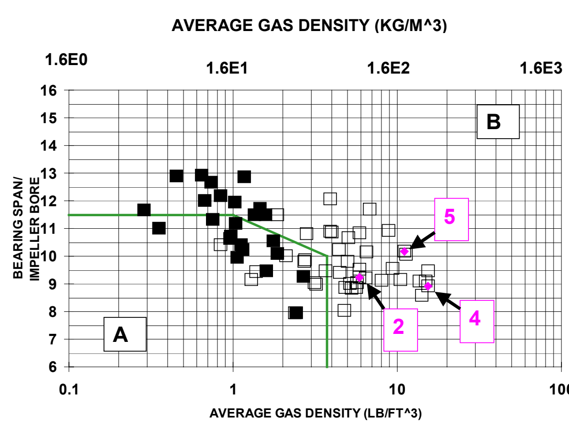

Figure 5B. BEARING SPAN/IMPELLER BORE *VS.* AVERAGE GAS DENSITY - WITH NON-DAMPER BEARINGS

Plots like Figures 4A/B were given in [31] by an end-user, with different dividing lines and without a cut-off density. See [14-16] with plots of flexibility ratio vs. average gas density with the experience of other companies.

Figures 5A (damper bearings) and 5B (non-damper bearings) are plots of bearing span/impeller bore vs. the average gas density. The bearing span is the distance from the journal bearing centerline at one end of the compressor to the journal bearing centerline at the other end of the compressor. The diameter of the shaft at the position at which the impellers are located on the rotor is used as the impeller bore. If the impellers have different bores then a weighted averaging system is used to determine an effective bore. In [4 and 8] it was shown that the plot with bearing span/impeller bore more accurately predicts the need for stability enhancements.

The ratio bearing span/impeller bore is an estimate of the slenderness of the shaft and is calculated with a minimum knowledge of the shaft geometry.

As with the flexibility ratio, it is non-dimensional. The effect of the slenderness of the shaft on the stability should be considered, no matter if the flexibility ratio is high or low.

Figures 6A (damper bearings) and 6B (non-damper bearings) are plots of discharge pressure times differential pressure across the case vs. flexibility ratio. The pressures are typically from the analytical calculated normal operating (certified) point. They are not from tests or experiments. From [4 and 8] there should be a cut-off discharge pressure times differential pressure across the case above which more stringent stability criteria should apply no matter what the flexibility ratio. Plots like Figures 6A/B were first given in [32] and then in [31], with different dividing lines and without a cut-off discharge pressure times differential pressure across the case.

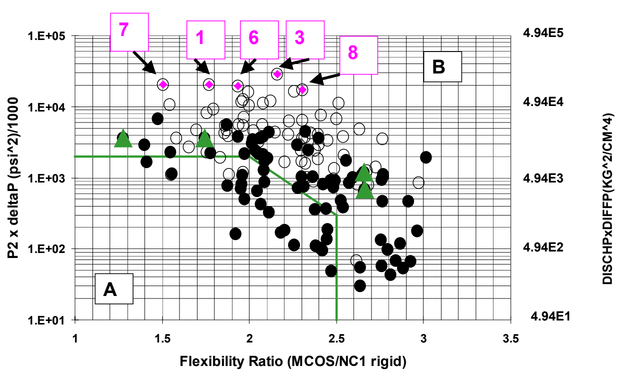

Figure 6A. DISCHARGE PRESSURE X CASE DIFFERENTIAL PRESSURE *VS.* FLEXIBILITY RATIO WITH DAMPER BEARINGS

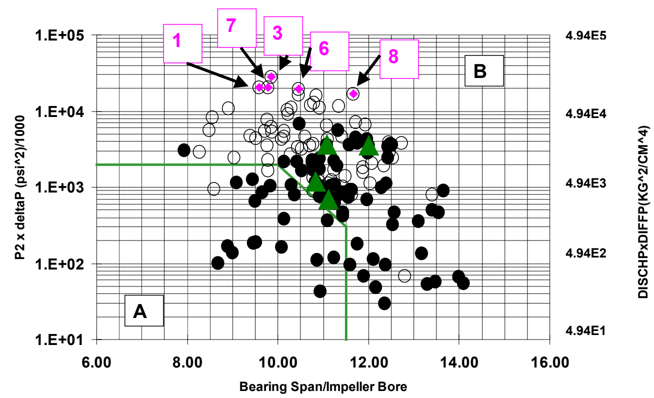

Figure 7A. DISCHARGE PRESSURE X CASE DIFFERENTIAL PRESSURE *VS.* BEARING SPAN/IMPELLER BORE WITH DAMPER BEARINGS

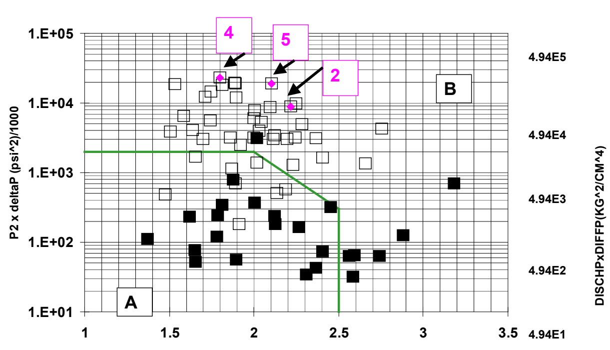

Figure 6B. DISCHARGE PRESSURE X CASE DIFFERENTIAL PRESSURE *VS.* FLEXIBILITY RATIO WITH NON-DAMPER BEARINGS

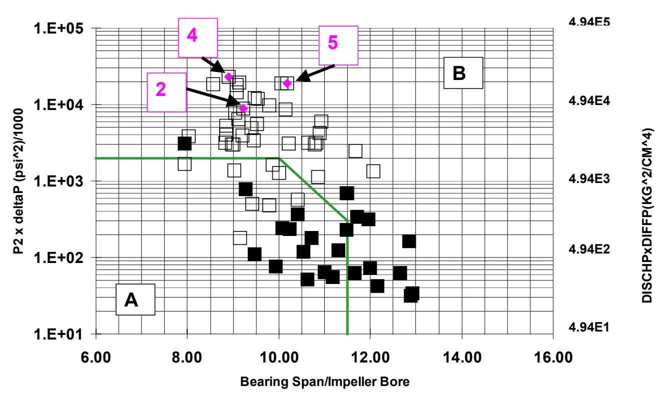

Figure 7B. DISCHARGE PRESSURE X CASE DIFFERENTIAL PRESSURE *VS.* BEARING SPAN/IMPELLER BORE WITH NON-DAMPER BEARINGS

Figures 7A (damper bearings) and 7B (non-damper bearings) are plots of discharge pressure times differential pressure across the case vs. bearing span/impeller bore. This type of plot was first given in [8]. As was discussed in [8] the plot with bearing span/impeller bore more accurately predicts the need for stability enhancements.

Conclusions from the study of these plots:

- There is extensive experience in the high density and high differential pressure regions with damper seals.
- Damper seals have extended the experience envelope in the high density and high differential pressure regions.
- With non-damper bearings for high density or high differential pressure applications all of the experience is with the use of damper seals.
- There is extensive experience with use of both damper bearings and damper seals.
- There is considerable experience of the concurrent use of damper seals and damper bearings.
- For high flexibility ratios or high ratios of bearing span/impeller bore the trend is to use damper bearings.
- The papers [4 and 8] showed that bearing span/impeller bore does better than flexibility ratio in assessing the need for stability enhancements.

ROTOR DYNAMIC PROGRAMS USED

The rotor dynamic programs that were used for the stability analysis analytical results given in this paper, besides the damper seal program described above, are: The author's company rotordynamic software suite [33], the stability program of Lund [27], the toothed labyrinth program by Kirk [34], and the tilt pad bearing program by Nicholas [35]. See the paper [7] for a description of the analytical basis of the programs. Also see [26], for a discussion of the usage of the programs, except for the programs for damper seals and toothed labyrinths.

STABILITY CASE HISTORIES WITH HOLE PATTERN SEALS AND DRY GAS CASING END SEALS

Stability case histories will be given of high-pressure compressors with hole pattern seals at the division wall or balance piston and dry gas casing end seals. All of them had full-load full pressure shop tests. All of them are natural gas injection compressors except for case history 8, a CO_2 compressor with very high discharge pressure, 303 bara (4400 psia), for a CO_2 compressor. All are on the outside edge of the experience envelope shown in Figures 4-7, except for case history 2, on the plots with bearing span/impeller bore.

All of them have teeth on stator. All have swirl brakes at the impeller eyes. All of them have shunt holes and swirl brakes at the division wall seal if back-to-back or at the balance piston if in-line. All of them have tilt-pad bearings. The compressors in Case Histories 1, 2, 5, 6 and 8 were tested with the use of a magnetic bearing exciter, as is shown in Figure 8.

The magnetic bearing exciter is put on the free end of the compressor to generate asynchronous forcing functions into the rotor at various aerodynamic conditions. See the papers [15, 16, and 36-39] for descriptions of the application of magnetic bearing exciters. The effect of pressure on the log dec of the first natural frequency of compressors with hole pattern seals at the division wall or balance piston is evaluated by the use of the magnetic bearing exciter. The magnetic bearing exciter measures but does not control the log dec. The log dec increases with pressure with the hole pattern seals, contrary to what is found with toothed labyrinth seals. Good agreement is found between the testing and the analysis.

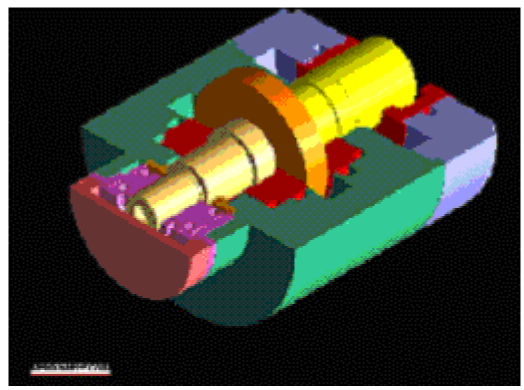

Figure 8. SOLID MODEL OF THE MAGNETIC BEARING EXCITER

Stability analyses, as in Level II of API 617 [28], including the toothed labyrinths and damper seals, are done as a matter of course by this OEM, whether or not [28] requires them. They were done for all of the Case Histories.

Case History 1 - 410 bara (5950 psia) Discharge Natural Gas Injection Compressor with Damper Bearings

This case history was discussed in [36]. The compressor has seven impellers and is a back-to-back design. It is driven by a gas turbine through a gear.

Figure 9 shows the measured and predicted log dec vs. discharge pressure from the magnetic bearing exciter tests. The log dec steadily increases with pressure and is very high at the final discharge pressure. The good values of log dec at low pressures is due to the damper bearings. There is good agreement between the measured and the analytical calculations. The vibration spectrum plot in Figure 10 shows no SSV during the class I test at the design discharge.

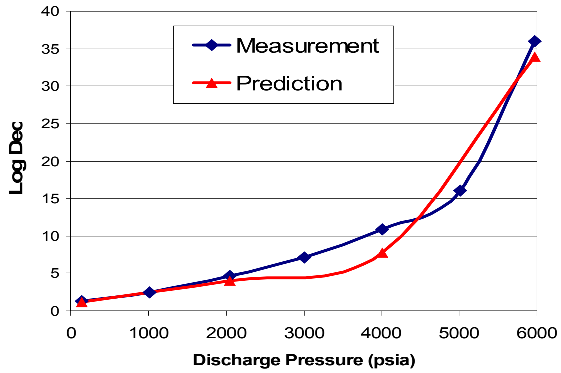

Figure 9. CASE HISTORY 1 –
LOG DEC *VS.* DISCHARGE PRESSURE

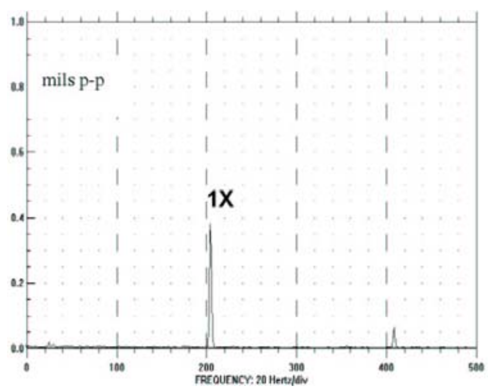

Figure 10. CASE HISTORY 1 –
VIBRATION SPECTRUM
AT FULL DISCHARGE PRESSURE

Case History 2 - 215 bara (3115 psia) Discharge Natural Gas Injection Compressor with Non-Damper Bearings

This case history was discussed in [37]. The compressor has eight impellers and is a back-to-back design. It is driven by a variable speed induction motor through a gear.

Figure 11 shows the measured and predicted log dec vs. discharge pressure from the magnetic bearing exciter tests. The log dec steadily increases with pressure and is high at the final discharge pressure. The lower log decs at low pressures is due to the lack of squeeze-film dampers. There is good agreement between the measured and the analytical calculations.

A smooth seal was tested with the same clearance as the hole pattern seal. As shown, the log decs were about the same but the leakage for the smooth seal was approximately 50 percent higher

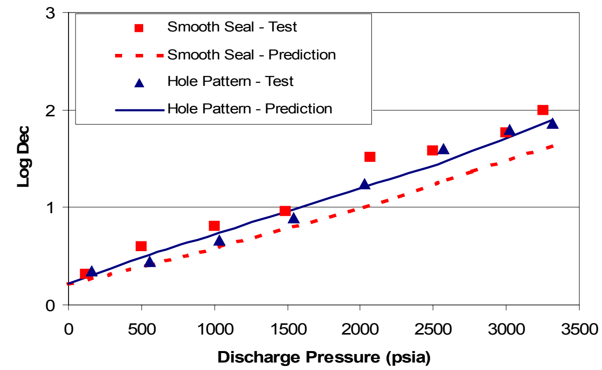

Figure 11. CASE HISTORY 2 –
LOG DEC *VS.* DISCHARGE PRESSURE

Case History 3 - 457 bara (6621 psia) Discharge Natural Gas Injection Compressor with Damper Bearings

This case history was discussed in [20] and earlier in this paper. The compressor has eight impellers and is a back-to-back design. It is driven by a gas turbine through a gear. There are four of these trains.

The results of a Level II stability analysis as given in 2.6.6 of Chapter 1 of API 617 7th edition [28] is shown in Figure 12. All of the toothed labyrinths and hole pattern seals are included in the analysis. The analysis was made for minimum and maximum clearance bearings and squeeze-film dampers and minimum and maximum hole pattern clearance at the division wall.

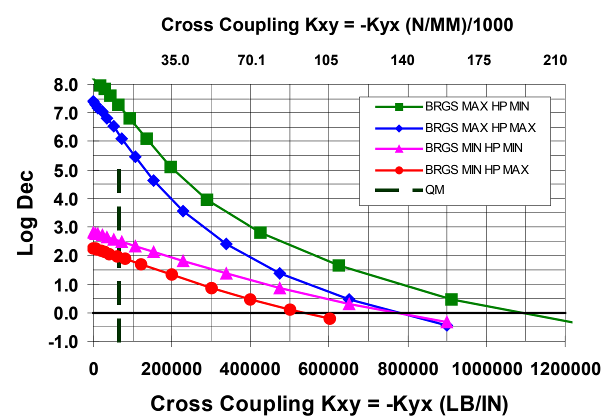

Figure 12. CASE HISTORY 3 – STABILITY ANALYSIS

Figure 12 is a plot of the log dec vs. additional values (in addition to the already included values for the toothed and hole pattern labyrinths) of cross coupled stiffness at the midspan. Shown on the plot is QM that is the modal sum of the anticipated cross coupling from the impellers as is given in [12]. The calculated log dec at the value of QM ranges from 2.0 to 7.3 and easily meets the goal of API 617 [28], that the log dec be greater than 0.1

Figure 13, the waterfall plot of the vibration spectrums, shows no SSV during the API 617 class I test at full load full pressure on hydrocarbon gas. There was no magnetic bearing exciter test for this compressor.

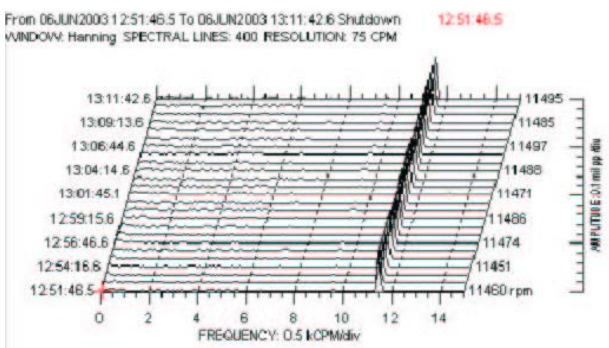

Figure 13. CASE HISTORY 3 – VIBRATION SPECTRUM

Case History 4 - 409 bara (5932 psia) Discharge Natural Gas Injection Compressor with Non-Damper Bearings

The compressor has six impellers and is a back-to-back design. It is the high-pressure compressor in a train driven by a variable speed synchronous motor through a gear first to the low-pressure compressor and then to the high-pressure compressor. There are two of these trains.

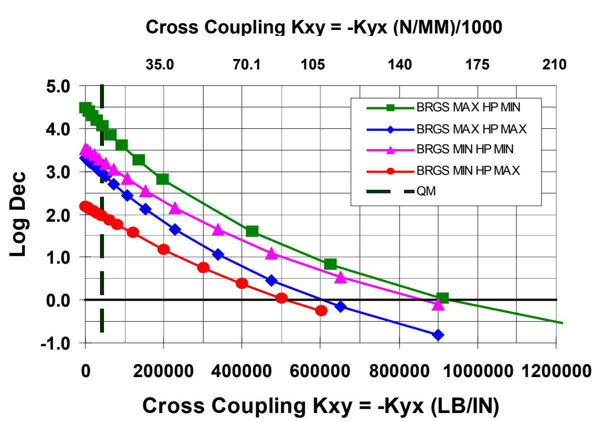

Figure 14. CASE HISTORY 4 – STABILITY ANALYSIS

The results of a Level II stability analysis as given in API 617 7th edition [28] are shown in Figure 14 for Case History 4. The same assumptions are made as for Case History 3. The calculated log dec at the value of QM ranges from 2.0 to 4.1 and easily meets the goal of API 617 [28], that the log dec be greater than 0.1.

The vibration spectrum plot in Figure 15 shows no SSV during the API 617 class I test at full load full pressure. There was no magnetic bearing exciter test for this compressor.

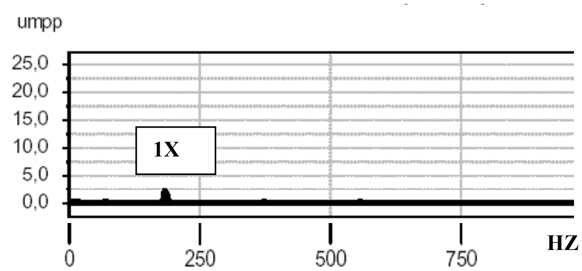

Figure 15. CASE HISTORY 4 – VIBRATION SPECTRUM AT FULL DISCHARGE PRESSURE

Case History 5 - 352 bara (5105 psia) Discharge Natural Gas Injection Compressor with Non-Damper Bearings

The compressor has six impellers and is a back-to-back design. It is driven by a variable speed induction motor through a gear.

Figure 16 shows the measured log dec vs. discharge pressure from the magnetic bearing exciter tests. The log dec steadily increases with pressure and is very high at the final discharge pressure.

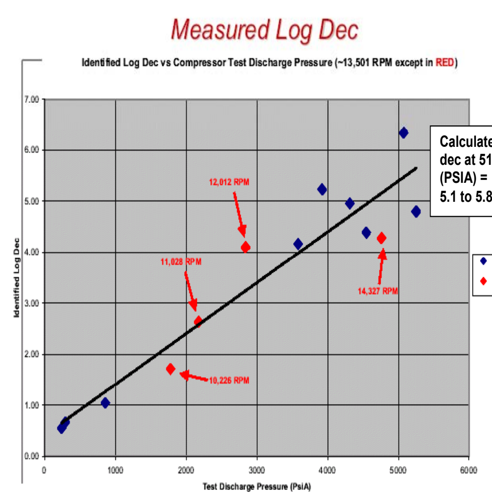

Figure 16. CASE HISTORY 5 – LOG DEC *VS.* DISCHARGE PRESSURE

At full pressure the log dec was calculated for the range of bearing, squeeze-film damper, and damper seal clearances and these values were added to Figure 16 and show an excellent agreement between the measured and the calculated values.

Figure 17 shows the magnetic bearing exciter results vs. pressure and how at high pressure the first natural frequency is completely damped out for Case History 5.

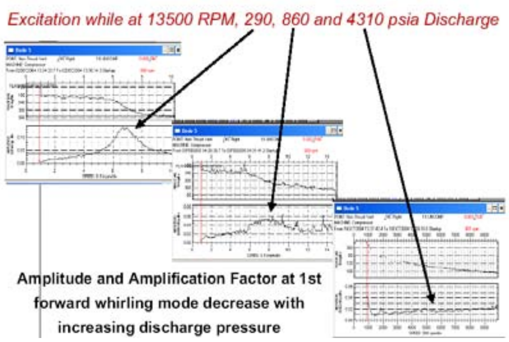

Figure 17. CASE HISTORY 5 – MAG BEARING RESULTS SHOWING NC1 *VS.* PRESSURE

Case History 6 - 380 bara (5515 psia) Discharge Natural Gas Injection Compressor with Damper Bearings

This case history was discussed in [38]. The compressor has nine impellers and is a back-to-back design. It is driven by a gas turbine through a gear.

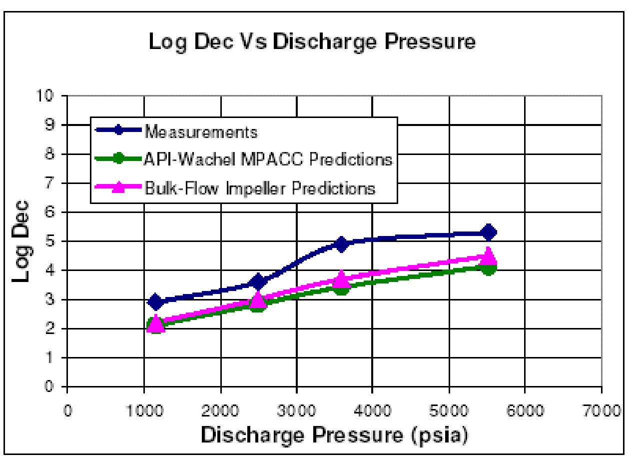

Figure 18. CASE HISTORY 6 – LOG DEC *VS.* DISCHARGE PRESSURE

Figure 18 shows the measured and predicted log dec vs. discharge pressure from the magnetic bearing exciter tests for Case History 6. The log dec steadily increases with pressure and is high at the final discharge pressure. There are two predicted curves, one with the API method [28] and the modal sum of the anticipated cross-couplings at the impellers and the other with bulk-flow predictions for the impellers. There is little difference in the predicted results and the predicted log decs are more conservative (lower) than the measured

Case History 7 - 485 bara (7030 psia) Discharge Natural Gas Injection Compressor with Damper Bearings

The compressor has five impellers and is an in-line design. It is driven by a gas turbine through a gear, first to a low-pressure compressor, then to a medium pressure compressor, and then finally to this high-pressure compressor.

The results of a Level II stability analysis as given in API 617 7th edition [28] are shown in Figure 19. The same assumptions are made as for Case Histories 3 and 4. The calculated log dec at the value of QM ranges from 1.2 to 1.7 and easily meets the goal of API 617 [28], that the log dec be greater than 0.1.

The vibration spectrum plot in Figure 20 shows no SSV during the API 617 class I test at full load full pressure. There was no magnetic bearing exciter test for this compressor.

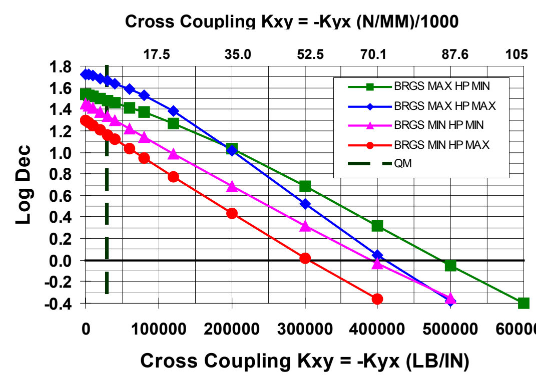

Figure 19. CASE HISTORY 7 – STABILITY ANALYSIS

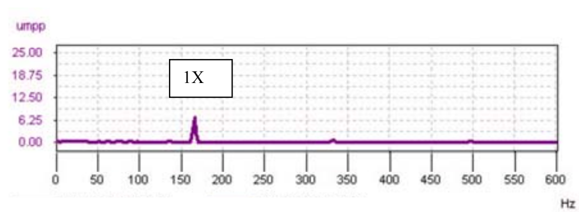

Figure 20. CASE HISTORY 7 – VIBRATION SPECTRUM AT FULL DISCHARGE PRESSURE

Case History 8 - 303 bara (4400 psia) Discharge CO_2 Compressor with Damper Bearings

This case history was discussed in [39]. The compressor has eight impellers and is a back-to-back design. It is the first body of a string of two centrifugal compressors driven by a constant speed induction motor through a gear.

Figure 21 shows the measured log dec vs. discharge pressure from the magnetic bearing exciter tests. The log dec steadily increases with pressure and is very high at the final discharge pressure. At full pressure the log dec was calculated for the range of bearing, squeeze-film damper, and damper seal clearances and there is excellent agreement between the measured and the calculated values. It is relatively high at low pressure because of the damper bearings. The vibration spectrum plot in Figure 22 shows no SSV during the API 617 class I test at full load full pressure.

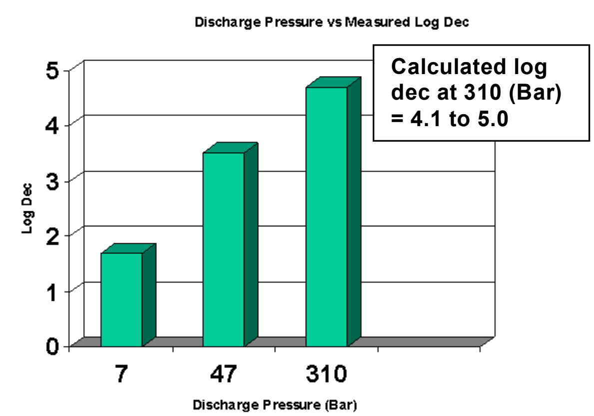

Figure 21. CASE HISTORY 8 – MEASURED LOG DEC *VS.* DISCHARGE PRESSURE

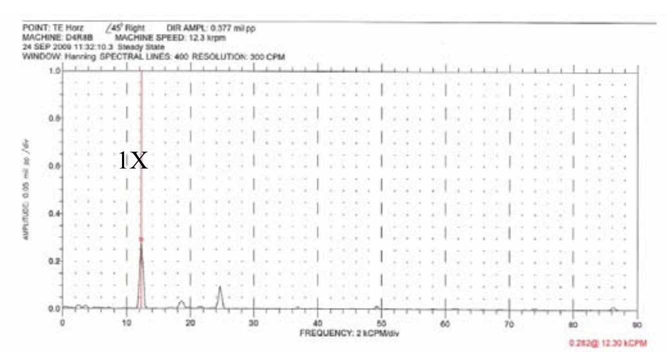

Figure 22. CASE HISTORY 8 – VIBRATION SPECTRUM AT FULL DISCHARGE PRESSURE

CONCLUSIONS

Damper seals have substantially extended the experience envelope of centrifugal compressors in high density and high-pressure applications. In such critical applications they can critically damp the first natural frequency, as is shown by the high value of the measured log dec of the first fundamental lateral critical frequency and the lack of SSV during full load full pressure tests.

The magnetic bearing exciter tests at full load and full pressure show the good correlation between the predicted and the tested values of the log dec of the first lateral critical frequency.

The additional experience plots provide an excellent supplement to the Level I screening criteria of API 617 [28]. The experience plots show that there is extensive experience with both damper seals and damper bearings and that both can be used in the same compressor. The experience plots give strong indications of when damper seals should be used.

ACKNOWLEDGMENTS

The author would like to thank the management of Dresser-Rand for its support and permission to publish this paper.

NOMENCLATURE

SSV = subsynchronous vibration
K = direct stiffness
k = cross-coupled stiffness
C = direct damping
Qeff = effective cross-coupled stiffness
Ceff = effective direct damping
ω = first lateral critical frequency
CFD = computational fluid dynamics
MCOS = maximum continuous speed

DISCLAIMER

The information contained in this paper includes factual data, technical interpretations and opinions which, while believed to be accurate, are offered solely for information purposes. No representation, guarantee or warranty of any kind is made concerning such data, interpretations and opinions, including the accuracy thereof.

12. REFERENCES

[1] De Santiago, O. & Memmott, E. A., 2006, "Stability of Centrifugal Compressors for Oil and Gas Service," 9th Latin America Congress and Exposition for Turbomachinery, Veracruz, Mexico, June 21-23

[2] Memmott, E. A., 1994, "Stability of a High Pressure Centrifugal Compressor Through Application of Shunt Holes

and a Honeycomb Labyrinth," CMVA, 13th Machinery Dynamics Seminar, Toronto, pp. 211-233, Sept. 12-13

[3] Moore, J. J. and Hill, D. L., 2000, "Design of Swirl Brakes for High Pressure Centrifugal Compressors Using CFD Techniques," 8th International Symposium on Transport Phenomena and Rotating Machinery, ISROMAC-8, Vol. II, Honolulu, Hawaii, pp. 1124-1132, March 26-30

[4] Memmott, E. A., 2002, "Lateral Rotordynamic Stability Criteria for Centrifugal Compressors," CMVA, Proceedings of the 20th Machinery Dynamics Seminar, Quebec City, pp. 6.23-6.32, October 21-23.

[5] Memmott, E. A. & Ramesh, K., 2008, " Application of Squeeze-film Dampers with a Centrifugal Compressor," IMechE, 9th International Conference on Vibrations in Rotating Machinery, Exeter, pp. 787-798, September 10-12

[6] Memmott, E. A. & Ramesh, K., 2009, "Application of Squeeze-film Dampers with a Centrifugal Compressor," CMVA, Proceedings of the 27th Machinery Dynamics Seminar, Vancouver, BC, October 28-30 (Updated Version of Paper 1)

[7] Memmott, E. A., 2010, "Application of Squeeze-film Dampers to a Large Centrifugal Compressor," ASME, Proceedings of ASME Turbo Expo 2010, Power for Land, Sea and Air, Glasgow, Scotland, UK, June 14-18, GT2010-23553

[8] Memmott, E. A., 2010, "Application of Squeeze-film Dampers to Centrifugal Compressors," CMVA, Proceedings of the 28th Machinery Dynamics Seminar, Universite de Laval, Ville Quebec, Canada, October 27-29

[9] Kuzdzal, M. J., Hustak, J. F., and Sorokes, J. M., 1994, "Identification and Resolution of Aerodynamically Induced Subsynchronous Vibration During Hydrocarbon Testing of a 34,000 HP Centrifugal Compressor," IFToMM, Proceedings of the 4th International Conference on Rotordynamics, Chicago, September 7-9

[10] Gelin, A., Pugnet, J-M., Bolusset, D, and Friez, P., 1996, "Experience in Full Load Testing Natural Gas Centrifugal Compressors for Rotordynamics Improvements," IGT & A C&E, Birmingham, UK, 96-GT-378

[11] Memmott, E. A., 1998-9, "Stability Analysis and Testing of a Train of Centrifugal Compressors for High Pressure Gas Injection," ASME Turbo Expo '98, Stockholm, June 2-5, Journal of Engineering for Gas Turbines and Power, July 1999, Vol. 121, pp. 509-514

[12] Memmott, E. A., 2000, "Empirical Estimation of a Load Related Cross-Coupled Stiffness and The Lateral Stability Of Centrifugal Compressors," CMVA, *Proceedings of the 18th Machinery Dynamics Seminar*, Halifax, pp. 9-20, April 26-28.

[13] Memmott, E. A., 2000, "The Lateral Stability Analysis of a Large Centrifugal Compressor in Propane Service at an LNG Plant," IMechE, *Proceedings of the 7th International Conference on Vibrations in Rotating Machinery*, Nottingham, England, pp. 187-198, September 12-14.

[14] Camatti, M., Vannini, G., Fulton, J. W., and Hopenwasser, F., 2003, "Instability of a High Pressure Compressor Equipped with Honeycomb Seals," Proceedings of the Thirty Second Turbomachinery Symposium, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas, pp. 39-48, September 8-11

[15] Bidaut, Y., Baumann, U., and Al-Harthy, S. M. H., 2009, "Rotordynamic Stability Of A 9500 Psi Reinjection Centrifugal Compressor Equipped With A Hole Pattern Seal- Measurement Versus Prediction Taking Into Account The Operational Boundary Conditions," Proc. of the Thirty-Eighth Turbomachinery Symposium, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas

[16] Bidaut, Y. and Baumann, U., 2010, "Improving the Design of a Compressor Casing with the Help of Finite Element Analysis to Ensure the Stability of High Pressure Centrifugal Compressor Equipped with a Hole Pattern Seal," ASME, Proceedings of ASME Turbo Expo 2010, Power for Land, Sea and Air, Glasgow, Scotland, UK, June 14-18, GT2010-222185

[17] Memmott, E. A., 2004, "Rotordynamic Stability Of The High Pressure Gas Injection Compressors At The North Slope," CMVA, Proceedings of the 22nd Machinery Dynamics Seminar, Ottawa, October 27-29

[18] Childs, D. W. and Wade, J., 2003, "Rotordynamic-Coefficient and Leakage Characteristics for Hole-Pattern-Stator Annual Gas Seals-Measurements Versus Predictions," ASME Paper 2003-TRIB-211, Proc. of 2003 STLE/ASME Joint International Tribology Conference, Ponte Vedra Beach, Florida, October 26-29

[19] Kocur, J. A., and Hayles, G. C., 2004, "Low Frequency Instability in a Process Compressor," Proceedings of the Thirty Third Turbomachinery Symposium, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas, pp. 25-32, September 20-23

[20] Tecza, J., Soulas, T., Eldridge, T., 2004, "Statistical Analysis Of Damper Seal Clearance Divergence And Impact Upon Rotordynamic Stability," ASME, Proceedings of

IMECE 2004, November 13-20, 2004, Anaheim, California USA, IMECE2004-60686

[21] Eldridge, T. M. and Soulas, T. A., 2005, "Mechanism and Impact of Damper Seal Divergence on the Rotordynamics of Centrifugal Compressors," ASME, Proceedings of GT2005, ASME Turbo Expo 2005: Power for Land, Sea and Air, Reno-Tahoe, Nevada, USA, June 6-9, GT2005-69104

[22] Kleynhans, G. F., and Childs, D. W., 1996, "The Acoustic Influence of Cell Depth on the Rotordynamic Characteristics of Smooth-Rotor/Honeycomb-Stator Annular Gas Seals," ASME International Gas Turbine and Aeroengine Congress and Exposition, June 10-13, Birmingham, UK

[23] Soulas, T., and San Andrés, L., 2002, "A Bulk Flow Model for Off-Centered Honeycomb Gas Seals," ASME Paper GT-2002-30286, Proc. of ASME Turbo Expo 2002, Amsterdam, The Netherlands

[24] Holt, C. G., and Childs, D. W., 2002, "Theory Versus Experiment for the Rotordynamic Impedances of Two Hole-Pattern-Stator Gas Annular Seals," ASME Journal of Tribology, 124, pp. 137-143.

[25] Chochua, G. and Soulas, T. A., 2006, "Numerical Modeling of Rotordynamic Coefficients for Deliberately Roughened Stator Gas Annular Seals," Proceedings of IJTC2006, 2006 International Joint Tribology Conference, San Antonio, Texas, October 23-26, IJTC 2006-12049

[26] Memmott, E. A., 2003, "Usage of the Lund Rotordynamic Programs in the Analysis of Centrifugal Compressors," *Jorgen Lund Special Issue of the ASME Journal of Vibration and Acoustics*, Vol. 125, October.

[27] Lund, J. W., 1974, "Stability and Damped Critical Speeds of a Flexible Rotor in Fluid-Film Bearings," Trans. ASME, *Journal of Engineering for Industry*, pp. 509-517, May.

[28] API Standard 617, 2002, "Axial and Centrifugal Compressors for Petroleum, Chemical, and Gas Industry Services," 7th Edition, July.

[29] Memmott, E. A., 1996, "Stability of an Offshore Natural Gas Centrifugal Compressor," CMVA, 15th Machinery Dynamics Seminar, Banff, Oct. 7-9.

[30] Memmott, E. A., 1992, "Stability of Centrifugal Compressors by Applications of Tilt Pad Seals, Damper Bearings, and Shunt Holes," IMechE, *5th International Conference on Vibrations in Rotating Machinery*, Bath, pp. 99-106, Sept. 7-10.

[31] Fulton, J. W., 1984, "Full Load Testing in the Platform Module Prior to Tow Out: A Case History of Subsynchronous Instability," Rotordynamic Instability Problems in High Performance Turbomachinery, NASA Conference Publication 2338, Texas A&M University, pp. 1-16, May 28-30.

[32] Kirk, R. G. and Donald, G. H., 1983, "Design Criteria for Improved Stability of Centrifugal Compressors," AMD Vol. 55, ASME, pp. 59-71, June

[33] Ramesh K., 2002, "A State-of-the-art Rotor Dynamic Analysis Program," The 9th International Symposium on Transport Phenomena and Dynamics of Rotating Machinery, Honolulu, Hawaii, February 10-14

[34] Kirk, R. G., 1990, "Users Manual for the Program DYNPC28 -- A Program for the Analysis of Labyrinth Seals," Negavib Research & Consulting Group, Virginia Tech, Blacksburg, VA, Jan.

[35] Nicholas, J. C., Gunter, E. J., and Allaire, P. E., 1979, "Stiffness and Damping Coefficients for the Five-Pad Tilting-Pad Bearing," *ASLE Transactions*, Vol. 22, 2, pp. 113-124, April.

[36] Moore, J. J., Walker, S. T., and Kuzdzal, M. J., 2002, "Rotordynamic Stability Measurement During Full-Load Full-Pressure Testing of a 6000 PSI Re-injection Centrifugal Compressor," *Proceedings of the Thirty First Turbomachinery Symposium*, Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas, September.

[37] Moore, J. J. and Soulas, T. A., 2003, "Damper Seal Comparison in a High Pressure Re-Injection Centrifugal Compressor During Full-Load, Full-Pressure Testing Using Direct Rotordynamic Stability Measurement," ASME, *Proceedings of the 19th Biennial Conference on Mechanical Vibration and Noise*, International 2003 DETC, Chicago, Illinois, DETC2003/VIB-48458, September 2-6.

[38] Gupta, M. K., Soulas, T. A., and Childs, D. W., 2007, "New Steps To Improve Rotordynamic Predictions Of Centrifugal Compressors," ASME, Proceedings of GT2007, ASME Turbo Expo 2007: Power for Land, Sea and Air, Montreal, Canada, May 14-17, GT2007-27501.

[39] Soulas, T., Desai, A. R., Colby, G., and Griffin, T., 2011, "CO2 Compression for Capture and Injection in Today's Environmental World," *Proceedings of the First Middle East Turbomachinery Symposium*, Doha, Qatar, February 13-16.