EXPERIENCES AND APPLICATIONS OF API 617
FULL STABILITY ANALYSIS (LEVEL-II) ON CENTRIFUGAL COMPRESSORS

by

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ABSTRACT

To cover the remaining operational life of an offshore platform, it was necessary to go from a single stage compressor to a two-stage compressor. The design of the new two-stage compressor required two different impeller arrangements to cover the range of suction/discharge pressures anticipated. One rotor has nine wheels and another one has 11 by adding one impeller at the outboard ends of both compression sections. With a rotor flexibility ratio around of 2.5, the new replacement compressor is a borderline machine on the API 617 (2002) stability screening map. The stability study indicated that the compressor failed to meet the Level-I stability criteria at the design conditions. A detailed stability investigation became necessary though the machine was predicted to be stable based on an original equipment manufacturer's (OEM) empirical method.

An independent party was contracted to conduct a full stability audit on both rotors. Compressor cross-coupling was evaluated uniquely and lumped at each impeller location in the analysis model. Additionally, the destabilizing forces from all labyrinth seals in the compressor were calculated and included in the detailed stability analysis. The study results indicated that both rotors would most likely be susceptible to severe instability problems even using a modified center labyrinth seal with a shunt-injection and swirl brakes. Consequently, the effect of damper seals on the rotor stability was further investigated independently. It was concluded that a honeycomb seal could bring the log-decrement to a distinctively positive value for the nine-wheel rotor, but not succeed in raising the log-decrement to positive for the 11-wheel rotor.

Because of faster than anticipated declines in well deliverability, it was decided to bypass the nine-wheel configuration and install the 11-wheel configuration. The 11-wheel rotor used a modified center labyrinth seal with a shunt/brake deswirl system. As a backup, the nine-wheel rotor and the inner casing were modified to fit with a pocket damper seal. The 11-stage compressor was successfully commissioned and has run well in the field for more than one year. Overall shaft vibration level was low and the predicted subsynchronous vibration was not observed. In this paper, a correlation study was carried out and a comparison was made to discuss how to calculate the Wachel number, and how to practically evaluate the contribution of the labyrinth seals and the gas deswirl devices in a detailed (API Level-II) rotor stability analysis.

BACKGROUND OF COMPRESSOR REPLACEMENT PROJECT

Steelhead Platform is located in the Cook Inlet region of Alaska and is operated by Unocal Alaska, part of Union Oil of California. Primary function of the Steelhead Platform is to recover natural gas from the Grayling Gas Sands (GGS) Reservoir. The platform was installed in the mid 1980s, and started production in 1987, at which stage there was a single 5000 hp gas compressor train (C-12). During the early 1990s, a second 6500 hp gas compression train (C-13) was added to the platform, to provide redundancy and the soon to be required parallel gas production train. In 1997, the original C-12 compressor package was removed and replaced with a 10,400 hp package comprising a gas turbine driving a single stage 25MB4/4 compressor directly.

In 2002, after more than 15 years of gas production from the Steelhead Platform, Unocal Alaska was faced with optimizing project end of life gas recovery operations, and the C-12 replacement project was initiated. The project goal was to attempt to cover remaining operations with a single compressor change, to complete the work with minimum downtime, and to minimize the risk of any post startup issues. Due to the continuous decline of the gas well pressures, it was necessary to optimize the design of the replacement compressor across multiple operating points.

The replacement compression train consists of a new back-to-back centrifugal compressor directly driven by the existing gas turbine, as shown in Figure 1. Considering the variation in the production conditions and the amount of gas to be transmitted, two exchangeable inner casings were designed for the same vertical assembled compressor outer casing. The first inner casing has an expendable two-stage rotor with four and five impellers in the first and second stages, respectively. The second rotor has a total of 11 impellers by adding one impeller at the outboard ends of the compression stages. Compared to the original back-to-back eight-wheel compressor, the maximum continuous operating speed (MCOS) is increased by 19 percent, but the machine power goes up only 3 percent because of the new design at a lower suction pressure and lower gas flow rate. On the other hand, the new compressor rotors have more impellers on the shaft and are more flexible. For convenience the two new compressor rotors are referred to as 25MB4/5I and 25MB5/6I in this paper, respectively. The main operating conditions for both rotors are given in Table 1 along with the compressor design parameters.
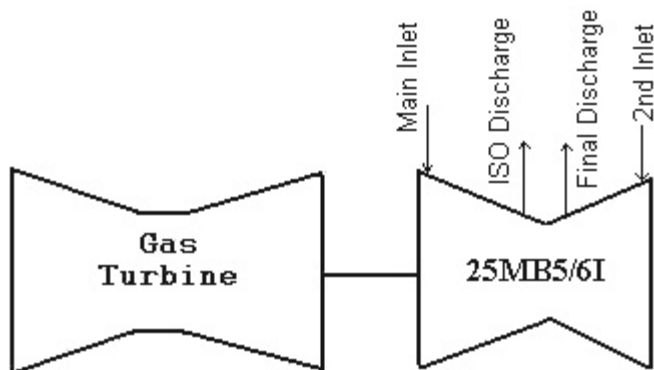

Figure 1. Platform C-12 Compression Train.

Table 1. C-12 Replacement Compressor Design Conditions and Features.

	25MB4/5I		25MB5/6I	
	1st stage	2nd stage	1st stage	2nd stage
Inlet P (psi)	275	587	115 ~ 165	329 ~ 434
Inlet T (°F)	70	90	70	90
Discharge P (psi)	597	1245	334 ~ 439	1065 ~ 1145
Discharge T (°F)	200	227	234 ~ 260	264 ~ 307
Gas Molecular Wt	16.25		16.25	
Flow (lbs/min)	3460 ~ 3910		1771 ~ 2400	
Design Speed (rpm)	9950 ~ 10240		10250 ~ 11620	
Max. Power (Hp)	10700		10700	
Max. Speed (rpm)	12000		12000	
Bearing Span (in)	73.34		73.34	
Flexibility Ratio	2.48		2.52	
No. of Impeller	9		11	

The compressor rotors are fitted with dry gas seals and supported on two conventional five-shoe, tilt-pad journal bearings. The diameter of the journal bearing is 5.0 inches and the bearing load is applied in the direction between two bottom pads. The detailed bearing geometry is provided in Table 2. Figure 2 is a cross-section sketch of the 25MB5/6I inner casing to show the impeller arrangement and the axial locations of seals. Shaft seals and impeller eye seals are of a typical straight-through labyrinth. Abradable stationary seal materials are used with rotating labyrinth teeth and allow tighter seal clearances to achieve a higher efficiency. It is tolerable for the rotating teeth to wear into the sacrificial stator material during transient periods without damage to the rotor. The back-to-back compressor has a long intermediate center labyrinth seal between the two compression stages. Compared to other labyrinth seals in the compressor the center seal would sustain the largest differential pressure. The original design of the center seal is a type of teeth-on-rotor labyrinth seal, which rides against an abradable stationary component with a smooth bore surface.

Table 2. Journal Bearing Parameters for C-12 Replacement Compressor.

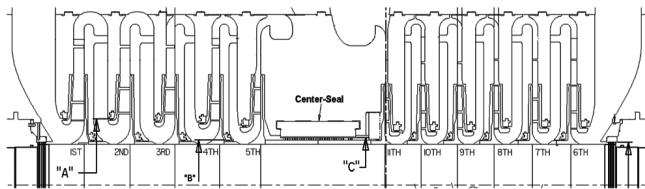| Diameter (in)	L/D	Bearing Preload	Assembled Diametral Clearance (in)	Pivot Location
5.00	0.425	0.402	0.007	55%

Figure 2. Cross-Section Sketch of C-12 Replacement Compressor 25MB5/6I.

The 25MB4/5I compressor successfully passed its mechanical test at the manufacturer's facility. However during the performance test with a partial load, a small subsynchronous vibration (0.04 mils peak-to-peak) showed up and the overall vibration level of the compressor was below 0.5 mils. As shown in Figure 3, the low level 70 Hz subsynchronous peak existed for all operating points away from surge line and its frequency was coincident with the first critical speed of the rotor. This phenomenon is not abnormal in performance tests and field operations with a loaded centrifugal compressor. Due to possible excitation sources such as normal flow turbulence, pressure fluctuation, and noise, some low-level forced vibration likely exists at rotor natural frequencies. Based on the aerodynamic excitation force calculated using the manufacturer's proprietary equation, rotordynamics analysis illustrated that the compressor would be stable at the full rated load.
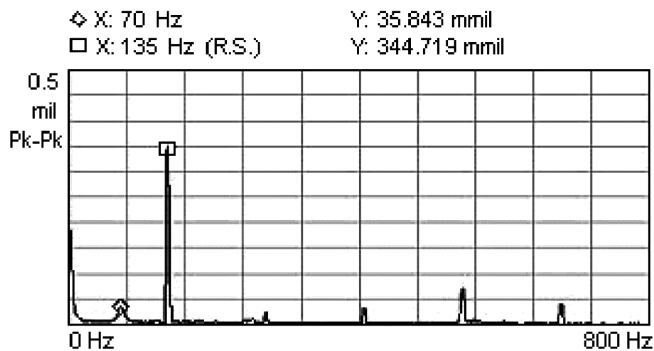

Figure 3. Vibration Spectrum at Partial Load Performance Test.

However, the potential instability problem was still a customer concern. An independent rotor stability audit was performed in accordance with the customer's request. Aerodynamic destabilizing forces including Wachel numbers and all labyrinth seals were considered in the full stability investigation. A unique procedure was adopted to estimate the Wachel numbers. The results of the study increased the concern related to an instability problem and indicated that a special damper device such as a honeycomb seal had to be installed to improve the rotor stability. Available analytical tools, application experience, and stability criteria were thoroughly discussed by the manufacturer and the customer. It was mutually agreed to install a hybrid shunt-brake system as a direct counteraction by modifying the discharge volute. The objective of the change was to reduce gas swirl velocity in the leading portion of the center labyrinth seal and consequently to reduce the destabilizing cross-coupling force.

However, due to well-pressure declines greater than forecasted, it was decided to first install the 11-wheel bundle 25MB5/6I with the added shunt-brake device in 2003. Meanwhile as a stability contingency plan per the customer's request, the spare bundle 25MB4/5I was changed to fit with a center damper seal. Although two kinds of gas damper seals, a honeycomb seal and a pocket seal, were available and both were analyzed as well, the pocket-damper seal was used finally in the spare bundle. In September 2003, the replacement compressor 25MB5/6I was run at various points to check rotor stability and confirmed no subsynchronous vibration

issues. The new compressor has smoothly run more than one year with a low level synchronous vibration.

STABILITY DESIGN AND AUDIT ON C-12 REPLACEMENT COMPRESSOR

According to the progress arrangement of the project, original rotordynamics review on both rotors 25MB4/5I and 25MB5/6I was finished prior to the release of the API 617, Seventh Edition (2002). After June of 2002, the customer decided to proceed with the second rotor with 11-wheels first. The rotor stability was reinvestigated including the effect of labyrinth seal, deswirl remedy, and damper seals. A stability study per API 617 Level-I requirements was performed on the rotors as well for internal reference usage. Meanwhile the user of the compressor contracted a third party to conduct an independent rotor stability design audit, and the manufacturer was responsible for providing rotor models and other required information except for the aerodynamic excitation forces to enable the third party to conduct the full stability analysis.

Original Stability Study on C-12 Compressor Rotors

For the C-12 compression train, the molecular weight of the process natural gas is 16.25 and the expected average gas density varies from 1.24 to 1.84 lb/ft³. The flexibility ratio CSR (MCOS/first rigid support critical speed) of the new rotor is about 2.5. Figure 4 shows the rated design positions of the 25MB4/5I and 25MB5/6I of the C-12 train in an API 617 Level-I screening map along with several reference cases. These referenced compressors have been operating well in the field. Apparently the new replacement rotors are more flexible than the original eight-wheel compressor rotor by increasing the CSR value from 2.0 to 2.5. The map also indicates that the subject two C-12 rotors are still within the envelope of previous design experience and applications of the OEM.
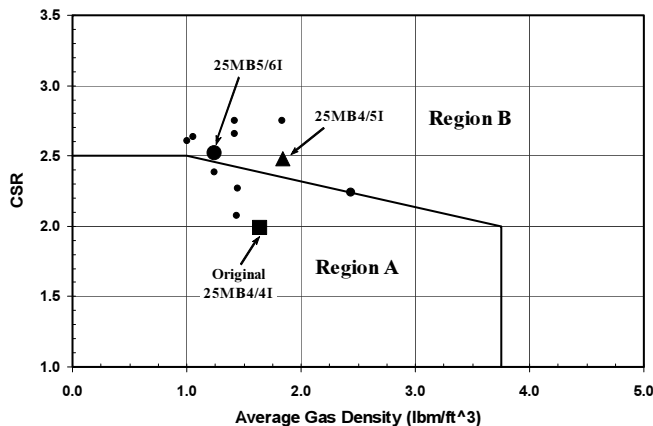

Figure 4. Platform C-12 Compressor Positions on API Screening Map.

In the original rotordynamics examination, the rotor unbalance response, critical speeds, and separation margins related to the operating speeds were calculated per API 617, Sixth Edition, and the results were acceptable. The rotor stability was reviewed based on the manufacturer's empirical method and criteria. To estimate the level of the load-dependent destabilizing force applied on the compressor rotors, aerodynamic cross-coupling stiffness was predicted using the manufacturer's proprietary equation in the following form:

$$Q = F\left(P_i, T_i, P_d, T_d, M_w, N_{\max}, N_{rl}\right) \qquad (1)$$

The total cross-coupling stiffness was estimated based on compressor design and operating conditions including the maximum operating speed (N_{max}), the gas molecular weight (M_w), the inlet and discharge gas pressures (P_i and P_d), the inlet and discharge gas

temperatures (T_i and T_d) of the compressor, and the rotor rigid support critical speed (N_{r1}). In this paper, a cross-coupling stiffness of 18,200 lb/in and 15,300 lb/in was predicted for the subject 25MB4/5I and 25MB5/6I rotors, respectively, using the above empirical equation. In general, this is an end-to-end approach to estimate the aerodynamic cross-coupling generated by all identified and unidentified destabilizing sources in the compressor. Therefore, the effect of labyrinth seals on rotor stability was not separately investigated originally.

The predicted total cross-coupling stiffness was evenly distributed at impeller locations in the log-decrement calculation to evaluate the compressor stability. Table 3 gives the rotor basic and aero log-decrements predicted at the maximum operating speed of 12,000 rpm and the average bearing clearance. Although the second rotor 25MB5/6I had two more wheels than the first rotor 25MB4/5I, the expected cross-coupling was lower because of a lower pressure range. Consequently the predicted aero log-decrement was slightly higher for 25MB5/6I. The predicted log-decrement values were within the experience envelope and accepted by following empirical criteria. Note the nonsynchronous coefficients of the journal bearing were adopted with the OEM's empirical method. People should be aware that the log-decrements are not directly comparable if different types of bearing coefficient are used in rotor stability analyses. Normally using synchronous bearing coefficient yields a larger log-decrement at the same aerodynamic excitation level.

Table 3. Rotor Stability/Log-Decrements Based on the OEM's Method.

Rotor	No. of Wheels	Total Aero Q (lb/in)	Basic Log-dec	Aero Log-dec
25MB4/5I	9	18,200	0.191	0.061
25MB5/6I	11	15,300	0.191	0.094

API 617 Stability Screening Study

Although manufacturers and users of turbomachinery have been familiar with load-dependent instability problems in centrifugal compressors for quite a long time, there was no uniform procedure for engineers to follow in compressor stability analyses prior to the release of API 617, Seventh Edition (2002). The Level-I stability study of API 617 provides a standardized method and the same criteria for both manufacturer and users to perform a preliminary evaluation on compressor stability.

As a practical procedure for industrial applications, the API 617 Level-I stability analysis sets up a guideline for what is the minimum analytical work to be done to ensure a rotordynamically stable compressor. Based on a simplified conservative procedure defined in API 617 Level-I, those compressors requiring a detailed stability study will be identified. A modified Wachel equation is provided in Level-I for the evaluation of the anticipated aerodynamic excitation Q_A. The Level-I stability study also requires using synchronous bearing coefficients if the rotor is fitted with tilt-pad journal bearings. If squeeze film dampers are used in the compressor, their effect should be included in the Level-I preliminary stability analysis. The destabilizing contributions of labyrinth seals will be further considered separately in the API 617 Level-II study when a detailed stability analysis is finally necessary.

According to the Level-I stability requirement, a minimum log-decrement of 0.1 must be achieved at the extremes of journal bearing clearance with the anticipated cross-coupling excitation, and certain stability margins are needed to account for any uncertainties in the modeling and analysis. The amount of the aerodynamic cross-coupling to generate a zero log-decrement is defined as Q_0. In the API 617 Level-I stability study, the ratio of (Q_0/Q_A) is called the safety factor of the compressor rotor stability. For a compressor within Region "A" in the API screening map, the

minimum safety factor should not be less than 2.0. If the compressor falls into Region "B," a more conservative safety factor of 10.0 is needed to meet the Level-I stability criteria.

As shown in Figure 4, the two rotors of the C-12 compressor, 25MB4/5I and 25MB5/6I, belong to borderline machines in the API screening map. A safety factor of 10.0 would be required for the subject rotors to satisfy API Level-I requirements. To the authors' knowledge, it is analytically almost impossible for such flexible compressor rotors to sustain so large a stability safety factor without using an exceptional damping device. Following the procedure in the API Level-I stability study, the rotor stability (log-decrement) was still calculated at the cross-coupling excitation levels Q_A and $2Q_A$, and the predicted results are shown in Table 4. Clearly, both rotors did not have a satisfactory safety factor and even their log-decrement could not reach the minimum acceptance value of 0.1 in the minimum and nominal bearing clearance cases at the excitation level of Q_A. It could be concluded that both 25MB4/5I and 25MB5/6I rotors did not pass the API 617 Level-I stability criteria. Note that the predicted aero log-decrements in Tables 3 and 4 are not comparable because these aero log-decrements were calculated with different bearing coefficients and aerodynamic excitation forces.

Table 4. Log-Decrements per API 617 Level-I Stability Study Approach.

Case	Total Q_A (lb/in)	Minimum Bearing Clearance		Average Bearing Clearance		Maximum Bearing Clearance	
		Q_A	$2Q_A$	Q_A	$2Q_A$	Q_A	$2Q_A$
25MB4/5	28,600	-.057	-.310	.012	-.253	.154	-.137
25MB5/6	26,000	-.068	-.293	-.010	-.245	.110	-.145

It should be pointed out that the stability analysis approach of API Level-I is conservative. Level-I study results do not directly reflect the actual operating conditions and the real dynamic behavior of the compressor investigated. Failing to meet the Level-I requirements does not mean an unacceptable stability design and just indicates a detailed stability investigation (Level-II) should be performed on these rotors. Obviously, the rotordynamic force coefficients of labyrinth seals in the compressor are essential and critical for conducting a full rotor stability analysis. Like the manufacturer's method discussed above, the API Level-I stability study does not explicitly investigate the contribution from any labyrinth seal and deswirl stabilizing device either.

An Independent Compressor Stability Design Audit

After reviewing the available results of the rotor stability study on the C-12 replacement compressor, the compressor user decided to contract a third party to perform an independent rotordynamics analysis. The investigation was focused on the stability design of both nine-wheel and 11-wheel rotors. Based on the basic mass-elastic data transferred from the OEM, the lateral model was rebuilt by including an additional stiffening effect to the shaft due to the impellers shrunk on the shaft. Bearing coefficients of the five-shoe, tilt-pad journal bearings were regenerated using a different bearing code developed by the research facility of a major university. The calculated rotor unbalance response results and plots, including critical speed map, damped mode shapes, critical speeds, and amplification factors, are close to the analysis results previously provided by the manufacturer. Therefore the correctness of the rotor models and analytical tools used for the rotordynamics study was confirmed. However, there was an apparent discrepancy in the evaluation results of the rotor stability between the manufacturer and the consultant.

The independent stability study was conducted at the rated compressor operating conditions and the blueprint assembled clearances

of the journal bearings and the labyrinth seals. Like the manufacturer's approach, nonsynchronous bearing coefficients were adopted in the stability calculation. Given below is the well-known Wachel's equation, which was used to estimate the overall aerodynamic cross-coupling applied on the compressor rotor in the stability analysis.

$$Q = 6300 \times \frac{HP \times M_{wt}}{D \times b \times N}\left(\frac{\rho_d}{\rho_s}\right) \qquad (2)$$

Wachel's cross-coupling equation resulted from correlation studies of rotor instability experienced by centrifugal compressors earlier and has taken into account all the destabilizing effects from impeller, labyrinth seals, oil seals, etc. Gas properties, horsepower absorbed, and impeller/diffuser geometries should be known to use this formula. To date, Wachel's equation is one of most common formulae used by compressor users and consultants when the rotor stability needs to be examined. However, there are various approaches of estimating the aerodynamic cross-coupling (Wachel number) in using this equation. The Wachel number can be calculated wheel by wheel with local gas density ratio and local horsepower transmitted at each impeller. On the other hand, a total cross-coupling can be first calculated and then distribute the total cross-coupling at individual impellers. The most conservative method in estimating the Wachel number is to use the last impeller/diffuser geometries with the overall compressor gas density ratio and the entire power absorbed.

In the independent stability investigation, a unique approach was followed to calculate the aerodynamic cross-coupling. Based on their experience, the consultant estimated a local Wachel number wheel by wheel using a combination of the local wheel horsepower and the overall compressor gas density ratio. Following this approach, a total aerodynamic cross-coupling Q_T of 48,600 lb/in was anticipated for the nine-wheel 25MB4/5I rotor and a larger total cross-coupling Q_T up to 74,500 lb/in was estimated for the 11-wheel 25MB5/6I. The estimated Wachel numbers at the impellers were used as a baseline to further perform a detailed rotor stability analysis, which would include all labyrinth seals in the compressor. The rotordyanmics force coefficients of the labyrinth seals were to be solved at the designed conditions. To meet this end, the compressor manufacturer further provided the third party the detailed gas properties at each labyrinth seal along with the seal geometry data. Considerable effort was made to calculate the cross-coupling stiffness of the labyrinth seals and complete the full stability study.

The independent full stability analysis was conducted at the nominal seal clearance only. Much more effort and engineering time would be required considering the variation of the assembled seal clearance. Including impeller eye seals, shaft seals, buffer seals, and the center seal, there are total of 23 labyrinth seals in the 25MB5/6I casing and 19 labyrinth seals in the 25MB4/5I casing. A thorough labyrinth seal analysis confirmed that the major seal destabilizing cross-coupling was generated by the center labyrinth seal, which has the longest axial length and sustains the maximum differential pressure of about 650 psig in the C-12 replacement compressor. Using a particular procedure allowed extraction of the cross-coupling of the labyrinth seals from the baseline Wachel numbers to avoid duplication of the seal effect in the full stability analysis. For the C-12 replacement compressor, it was found that the labyrinth seals contributed about 10 percent to the total aerodynamic cross-coupling. A 10 percent contribution from the labyrinth seals in a back-to-back compressor was considerably low and no sound explanation was given by the auditor. It was claimed that the labyrinth seal forces could provide around 35 percent of the overall aerodynamic cross-coupling in other centrifugal compressors.

As shown in Table 5, rotor stability (log-decrement) was calculated at the excitation levels 100 percent and 90 percent of the Q_T to examine the effect of the labyrinth seals. In order to anticipate the potential maximum effect of the labyrinth seals on the rotor

stability, the log-decrement was calculated at 65 percent of the Q_T as well. It seems that the influence of the labyrinth seals on the stability was limited because the predicted log-decrement was still negative by removing 35 percent aerodynamic cross-coupling from the rotor models.

Table 5. *Predicted Cross-Coupling and Log-Decrements at the Rated Speeds.*

Rotor	Cross-Coupling Excitation Q_T (lb/in)	Log-dec (100% Q_T)	Log-dec w/o Seals (90% Q_T)	Log-dec (65% Q_T)
25MB4/5I	48,600	-0.150	-0.111	-0.013
25MB5/6I	74,500	-0.384	-0.340*	-0.200*

* Actual calculation point was not available and the log-dec value was curve fitted data.

Using shunt injection/swirl brake to retard the preswirl gas at the center labyrinth seal entrance was considered and its influence was evaluated in the independent stability audit. The study results indicated realistically it was not possible to bring the aerodynamic log-decrement to the positive for both rotors without a damper. A further modification was proposed to replace the existing center labyrinth seal with a honeycomb damper seal. Table 6 theoretically shows the effect of the proposed honeycomb seal on the rotor stability. Unfortunately, the calculated rotordynamic coefficients of the damper seal and the labyrinth seals were not included in the independent audit report. Based on the auditor's judgment, a well-designed honeycomb seal should be able to achieve a sufficient stability margin for the nine-wheel 25MB4/5I rotor. However, for the 11-wheel rotor, even using a honeycomb seal with shunt injection could not raise the stability to an acceptable level. Exceptional countermeasures were predicted to be required to stabilize the 25MB5/6I rotor.

Table 6. *Stability Enhancement Study Results.*

Rotor	Log-dec using shunt/brake (90% Q_T)	Log-dec using Honeycomb seal (90% Q_T)
25MB4/5I	-0.052	0.138
25MB5/6I	-0.333*	-0.006

A meeting was held in the manufacturer's facility to fully review the rotordynamic audit results and discuss the potential counteractions. Because the well declined very quickly, the user made a decision to proceed with the 11-wheel bundle 25MB5/6I first with the original modification proposed by the manufacturer. The proposal of the OEM was to add a hybrid shunt/brake deswirl system at the center labyrinth seal between the two compression stages. The second rotor 25MB4/5I (nine-wheel bundle) was further modified to fit a center pocket damper seal for further stability enhancement and used as a stability contingency. The labyrinth teeth on the rotating sleeves were machined off and the original abradable seal ring was replaced to accommodate the pocket damper seal at the location of the original center labyrinth seal. This was a permanent change on the 25MB4/5I rotor, and the rotor would not be exchangeable with another rotor without restaging and using new teethed center sleeves.

FIELD OPERATION AND RECONSIDERATION OF THE STABILITY ANALYSIS

Operation Results on Site

In 2003, both the 25MB5/6I contract and 25MB4/5I spare inner bundles were transported to the offshore platform. Shown in Figure 5 is the 11-wheel 25MB5/6I casing first installed and the original

eight-wheel inner casing/rotor was changed out from the C-12 compression train. In the startup check period, the new replacement compressor was run at various operation points to verify its rotordynamic behavior. During the commissioning, the compressor reached a maximum operating speed of 11,090 rpm using the maximum turbine power output. The rated pressure ratios of the two stages and the design maximum discharge pressure were achieved.

Figure 5. Platform C-12 Compressor 25MB5/6I.

Figure 6 shows a typical peak-hold vibration spectrum of the compressor, which was measured in a field test period of 49 minutes. Operations at various possible conditions indicated that there was no concern about the rotor instability problem with this 11-wheel rotor. Note that this rotor was only fitted with a manufacturer's hybrid shunt/brake system and without an exceptional damper. The rotor vibration was dominated by a small synchronous vibration from 0.4 ~ 0.5 mils. Like the phenomenon observed in the partial load runs performed on the manufacturer's test floor, a small nonsynchronous vibration still showed up around the rotor natural frequency of 74 Hz, and its magnitude was stable and less than 0.02 mils.
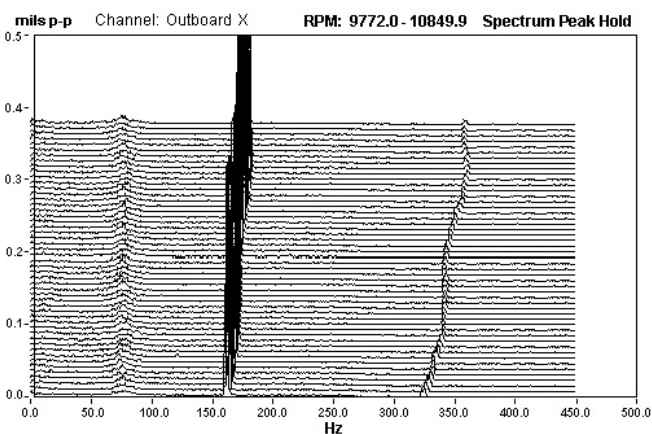

Figure 6. Compressor 25MB5/6I Vibration Spectrum in Commissioning.

The replacement 25MB5/6I compressor was also pushed near the surge line several times to verify the design limit. The spectrum plot in Figure 7 clearly shows that a low frequency at 23 Hz emerged when the compressor was forced to run near its surge line. This low frequency vibration was expectantly excited by a rotating

stall in the compressor. Although the compressor was upset by the rotating stall, the vibration peak at the rotor natural frequency was still stable, which illustrated that the first forward mode of the compressor rotor was well damped.
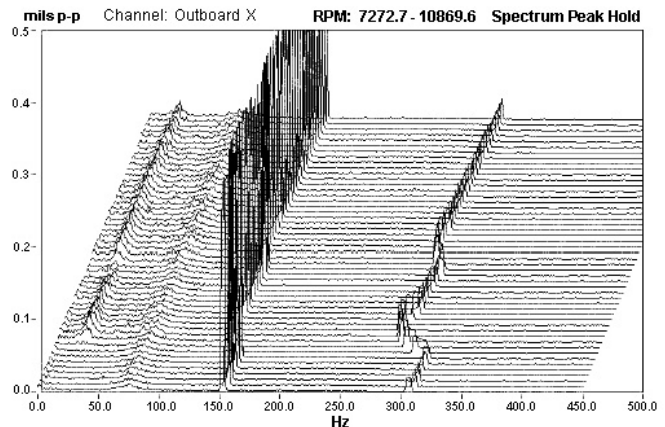

Figure 7. Compressor 25MB5/6I Vibration Spectrum near the Surge Line.

After the commissioning was successfully performed, the C-12 compression train was brought inline to transmit 84 mmscfd natural gas from the offshore platform at the design discharge pressure. Both the user and the OEM were satisfied with the performance and the vibration level of the new replacement compressor. Considering the exchangeability between the two rotors in the future, the nine-wheel 25MB4/5I rotor/casing would be sent back for restaging and change out the damper seal with the deswirl-remedied center labyrinth seal. Although the independent stability study seemed conservative and did not accurately predict the stability in this application, their quantitative results did encourage both the customer and the manufacturer to install the deswirl system in the compressor and helped in preventing the compressor from any potential rotor instability.

The full rotordynamics audit could not accurately predict the rotor stability by including the contribution of the labyrinth seals in this case. However, the empirical method of the OEM can be further improved to enable direct consideration of the influence of labyrinth seals and deswirl devices in the stability calculation. API 617 Level-I stability study has standardized the methodology and the acceptance criteria of a preliminary stability analysis. However, the primary goal of the Level-I study is to screen those compressors that need a detailed analysis (Level-II). There still are some pending issues in the Level-II stability procedure recommended due to the existing uncertainties in the analytical tools and the conditions assumed in the stability calculation.

With the implementation of API Level-I stability study requirements, more and more compressors need a Level-II study including the labyrinth seals in the rotordynamics model. But sometimes it is not possible and not necessary to examine all of the labyrinth seals in a centrifugal compressor. For the compressor manufacturers and users, the analysis workload would become overwhelming and the engineering time would not be affordable to investigate every seal, especially in the quote stage of a compressor unit. From the industrial practice point of view, a quick and reliable detailed study is expected.

Evaluation of Aerodynamic Excitation
Using Wachel Number Formula

Although Wachel's cross-coupling formula has been widely used more than 20 years, the interpretation of the variables in the equation is quite liberal. Based on the experience from users with different backgrounds, some engineers use the entire horsepower and gas density ratio while others would like to use the local power

and density ratio at individual wheels. The minimum flow path could be defined at the impeller tip or the diffuser channel. The application in the new C-12 replacement compressor indicates that using the overall gas density ratio would result in a conservative estimation of the destabilizing aerodynamic excitation in a compressor. The conclusion is consistent with the authors' experience that the Wachel formula would give a more reasonable amount of aerodynamic excitation using the local absorbed power and gas density ratio.

The procedure using the entire gas density ratio to estimate cross-coupling at each impeller station is one of the considerable conservative approaches. In the nine-wheel 25MB4/5 case, the estimated aerodynamic cross-coupling was 48,600 lb/in, which was about 2.7 times the excitation predicted by the OEM and 70 percent more than the Wachel number yielded by API's formula. The total Wachel number even increased up to 74,500 for the 11-wheel 25MB5/6I due to a decline in the suction pressure. It would be of interest to note that both OEM and API's equations showed the aerodynamic excitation force to be lower for the 11-wheel rotor. Due to an overestimation of aerodynamic excitation baseline in the detailed stability analysis discussed above, the destabilizing contribution of the labyrinth seals appeared minor. Only the effect of the center labyrinth seal was noticeable, but secondary. Apparently the exaggerating effect in the compressor cross-coupling would be more severe for a low suction gas density application such as the case studied in this paper.

The Effect of Labyrinth Seals on Stability

Labyrinth seals have been demonstrated to be one of the major sources of destabilizing forces resulting in rotor instability vibration in centrifugal compressors. The forces of the labyrinth seal are compressor load-dependent and dependent on the seal geometric configuration as well. A well-known fact is that the rotordynamic force of the labyrinth seal is dominated by its cross-coupled stiffness coefficient K_{xy} because other force coefficients (K_{xx}, C_{xx}, C_{xy}) are relative small. The seal cross-coupling stiffness is strongly determined by the preswirl velocity of the process gas at the labyrinth seal entrance in the direction of rotation. The effective damping of a labyrinth seal can be defined in the equation below,

$$C_{ef} = C_{xx}\left(1 - \frac{K_{xy}}{\Omega C_{xx}}\right) \qquad (3)$$

where Ω is the precession frequency of the first forward mode of a flexible rotor in a stability study scope. From the above equation, it can be seen that any contribution to promote the cross-coupled stiffness would reduce the effective damping of the labyrinth seal. It should be pointed out that the positive direct damping C_{xx} of the labyrinth seal is small, but it is critical to keep the rotor in the stable region when the compressor is a borderline machine.

It has been recognized that most of the seal cross-coupling is generated by the center labyrinth seal in a back-to-back compressor or the balance piston seal in a straight-through compressor. The center seal and the balance piston seal normally take more axial space and have more labyrinth teeth than other impeller eye seals and shaft seals. In the centrifugal compressor, the center labyrinth seal or the balance piston seal sustains the largest differential pressure inherently. For a typical subsynchronous vibration associated with the first forward mode of the flexible rotor, the effect of the labyrinth seals on the stability is substantial because the magnitude of the destabilizing force of the labyrinth seal is proportional to the shaft precession amplitude at the seal location.

Prior to adopting gas damper seals such as honeycomb seals and pocket-damper seals, it is easier to enhance rotor stability by reducing the cross-coupling of the labyrinth seal than increasing its direct damping. Reducing the circumferential flow in the labyrinth seal is a direct and effective approach to reduce the seal cross-coupling. Using shunt-injection and installing swirl-brakes are two popular options for compressor engineers to break up the circumferential swirl flow in the labyrinth seal and to solve the load-dependent subsynchronous vibration problems. Apparently, a shunt-injection would slightly degrade the compressor efficiency due to an increase in the flow leakage through the labyrinth seal. Normally the impact of the shunt-injection on the efficiency is minimal if the shunt-injection is used with an abradable labyrinth seal. Sometimes using shunt-injection or swirl brakes cannot completely eliminate the subsynchronous vibration. Field observations and analytical predictions are not consistent. An obvious reason is the improper design and installation of the deswirl system. Another potential problem could be that it is not sufficient to only use one deswirl device, shunt or brake. A hybrid shunt/brake design is developed and applied in centrifugal compressors to deal with complicated rotor instability problems in the authors' company. A detailed description of this kind of stability-enhancing mechanism is beyond the discussion scope of this paper.

No doubt the effect of the major labyrinth seals and deswirl devices should be included in a detailed compressor rotordynamic stability analysis. Besides the diameter, length, and number of teeth of the seal, the seal clearance is another important parameter to be considered in the cross-coupling evaluation of the labyrinth seal. With a positive gas swirl velocity at the seal entrance, typically the larger the seal clearance, the larger the seal cross-coupling stiffness. When performing a detailed stability investigation considering the contribution from the labyrinth seal, the seal cross-coupling and the log-decrement should be examined at the extremes of the seal clearance.

A Practical Procedure to Evaluate Compressor Rotor Stability

Based on the authors' experience, the cross-coupling of the major labyrinth seal such as the center seal or the balance piston seal should be calculated directly in a full rotor stability analysis. The cross-coupling of other labyrinth seals may not be solved explicitly due to its time-consuming nature and the uncertainties in the actual conditions. Their contribution can be included in the analytical model by using partial Wachel numbers. The objective is to avoid the possible duplication of the effect of the labyrinth seal in the detailed stability analysis. The simplified full stability study should be able to save engineering time and not lose the reliability of the high-level stability investigation.

A rotor instability vibration is a kind of self-excited subsynchronous vibration. Normally the instability vibration most likely occurs when the rotor running frequency is higher than the first forward natural frequency of the rotor. The damping forces of the journal bearing and the damper seals (if any) applied on the rotor are proportional to the frequency of the potential instability vibration at a nonsynchronous frequency. Therefore, the authors used the nonsynchronous coefficients of the journal bearings in a proposed full stability study to predict the real dynamic behavior of the compressor rotor. Since the effect of the center seal or the piston labyrinth seal is split with other aerodynamic excitations in the proposed procedure, the contribution of the major labyrinth seals in the compressor can be investigated in depth. The following steps would give a simplified procedure to conduct a detailed stability study on the compressor rotors.

● Use the Wachel number formula in Equation (2) to calculate the aerodynamic excitation (Wachel number) at each impeller based on the local gas density ratio, local wheel horsepower absorbed, and gas molecular weight at the rated operating conditions. The flow path width is the minimum from the diffuser channel and impeller tip.

● Calculate the rotordynamic force coefficients of the major labyrinth seals at the extremes of the seal clearance with the rated gas properties around the seals and the given gas preswirl velocity ratios.

● Adjust the magnitudes of the predicted Wachel numbers down with a multiple factor around 0.65 ~ 0.75. The objective is to separate the excitation of the major labyrinth seal from other aerodynamic excitation for conducting a detailed stability analysis.

● Incorporate the adjusted local Wachel numbers at the corresponding impellers and the predicted cross-coupling of the major labyrinth seals into the model of the compressor rotor. If a damper seal is used at the center seal location or the balance piston location, the seal direct stiffness and damping items should be included in the analytical model as well.

● First calculate the average aerodynamic log-decrement at the nominal bearing clearance and the maximum continuous operating speed, then the minimum aerodynamic log-decrement at the extremes of the journal bearing clearance and the supply oil temperature. In the proposed study procedure here, nonsynchronous bearing coefficients should be used in the log-decrement calculation if the compressor is fitted with tilt-pad journal bearings. For typical compressor applications with an average gas density less than 4.0 lb/ft^3, the stability design of a centrifugal compressor would be considered as acceptable if the average log-decrement is not less than 0.05 and the minimum log-decrement is positive.

Following the stability analysis procedure above, both rotors 25MB4/5I and 25MB5/6I of the C-12 replacement compressor were studied and the results are presented below. Table 7 shows the adjusted local Wachel numbers for the C-12 compressor at the rated operating conditions. The cross-coupling of the center labyrinth seal was calculated separately considering with or without a deswirl remedy and predicted data were provided in Table 8. Including the aerodynamic excitations in Tables 7 and 8, the log-decrements were calculated using the nonsynchronous bearing coefficients at the MCOS of 12,000 rpm, and the final analysis results are shown in Table 9. The stability analysis results illustrated that the stability design of the C-12 replacement compressor would be accepted with a shunt/brake deswirl system, and the field operation also confirmed that the compressor vibration was well damped without the instability issue.

Table 7. Adjusted Wachel Numbers (65 Percent) in C-12 Compressor.

Impeller		9-Wheel Rotor (25MB4/5I)	11-Wheel Rotor (25MB5/6I)
Stage I	1st	602	341
	2nd	702	416
	3rd	814	487
	4th	1333	563
	5th		900
Stage II	1st	883	690
	2nd	987	787
	3rd	1058	892
	4th	1223	968
	5th	2474	1099
	6th		2021
Total		10074	9165

Table 8. Predicted Cross-Coupling of the Center Labyrinth Seal.

Seal Cross-coupling (lb/in)	9-Wheel Rotor (25MB4/5I)		11-Wheel Rotor (25MB5/6I)	
	Min. Clearance	Max. Clearance	Min. Clearance	Max. Clearance
Seal w/o Shunt/Brake	8980	10722	9194	10999
Seal with Shunt/Brake	777	-1468	884	-1462

Table 9. Predicted Average and Minimum Aerodynamic Log-Decrements.

Aero-Log Dec	9-Wheel Rotor (25MB4/5I)		11-Wheel Rotor (25MB5/6I)	
	Average log-dec	Minimum log-dec	Average log-dec	Minimum log-dec
Seal w/o Shunt/Brake	0.02	-0.06	0.03	-0.05
Seal with Shunt/Brake	0.11	0.03	0.12	0.04

CLOSURE

The above stability study procedure is practical in engineering time-consumption and straight forward for the compressor manufacturers and the users. It permits a thorough investigation of the rotor stability quickly without missing any main points associated with the load-dependent rotor subsynchronous vibration. This procedure may be considered as an equivalent full stability study per API requirements before sound validated analytical tools are well developed and become available to accurately estimate the aerodynamic excitation of the compressor impeller. A minimum log-decrement of 0.05 is always preferred in the stability analysis. Based on the authors' experience, a rotor having a minimum log-decrement of 0.05 with nonsynchronous bearing coefficients normally can reach the log-decrement value 0.10 required by API Level-II using synchronous bearing data. If the predicted minimum log-decrement becomes negative, an exceptional damper such as a honeycomb seal may be further considered and the seal rotordynamic force coefficients should be solved as a function of the rotor whirling frequency at the given operating conditions. It is noted that the study procedure is developed for current compressors fitted with dry gas face seals. If the centrifugal compressor has oil seals or fitted with squeeze film damper devices, more special considerations should be devoted in the stability analysis and the acceptance value could be different.

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