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PREDICTING, UNDERSTANDING AND AVOIDING THE EKOFISK ROTOR INSTABILITY FIFTY YEARS LATER

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ABSTRACT

A re-examination of the well-known Ekofisk re-injection centrifugal compressor is presented. Several opportunities make this an ideal candidate to assess how well (or not) current design and analytical methods have evolved to avoid shaft whip instability. The Ekofisk compressor represented a major step forward in the discharge pressure and subsequently, the challenges faced by the vendor and user concerning rotor lateral stability. Open literature contains a historical record of the efforts to address the instability. New information concerning the application and machine specific information is provided by the compressor vendor. To address the advances made in component and system design, the Ekofisk re-injection compressor is reconfigured as would be selected with today's methods and components. The rotordynamic behavior of this modern machine is compared against the original.

INTRODUCTION

The continual improvement process (CIP) plays a critical role in successful engineering companies. Used to avoid past mistakes, reduce manpower requirements and improve upon successful products, CIP is a time proven method. The basic tenet of many CIP's involve identifying opportunities, planning for change, implementation of the modified process or product, assessing the impact and actively implementing the change on a widespread basis if successful [1]. The modified process or product is then monitored continually with the CIP repeating.

In the turbomachinery industry, a similar continual improvement process can be witnessed as we examine the evolution of the centrifugal compressor product design and behavioral prediction methodology [2]. While not always a structured or conscious process, competition has forced turbomachinery processes to undergo a continuous improvement over time out of necessity. (Creating an environment and structured process within the organizations may have saved some pain and expense but such is the advantage of hindsight.)

Identification of opportunities to assess the impact of the evolution of the design and analytical methods permits the vendor and user alike the possibility to determine the effectiveness of these improvements. Literature offers the investigator many challenges spanning all aspects of turbomachinery design. From general application in industrial processes to the individual design of components comprising the machine assembly, much has been written about the difficulties that have faced the industry. The paper will focus the re-examination of the past on the prediction of rotordynamic behavior and the machine design modifications implemented to improve such behavior. Specifically, the lateral stability of centrifugal compressors, that drove the industry to many design and predictive improvements, will be investigated.

The Ekofisk re-injection centrifugal compressor is selected as a perfect example of the evolution of compressor design and analytic methodologies. Several opportunities were present that make this an ideal candidate. Namely:

- The compressor application represented a major step forward in the discharge pressure and subsequently, the challenges faced by the rotor lateral instability.
- The application is well known and continues to be referenced as a learning tool.
- The project and high pressure compressor has been the topic of numerous papers [3–11].
- A limited amount of information concerning the application difficulties is available in open literature [6, 10].
- Application and machine specific information was made available by the vendor.

To assess the improvement effectiveness of component and rotor design and the methods to predict to the dynamic behavior of each, the paper will investigate the following topics:

- How well current analytical tools (methods/theories) predict the instability behavior that was observed during the original installation of the compressor.
- How well current analytical tools (methods/theories) predict the stable behavior that was observed during the final configuration of the compressor.
- How important is the consideration of frequency dependence of the seal and bearing behavior in the stability prediction of the rotor when compared against field experience.
- Would the current industry stability analysis methodology (API 617 Level I and Level II) [12] have identified the original configuration of this machine as a risk and the final configuration as acceptable?
- Can current design features/technologies produce a better design than the modifications that were considered during the initial application?

An examination restricted only to the centrifugal compressor configuration as based on 1970's aerodynamic technology ignores the advances made in impeller and compressor design. The current rotor configuration process includes factors to optimize both aerodynamic performance and rotordynamic behavior which tend to work toward opposing design goals. To complete this cursory review of the impact of compressor design improvements, the Ekofisk reinjection compressor is reconfigured as would be selected with today's methods and capabilities. The rotordynamic behavior of the newly designed machine is compared against the original.

HISTORICAL REVIEW

In 1972, Phillips Group Norway selected a centrifugal compression system designed for gas reinjection in the Ekofisk reservoir [3]. The compressors were part of an enhanced recovery/energy conservation system for the Ekofisk platform. The associated natural gas from the oil field was to be injected into the reservoir minimizing the need for flaring and assisting in the recovery of oil. Due to governmental restrictions on flaring and the volume of gas recovered from the oil, reciprocating compressors were not economically viable or practical as previously used in the industry.

To maintain the oil field pressure at a minimum of 47.3 MPa (6850 psia), 48.3 MPa (7000 psia) pressure at the reservoir 3350 m (11,000 ft) down was required [8]. At these depths, a reinjection pressure of 63.4 MPa (9200 psia) was necessary. The injection gas was delivered to the platform from the onshore separation facility at 6.72 MPa (975 psia). The first stage compressor, an Elliott 25 MBH, boosted this to 24 MPa (3500 psia). The gas was then intercooled and sent to the second stage compressor, a two section Elliott 25 MBHH, to achieve the desired 63.4 MPa (9200 psia) discharge pressure.

At the time, the discharge pressure of the 25 MBHH was 14 MPa (2000 psi) higher than any centrifugal compressor had yet to achieve. This presented major design challenges and their solutions helped advance the technology's state-of-the-art. These included finite element stress analysis of the casing, advanced equation of state for gas properties, first application of Lund's transfer matrix code on an industrial machine and one of first log dec calculations made on a problematic compressor. A cross-section of the compressor is shown in Figure 1. The specifications for the compressor are shown in Table 1.

The compressor was driven by a GE Frame 5 gas turbine through a single helical, speed increasing gear (1.7346 gear ratio). The compressor train is shown on Figure 2. Two trains were put into service. Photos of the compressor ready for shipment and installed on the platform are shown in Figure 3.

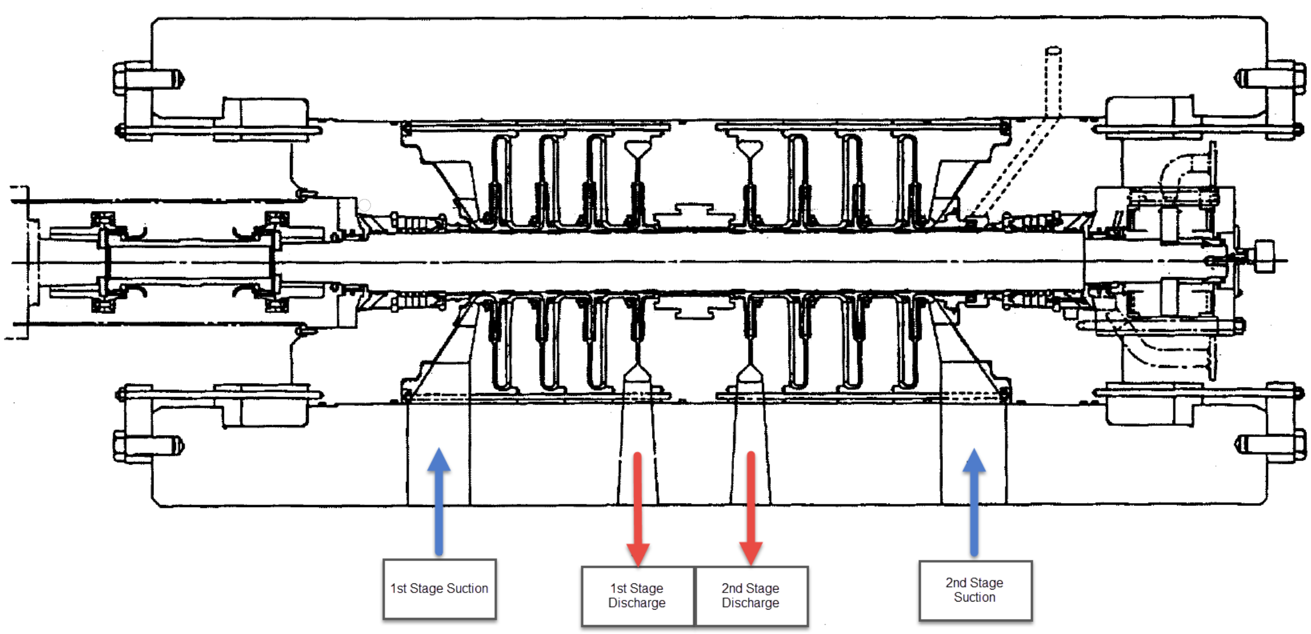


Figure 1: Injection compressor cross-section (adapted from [6])

The gas injection train was initially started under load in June 1974. Immediately upon reaching minimum governor, the compressor tripped on high vibration levels. Examination showed the majority of the vibration was occurring at frequency slightly higher than the stiff support undamped critical speed of 4380 rpm and much higher than the predicted first flexible

Table 1: 25 MBHH compressor specifications [6]

Weight Flow [kg/s (lbm/min)]	63.9 (8450)
Inlet Volume Flow [Im^3/s (ICFM)]	16.8 (594)
Molecular Weight	20.92
1st Stage Suction Pressure [MPa (psia)]	24.1 (3495)
1st Stage Suction Temperature [°C (°F)]	49 (120)
2nd Stage Discharge Pressure [MPa (psia)]	63.4 (9200)
Design speed (rpm)	8426

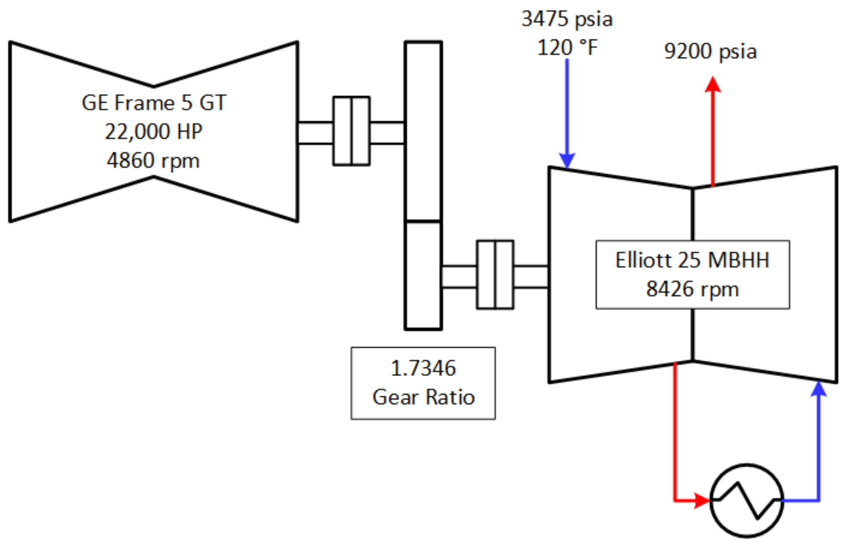


Figure 2: Ekofisk injection compressor train

bearing critical speed of 3462 rpm [4]. The vibration was found to be strongly associated with the suction pressure and rotor speed. Identified early on as a rotor instability, the investigation was extensive and included personnel from Phillips, the compressor vendor and two consultants. So complicated was the issue and solution that the final field design was arrived at in four phases (the third being dismissed in favor of the fourth) [6]. The original configuration was labeled Phase I.

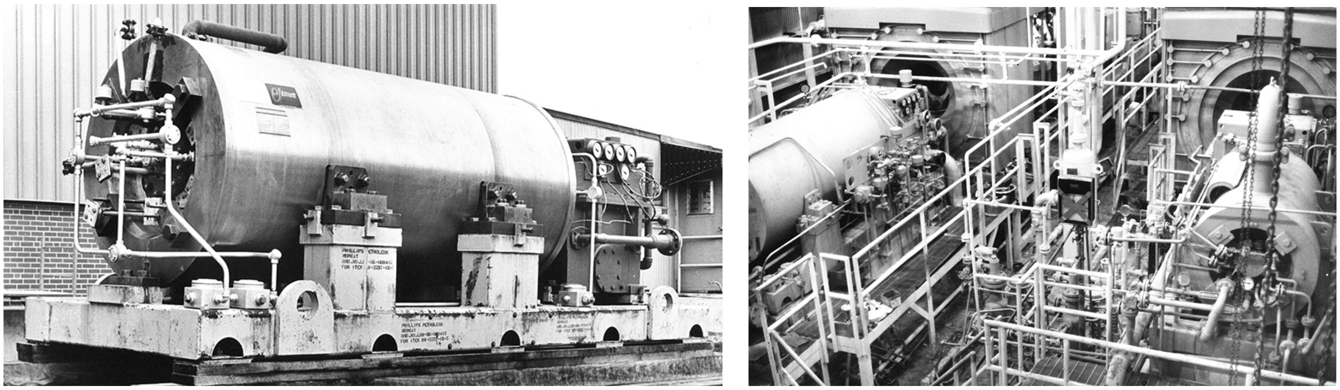
Data, drawings and vibration plots of a nearly 50 year old problem are hard to obtain. However, J. C. (Buddy) Wachel [10, 11] and C. H. Geary *et al.* [6, 7] provide an excellent review of the problem in their respective papers. A summary of their findings is shamelessly included here.

VIBRATION PROBLEM

Vibration problems were experienced during the initial attempts of loaded operation. Trip levels were exceeded at minimum operating speed at a reduced suction pressure of 10.3 MPa (1500 psia). The nature of the vibration was dominated by a subsynchronous component at 4400 cpm. Both rotordynamic instability and rotating stall were suspected causes of the subsynchronous vibration. Vibration magnitude would jump to 125–150 μm (5–6 mils) [6]. Figure 4 illustrates a typical waterfall plot during this time. Even though protection systems had danger levels set at 63.5 μm (2.5 mils), due to the suddenness of the onset of vibration, levels approaching the bearing clearance were experienced. This sudden increase in vibration without warning is a common trait of rotor instability.

The fact that the subsynchronous vibration frequency of 4400 cpm was higher than the rigid support critical speed of 4200 rpm was attributed to the oil seal dynamic behavior. Furthermore, the subsynchronous vibration amplitude was highly sensitive to the differential pressure across the oil seal, which was the compressor suction pressure. It was theorized that with oil seals locked in a fixed position, they could act like bearings and effectively reduce the bearing span raising the first critical speed. An oil film stiffness of 87,500 N/mm (500,000 lbf/in) was required to raise the natural frequency to that witnessed during operation. This reduced the predicted log dec from 0.3 to 0.08 and confirmed that the oil seals played a significant part in the stability of the compressor [10].

It is important to note here that many analytic tools currently available did not exist or were just starting development 45 years ago. These include analysis of liquid ring seals in eccentric operation, turbulent dynamic behavior of gas labyrinth



(a) Ready for shipment [4] (b) Compressor (left) on the platform

Figure 3: Phase I HP compressor photos

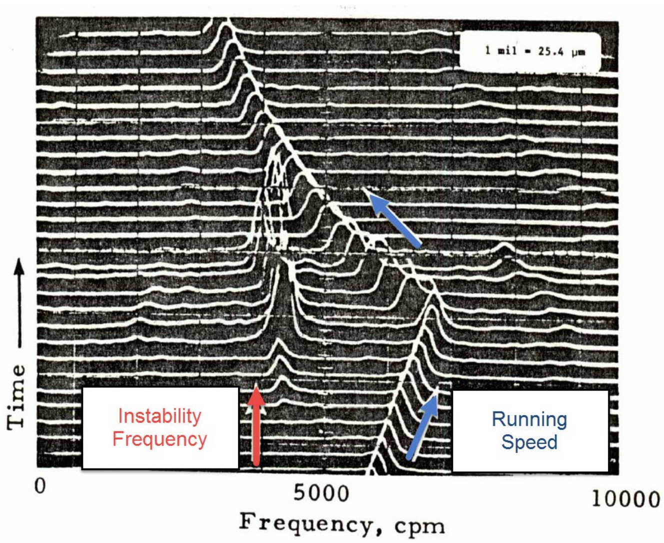


Figure 4: High subsynchronous vibrations occurring during operation with Phase I design (adapted from [11])

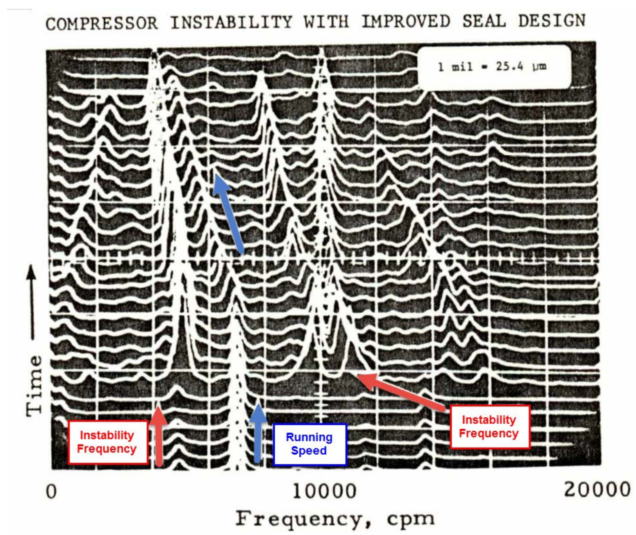


Figure 5: Subsynchronous and supersynchronous vibrations with improved oil seals (adapted from [10])

seals, advanced thermal/mechanical bearing behavior, and squeeze film damper optimization. Lacking these tools, designers would rely on past experience, prior problem solving practices and basic engineering knowledge. For the vast majority of applications, this proved adequate. However, for those machines that truly pushed the limits of technology, the unknowns could become significant.

To combat the destabilizing influences and lock-up potential of the seals, two circumferential grooves were added to the sealing surface, pressure balance was improved and the coefficient of friction at the axial face was reduced. The first modification was intended to reduce the dynamic coefficients produced by the seal, while the latter two modifications were aimed at minimizing the radial locking force of the seal and minimize eccentric operation. These modifications, while not common at the time, are used to correct stability problems in later years [13–15].

The complexity of the oil seal design, illustrated in Figure 6, leads one to understand the difficulties in predicting the locked position of the individual rings. The four rings during a typical start could potentially take an almost random distribution of eccentric positions. This helps explain the difficulties encountered with the Ekofisk compressor in stabilizing the influence of the oil seals. Instability at whirl frequencies less than two were encountered, as well as, the supersynchronous frequency shown in Figure 5. Figure 7 illustrates the numerous seal designs tested in an effort to stabilize the injection compressor. None were successful in completely eliminate the nonsynchronous vibrations. Wachel [10] provides an excellent review of the different attempts including their impact on the vibration levels.

These modifications did not eliminate the compressor instability. As noted in [10], the subsynchronous vibration remained and an apparent instability of a supersynchronous mode was introduced, Figure 5. Whether this was due to the proximity of the higher mode to twice the frequency of the first mode or the strength of destabilizing influence of the oil seals, it remains one of the few incidents of a supersynchronous instability in a compressor. Field tests showed that with slowing of the ramp rate during starting, normal operation could be obtained. The subsynchronous magnitude was found to be very sensitive to suction pressure at constant speed, Figure 8, and almost linear over a limited speed range at a constant suction pressure, Figure 9, as the limits of stability were tested in the field [10]. Subsynchronous frequencies also varied from 4400 to 5160 cpm during this testing confirming the varying influence of the oil seals.

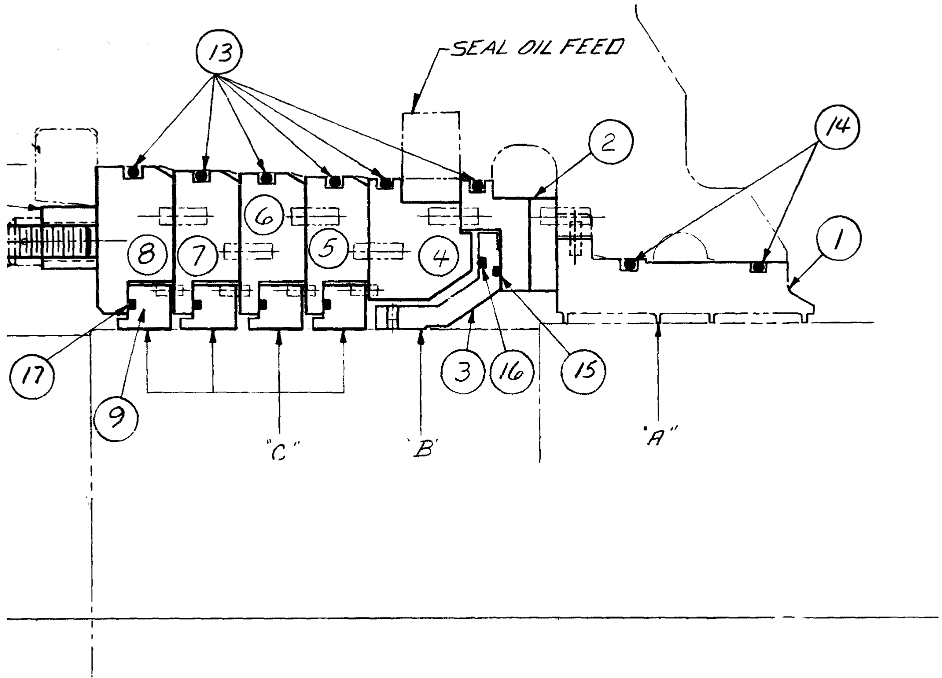


Figure 6: Phase I oil seal drawing

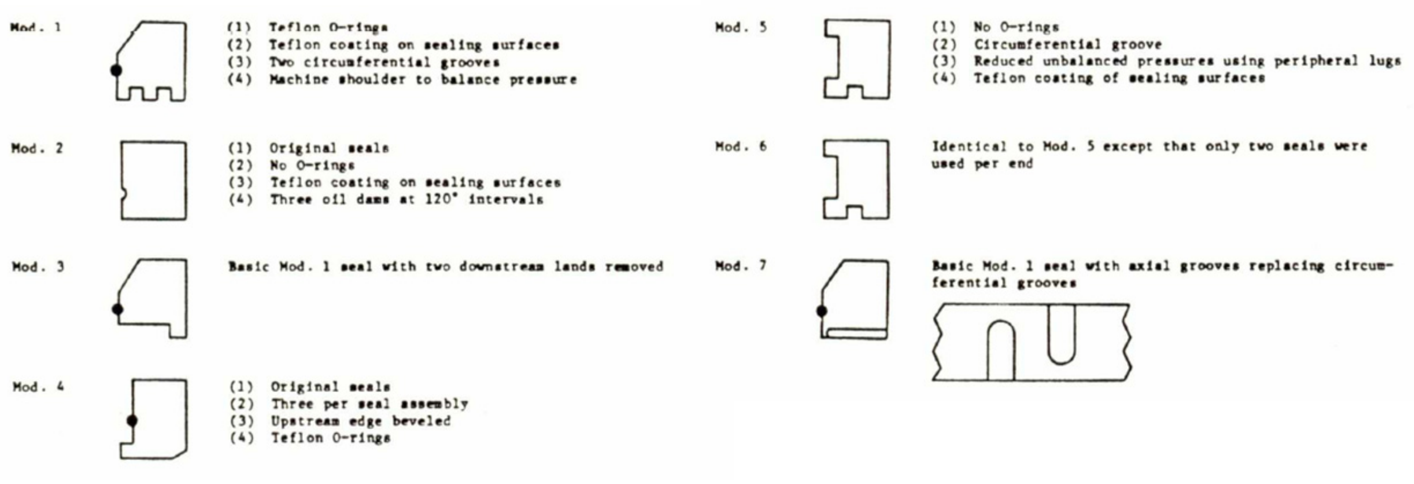


Figure 7: Seal modifications tested in an attempt to stabilize the Phase I rotor (adapted from [10])

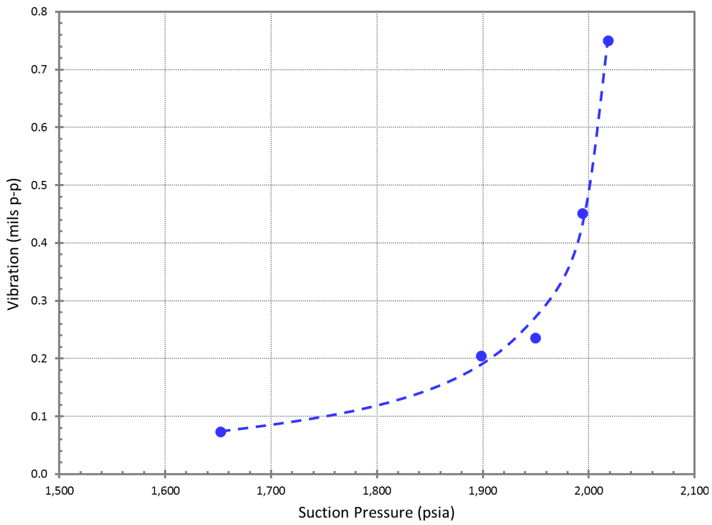


Figure 8: Subsynchronous vibration vs. suction pressure

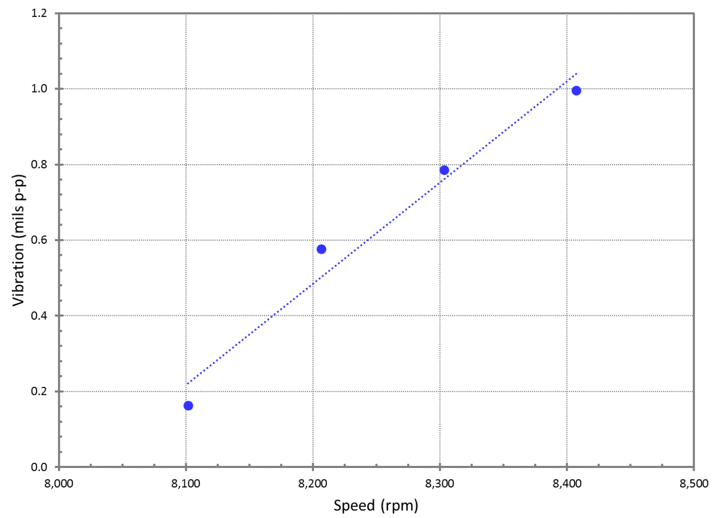


Figure 9: Subsynchronous vibration vs. speed

The Phase II of the Ekofisk design included the following modifications [7]:

- Redesigned oil seal (see Figure 10)
- Bearing span reduced by 140 mm (5.5") and loading was placed between pads
- Tilt pad bearings with near zero minimum preload
- Removal of balance rings on the impellers
- Increase in impeller back wall clearances
- Diffuser pinch was further blended for better transition of the flow
- Thrust collar attachment was changed to further increase rotor stiffness
- Diffuser widths were decreased to increase flow angle
- Diffuser vanes were introduced into the 4th and 8th stage
- Laby seal clearances were increased

Finally, the Phase II configuration of the injection compressor included squeeze film damper (SFD) supported radial bearings. While the initial attempt at a SFD supported radial bearing still experienced subsynchronous vibration exceeding trip levels, Figure 11, this was attributed to the inadequate analysis tools to properly design SFDs. Industrial machines incorporating damper supported radial bearings required tuning to optimize the SFD design.

The Phase II configuration was started up in stages with the first stage eliminating the use of the SFD bearings. This was used to determine the effects of the seal, bearing and rotor modifications. The second stage examined the impact of the pressure balanced seals by lowering the sour gas drain pressure. In both stages, the instability reappeared as attempts to achieve rated conditions failed. This led to the installation of the SFD supported radial bearing at the coupling end.

The damper bearings when properly "tuned" permitted both injection trains to be put into service reaching discharge pressures above 52 MPa (7500 psia) during operation. However, trips due to process related events could be accompanied by surging of the compressor. This led to elevated response levels due to the increased flexibility of the damper bearings. These bearings were found to be prone to fatigue. To further increase reliability, a Phase IV configuration of the compressor was pursued without damper bearings (Phase III was discarded in favor of Phase IV. The Phase III configuration was not run in the field and is not discussed in this paper.).

The Phase IV configuration consisted of the following changes [6, 7]:

- Same bearings and oil seals as Phase II
- Larger main shaft diameter, 170.5 to 229 mm (6.7125 to 9 in)

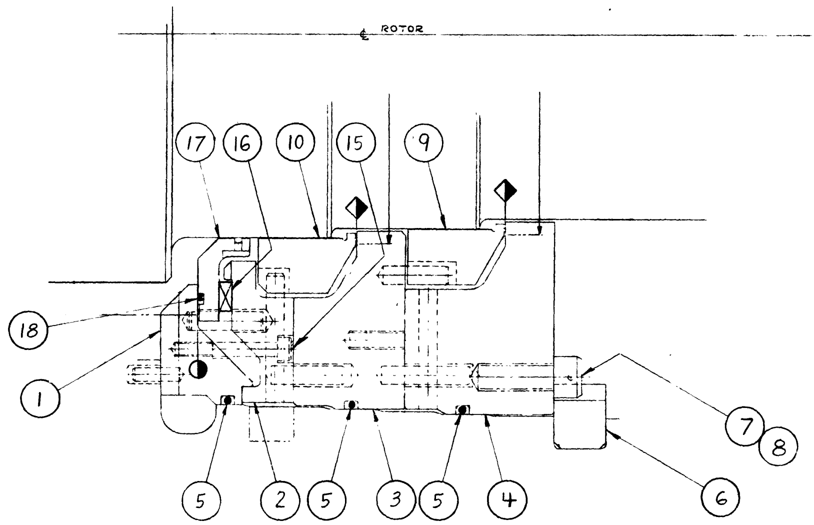


Figure 10: Phase II and Phase IV oil seal design

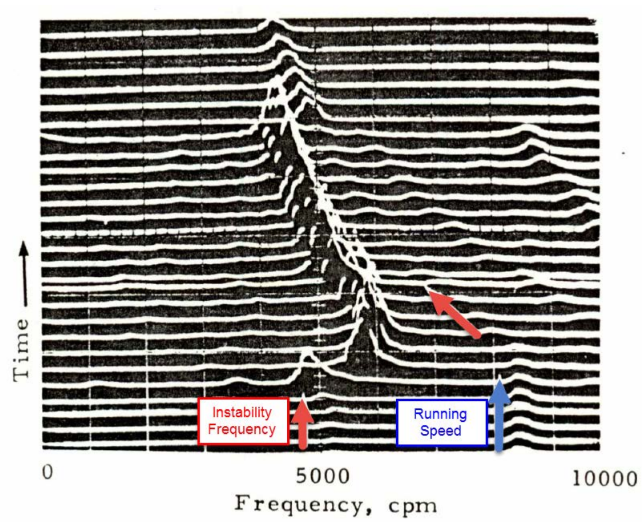


Figure 11: Vibration trip with damper bearings installed (adapted from [10])

- New aerodynamic stages with larger bore diameter and larger tip diameter
- Reduction in number of impeller blades from 19 to 17
- Increased blade tip angle
- All flow surfaces were polished
- Drilled leak offs from the 8th stage diffuser to the center seal at about $1/6^{th}$ the seal length
- Stepped impeller eye laby seals.

It is of particular interest to note the use of a shunted center seal in the Phase IV arrangement. The purpose of the shunt was to "...improve the (aerodynamic) stability of the 8th stage diffuser because the mass flow through the diffuser is then higher than the inlet mass flow at the stage inlet."[1] While frequently used now to minimize the destabilizing influence of the center seal, improvement of the rotordynamic stability was not the intent of the redesign.

[1] A. M. Badawi and F. J. Wiesner, "Aerodynamic Investigations and Test Results of the Phillips-Ekofisk 25MBHH Compressors," Technical Memorandum No. 284, Elliott Company, October 29, 1975.

In September 1975, the first Phase IV compressor was successfully proof tested out to 60.8 MPa (8815 psia). The compressor was then placed into operation with no signs of the subsynchronous vibration during its normal operation at discharge pressures ranging from 53 to 54 MPa (7700 to 7800 psia), and occassionally reaching 57 MPa (8300 psia). The second Phase IV compressor was delivered in March 1976 and placed into operation within 16 hours.

To examine the evolution of the centrifugal compressor design and behavioral prediction capabilities, the authors will present the following:

- A re-examination of the rotordynamic behavior of the original and final (Phase I and IV) configurations of the Ekofisk injection compressor.
- Reselection of the compressor given today's improved aerodynamic, mechanical and component knowledge. The improved design will take advantage of minimized dimensional requirements given a better stress knowledge, elimination of the oil seals in favor of dry gas seals and advances with internal damper seals. The configuration will also show the increased stability margin attributed to these advances.

PHASE I ROTORDYNAMICS

Modeling and Analysis Aspects

Following the typical methods and approaches commonly used today, individual dynamic models were developed for the following components: rotor assembly, tilting pad journal bearings, casing oil bushing seals (outer), center labyrinth seal, interstage and eye seals as well as equalizer laby seal. Each of these component models will be discussed, highlighting the key differences between this investigation's model and the state-of-the-art in 1973.

Figure 12 illustrates the Phase I rotor model developed for this current investigation. The rotor assembly has been discretized into 91 cylindrical beam elements, almost three times the number elements considered when the machine was originally designed. This increase in modeling fidelity/complexity is representative/indicative of the computational power advances in the last 40 years. However, this higher fidelity rotor model is not considered to be significantly more accurate than the past coarser models, but does offer some additional analytic flexibility.

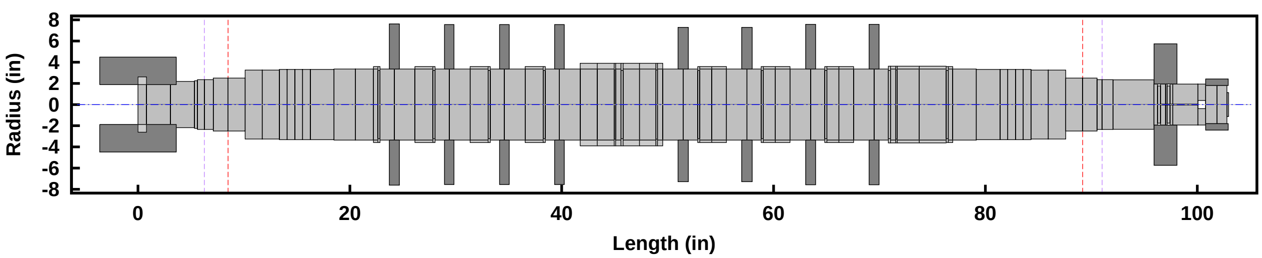


Figure 12: Rotor model of Phase I design

No stiffening contributions from the impellers and other sleeves have been considered in this investigation, although some OEMs incorporate this effect in their modeling to improve apparent modal matching [16]. A shrink fit's influence/importance is highly dependent on the mounted component's L/D, amount of shrink, and its modal location.

No support stiffness effects are included in this investigation. The impact of support flexibility on the stiffness and damping of radial bearings has been studied extensively [17, 18]. However, it has been common practice in the industry to ignore the support stiffness for centrifugal barrel compressor configurations. This is due to the fact that the support structure tends to be close to an order of magnitude higher in stiffness than the radial bearings. API 617 recommends, as a rule of thumb, to include support stiffness effects when the ratio is less than 3.5 [12].

When the Phase I machine was being designed, eleven different bearings were analyzed and a load on pivot with offset pivots bearing was recommended based on rotordynamics and bearing temperature rise [19]. In 1973, the state-of-the-art for modeling tilting pad bearings' dynamic performance was based on the method developed by Jørgen Lund [20]. This method provided the first theoretical basis for determining these bearings' dynamics [21]. However, the method excluded detailed thermal modeling, such as pad thermal deformations, which may be important for this application due to its relatively high temperature.

The compressor's tilting pad journal bearings were modeled for this investigation using a modern algorithm based on a two-dimensional, finite element formulation [22]. Physical effects considered in the modeling included conduction, variable viscosity, turbulence, and deformations of the pads and pivots. Presently, the most commonly-used models for industrial API 617 compressors are based on one-dimensional techniques, such as those described in [23, 24]. The two-dimensional, finite element algorithm utilized in this investigation, while beyond what is typically applied, has been shown to give reliable and accurate predictions [25, 26].

Table 2 presents the predicted performance for the inboard journal bearing at average operating conditions (clearance, preload and oil supply temperature) at MCS. Almost identical performance was predicted for the outboard, thrust end bearing.

Table 2: Predicted inboard, coupling end bearing performance for Phase I design
(average conditions, MCS)

ε (dim)	0.39
K_{xx} [N/mm (lbf/in)]	1.156e5 (6.6e5)
K_{yy}	1.73e5 (9.5e5)
C_{xx} [N-s/mm (lbf-s/in)]	103 (589)
C_{yy}	115 (654)
Min. Film Thickness [mm (in)]	0.0305 (0.0012)
Probe Temperature [°C (°F)]	100 (212)
Power Loss [kW (hp)]	5.75 (7.71)

Measured bearing temperatures during the machine's mechanical run test were compared with those predicted here. Good agreement (within 2.5°C (5°F)) was obtained, providing some qualitative confidence in the predicted dynamic coefficients' accuracy, since they are derived from the bearings' steady state performance.

The coefficients in Table 2 are synchronously reduced coefficients (SRC), which are a simplified set of 8 stiffness and damping coefficients (cross-coupled coefficients are determined, but not shown in the table) derived from a 5 pad bearing's set of 58 full coefficients (FC). This simplification was initiated by Lund in his original work to manage the computational limitations at the time, since including all the FCs for a bearing was not practical for stability analysis. While Lund recognized that SRCs were not appropriate for stability analyses [27] and the computational limitations for using FCs have been eliminated, many in industry continue to use SRCs for stability calculations, creating an ongoing debate.

Recent investigations [28, 29] have illustrated the dramatic stability prediction differences between these competing models and the better predictive accuracy obtained by using FCs that allow for any frequency dependencies. Furthermore, numerous experimental studies have shown bearings' radial dynamic stiffness (which equals the real part of the dynamic stiffness) acting on the shaft can possess a strong frequency dependency [30–32]. The authors have chosen to use FCs due to the nature of the rotordynamic codes employed. In the subsynchronous region of interest, the FC representation and the bulk flow, second order representation [30] produce comparable radial dynamic stiffness magnitude that exhibits a frequency dependence.

Fortunately, there will be little debate about which model's (SRC vs FC) predictions are more accurate for this Phase I design because this design incorporates 54% offset bearing pivots. Originally discovered by White and Chan [33], bearings with greater than 50% offset pivots have little difference between the two models' stability predictions.

With the dominant application of dry gas seals since the late 1990s, there has been a few advancements in the oil seal modeling techniques since Baheti and Kirk introduced a finite-element approach in 1994 [34]. One, in particular, is the analytical work by San Andrés and Delgado [35]. Their efforts were to include inertia effects from a bulk flow analysis of the oil seal ring operating with Reynolds number less than 2000. A key finding of the analysis illustrated that circumferential grooves, regardless of the depth, did not isolate the individual lands of the oil seal ring. The previous belief that circumferential grooving reduced cross-coupled stiffness and principle damping by the square of the number of lands was shown to be overly optimistic. The success in stabilizing compressors by grooving seal rings can be seen by the reduction in these terms and the decrease in load capacity at eccentric operation.

The oil seal bushings were individually modeled, using the same two-dimensional finite element formulation for the film as that used for the journal bearings, but with the added complexity of an axial pressure gradient and the need to iterate on the bushing's operating position based on additional forces (seal face friction and gravity) acting on the floating bushing. The pressure differential across each bushing was adjusted until the axial leakage flow was equal across all four bushings. This flow matching approach follows that introduced by Reedy and Kirk for multi-ring liquid seals [36].

At the time this compressor was designed, oil seal modeling was very crude with the typical approach representing the multiple bushings as a single ring. This single bushing would then be modeled as an unloaded (*i.e.*, fixed and centered), uncavitated, plain journal bearing. Various effects, such as thermal effects in the film, its axial pressure distribution as well as frictional and gravitational forces on the bushing, were ignored.

Table 3 summarizes the predicted operating eccentricity and dynamic characteristics for the four oil seal bushings at average clearance and oil supply conditions with the compressor operating at MCS and rated suction pressure. While each of the bushings is predicted to have a somewhat different operating equilibrium position, all of them are predicted to have relatively high total eccentricities near 90% of their clearance. This results in direct stiffness and damping levels that are larger

Table 3: Predicted performance of Phase I oil seal bushings
(average conditions, MCS)

Bushing	1 (most inner)	2	3	4
ε (dim)	0.87	0.87	0.87	0.87
K_{xx} [N/mm (lbf/in)]	1.14e6 (6.5e6)	8.4e5 (4.8e6)	7.7e5 (4.4e6)	3.9e6 (2.2e7)
K_{xy}	0.65e5 (0.37e6)	-1352 (-7729)	-0.17e6 (-0.99e6)	-2.6e5 (-1.5e6)
K_{yx}	-1.38e6 (-7.9e6)	-9.6e5 (-5.5e6)	-8.4e5 (-4.8e6)	-6.65e5 (-3.8e6)
K_{yy}	1.24e6 (7.1e6)	1.35e6 (7.7e6)	2.98e5 (1.7e6)	6.3e5 (3.6e6)
C_{xx} [N-s/mm (lbf-s/in)]	2891 (16507)	1872 (10692)	1093 (6242)	334 (1910)
C_{xy}	-401 (-2292)	-367 (-2094)	-488 (-2789)	-371 (-2120)
C_{yx}	-417 (-2382)	-378 (-2157)	-488 (-2786)	-370 (-2115)
C_{yy}	361 (2065)	275 (1573)	368 (2103)	568 (3244)

than those predicted for the journal bearings (see Table 2). As expected, cross-coupled stiffness values are also significant for each bushing.

Modeling of the labyrinth seals was accomplished using a bulk flow, single control volume method [37]. Based on the works of Iwatsubo [38, 39], the method assumes an ideal gas and ignores any thermal effects. More advanced techniques, incorporating more control volumes or utilizing computational fluid dynamics (CFD) and real gas properties, are currently available for modeling laby seals. However, the approach used in this investigation is the one of the most common currently used in industry.

In 1973, no such modeling techniques were commonly available to estimate a labyrinth seal's dynamic characteristics. As discussed earlier and by Pettinato *et al.* [40], this absence encouraged the development of empirical methods to estimate the "aerodynamic" destabilizing mechanisms within centrifugal compressors [40]. Such simple empirical correlation does not directly model the fluid dynamics of the seal, and thus, ignores the important specifics, such as the seal type, clearance, preswirl, etc.

For this investigation, the labyrinth seals were analyzed at their minimum clearances according to API 617 Level 2 requirements. Preswirl values were assumed based on the authors' experience and no preswirl control devices, *e.g.* shunt bypass or swirl brakes, were incorporated in the Phase I design. Gas boundary conditions for each labyrinth were based on the compressor's rated condition, extrapolated to MCS.

Any frequency dependency within the labyrinth seal coefficients was assumed to be relatively weak and within the uncertainty of the overall predictions. Although Kirk suggested [41] and others have since verified the presence of frequency dependency in some labyrinth seals [42–44], incorporating such frequency dependency is not currently a common practice.

Only cross-coupled stiffness and direct damping predictions have been utilized for each labyrinth seal model. Instead of presenting all these coefficients for the various labyrinths, Figure 13 compares their effective cross-coupled stiffness as defined by:

$$Q_{eff} = K_{xy} - \omega \cdot C_{xx} \tag{1}$$

where ω is the vibration frequency of interest. In Figure 13, this frequency has been set to be 4700 cpm, the observed subsynchronous vibration frequency.

As shown in Figure 13, all of the interstage shaft labyrinths are expected to be slightly stabilizing, a common result. The center seal, equalizing seal, and individual eye seals should destabilize the forward mode.

To compute the first forward rotor mode's stability, the various component models' dynamics were assembled into a first order, system matrix using a finite element formulation. First applied to rotor systems by Ruhl and Booker [45], the system matrix A is given by the following:

$$A = \begin{bmatrix} [0] & [I] \\ -[M]^{-1}[K] & -[M]^{-1}[C] \end{bmatrix}$$

where $[M]$, $[C]$ and $[K]$ are the assembled mass, damping and stiffness matrices, respectively. Solving for the damped eigenvalues and eigenvectors of the A matrix provides characteristics for all the modes in the system. Of main concern here, the log decrement δ for each mode is computed from the real part and the imaginary part of each complex eigenvalue λ according to:

$$\delta = \frac{-2\pi\,\mathrm{Re}(\lambda)}{|\mathrm{Im}\,(\lambda)|} = \frac{-2\pi\,\mathrm{Re}(\lambda)}{\omega_d}$$

When the Phase I machine was designed, log decrement values could not be calculated because a damped eigenvalue solution for rotor systems was not commercially available. In other words, the real part of each eigenvalue, that determines

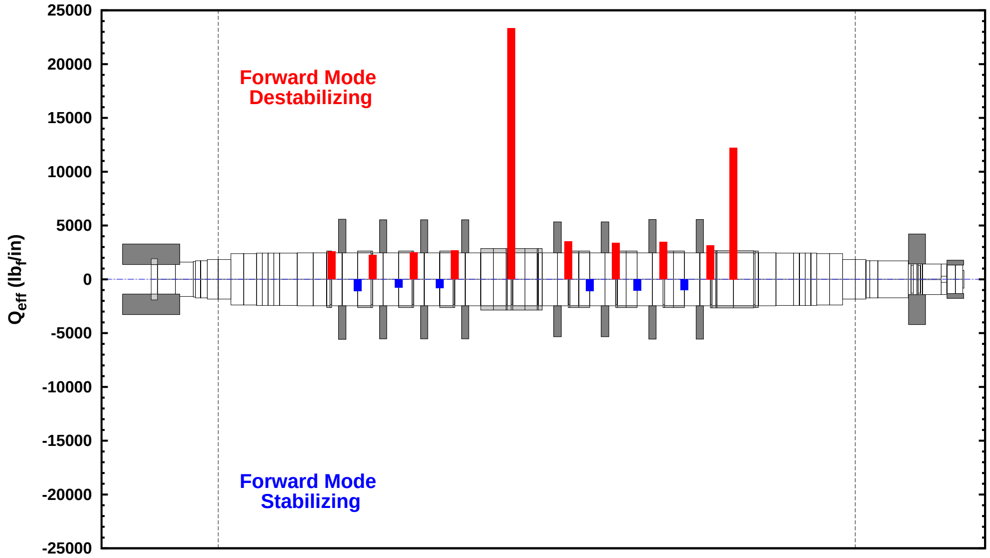


Figure 13: Predicted Q_{eff} values from Phase I labyrinth seals

whether a mode is stable or not, was unknown at any particular speed or operating condition. Only the stability threshold speed, *i.e.*, the speed at which the compressor would go unstable, and the whirl frequency could be estimated [46]. During design, the machine's stability threshold speed was predicted to be over 15,000 rpm, well above operating speed, meaning any rotor instability was unlikely [19]. However, this threshold speed prediction was based on component models with the limitations and unknowns outlined above, particularly, with respect to the floating ring seals and labyrinth seals.

Just nine months before the Phase I compressor was started in the field, Lund presented his landmark work (published in [47], one month before startup) that solved the damped eigenvalue problem. This work introduced the concepts of damped natural frequency, damping exponent (= $\mathrm{Re}(\lambda)$), and log decrement to the rotating machinery industry. Because of the computational limitations at the time, calculating the eigenvalues directly from the system matrix A was not practical. So, Lund's technique utilized the transfer matrix method and searched in the complex frequency domain for the damped eigenvalues.

To illustrate application of the method, the paper investigated the stability characteristics of an 8 stage industrial compressor, "typical of machinery for chemical processing plants." The compressor was actually the Ekofisk high pressure machine. Although the paper did not analyze the machine's stability with the actual bearings and oil seals for the Phase I design, it reached an insightful premonition, *i.e.*, the machine will go unstable before reaching its design speed.

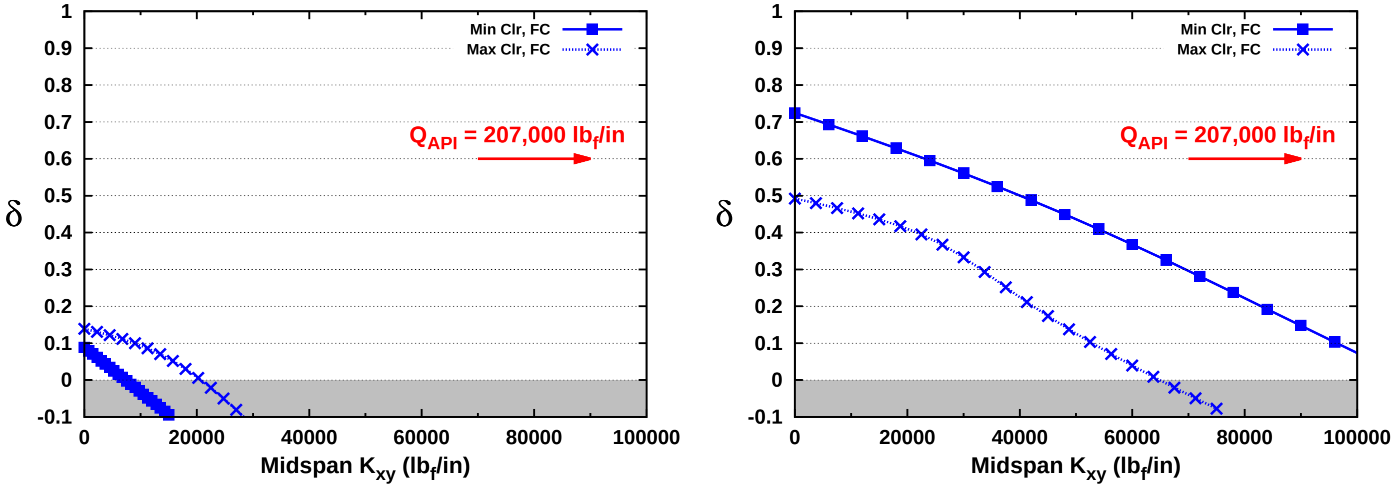
API 617 Level 1 Stability Screening Analysis

Following current API requirements, the compressor was examined to assess whether or not a more detailed, Level 2 analysis would be prudent. Since we know that's the case for this problematic machine, performing such a screening analysis helps to validate (or not) the screening analysis itself and its criteria.

Excluding the influence of the oil seals, Figure 14(a) presents the machine's stability sensitivity curve at the two extreme bearing conditions. The base stability without any destabilizing cross-coupling at the midspan is very low, with an approximately 0.1 log decrement (1.67% damping) at minimum bearing clearance conditions. The aerodynamic cross-coupled stiffness estimated using the API approach Q_{API} is approximately 36,250 N/mm (207,000 lbf/in), more than an order of magnitude higher than the predicted stability threshold (approximately 1,225 N/mm (7,000 lbf/in) at minimum clearance bearing conditions).

Because of their often strong influence, the oil seals must be included to assess the compressor's stability characteristics. Figure 14(b) shows that the oil seals are predicted to significantly improve the machine's stability robustness, increasing the base stability as well as the stability threshold. Such behavior is a result of the very high direct stiffness predicted by the oil seal model, which effectively reduces the machine's bearing span. However, even with this improved robustness from the oil seals, the estimated cross-coupling present Q_{API} (36,250 N/mm (207,000 lbf/in)) remains well above the predicted stability threshold, indicating a more detailed analysis is warranted.

While the above Level 1 screening analysis results are sufficient to justify a more detailed, Level 2 examination, it is informative to examine this machine relative to another Level 1 screening criteria that is based on industrial experience with compressor rotor instability problems. Derived from one originally conceived by Sood [48], Figure 15 illustrates two regions



(a) Rotor and bearings only

(b) Including oil seals

Figure 14: Phase I stability sensitivity curves

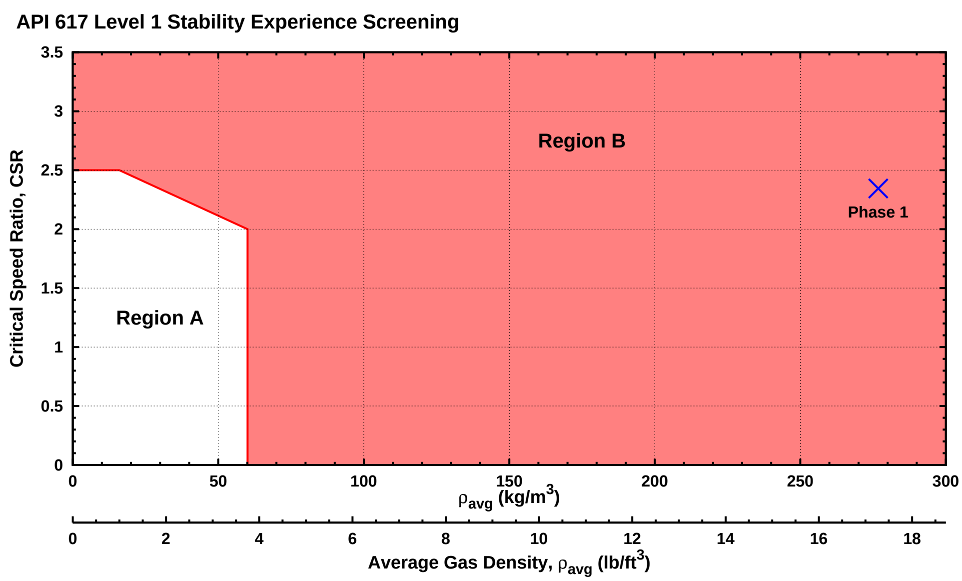


Figure 15: API 617 Level 1 experience screening plot for Phase I design

of risk that are dependent on the rotor critical speed ratio CSR, an indicator of rotor flexibility, and the average gas density within the compressor, an indicator of the amount of destabilizing excitations. Machine designs falling within Region A are expected to have a low risk of rotor instability problems based on past experience. Ekofisk's Phase I design is shown to be in Region B where there is a high risk of instability, having both a relatively flexible rotor and a high gas density. API 684 [49] provides a good discussion on the development of this experience plot.

To summarize, the Phase I design is shown to exceed several of the API 617 Level 1 criteria:

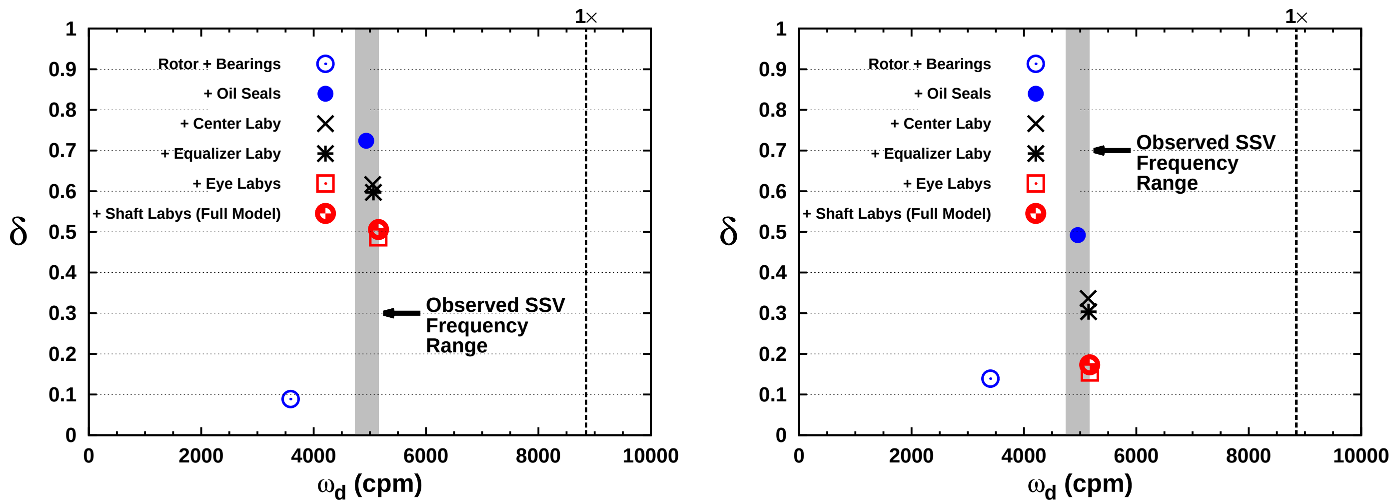
- The stability margin Q_{API}/Q_0 is less than two,
- The anticipated log decrement is less than +0.1, actually it's highly negative,
- The machine also falls within Region B of the experience screening plot.

So, the Level 1 screening process has performed as desired by identifying this machine design as one that deserves greater scrutiny through a detailed Level 2 analysis.

API 617 Level 2 Detailed Stability Analysis

Unlike the previous Level 1 analysis, which estimated the destabilizing effects of the compressor's labyrinth seals using API's empirical modified Wachel equation, direct damping and cross-coupled stiffness predicted for each seal was included in this Level 2 analysis using the bulk flow approach described earlier.

The stability spectrums in Figure 16 illustrate different components' influence on the frequency and log decrement of the first forward mode as they are added to the overall system rotordynamic model. With their large direct stiffness, addition of the oil seals is predicted to shift the mode significantly higher in frequency, very close to the subsynchronous vibration's frequency range observed on the platform. The results suggest that, without the influence of the oil seals, the 1F mode would be much lower in frequency.



(a) Minimum clearance

(b) Maximum clearance

Figure 16: Level 2 stability spectra for first forward mode of Phase I design

As expected, addition of the center labyrinth causes a reduction in stability. However, the eye seals cause a similar level of reduction. The equalizer labyrinth and interstage shaft labyrinths are shown to have a relatively benign influence on the 1F mode's stability.

These Level 2 results indicate that the machine's stability looks very acceptable with a minimum log decrement of 0.17 for the full model, above the API 617 minimum of 0.1. Unfortunately, this is quite a different picture than suggested by the Level 1 analysis and from what was experienced in the field. So, why does the machine's model fall so short of accurately predicting the shaft whip problem? Are all the component models really that poor?

Whenever such deviations exist between predictions and reality, it is good practice to scrutinize where the uncertainties lie within the model. All the component models have some level of uncertainty associated with their dynamics. For this machine, we have the most confidence in the rotor assembly model, and the least in the labyrinth and oil seal models. Therefore, it's likely these annular seal models could be the main source of the poor correlations.

Indications that the laby seal dynamic analyses were not reliable is recorded in an internal manufacturer's report. It stated: "....It was also evident that the machine with tight seal clearance was more unstable and when the seals are rubbed due to rotor vibration, the machine appeared to become more stable. This can be explained as increased seal leakage increases the flow in all parts of the stages. One may also look at the action of these tight seals in this high density gas environment as a source of forces that may contribute to the shaft instability (acts like bearing). For these reasons, it was decided to increase the clearances of all labyrinth seals. . . ."[2]

Many factors could be impacting the fidelity of the labyrinth seal models. Candidates include the assumed preswirl into each seal, tighter than expected operating clearances, gas properties, and limitations of the bulk flow modeling approach. Fortunately, the labyrinths' geometries are fairly conventional and not drastically different than what the tool was developed for. All these factors mean that the labyrinth models may be underpredicting the seals' destabilizing influence. However, it seems unlikely that they could be radically underpredicting sufficient to drive the machine well below the stability threshold. One other factor not examined is the destabilizing effects of the impeller-shroud interaction and secondary flow within the

[2] A. M. Badawi and F. J. Wiesner, "Aerodynamic Investigations and Test Results of the Phillips-Ekofisk 25MBHH Compressors," Technical Memorandum No. 284, Elliott Company, October 29, 1975.

impeller. While considered by the authors to be a second order effect, especially in comparison to the dynamic behavior of the oil seals, there is recent evidence that in some cases this may not be true [50].

Without the benefits of the oil seals' additional stiffness and damping at high eccentricities, it's pretty clear that the machine would be predicted to be unstable with log decrements significantly negative. In fact, the oil seals appear to have a more dominant influence on the machine's stability robustness than do the tilting pad journal bearings. However, when one considers the complexity of their design coupled with how they are modeled, it becomes questionable as to how accurately their behavior can be predicted.

Table 3 indicates that the individual oil seal bushings are predicted to operate at eccentricities approaching 90% of their clearance. This relatively high eccentricity is a direct result of the bushings' poor pressure balance, which creates large frictional contact forces. The modeling approach determines when the frictional and gravitational (minor in this application) forces match the oil film's hydrodynamic forces and establishes the bushing's eccentricity at this force equilibrium.

However, this predicted eccentricity should really be considered somewhat of a *maximum expected* eccentricity. Each bushing may settle or lock at a lower eccentricity whenever the friction forces are greater than the hydrodynamic ones. In other words, each bushing in this compressor could lock at a much smaller eccentricity, basically anywhere below the 90% predicted eccentricity. Therefore, each bushing may generate a huge range of possible dynamics, from those at centered, zero eccentricity to those at near 90% eccentricity.

The potential for friction forces to lock each bushing at a much lower eccentricity is very high in this compressor design. Several design features of the oil seals create this potential. First, as mentioned earlier, the individual bushings are not well pressure balanced. Second, each bushing has an O-ring that rides against the stationary housing's lapped face. Finally, there is very little axial clearance for each bushing to float radially, especially after one considers the possibility of O-ring swell and deformation of the bushings and adjacent housings.

These additional, but likely, restrictive forces created by the O-rings and deformations are not considered in the modeling process and would be highly difficult to do so with many non-linearities and initial condition dependence. Therefore, the reality is that we probably have little confidence in predicting the actual dynamics of an individual bushing, and much less for eight of them (four in each oil seal), all independent. The best we can do, practically, is try to bound their dynamics.

If we assume that the oil seal bushings behave as desired (operating centered), not as predicted at high eccentricity, Table 4 indicates the drastic reduction in stability that occurs. Centered bushings are typically considered the ideal situation because the cross-coupled stiffness and axial leakage are minimized here. However, centered oil seal bushings generate no direct stiffness causing the nodal points of the rotor mode to fall close to the journal bearings. If any of the bushings are significantly eccentric, large direct stiffness and direct damping are created that shift the 1F mode's node points closer to the oil seals, making the bearings' damping more effective. This behavior of the oil seal rings, *i.e.*, destabilizing at low eccentricities and stabilizing at highly eccentric operation, was confirmed in test data by Childs *et al.* [51, 52].

Several variables were noted as key to the instability, these were ramp rate, suction pressure at startup and during operation, amount of recycle, and rotor speed. Lowering suction pressure prior to start will have a significant impact on the locking position of the oil rings and, thus on the dynamic behavior of the rotor. Additionally, the suction pressure can be impacted by recycle amount and a slower ramp rate can permit better centering of the oil rings. These field observations tend to indicate the primary influence of the oil seals on the rotor stability.

Table 4: The influence of oil seal position on stability (Phase I)

Model	Oil Seal ε (dim)	log dec δ (dim)	
		Min Clr	Max Clr
Full Model w/ Predicted Oil Seal Performance	0.87	0.51	0.17
Full Model w/ Centered Oil Seals	0.00	-0.67	-0.82

Many publications tend to suggest that the main cause of this machine's rotor instability problems were the internal labyrinth seals and their high cross-coupling levels at the gas densities involved. However, the results presented above suggest that the oil seals were just as much, if not more, destabilizing as the labyrinth seals. Greater attention may have been placed on the labyrinth seals' effects because their impact was relatively poorly understood and unpredictable at the time.

The authors have shown the oil rings possess significant dynamic coefficients. Couple this with the uncertainty in the prediction of the locked position of the rings and the influence on stability, it becomes quite clear that the challenge to produce a stable compressor the first time was extremely daunting.

PHASE IV ROTORDYNAMICS

Design and Modeling Aspects

As discussed earlier, a variety of design changes were implemented to help eliminate the rotor instability problems. Illustrated by its model in Figure 17 and photo in Figure 18, the rotor assembly was changed significantly, *e.g.*, reducing the bearing span by approximately 125 mm (5 in) and increasing the shaft diameter under the impellers by more than 50 mm (2 in). These changes decreased the rotor L/D from 12 to 8.4, resulting in large shifts in the critical speed characteristics as illustrated in the critical speed map comparison of Figure 19. The rotor's first rigid bearing critical increases by approximately 2000 cpm, yielding a lower critical speed ratio (see Figure 20).

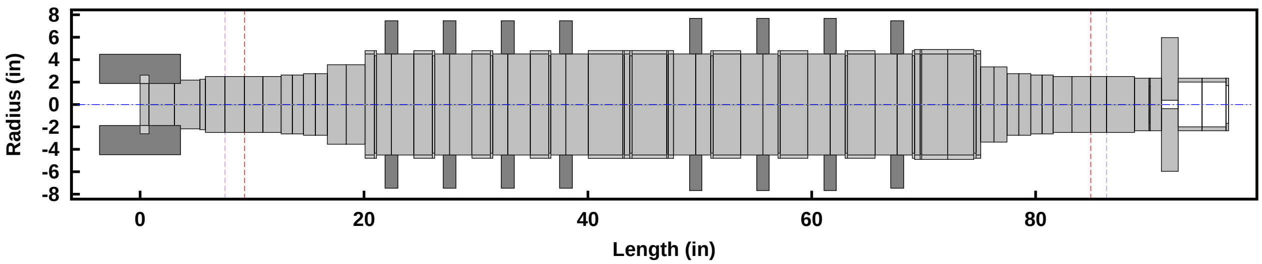


Figure 17: Rotor model of Phase IV design

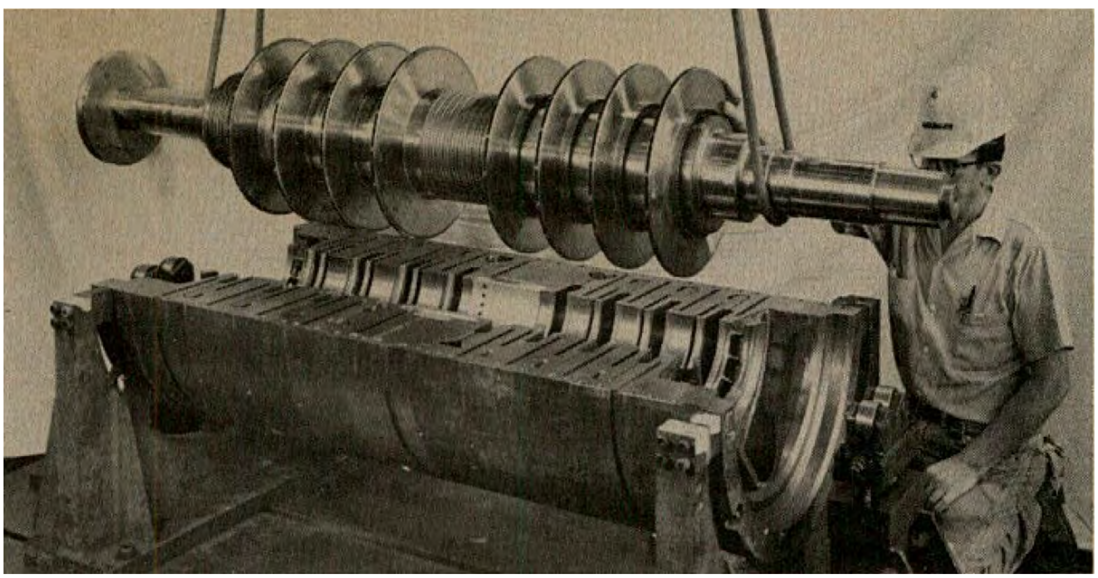


Figure 18: Photo of Phase IV rotor assembly [4]

The journal bearings' design was also changed in several ways. Their original on-pad loading configuration was switched to between pads and the pads' offset was reduced to centered, *i.e.*, 50%. Furthermore, an assembled clearance increase from 20–30 μm/mm (0.8–1.2 mils/in) to 25–35 μm/mm (1.0–1.4 mils/in) was accompanied by a preload reduction, from 0.29–0.53 to 0.07–0.38. This may be one of the first instances where the stability benefits of increased bearing clearance and reduced preload were recognized. A few years later, the physical reasoning behind their stability benefit was explained; both design trends help increase a tilting pad journal bearing's effective damping [53].

Table 5 compares the Phase IV bearing's design performance characteristics with those from the original, Phase I design. While there are no radical shifts in performance, one should notice the overall reduction in stiffness and damping. Combined with the rotor critical speed increases in Figure 19, the bearing stiffness reduction should greatly improve the bearing's effective damping.

The outer, atmospheric side, oil seals also underwent major modifications. The four bushing design was replaced with two bushings, incorporating greatly improved pressure balancing and the elimination of the O-rings at the contact faces. Each bushing's Babbitted sealing land was machined with three circumferential grooves and a 25 μm (1 mil) increase in diametral clearance.

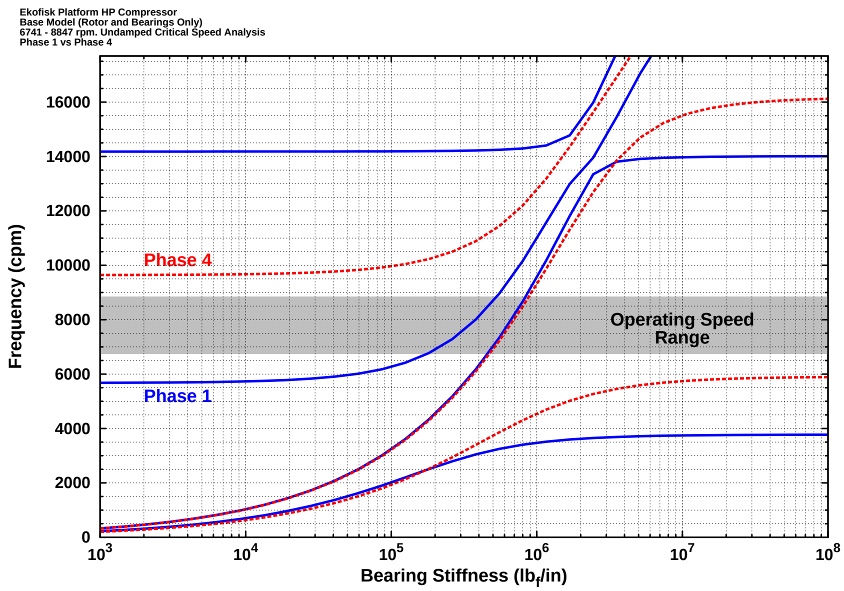


Figure 19: Critical speed maps of Phase I and Phase IV rotor design

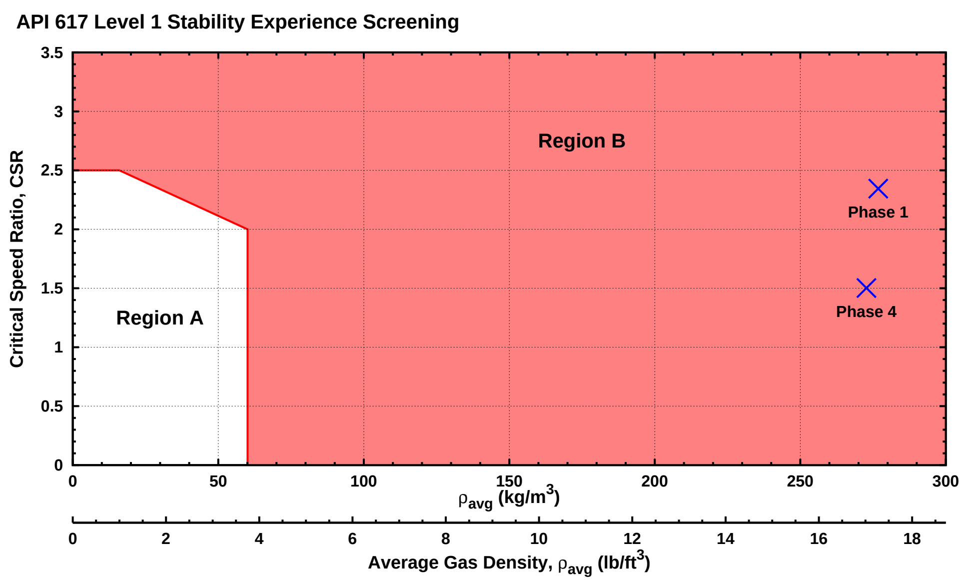


Figure 20: API 617 Level 1 experience screening plot for Phase I and IV designs

Table 6 summarizes the predicted performance of these two oil seal bushings at average clearance and oil supply conditions with the compressor operating at MCS and rated suction pressure. Compared to the Phase I oil seal predictions in Table 3, the Phase IV bushings effectively provide no direct stiffness and damping. This is due to their somewhat lower eccentricity and the presence of the circumferential grooves. The oil seal bushings continue to generate significant levels of cross-coupled stiffness, but are several orders of magnitude smaller than their Phase I counterparts. With better pressure balance and no complications from O-rings, there is much more confidence in the predicted behavior of this oil seal design.

Like the journal bearings and oil seals, the center seal's clearance ratio was also increased, specifically, from 91 μm/mm (3.6 mils/in) to 137 μm/mm (5.4 mils/in). The number of teeth, their geometry and location (on-rotor) were maintained in the center seal. The eye seal labyrinths' clearance ratio was also increased.

As mentioned earlier, in order to help address suspected diffuser stall issues, some discharge gas off the last stage diffuser was internally routed back into the center seal's entrance, just downstream of the first few teeth. Since the importance of entry swirl on labyrinth seals' rotordynamic behavior had yet to be identified by Benckert and Wachter [54], this was a very serendipitous modification that is, in essence, the shunt bypass that is commonly used today. This experience was over a decade prior to the first published accounts of shunt bypass designs being implemented in centrifugal compressors [41, 55].

Table 5: Predicted inboard, coupling end bearing performance for Phase I and IV designs
(average conditions, MCS)

	Phase I	Phase IV
ε (dim)	0.39	0.489
K_{xx} [N/mm (lbf/in)]	1.16e5 (6.6e5)	8.58e4 (4.9e5)
K_{yy}	1.66e5 (9.5e5)	1.33e5 (7.6e5)
C_{xx} [N/mm (lbf-s/in)]	103 (589)	78 (446)
C_{yy}	115 (654)	81 (462)
Min. Film Thickness [mm (in)]	0.0305 (0.0012)	0.0356 (0.0014)
Probe Temperature [°C (°F)]	100 (212)	97 (206)
Power Loss [kW (hp)]	5.75 (7.71)	5.64 (7.56)

Table 6: Predicted performance of Phase IV oil seal bushings
(average conditions, MCS)

Bushing	1	2
ε (dim)	0.62	0.66
K_{xx} [N/mm (lbf/in)]	-6.48 (-37)	-1.4 (-8)
K_{xy}	5,453 (31,138)	3,789 (21,636)
K_{yx}	-8,482 (-48,432)	-6,765 (-38,631)
K_{yy}	-34.7 (-198)	-27.5 (-157)
C_{xx} [N-s/mm (lbf-s/in)]	20.5 (117)	16.1 (92)
C_{xy}	0	0
C_{yx}	0	0
C_{yy}	11.8 (67)	8.2 (47)

The manufacturer did not realize the rotordynamic importance of the shunt in 1975, but they kept empirically applying the concept to fix a series of subsequent rotordynamic instability problems until finally adopting it as a standard feature in its back-to-back lineup in the early to mid 1980s.

Figure 21 summarizes the dramatic reduction in effective cross-coupled stiffness between the Phase I and Phase IV center seal designs. In fact, due the presence of the shunt bypass, the center seal's Q_{eff} has switched signs such that it should help stabilize the forward mode. For this investigation, the Phase IV equalizing and shaft seals were not modeled due to their expected trivial influence.

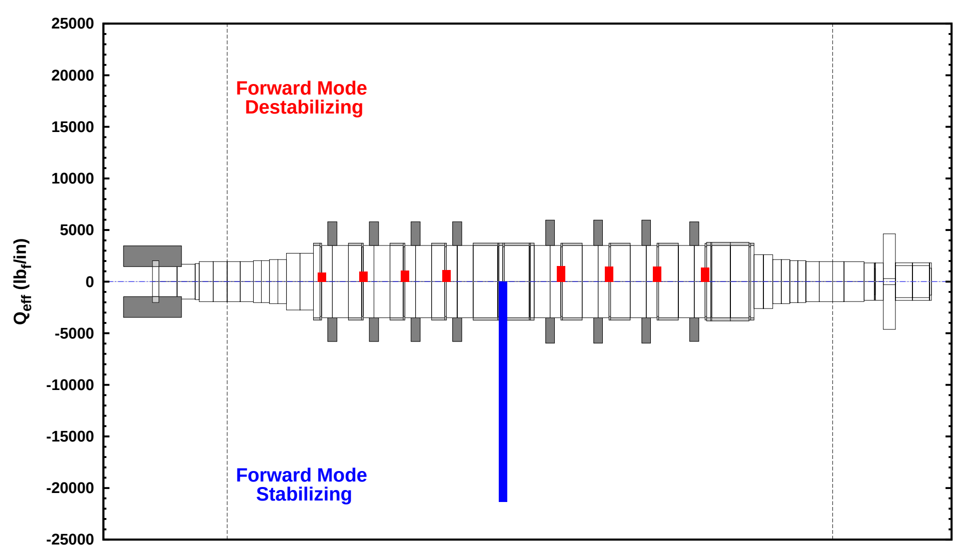


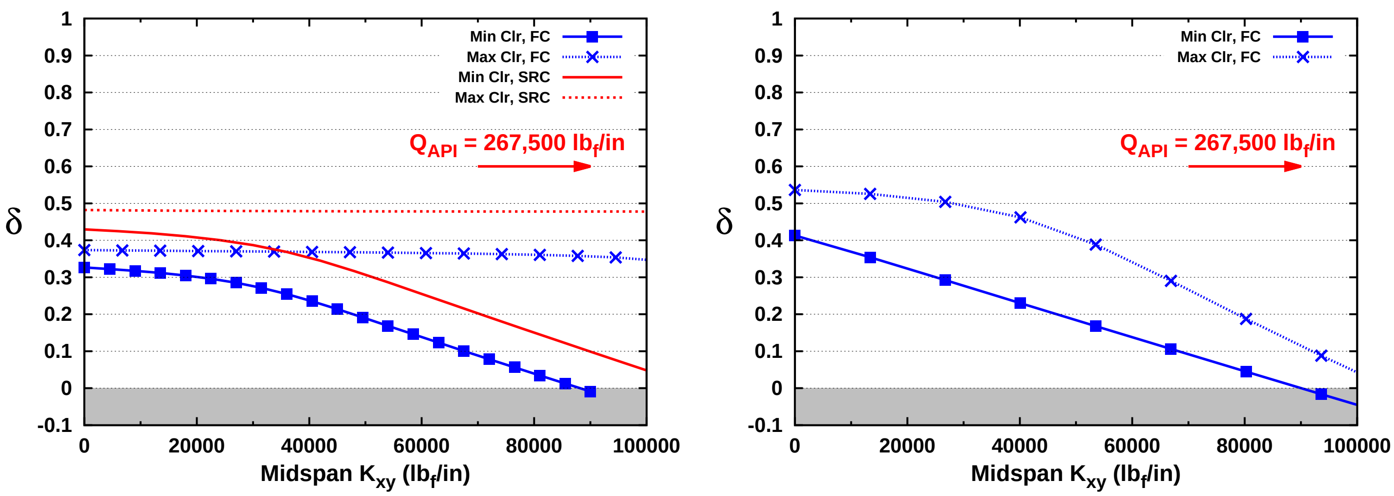
Figure 21: Predicted Q_{eff} values from Phase IV labyrinth seals

API 617 Level 1 Stability Screening Analysis

The significant benefits obtained by Phase IV's rotor and bearing modifications can be seen by comparing the stability sensitivity curve results in Figure 22(a) with those from Phase I in Figure 14(a). Relative to the Phase I rotor-bearing design, the base δ of the machine has increased by a factor of three and the stability threshold has increased by more than an order of magnitude.

Figure 22(a) provides the stability sensitivity curves based on the tilting pad journal bearings' full coefficients (FC) and synchronously reduced coefficients (SRC). Unlike with the Phase I design, there is a noticeable difference in the two competing models' predictions. This is an expected result given the conversion from offset pads with relatively high preloads to centrally pivoted pads with low preloads.

With the oil seals included, Figure 22(b) shows that the machine's base stability increases from the oil seals' additional damping. However, the machine's sensitivity characteristics are also altered. In particularly, the robustness to cross-coupled stiffness is decreased under maximum clearance conditions.



(a) Rotor and bearings only

(b) Including oil seals

Figure 22: Phase IV stability sensitivity curves

API 617 Level 2 Detailed Stability Analysis

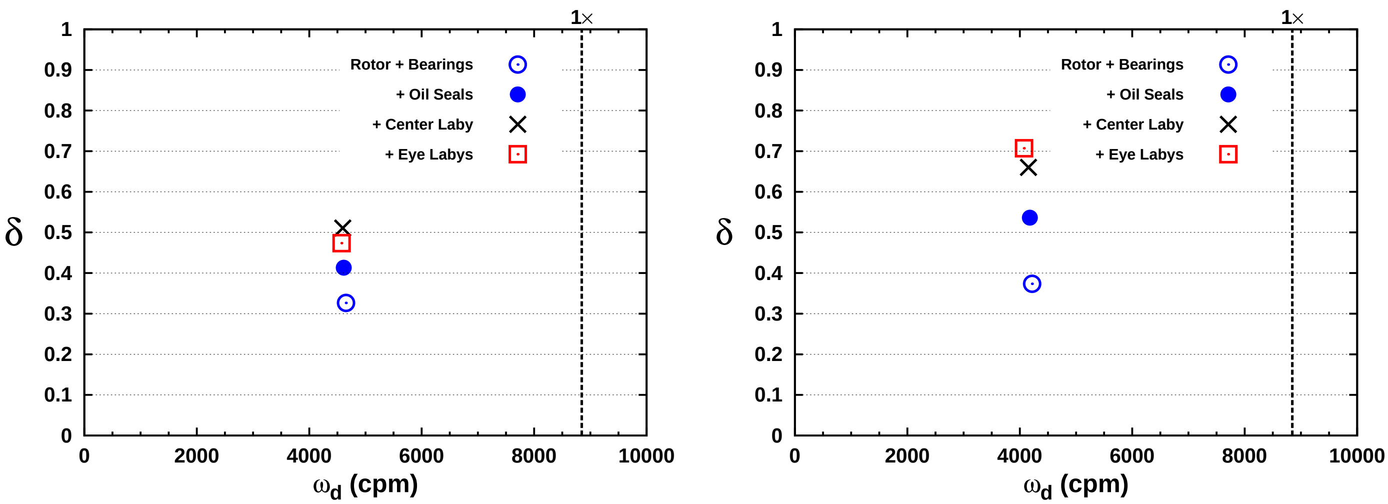
As done with the Phase I design, the first forward mode's stability was examined as different components were added to the overall system rotordynamic model. The stability spectrums in Figure 23 indicate several interesting findings. First, unlike the Phase I design, the Phase IV oil seals do not significantly shift the location of the mode. The mode's ω_d is very similar to the Phase I design's, but its location is not sensitive to the oil seals' or the other seals' as-predicted dynamics. The stiffer rotor design and modified bearings are dictating its location and behavior. With a stiffer shaft and softer bearings, we have increased the bearings' effective damping and increased control of the instability.

Also unlike the Phase I machine, the center labyrinth with its shunt actually increases the log decrement, stabilizing the mode, in agreement with what was expected from its negative Q_{eff}. Eye seals effects are predicted to have a relatively small effect on the mode's stability.

The most significant finding from the detailed stability analysis is that the machine is predicted to be highly stable with log decrement values well above the API minimum of +0.1. Given the prior experience with the Phase I design, it is prudent to once again consider the machine's stability with the oil seals centered. Table 7 indicates that there is predicted to be little change in Phase IV machine's stability, if the seals are operating in their ideal position.

Table 7: The influence of oil seal position on stability (Phase IV)

Model	Oil Seal ε (dim)	δ	
		Min Clr	Max Clr
Full Model w/ Predicted Oil Seal Performance	~0.64	0.47	0.71
Full Model w/ Centered Oil Seals	0.00	0.48	0.76



(a) Minimum clearance

(b) Maximum clearance

Figure 23: Level 2 stability spectra for first forward mode of Phase IV design

MODERN ROTORDYNAMICS

Design and Modeling Aspects

A true reselection of the Ekofisk application would include both the low and high pressure compressors optimizing the overall cost and efficiency to compress the gas from 6.72 MPa (975 psia) to 63.4 MPa (9200 psia) within the two compressor bodies. However, due to the intent of this paper, only the high pressure compressor was reselected. An approximate figure of the reselected compressor is shown in Figure 24. A number of similarities can be immediately spotted. However, there are some important differences between the 1975 Phase IV compressor and the reselection, and some of these are as follows:

- Frame size reduction
- Operating speed increase
- Number of impellers reduced from 8 to 6
- New aerodynamic stages
- Abradable impeller seals
- Vaneless diffusers
- Tandem dry gas seals (not pictured)
- Hole pattern center seal
- Journal bearing upgrades
- Reduced weight coupling

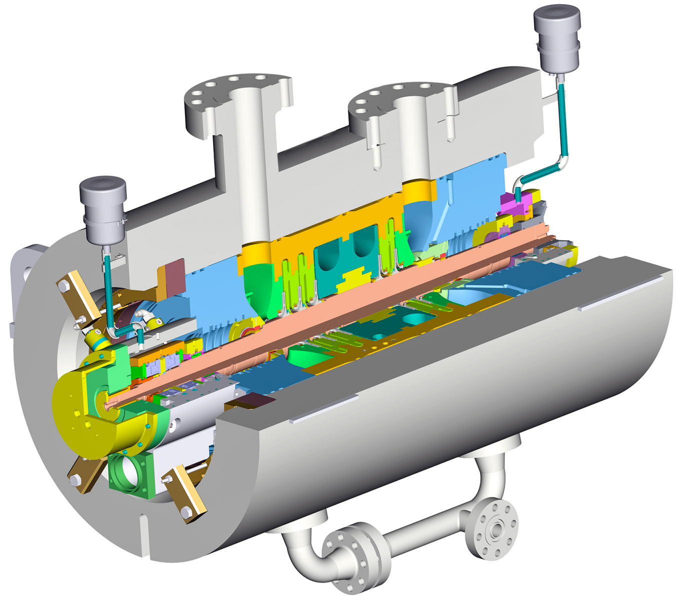


Figure 24: Approximate cross section of reselected, modern compressor design

The reselection reduced the frame size to a 20MBHHH having an increased shaft speed. This compressor is a barrel design, also known as a vertically split design with an extreme pressure rating. Only six (6) stages are required, which is a reduction of two stages. The stages are in a back-to-back arrangement. There are three (3) stages on either side of the division wall with external inter-cooling occurring between the 3rd stage discharge and 4th stage inlet. The back-to-back arrangement assists with thrust balancing. As with the original design, an equalizing seal is used as a second thrust balance, in addition to the center seal arrangement.

The aerodynamic stages are selected from a standard design family having well over ten years of application. Aerodynamic improvements enable a 9 point gain in efficiency over the 1975 Phase IV compressor. Some of this gain is due to application of abradable seals, which were originally developed and applied to compressors in the 1980s. Vaneless diffusers are used throughout the machine.

Figure 25 and Table 8 compare the rotor designs of previous phases with this modern, reselection. The rotor has flexibility characteristics similar to that of the Phase I design with a rotor L/D ratio near 11 and CSR of approximately 2.6. Both of these values are an increase as compared to the 1975 Phase IV compressor.

This return to a more flexible rotor design is influenced by the higher operating speeds as well as the use of dry gas seals. While the dry gas seals have eliminated the highly influential oil seals, they also demand significant axial space, affecting the bearing span. Fortunately, with the exception of their rotating assemblies' added mass, their specific rotordynamic coefficients are negligible.

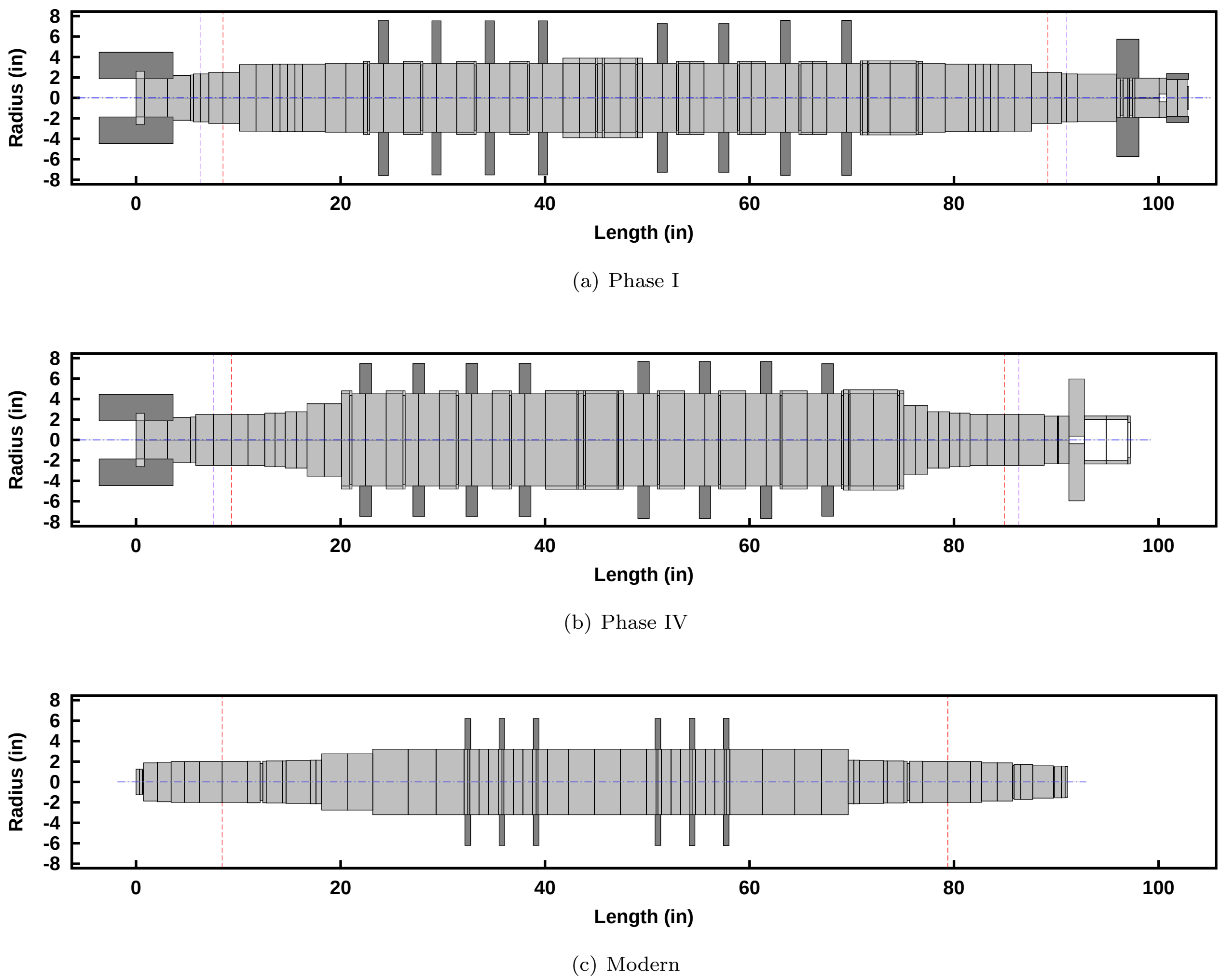


Figure 25: Rotor design comparison

Table 8: Rotor characteristics

	Phase I	Phase IV	Modern
Bearing Span (L)	2049 mm (80.7 in)	1920 mm (75.6 in)	1803 mm (71.0 in)
Main Diameter (D)	171 mm (6.7 in)	229 mm (9.0 in)	163 mm (6.4 in)
Shaft L/D	12.0	8.4	11.1
OSR (rpm)	6741–8847	6741–8847	9636–12647
CSR (dim)	2.35	1.5	2.59

Summarized in Table 9, the journal bearings have gone through a series of upgrades, both in terms of the design and in terms of manufacture. The selected bearings are a self-aligning, spherical pivot design. Optimization is performed on the bearing length-to-diameter (L/D), assembled clearance, preload, and pivot offset. The most notable differences in the theoretical design are the use of a 0.5 L/D, higher assembled clearance, and load-between-pad orientation to achieve both rotordynamic and temperature objectives. The general rule of thumb concept of the early 1970s of stability enhancement through high bearing asymmetry (0.3 L/D and load-on-pad) are rejected in favor of a more analytical approach. These changes in approach are corroborated by Gunter and Weaver in their recent examination of the Kaybob instability [56].

Table 9: Tilting pad journal bearing design characteristics

	Phase I	Phase IV	Modern
Loading Direction	LOP	LBP	LBP
L/D	0.284	0.296	0.5
$c_b^{\varnothing}/D$ ($^\circ/_{\circ\circ}$)	0.8–1.2	1.0–1.4	1.7–2.2
m_p (dim)	0.29–0.53	0.07–0.38	0.19–0.40
Pivot Offset (%)	54	50	50

As shown in Figure 26, the compressor continues to remain in a region of gas density and critical speed ratio where the OEM requires additional log dec and countermeasures as related to rotordynamic stability. In this case, a hybrid hole pattern center seal is applied, having an upstream swirl brake and downstream abradable labyrinth seal. Unlike the 1975 Phase IV compressor, there is no shunt in this reselection.

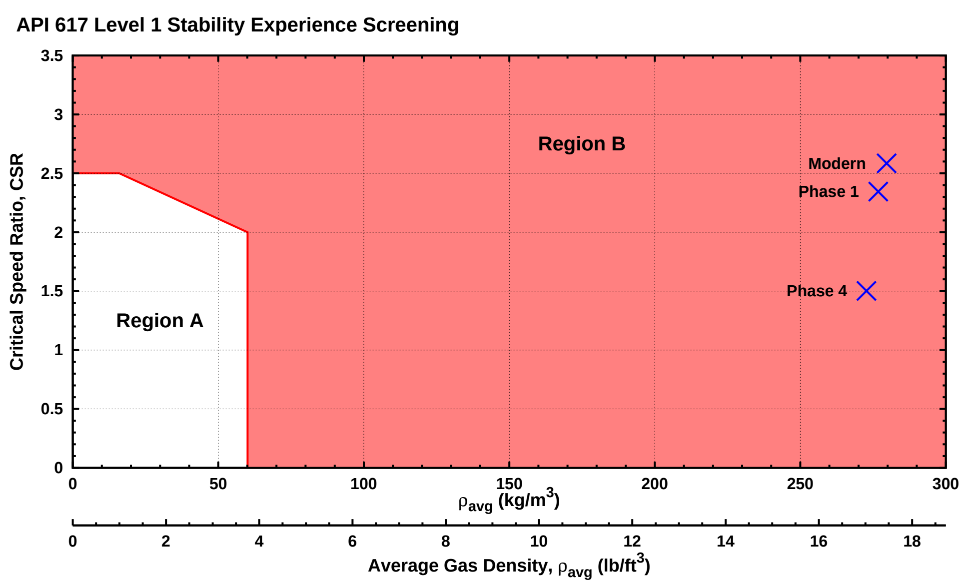
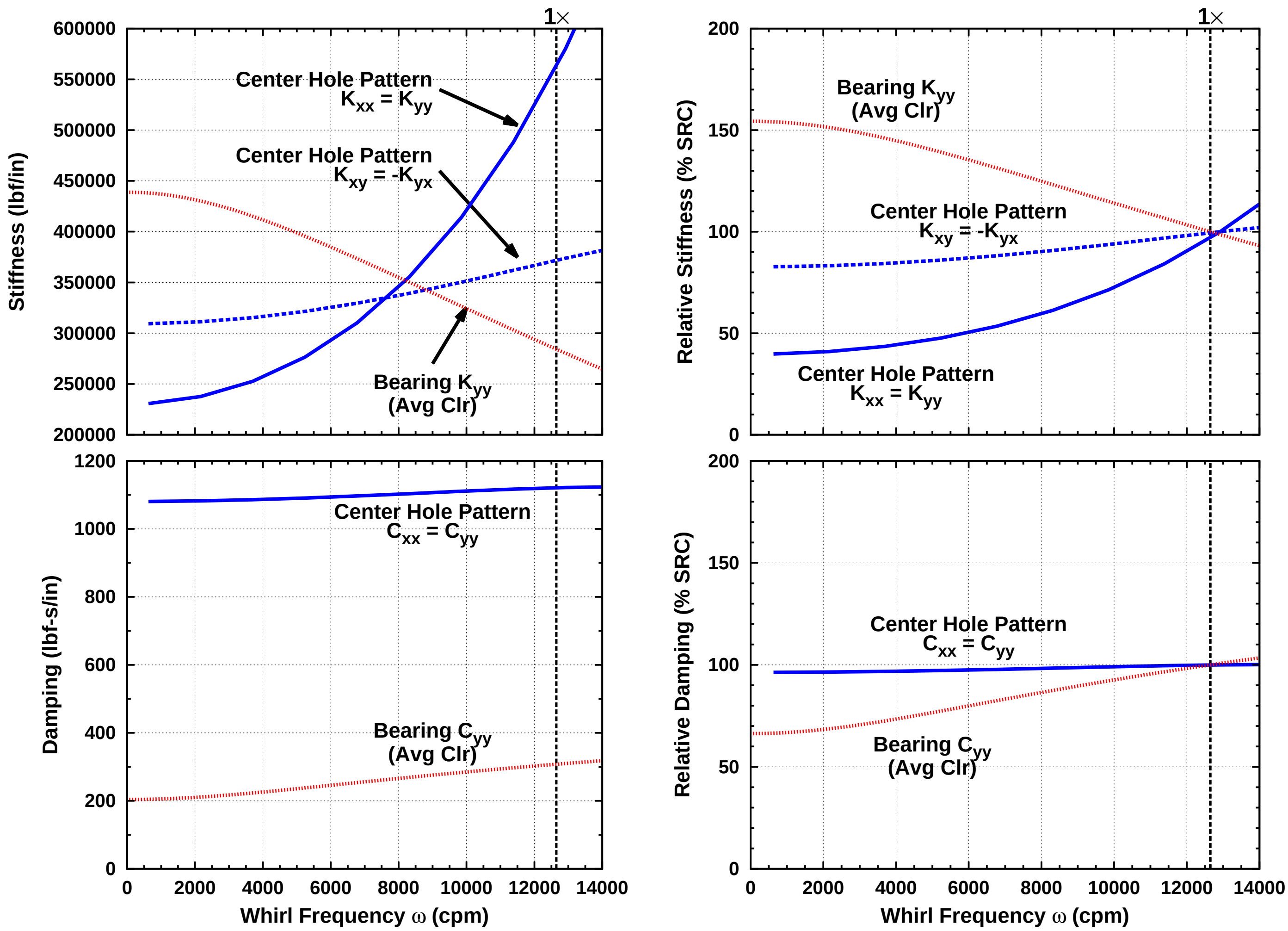


Figure 26: API 617 Level 1 experience screening plot for all designs

Dynamic performance of the center seal's hole pattern section was calculated based on the approach originally developed by Kleynhans and Childs [57]. This approach employs a two control volume, bulk flow model that is the current industry standard for modeling honeycomb and hole pattern seals. The tool employed for this investigation has shown to agree well with Kleynhans' and Childs' predictions, and with experiments by Childs and Wade [58].

Figure 27 presents the hole pattern seal's predicted dynamic characteristics and, for comparison, some of those from the tilting pad bearings. One immediately notices the hole pattern dynamics' dependency on whirl frequency, a well-known behavior for these seals. In Figure 27(a), the hole pattern seal design is shown to be stiffer than the bearing at synchronous 1× whirl frequency, but becomes softer than the bearing at subsynchronous whirl frequencies below 8000 cpm. Even though the hole pattern seal generates relatively large amounts of cross-coupled stiffness, the direct damping that it provides is almost four to five times that of the bearing. In terms of Q_{eff}, the hole pattern generates -24,340 N/mm (-139,000 lbf/in) at 4000 cpm, meaning it should be highly effective in stabilizing the first forward mode. Mounting a "gas film damper" at the midspan of the rotor was discussed during the original vibration problems and ruled out as impractical [59]. Fifty years later, such a damper is now a reality through the use of a hole pattern seal.



(a) Absolute

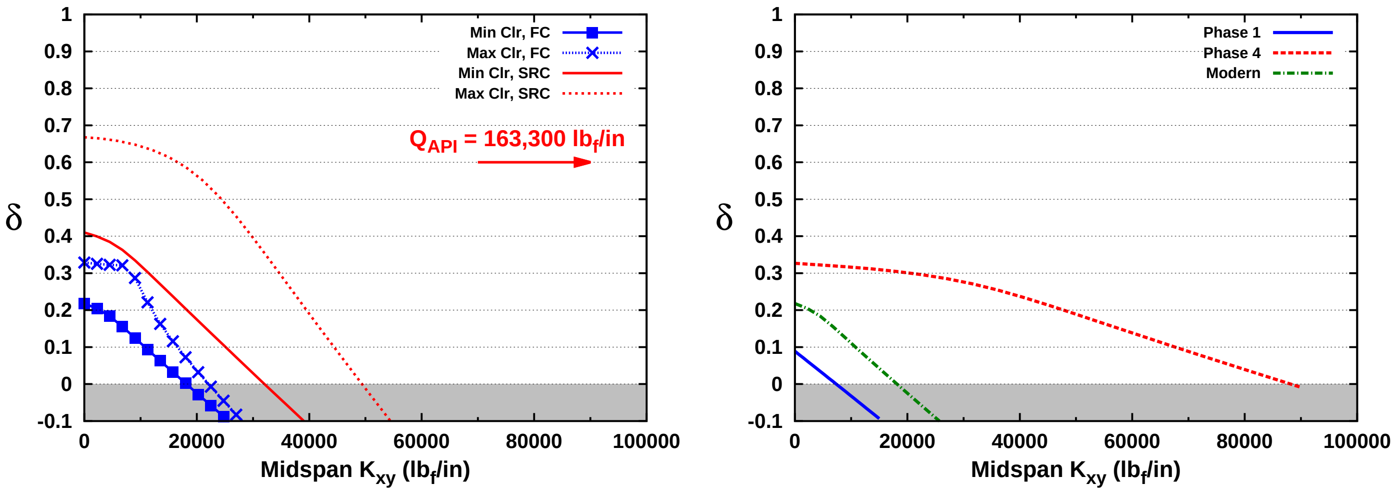
(b) Relative to SRC

Figure 27: Modern design's center hole pattern seal and tilting pad bearing dynamics vs. whirl frequency

The need to consider these frequency dependent dynamics of hole pattern seals in stability analysis are well accepted. For this particular hole pattern seal, Figure 27(b) shows that the seal's direct stiffness in the vicinity of where the 1F mode is expected (∼4000 cpm) is less than 50 percent of that produced at synchronous whirl frequency. Meanwhile, the bearing's direct stiffness goes in the opposite direction, increasing by almost 50 percent from the synchronous coefficient. Ignoring such frequency dependencies and always relying on the synchronous coefficients for stability prediction is inappropriate for these components containing significant internal dynamics (*e.g.*, gas oscillations in cell holes, pads tilting). Active magnetic bearings are another such component where frequency dependent dynamics must be considered [60].

API 617 Level 1 Stability Screening Analysis

Stability sensitivity results of the modern design in Figure 28(a) indicate that a detailed Level 2 analysis is warranted given that the anticipated cross-coupling Q_{API} is well beyond the predicted stability threshold. As with the Phase IV design, the stability characteristics are highly influenced by the choice of tilting pad journal bearing representation. δ predictions based on full coefficients are significantly lower than those from the simplified approach using the synchronously reduced coefficients. Considering the bearings' frequency dependencies observed in Figure 27, it is clear why such stability prediction differences will exist. By allowing for any frequency dependency, the FC representation accounts for the increased stiffness and lower damping produced by the bearing design at subsynchronous frequencies. The simplified representation with SRCs assumes constant, non-frequency dependent dynamics that optimistically yield higher effective bearing damping (low K, high C) and, thus, higher predicted stability levels.



(a) Rotor and bearings only

(b) Relative to earlier designs (rotor + bearings only, FC, Min Clr)

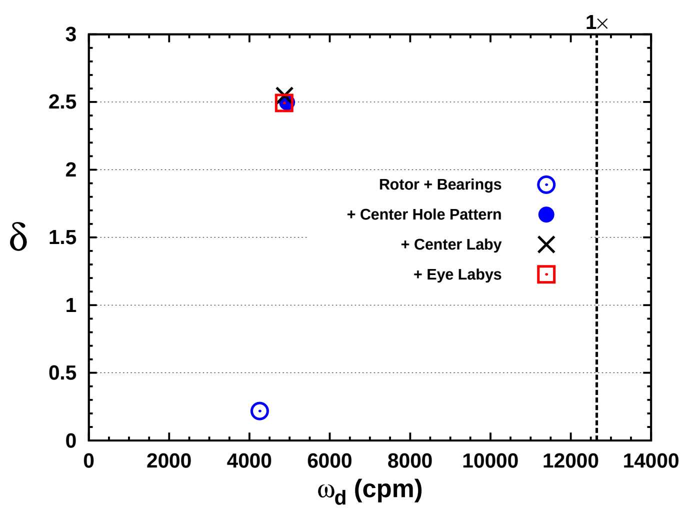
Figure 28: Modern design's stability sensitivity curves

Figure 28(b) compares the stability sensitivity characteristics of the three different rotor-bearing system designs. Even though it possesses similar rotor flexibility characteristics (CSR and rotor L/D), the modern design has higher base stability and a higher stability threshold than the original, Phase I design. These stability increases are a direct result of a better optimized bearing design with higher effective damping from the various preload, L/D, clearance and offset changes. The stability robustness benefits of Phase IV's larger, stiffer rotor design and optimized bearings are pretty apparent in its higher base stability and stability threshold.

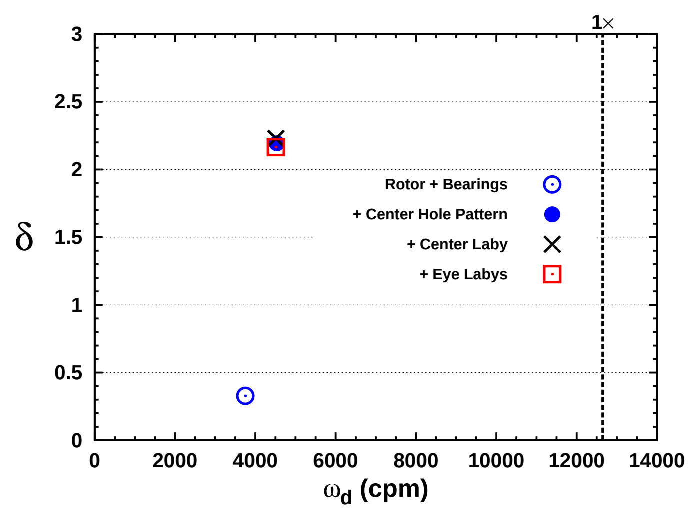
API 617 Level 2 Detailed Stability Analysis

Detailed analysis of the modern design was performed in the same fashion as the Phase I and Phase IV designs. Figure 29 shows the stability spectrum results as components are added to the system model. Relative to the earlier machine designs, the stability effects from the center labyrinth and eye labyrinth seals are secondary.

The only significant change occurs with the presence of the center hole pattern seal. As expected from its predicted, large negative Q_{eff}, the hole pattern seal increases the 1F mode stability to log decrement levels almost an order of magnitude higher than the rotor and bearings alone. Such behavior is why such "damper seals" have become popular amongst some manufacturers, especially in high density applications. It's interesting to note that the machine has such a high level of stability even though it is running almost three times above the 1F mode, a situation that was not conceivable fifty years ago.



(a) Minimum clearance



(b) Maximum clearance

Figure 29: Level 2 stability spectra for first forward mode of modern design

CONCLUSIONS AND RECOMMENDATIONS

The authors have presented an extensive rotordynamic examination of the Ekofisk injection compressor. The paper started with a review of the compressor history, the field vibration problems experienced and discussion on the step-outs regarding the compressor and component design. Difficulties in the dynamic behavioral analysis of the compressor were discussed, in part, resulting from a lack of analytic tools.

Analytical tools commonly used by the turbomachinery industry were applied to better predict and understand where areas of uncertainties remain. As discovered, the as-designed oil seals remain a principle source of this uncertainty. Given the application, pressures and rotational speeds involved, the oil seal behavior dominates the dynamic stability of the rotor. Depending on the eccentric position of the four rings (which can act independently), the stability, as represented by the log dec, can be calculated to be either very stable to very unstable. This was in part confirmed by the operational experience were variations in startup speed, seal design, and suction pressure, affected whether vibrations associated with rotor whirl were noted.

To address the original intent of the investigation concerning the constant improvement process of the industry, the initial questions asked will be discussed here:

- *How well current analytical tools (methods/theories) predict the instability behavior that was observed during the original installation of the compressor?*
  Answer: Good to a point
  It is easily argued that bearing, gas/liquid seal and squeeze film damper behavior prediction tools have improved (or have actually been created) during the last 45 years. There is readily available experimental data to validate a significant portion of these methods and results. In addition, the API 617 requirements would have easily directed the rotordynamics into a detailed stability analysis. It appears likely that the significance of the destabilizing effects of the laby seals would be readily understood. However, the impact of the oil seals would have remained in question. Perhaps one could hope that other factors (*e.g.*, increased independent audits, full load testing) would be used to quantify/understand the oil seal behavior before operation in the field.

- *How well current analytical tools (methods/theories) predict the stable behavior that was observed during the final (Phase IV) configuration of the compressor?*
  Answer: Very well.
  With the redesign of the rotor, bearings and seals, the current tools not only identify the improvements to the rotor stability but also explain why the improvement was achieved. The resulting components used in the Phase IV configuration were proved over the course of time as the correct direction to be taken. Not only do the current tools identify the increased stability of the compressor, they also show the reduced impact of the position uncertainty of the oil seals on the dynamic behavior. With this in mind, the authors' analyses confirm the improvement of the Phase IV design over the original compressor configuration.

- *How important is the consideration of frequency dependence of the seal and bearing behavior in the stability prediction of the rotor when compared against field experience?*
  Answer: Mixed.
  Using the Ekofisk compressor as an indicator of the importance of (or even the existence of) frequency dependent terms in tilt pad radial bearings is problematic because of the dominating and uncertain nature of the oil seals. The original (Phase I) compressor included tilt pad bearings with an offset pivot that are known to have a much smaller difference in the stability prediction between full vs. synchronously reduced coefficients. The Phase IV configuration was shown to be stable using either FC or SRC. The authors attribute this to the improvements made to the oil and center labyrinth seals. Frequency dependence in the laby seals remains a minor influence in their impact on rotor stability.

- *Would the current industry stability analysis methodology (API 617 Level I and Level II) [12] have identified the original configuration of this machine as a risk and the final configuration as acceptable?*
  Answer: Yes.
  API 617 specifications would have easily required a detailed stability analysis of the Ekofisk compressor failing the Level I requirement. Would today's tools have identified the original compressor as being at risk for a rotor instability? The answer to this lies in the simple truth that the tools alone are insufficient to guarantee that any design will be successful. A knowledgeable user of those tools is still required, as many have discovered over the years. "Idiot proof" still has not been achieved for any analytical method. Given today's tools, the impact of the oil seal behavior and the uncertainties associated with it are readily quantified. This should lead to the discussion of risks and mitigation strategies associated with the unacceptable risks. The Phase IV design is shown to reduce the risks of the uncertainties in the compressor design, to the point where the risk of instability is basically eliminated.

- *Can current design features/technologies produce a better design than the modifications that were considered during the initial application?*
  Answer: Most assuredly yes.
  In high pressure/high head applications, the use of anti-swirl devices, damper seals (*e.g.*, honeycomb, hole pattern) have eliminated the destabilizing influence of the laby seal used at impeller eye and center seal location. In addition, hole pattern/honeycomb seals have also been shown to develop significant positive stiffness. When applied at the center location, such damper seals create an effect similar to a three bearing rotor, significantly raising the first natural frequency. Given the simple relationship that,

  $$Q_{eff} = K_{xy} - \omega_{d1} \cdot C_{direct} \tag{2}$$

  one can easily see that, as the first mode's natural frequency increases, the effective destabilizing influences on the rotor decrease. The extended use of dry gas seals at higher pressures also eliminates the potential destabilizing influences of the oil seals. Improvements in the rotordynamic design also result from the aerodynamic achievements. The modern configuration of the Ekofisk compressor has fewer impellers which reduce the bearing span and the L/D ratio. This raises the bending stiffness of the rotor and results in a higher first natural frequency and more stable rotor given everything else being the same.

Perhaps a more appropriate question to ask at this date in time is: "Could today's technology and analytic tools be used to design a compressor that greatly exceeds experience limits?" Perhaps, but even with today's methods, sole reliance on analytical predictions for designs exceeding experience is not recommended. In the past, with the restricted capabilities of the analytical methods, reliance on past experience was significant. This was typically followed by extensive testing. While we can focus the testing more efficiently today, the need still exists. Given the uncertainties in the behavior of gas seals (laby/damper), impeller/shroud interaction forces and the required reliability of turbomachinery in our industry, verification testing remains an important tool. Today, verification testing of the rotor stability and associated threshold is available and recommended [61, 62]. As the trend towards higher power and pressure continues, using analytical methods to identify risks and verification testing to mitigate those risks is a good combination to greatly improving the chances of a successful design.

NOMENCLATURE

1F	=	First forward mode	
API	=	American Petroleum Institute	
$C_{..}$	=	Lateral damping	lbf-s/in
CSR	=	Critical speed ratio	dim
$c_b^{\varnothing}/D$	=	Bearing assembled clearance ratio	$^\circ/_{\circ\circ}$
FC	=	Full coefficients of a tilting pad journal bearing	
$K_{..}$	=	Lateral stiffness	lbf/in
LBP	=	Load between pads	
LOP	=	Load on pad	
MCS	=	Maximum continuous speed	rpm
m_p	=	Pad preload	dim
OSR	=	Operating speed range	
Q_0	=	Stability threshold cross-coupled stiffness where $\delta = 0$	lbf/in
Q_{API}	=	Anticipated cross-coupled stiffness based on API	lbf/in
Q_{eff}	=	Effective cross-coupled stiffness	lbf/in
SRC	=	Synchronously reduced coefficients of a tilting pad journal bearing	
SSV	=	Subsynchronous vibration	
δ	=	Logarithmic decrement	dim
ε	=	Eccentricity ratio	dim
ω	=	Vibration whirl frequency	
ω_d	=	Damped natural frequency	cpm
ρ_{avg}	=	Average gas density	lb/ft^3

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