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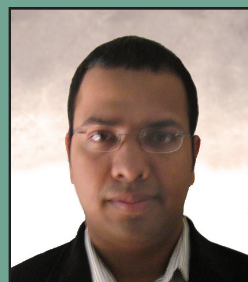

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New Steps to Improve Rotordynamic Stability Predictions of Centrifugal Compressors

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INTRODUCTION

Re-injection compressors, used to inject natural gas into oil wells at pressures ranging from 100 to 700 bar, have traditionally created rotordynamic challenges due to high pressure and density. As a rule, the instabilities are load versus speed dependent. Specifically, in the past, above a limiting load condition (head rise), the rotors could display sub-synchronous unstable motion at the rotor's first natural frequency. Rotordynamicists over the years have successfully followed the dual approach of excitation source elimination and vibrations absorption.

The "Kaybob" compressor instability was eliminated by stiffening the rotor (Smith [2], and Fowlie and Miles [3]). "Ekofisk" injection compressor instabilities were eliminated by several changes including shortening the bearing span by 140 mm (5.5 in.), modifying the bearings, increasing the "backwall" clearances on both sides of the impellers, using a squeeze-film damper at the coupling-end bearing, and building a new rotor (Geary et al. [4], and Cochrane [5]). Several other improvements such as reducing the bearing span, removing vane diffusers, reducing labyrinth diameters, providing shunt hole injection, building swirl brakes, and using hole pattern damper seals have greatly improved rotordynamic stability. While most of those enhancements have been towards bearings and seals, very little effort has been made in studying impellers and their influence on rotordynamic stability. Although several validated commercial codes are available for predicting rotordynamic characteristics of seals and bearings, none exists for impellers. Over the years, the prediction tools for seals and bearings have gained several validations and have been greatly improved. Unfortunately, this is not true for impellers from a rotordynamic standpoint. Predicting accurate impeller forces continue to be a challenge; thus, a complete rotordynamic analysis is difficult to perform.

The only modeling tool widely used in predicting the impeller aero-excitation comes from Wachel's [6] purely destabilizing empirical model (equation 1) for the aerodynamic forces (inch-lb-sec units) and is given in equation (2)

$$-\begin{Bmatrix} f_X \\ f_Y \end{Bmatrix} = \begin{bmatrix} 0 & Q \\ -Q & 0 \end{bmatrix} \begin{Bmatrix} X \\ Y \end{Bmatrix}; \qquad (1)$$

$$Q = [\frac{6300 \times HP \times MW}{D \times h_t \times RPM}] \times \frac{\rho_D}{\rho_S} \qquad (2)$$

where HP is the horsepower, *MW* is the molecu-

ABSTRACT

Improved rotordynamic stability is desired by end users, and centrifugal compressor manufacturers are expected to meet, if not exceed, this expectation. Compressor manufacturers are required to design and build machines that are rotordynamically stable on the test stand and in the field. Confidence has been established in predicting the excitation forces from seals and bearings, but impeller aerodynamic excitation forces continue to be a challenge. While much attention is paid to impellers from an aerodynamic performance point of view, more efforts are needed from a rotordynamic standpoint. A high-pressure, re-injection centrifugal compressor is analyzed in order to predict rotordynamic stability using the best available resources for seals and bearings. Impeller shroud forces are predicted using the bulk-flow model developed by Gupta and Childs [1]. Each impeller stage is analyzed and an attempt is made to improve the estimation of impeller aerodynamic excitation forces. Logarithmic decrement (log dec) predictions for the full rotor model consisting of all the stages and seals are compared to the full-load full-pressure test measured values using a magnetic bearing exciter. A good correlation is obtained between the measured test results and analytical predictions.

lar weight of the gas, D is the impeller outside diameter, h_t is the impeller tip opening at discharge, and ρ_D and ρ_S are the fluid densities at discharge and suction, respectively. The current API standard uses MW=30 for the calculation of cross-coupled stiffness, and is a modified Wachel's formula. It combines all possible stabilizing actions caused by damping from the impellers and all destabilizing force actions from the impellers into one purely destabilizing element to be applied at the center of the rotor model. Some analysts have tried to further refine the Wachel model's application by calculating labyrinth seal forces and then using Wachel's model separately to account for the unknown forces that are required to explain observed instabilities.

Memmott [7,8] introduced the Modal Predicted Aero Cross Coupling (MPACC) number. This number is defined as

$$MPACC = 189000 \times \sum_{j=1}^{NI} \frac{HP_j}{N \times D_j \times h_j} \left(\frac{\rho_d}{\rho_s} \right)_j x_j^2 \qquad (3)$$

where x_j is the modal co-ordinate of the stage. By taking a modal sum based on the first forward whirling mode shape, an effective aero-cross-coupling coefficient (Kxy) is calculated and applied at the mid-span of the rotor. This has been benchmarked on numerous test cases operating with different mole weight gases, and has been successfully used by the author's company.

Since the introduction of Wachel's destabilizing aero excitation formula, research interest has slowly arisen in the study of impeller dynamics and rotordynamic characteristics. Bolleter *et al.* [9] presented rotordynamic-coefficient data for several pump impellers. Childs [10] then compared Bolleter's test data with reasonable success to his bulk flow model for pump impellers. Yoshida *et al.* [11] made flow and pressure measurements in the back shroud/casing clearance of an inclined precessing centrifugal impeller and integrated the unsteady pressure distribution to obtain the fluid moment on the precessing impeller shroud, showing good correlation between bulk flow model and test measurements. However, they never made force measurements on the shrouded impeller.

Gupta and Childs [1] presented a bulk-flow model for the annular flow paths between the impeller shroud and its adjacent housing based on Childs [10] earlier work for pump impellers. They predicted both the force and moment coefficients for a compressor impeller shroud surface using the reaction-force/moment model of the form given in Equation (4), where (X,Y) and (α_Y, α_X) are components of the impeller's displacement and rotation vectors, and (f_X, f_Y) and (M_Y, M_X) are components of the impeller reaction force and moments.

$$-\begin{Bmatrix} f_X \\ f_Y \\ M_Y \\ M_X \end{Bmatrix} = \begin{bmatrix} K & k & K_{\varepsilon\alpha} & -k_{\varepsilon\alpha} \\ -k & K & -k_{\varepsilon\alpha} & -K_{\varepsilon\alpha} \\ K_{\alpha\varepsilon} & k_{\alpha\varepsilon} & K_\alpha & -k_\alpha \\ k_{\alpha\varepsilon} & -K_{\alpha\varepsilon} & k_\alpha & K_\alpha \end{bmatrix} \begin{Bmatrix} X \\ Y \\ \alpha_Y \\ \alpha_X \end{Bmatrix} + \begin{bmatrix} C & c & C_{\varepsilon\alpha} & -c_{\varepsilon\alpha} \\ -c & C & -c_{\varepsilon\alpha} & -C_{\varepsilon\alpha} \\ C_{\alpha\varepsilon} & c_{\alpha\varepsilon} & C_\alpha & -c_\alpha \\ c_{\alpha\varepsilon} & -C_{\alpha\varepsilon} & c_\alpha & C_\alpha \end{bmatrix} \begin{Bmatrix} \dot{X} \\ \dot{Y} \\ \dot{\alpha}_Y \\ \dot{\alpha}_X \end{Bmatrix} + \begin{bmatrix} M & m & M_{\varepsilon\alpha} & -m_{\varepsilon\alpha} \\ -m & M & -m_{\varepsilon\alpha} & -M_{\varepsilon\alpha} \\ M_{\alpha\varepsilon} & m_{\alpha\varepsilon} & M_\alpha & -m_\alpha \\ m_{\alpha\varepsilon} & -M_{\alpha\varepsilon} & m_\alpha & M_\alpha \end{bmatrix} \begin{Bmatrix} \ddot{X} \\ \ddot{Y} \\ \ddot{\alpha}_Y \\ \ddot{\alpha}_X \end{Bmatrix} \qquad (4)$$

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The model accounts for shroud forces but does not consider potential impeller-diffuser interaction forces. Gupta and Childs [1] reported a more stable stage than that predicted using Wachel's model. This was in direct agreement with Memmott [7] for a large centrifugal compressor. Memmott found poor correlation between predictions and test and field experience using the specific Wachel's formula. While Wachel's model predicted an unstable compressor for Memmott's large centrifugal compressor, the compressor was stable in operation. Qualitatively, the impeller-shroud model of Equation (4) did a good job in predicting the frequency characteristics of the measurements and an adequate job in predicting the cross-coupled stiffness k and direct damping C.

To date, there has not been a clear validated rotordynamic program for impeller-shroud force predictions. Furthermore, there has been no comparison between rotor stability predictions using the impeller bulk-flow model and test results. Using the log dec measurement technique described by Moore, Walker, and Kuzdzal [12] and Moore and Soulas [13], log dec for a re-injection machine is measured and comparisons are made between the test results and the predictions.

NOMENCLATURE

MPACC = Modal Predicted Aero Cross-Coupling

NI = Number of Impellers

ω_d = Unloaded First Damped Natural Frequency (rad/s)

δ = Logarithmic Decrement

x_j = Modal Co-ordinate of the Stage

$(k_{xy})_{Eff}$ = Effective Aero-Cross Coupled Stiffness

HP_j = Horsepower of Impeller j (hp)

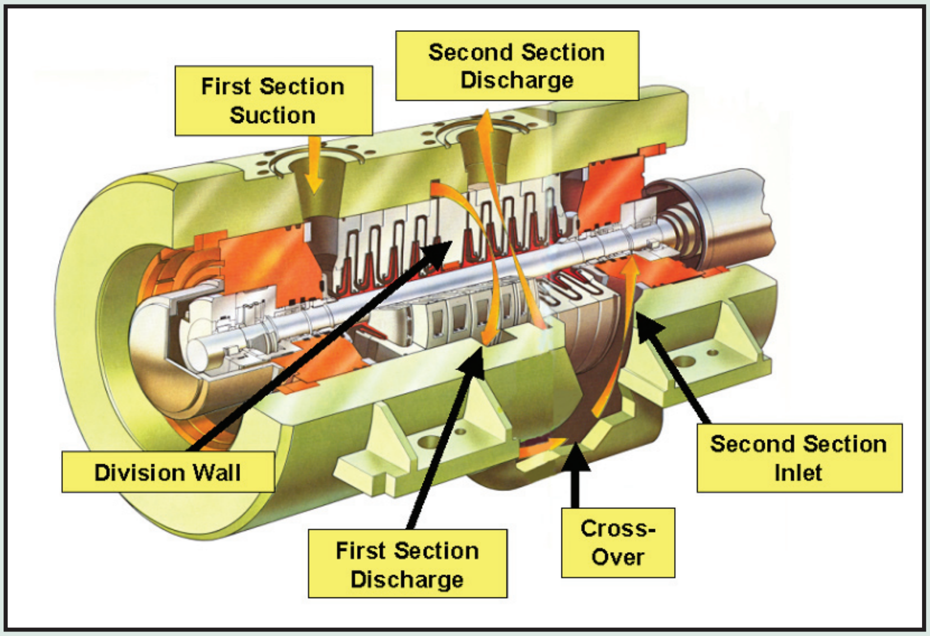

Figure 1 - Sketch of high-pressure, back-to-back compressor

RE-INJECTION COMPRESSOR DESCRIPTION

The compressor used in this study is a nine-stage, noninter-cooled, back-to-back centrifugal compressor. A sketch of a typical back-to-back centrifugal compressor is shown in Fig 1. The center division wall seal is a damper seal (hole pattern seal). The compressor has tilting-pad radial journal bearings in series with squeeze film dampers, a tilting-pad thrust bearing, and dry gas seals. This unit has a 7200 psi case pressure rating. The advantages of using a back-to-back machine for high-pressure re-injection applications have been mainly better thrust balance ability especially at off-design conditions, elimination of large diameter balance piston resulting in less leakage, and higher damping because of the optimum location of hole pattern seal.

ANALYTICAL MODELING

Rotordynamic modeling of this nine-stage, back-to-back, high-pressure re-injection compressor is done by considering only the reaction

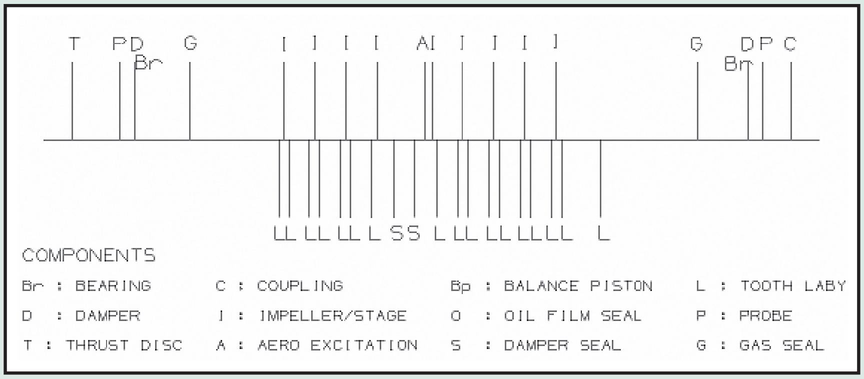

Figure 2 - Schematic of the rotor model

forces and no moments. A model of the rotor is built using a rotordynamic analysis software developed by Ramesh [14]. A schematic of the rotor model is shown in Fig 2. For the complete lateral analysis of this compressor, the following components are modeled: tilt-pad journal bearings in series with squeeze film dampers, hole pattern division wall seal, impeller eye and interstage stationary tooth labyrinth seals, second-section gas balance stationary tooth labyrinth seal, and all nine impeller stages.

Tilt-pad bearing coefficients are computed using the work of Nicholas, *et al* [15]. Hole pattern damper seal coefficients are determined using the ISOTSEAL program developed by Kleynhans and Childs [16]. The toothed labyrinth seals are modeled by the program of Kirk [17]. Impeller coefficients are calculated using both the modified form of Wachel number (MPACC) and the bulk-flow impeller code developed by Gupta and Childs [1]. The impeller bulkflow code solves the turbulent bulk-flow continuity and momentum equations to obtain full force and moment reaction matrices. Therefore, a reduced reaction force only matrix of the form given in Equation (5) is used for the analysis,

$$-\begin{Bmatrix} F_X \\ F_Y \end{Bmatrix} = \begin{bmatrix} K & k \\ -k & K \end{bmatrix} \begin{Bmatrix} X \\ Y \end{Bmatrix} + \begin{bmatrix} C & c \\ -c & C \end{bmatrix} \begin{Bmatrix} \dot{X} \\ \dot{Y} \end{Bmatrix} + \begin{bmatrix} M & m \\ -m & M \end{bmatrix} \begin{Bmatrix} \ddot{X} \\ \ddot{Y} \end{Bmatrix} \quad (5)$$

For seals and bearings, the mass matrix is negligible and is often eliminated in modeling.

The current approach to compute the total impeller aero-excitation involves computing the aero excitation for each stage and then taking the modal sum by squaring the normalized deflection of the first fundamental frequency. A similar approach is followed when using the impeller bulk-flow code. The bulk-flow code produces both cross-coupled stiffness *Kxy* and direct damping *C*; therefore effective cross-coupled stiffness defined in Equation (6) is used to compute the net aero cross coupled stiffness:

$$(k_{xy})_{Eff} = k_{xy} - C\omega_d \quad (6)$$

The first forward damped natural frequency ω_d in the above equation 6 is taken as the unloaded damped frequency. In this particular study, the loaded damped natural frequency differed only slightly from the unloaded damped natural frequency.

MAGNETIC BEARING EXCITER TEST SETUP

The Type 1 test in accordance with ASME PTC-10 test specification consisted of eight head-capacity points (1-8), and three additional data points (9-11) at the end of the test, all eleven points in co-ordination with magnetic bearing exciter sweeps. The three last test points were taken during the evacuation of the gas from the test loop to obtain data at decreasing density profiles across the compressor. These three additional points provide further information about the effect of density on the stability of the system. The magnetic bearing exciter was attached to the free end of the rotor, and an asynchronous force was injected into the rotor system to excite the first forward whirling mode. This technique measures the rotor's logarithmic decrement (log dec) as described by Moore, Walker and Kuzdal [12]. A solid model assembly with the magnetic bearing exciter installed on the shaft is shown in Fig 3 and the magnetic bearing exciter used is shown in Fig 4.

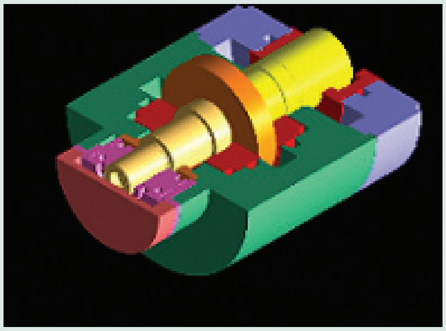

Figure 3 - Solid model of the magnetic bearing exciter

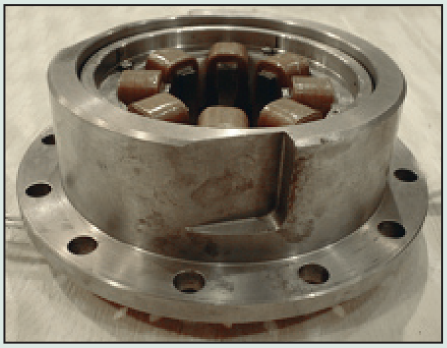

Figure 4 - Magnetic bearing used for the excitation

COMPARISION BETWEEN TEST DATA AND ANALYTICAL RESULTS

Fig 5 shows predicted impeller cross-coupled stiffness as a function of the discharge pressure for constant speed data points 6, and 9-11. The bulk-flow impeller code predicts the net aero-cross coupling forces to increase with the increase in discharge pressure. Comparison between measurement and predictions of the rotor log dec as a function of discharge pressure is shown in Fig 6 for the constant speed data points used in Fig 5. Fig. 6 illustrates that system stability increases with increasing discharge pressure. This result can be attributed mainly to the "dominance" of the hole pattern seal at higher discharge pressures. Thus, even though the impeller generated aero- cross cou-

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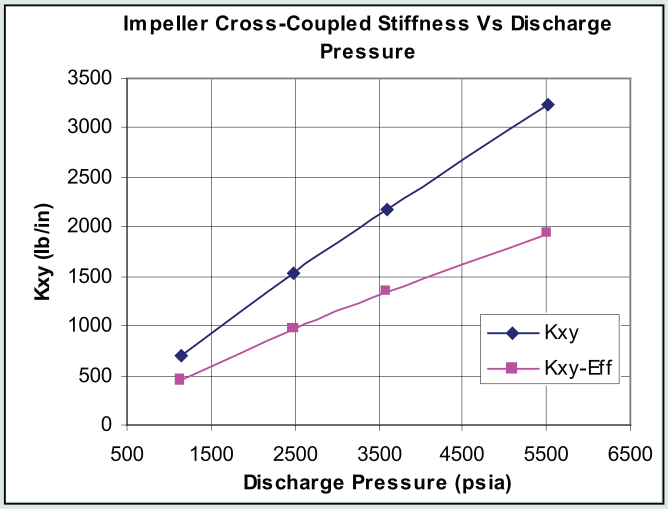

Figure 5 - Bulk-flow impeller code predicted cross-coupled stiffness vs. discharge pressure

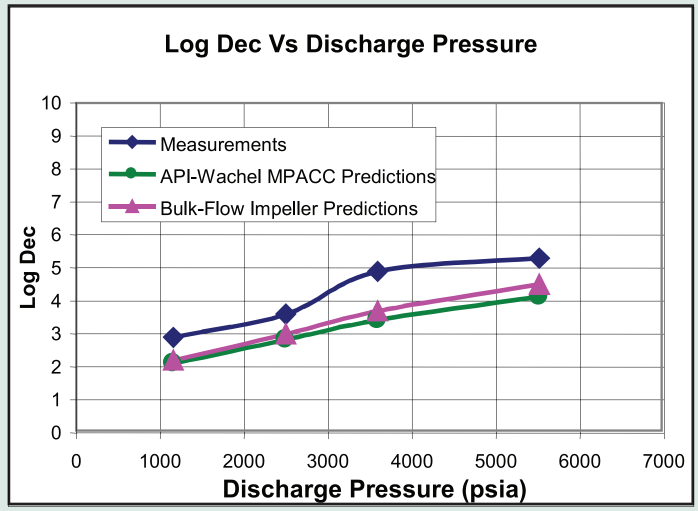

Figure 6 - Test and predicted rotor log dec vs. discharge pressure

pling force increases with discharge pressure, hole pattern seals provide net positive effective damping at a faster rate than the excitations. A good correlation is obtained between the measurement and predictions, especially considering the stability trend with discharge pressure. The small under-prediction in the log dec can be attributed to various uncertainties and the conservative nature of predictions.

There is a small improvement in the predictions using the bulk-flow impeller model. The impeller force excitation does not have a significant impact on the overall system stability for this high-pressure re-injection machine.

The impeller bulk-flow code correctly predicted the increased stability and provided a slightly better match than API-Wachel based MPACC, but the impeller coefficients are significantly small when compared to the hole pattern seal. Complete predictions of all the test points covering a wide range of operating conditions are given in Table 1.

Point #	Inlet Pressure psia	Discharge Pressure psia	Discharge Density Lbs/ft^3	Gas Power HP	Speed rpm	Measured Log Dec δ	API-Wachel MPACC Log Dec δ	Bulk-Flow Log Dec δ
1	2327.98	5586.58	20.08	11633	7234	4.8	5.5	6.1
2	2313.80	5528.14	20.49	7993	6494	6.2	6.3	6.8
3	2333.81	5559.78	20.53	5875	6153	6.8	7.3	7.8
4	1756.30	5352.03	13.11	14704	9495	5.8	3.9	4.4
5	2010.38	5526.35	13.57	14313	9150	5.2	3.8	4.2
6	2016.08	5514.50	13.68	10487	8486	5.3	4.1	4.5
7	2050.88	5574.98	13.68	8276	8199	5.7	4.3	4.5
8	1469.51	5408.53	12.27	10464	9492	5.3	4.3	4.6
9	1205.14	3590.58	9.84	6808	8480	4.9	3.4	3.7
10	831.70	2494.42	7.19	4701	8471	3.6	2.8	3
11	402.59	1156.04	3.35	2149	8504	2.9	2.1	2.2

Table 1 - Test operating conditions and log dec values

Table 1 clearly shows that system stability is maintained, even when running at off-design conditions. Points 1 and 2, although operating at similar discharge pressure and gas density, differ in running speed. This change in speed produces less damping for point 1, which is close to the overload limit. Small over-prediction occurs at points 1, 2 and 3 as shown in the Table 1. Also note that the high log dec measured at low discharge pressure is due to external squeeze film dampers in series with the tilt-pad journal bearings.

SUMMARY, DISCUSSSION, AND CONCLUSIONS

A complete analysis of the rotor model is done using the state-of-the-art tools available, including the newly developed impeller bulk-flow code. Stability predictions were made for a wide range of conditions to cover both the head changes and density influence. Predicted results using a bulkflow model and the MPACC numbers were compared to the measured test data results. A good correlation is obtained between measurement and predictions. Actual impeller force coefficients were predicted by integrating the dynamic pressure and shear stress field in the shroud-casing clearance, thus providing a reasonable estimate of the impeller contribution to the overall rotordynamic stability. This new model has helped in rational estimation of the impeller shroud forces. The results show that increasing gas density yields increased stability when hole pattern seals are used. The measurements provide further validation in the analytical tools and have helped in further validating the bulk-flow code predictions. The results presented demonstrate the low impact of impeller produced aeroexcitation on the rotordynamic stability when hole pattern seals are used at high discharge pressure. Clearly, hole pattern seals are the most dominant element at high discharge pressure, and knowing the exact running clearances in operation, although difficult to achieve, could further improve the predictions. It has been shown that the compressor is stable for a wide range and can operate satisfactorily under off-design conditions from a rotordynamic stability standpoint.

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