FULL LOAD TESTING IN THE PLATFORM MODULE PRIOR TO TOW-OUT:

A CASE HISTORY OF SUBSYNCHRONOUS INSTABILITY

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A recent project bought an electric motor driven centrifugal compressor to supply gas for further compression and reinjection on a petroleum production platform in the North Sea. Review of the compressor design, after manufacture was completed, raised concerns about susceptibility to subsynchronous instability. Consideration of log decrement, aerodynamic features, and the experience of other compressors with similar ratios of operating-to-critical speed ratio versus gas density led to the decision to full load test. Mixed hydrocarbon gas was chosen for the test to meet discharge temperature restrictions. This choice of gas led to the use of the module as the test site. On test, subsynchronous vibrations made the compressor inoperable above approximately one-half the rated discharge pressure of 14500 kPa (2100 psia).

Extensive modifications, including shortening the bearing span, changing leakage inlet flow direction on the back-to-back labyrinth, and removing the vaned diffusers on all stages were made simultaneously to meet project needs. Retest at 105 percent speed and overpressure proved the modifications successful, and established a margin of stability. The compressor was commissioned on schedule and is operating with satisfactory vibration levels.

The objectives of this paper are threefold:

- To provide the available case history for students of subsynchronous instability;

- To discuss empirical criteria for subsynchronous instability;

- To consider the benefits of pre-commissioning tests at rated speed and rated gas conditions.

The case history is a practical example to the petroleum industry. The case is also of theoretical interest because of two aerodynamic features of the initial test: the first feature was vaned diffusers on all stages, and the second was a blocked flow passage on one impeller. Both raise a speculative counterpoint to the current trend toward concentrating on labyrinth forces as the main cause of subsynchronous instability in centrifugal compressors. (For example, see Ref. 1.)

Empirical stability criteria compare such things as rotordynamic parameters and gas density for stable versus unstable compressors. In this case, several empirical criteria gave poor correlations. As a consequence, their utility is limited to serving as a rule of thumb before detailed design evaluations are possible.

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The pre-commissioning test proved beneficial. The current state-of-the-art for rotor stability poses considerable risk of inoperable compressors to certain oil and gas projects, which can be addressed by such tests.

Equipment Description

The oil production platform is designed for crude oil production with reinjection of associated gas, which must be compressed from 19 psia to 6600 psia. Two compression systems, each handling half the total flow, are mounted in a single pre-assembled module, designed to be lifted onto the platform deck, complete with a local control room for each system. In each system, the gas first flows through a centrifugal compressor train and then into a reciprocating compressor. The centrifugal train has a motor and gear driving through the low pressure casing to the higher pressure casing. The higher pressure casing is a barrel type. Figure 1 is a photograph of the completed platform.

The subject of this paper is the high pressure casing of the centrifugal train. It is rated to compress 1650 m3/h (970 ACFM) of 27 molecular weight gas from 3030 to 14500 kPa (440 to 2100 psia) at 7732 RPM, absorbing 4200 kW (5600 HP). It has 10 impellers, in 2 groups of 5 each, arranged back-to-back. The flow from the first section to the second section is through an integral crossover pipe without an intercooler. A large labyrinth at the shaft midspan separates the fifth from the tenth stage, and contributes to thrust balance forces. In common with many compressors in similar duty, the aerodynamics are characterized by a low specific speed configuration, by relatively low (1520 m (5000 ft.)) head per stage, and by low Mach numbers. Figure 2 is the general cross-sectional arrangement of the compressor.

Vaned diffusers were used on all 10 stages. The vendor had not previously used vaned diffusers on every stage for compressors with gas densities as high as our rated gas densities. Also, they had not used this particular seal design, which uses face-contact inner seals, at such a high suction pressure. The compressor is equipped with 5-pad tilting pad bearings which have no cross-coupled spring coefficients to excite subsynchronous vibrations.

Background

Publication of other cases of subsynchronous vibration in the petroleum production industry provided the background for recognizing the risk of subsynchronous vibration occurring and potentially causing a major project delay. Accounts of Kabob (Ref. 2) and Ekofisk (Ref. 3, 4) demonstrated that the vibration would not be found until commissioning with rated density and pressure gas, and that the remedy, major mechanical redesign, could cause expensive project delay. Hawkins (Ref. 5), which was familiar, being an Exxon plant, pointed out the risk of subsynchronous vibration at more moderate gas pressures.

The phenomena of concern is self-excited vibration as defined in Ref. 6. That is, the force which causes the vibration depends on the motion of the rotor. This phenomena will be referred to as subsynchronous instability. As discussed in Ref. 6, subsynchronous instability is essentially independent of the frequency of any external stimulus. Instead, it occurs at the natural frequency of the rotor system.

It was recognized that aerodynamic instabilities could cause forced vibrations in a rotor system. Frequencies in the range of 5 to 15 percent of rotor

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speed have been reported (Ref. 7) for that phenomena. Although a test conducted at full scale aerodynamic conditions would have the benefit of demonstrating aerodynamically forced vibrations, they were not the main concern. Neither were parametric instabilities, which typically occur at 1/3, 1/2, etc. of rotor speed.

Evaluating the Risk of Subsynchronous Instability

Due to the relatively low discharge pressure, no special concerns about subsynchronous vibration had been raised by the project team during the design phase. However, when an unrelated issue necessitated review of the design, concerns were raised because of design extrapolations. Further review concentrated on the following three areas:

(1) Low log decrement compared to conventional criteria;
(2) Empirical stability correlations;
(3) Vaned diffusers unproven in high density gas service.

The log decrement is a measure of the rate of decay of free vibration at the rotor natural frequency. Its calculation is covered in the classic paper by Lund (Ref. 8). Because we could not quantify the destabilizing aerodynamic forces acting on the rotor, we chose to compare the log decrements calculated for zero cross-coupled aerodynamic forces. Here we shall refer to the log decrement calculated for zero aerodynamic forces as the "basic" log decrement. The "basic" approach reduces the log decrement concept to an empirical criteria which requires a positive value of the log decrement as a design margin of safety against instability. A common recommendation for a safe "basic" log decrement is 0.3. Nowadays, we would attempt to quantify the destabilizing force at the labyrinths, but this technology was not available when the decision to test was made.

Figure 3 compares basic log decrement values for various compressors. The vendor calculated that the subject compressor had a basic log decrement of 0.16, which compared unfavorably to others, especially an inert gas compressor which was unstable at a lower gas density than the subject compressor's rated point. This implied a significant risk of subsynchronous instability. The vendor did not consider the risk to be high and cited a lower log decrement (approximately 0.14) for a very similar compressor at higher pressure, which they stated was marginally stable. The main features which were different from the similar compressor were vaned diffusers on every stage of our compressor, and oil-seals with a different atmospheric side sealing ring design.

The second empirical stability correlation used was proposed by Sood in Ref. 9. It plots stable and unstable compressors on two coordinates to define a threshold line for instability. The coordinates are "flexibility ratio" versus average gas density in the compressor. The flexibility ratio is defined as the "maximum continuous speed" divided by the "first critical speed on stiff supports." Although no numerical values are given in Ref. 9, several example points are available from the literature and from our experience. Some of these are shown on Figure 4.

We have drawn a line representing the approximate threshold where subsynchronous vibrations were believed to occur. Going above the line indicates an increasing tendency for subsynchronous vibration to occur, while staying below the line indicates that such vibration is less likely to occur. The line slopes down with increasing gas density because the higher gas forces on the rotor require a stiffer rotor to resist subsynchronous vibration. A stiffer rotor has a higher

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first critical speed and thus a lower "flexibility ratio" for the same threshold speed. This line, labeled "Typical Threshold Line," is drawn through the scattered points labeled A, B and C which represent compressors at the threshold of subsynchronous vibration, as known at the time of the decision.

As indicated by point D in Figure 4, our intended operating conditions were marginally on the stable side of the typical threshold line. However, the inert gas compressor was also in the marginally safe zone (point B), when it first encountered subsynchronous vibrations. When the marginal position on this curve was considered simultaneously with the low log decrement, the risk of subsynchronous vibration became substantial. Note that the rotor flexibility ratio versus density curve does not consider the effect of bearing damping, being based only on shaft mass-elastic properties, as implied by the rigid bearing critical speed. Furthermore, no other compressor on Figure 4 had vaned diffusers.

Two separate effects may be produced by the vaned diffusers. Firstly, they might cause a forced vibration due to an aerodynamic instability, such as rotating stall in the diffuser, especially if the impeller and diffuser were mismatched. Secondly, the diffuser might interact with the impeller to produce aerodynamic cross-coupled forces on the impeller which could cause rotordynamic instability. We had practical interest in either case, even though rotordynamic instability was the main concern.

Vaned diffusers had not been used before by the vendor on all stages of a compressor for high density gas. The bulk of the vendor's experience with vaned diffusers on all stages was in hydrogen rich service, at much lower gas densities. Ref. 4 mentioned that vaned diffusers were tried at Ekofisk on the fourth and eighth stages with neutral results on that rotordynamic instability problem. Although the intent there was to cure the rotor instability, it was not clear that the vane's effect always had to be an improvement, especially if the aerodynamic design was off target. Our concerns on this point were raised due to the unexpectedly high head found on our ASME PTC-10 Class 3 performance test.

Based on the above risks of subsynchronous vibration and the large costs and delay if the problem remained undemonstrated until start-up in the field, a full-load test was recommended. By testing prior to tow-out of the module, any problems with the compressor or the module would be discovered a year in advance of commissioning, which allowed time for correction, before incurring start-up delays. Contingency modifications to reduce the risk were considered. However, attempts to plan reduced scale testing, or to make contingency modifications, were defeated by the lack of basic engineering knowledge or proven calculation techniques addressing the mechanism of subsynchronous instability.

Testing Requirements For Subsynchronous Instabilities

Testing for subsynchronous instability requires virtually full scale operating conditions. Full speed is required to obtain the same rotordynamic behavior during testing as during service conditions. The cross-coupled aerodynamic forces must also be reproduced at full scale to cause the instability. Using, as one example, labyrinth forces due to gas entry swirl, it can be shown that at least three variables, gas density, gas flow angles and differential pressure across the stage, must be the same for test as for service (see Ref. 10). Therefore, the compressor must be operated at full power and with aerodynamic simultude. Some deviation from rated suction and discharge pressures is permissible so long as the differential pressure for the test is the same as rated. The only significant

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choice left after satisfying these three variables is whether a suitable non-flammable, non-toxic gas can be found to simulate the service gas within a reasonable tolerance.

For our test the additional constraint of a 204°C (400°F) casing design temperature limit defeated the search for a non-flammable gas. Meeting the suction and discharge densities for full speed and rated flow caused the discharge temperature to exceed the limit for all safe gas mixtures we tried.

The above considerations eliminated the possibility of testing with inert gas. The requirements of testing with hydrocarbon gas, plus the advantages of testing and precommissioning the other equipment in the module, led the project toward a test in the module.

The hydrocarbon gas used for the test was nominally 76 percent ethane and 24 percent methane. The intent was to be near ASME PTC-10, Class 1 conditions, with suction and discharge densities within 5 percent of rated, at close to the rated suction and discharge pressures. Literally blending the rated composition, shown on the compressor datasheet, was not practical because that gas is saturated and would condense if stored in a single vessel at high pressure and practical temperatures.

Test Conducted in Platform Module

The module is an enclosed structure designed to be lifted as a single unit when it is installed onto the oil production platform. Installation takes place by towing the module on a barge to the platform which has been previously erected in the North Sea. The module contains the two centrifugal compressor trains, the two reciprocating compressor trains, two local control rooms, the process piping, valves, and the compressor suction vessels.

The module formed a natural test site because the compressor trains and much of the equipment required to operate them were assembled in working order. Testing in the module had the additional advantage of pre-commissioning the ancilliary equipment in the module prior to its being installed in the North Sea, where the logistical burden of any corrections would be greater.

For the test, the module was placed on its tow-out barge, which was in the River Tyne, at Newcastle, U.K. A photo of two modules on the barge is shown in Figure 5. The module containing the compressor is on the left.

The following facilities were required in addition to the equipment permanently installed in the module:

● Nitrogen to inert and purge the hydrocarbon gas piping;

● Cooling water pumps;

● A hydrocarbon gas flare stack with associated liquid surge drums for safe release of the test gas;

● An electric power generator to supply power for driving the compressor motor. We used one of the gas turbine generators intended for permanent installation on the offshore platform. This stand-alone package is shown in Figure 6. Because our electric power system was independent of any other power system, we were able to vary the power supply frequency from the rated 60 Hertz to vary

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compressor speed.

Results of the Testing

The testing for subsynchronous instability was conducted in four conceptual steps as follows:

A. A test of the compressor design configuration as purchased, at rated speed, by increasing suction and discharge pressure until the onset of subsynchronous instability;

B. As (A) except the journal bearing orientation was switched from having the rotor weight between 2 of the 5 tilting pad to having the weight on 1 of the 5;

C. After extensive design modifications and at rated speed and pressure;

D. At proof test conditions not obtainable in the actual duty, for the purpose of demonstrating a margin of safety against subsynchronous instability.

The highlights of the test results are shown in Figure 7. Step A of testing showed severe subsynchronous instability at suction and discharge pressures above approximately 1580 kPa (230 psig) and 6500 kPa (940 psig) respectively. The instability barrier limited further pressure increases by causing the subsynchronous shaft vibrations near the bearings to exceed the 100 micron (4 mils) peak-to-peak shutdown limit, stopping the compressor. The initial test point is shown as point E on the Sood coordinates, Figure 4.

Step B consisted of modifications that could be made to improve stability without removing the rotor and stator from the barrel casing. The contact face of the atmospheric side oil seal ring was improved in order to reduce the radial loads that could be transmitted to the shaft. Based on the wear pattern on the contact surface, it was theorized that the ring was not moving freely in the radial direction, as the designer intended, and therefore, the oil ring could act similarly to a cylindrical bearing and introduce cross-coupled forces onto the rotor.

The tilt pad bearings were modified as tabulated in Figure 8 (adapted from Ref. 12) to increase the effective damping and to introduce more asymetry between the vertical and horizontal spring coefficients. Asymetry is expected to increase stability according to Ref. 11.

The results from Step B were disappointing, with only a slight increase in discharge pressure, from 6500 kPa (940 psig) to 8200 kPa (1190 psig) before encountering instability again. The vendor's prediciton of a positive log decrement, as shown in Figure 8, was not born out in practice. We do not know whether the assumed cross-coupled stiffness of 810,000 N/M (4644 lb/in.) was too low, or whether the bearings were less effective than expected, or whether other unaccounted factors caused the analysis to fail.

Step C was made after extensive modifications described below. The compressor operated with only traces of subsynchronous vibration. Those traces, at 4575 CPM and 4800 CPM are of some theoretical interest compared to the calculated first natural frequency of 4400 CPM. The second frequency of 4800 CPM was not explained, but is clearly not due to some other harmonic or sub-harmonic frequency of the rotor, gear, or electrical system because the 4800 CPM remained unchanged

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when the electrical supply frequency and motor speed was changed between steps C and D. Step C is shown as point F1 on the Sood coordinates, Figure 4.

Step D, the proof test, was entirely successful, with no significant subsynchronous components in the operating flow range. Step D was done by running at 105 percent rated RPM. 105 percent speed was achieved by increasing the power supply frequency from 60 Hertz to 63 Hertz to circumvent the limitation of having a constant speed motor drive. Step D is shown as point F2 on the Sood coordinates, Figure 4.

Possible Causes of the Subsynchronous Vibration

It is important to note that such a costly full scale test was necessary, because of the business risk created by engineering ignorance. No one knew for sure that the subsynchronous vibration would occur, but instead the risk was recognized and the options properly weighed. Typically, the business considerations which justified the test also required that all the changes, which might possibly help cure the problem, be made simultaneously. Therefore, the actual causes of the problem cannot be isolated or proven. We will give our opinion on the causes and try to rank their importance. The changes which are believed to have significant effect on subsynchronous vibration are as follows (in order of decreasing significance):

1. Bearing span was reduced by 6.5 inches.

    Reduced bearing span, which stiffens the fundamental bending mode, was estimated beforehand to have been sufficient to reach 1900 psi discharge versus the 2500 actually achieved during proof test at 105 percent speed. The estimate was based on a single mass (Jeffcott) rotor according to Figure 10 of Reference 11, by assuming the destabilizing out-of-phase coefficient is proportional to gas density.

2. Swirling flow entering into the central division-wall labyrinth was eliminated by introducing gas between the teeth at elevated pressure. (The higher pressure reverses the flow across the first two teeth so that gas that flows out of the high pressure side of the labyrinth instead of entering there.)

    Elimination of swirling flow into the most influential labyrinth is a common fix and is probably quite significant. The swirl in the labyrinth is not totally eliminated, because viscous forces reintroduce the swirl some number of teeth downstream. No estimate of labyringth cross-coupled forces could be made at the time.

3. All vaned diffusers were removed and narrower parallel wall vaneless diffusers were substituted.

    Removal of the vaned diffuser was prompted by the observation that the "similar compressor" operated without major subsynchronous instability at a less favorable point on the density speed-ratio plot. (See Figure 4, points A versus E.) The major difference between the designs was having vaned diffusers on all stages in our case. Excess head and efficiency were in hand at this time, so the loss in efficiency, caused by removing the vanes, could be tolerated.

    Removal of the vanes required reducing diffuser widths drastically to maintain

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the meridional velocity of the flow and the incidence angle at the return vanes. For example, on the last stage the diffuser width was reduced from over one quarter inch to less than one tenth of an inch, which is even less than the normal width for this staging. The narrow width enhances aerodynamic stability at the expense of aerodynamic efficiency. Reduced widths were used for enhanced stability on another subsynchronous vibration problem as discussed in Reference 4.

4. Axial forces on the breakdown-side oil seal were reduced by an improved pressure balance configuration. The seal design used face contact inner seals. The particular seal design had a breakdown bushing which had limited experience at over 400 psi. The vendor chose to improve the axial balance, but no test was made to demonstrate how much influence on subsynchronous vibration the breakdown bush might have had by acting as a cylindrical bearing with oil whirl.

5. A foreign object was found blocking one vaned passage of the ninth impeller. It was removed for the retest.

   A consultant raised the point that the asymetric blockage might cause whirl. No practical evaluation of this point was possible. The blockage was removed simultaneously with the other changes and did not reoccur during the successful retest.

6. Division wall labyrinth and balance piston diameter were reduced in diameter from 9 1/4 inches to 8. Reduction of the balance piston diameter reduced area under influence of the destabilizing gas swirl pressures. The reduction also helped raise the first critical speed by removing mass from the center of the span. As an aside, the thrust balance was improved. We are aware of other load tests which have also revealed thrust problems, which cannot be assessed during API mechanical run tests at low pressure.

Evaluation of Empirical Stability Criteria

The test results provide additional comparisons for the empirical stability criteria. Here we will evaluate the "basic" log decrement, and the rotor stability plot on Sood's coordinates, plus a criteria which was published later, without reference to our test, by Kirk and Donald (Ref. 13). Generally speaking, all three empirical criteria show so much scatter that they are only useful as rules-of-thumb. The scatter is not surprising because these empirical citeria do not correlate a specific physical cause, but instead represent the results achieved by diverse design features.

The basic decrement of 0.3, often recommended for high gas density service, is reaffirmed by the test results. The log decrement before the major modifications, 0.16, was shown to be insufficient by the occurrence of subsynchronous instability, while the basic log decrement after major modifications, 0.32, was proven adequate. However, the 0.14 log decrement of the similar compressor from the same vendor, as shown in Figure 3, demonstrates that 0.3 is not always necessary.

When the initial test point is put onto Sood's coordinates for rotor stability, point E on Figure 4, that point represents a worst case. Compared to the "typical threshold line" in Figure 4, the compressor was much less stable than expected. We have drawn a line through the initial test point and parallel to

typical threshold line, and labeled it "worst case threshold line," because it represents the least stable case we know of using the Sood coordinates.

The scatter that can be introduced (into Figure 4) by special design features is demonstrated by point "G" which is for a compressor with "deswirl cascades" in front of the impeller eye labyrinths of each stage. The deswirl cascades reduce labyrinth cross-coupled forces and is similar to the devices covered by U.S. Patent 4,370,094, dated 1983. The point "G" compressor was demonstrated to European industry users as an example of high pressure centrifugal compressor technology.

Kirk and Donald proposed an empirical criteria using the product of the discharge pressure with the differential pressure of the compressor versus the critical speed ratio (operating speed divided by first rigid bearing critical speed, which is identical to the "flexibility ratio"). Because our initial and final test points, plus the Hawkins points, all have back-to-back staging, it is fair to plot them on Figure 9, which is adapted from Ref. 13. Our compressor is again a worst case, and we have drawn a "worst case" parallel line. The "Point G" compressor from Figure 4 is indicated as well to represent a best case.

The scatter found in these three empirical stability criteria reduce their utility where accuracy is required. However, these criteria do serve to indicate the general area of concern for subsynchronous instability for industrial centrifugal compressors. Purchasers of these compressors should find the worst case lines in Figures 4 and 9 a commercially useful criteria if they adopt the following strategy: we recommend that proposals for compressors on the unstable side of the worst case line be required to include prices for an optional full speed, full gas density test with rated differential pressure across the compressor. The final decision to test could then be made later when the information from complete analytic studies is available. For example, the decision to full-load test the compressor at point "H" in Figure 4 was influenced by similar criteria.

Conclusion

The cause of the subsynchronous instability was not determined. Perhaps the changes to the bearing span, the center labyrinth and the atmospheric-side oil seals would have been sufficient to eliminate the instability. With state-of-the-art analysis techniques for labyrinth forces, it may soon be possible to estimate whether those changes would have been sufficient, given enough detailed design data. However, compared to the very similar compressor from the same vendor, the instability threshold is inexplicably low. Therefore, the vaned diffusers and the non-axisymetric passage blockage of the ninth impeller cannot be dismissed out of hand.

Both empirical stability criteria, one using the Sood coordinates, and the other by Kirk and Donald, are shown to have an order of magnitude of scatter with respect to their aerodynamic force parameters. Therefore, the utility of these criteria for comparing different designs is reduced to applications which can tolerate the scatter, and/or the pessimism of using the "worst case" line. The "worst case" lines given should be a useful rule-of-thumb for indicating a threshold-of-concern for subsynchronous instability in similar industrial centrifugal compressors.

The test was a technical and commercial success. Subsynchronous instability was found and quickly remedied, preventing any delay in bringing a

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valuable project onstream. Although the test was very expensive at approximately 3 million Pounds Sterling (see Ref. 12), the conservatively-estimated potential losses would have been substantially more than that amount, had subsynchronous instability delayed oil production.

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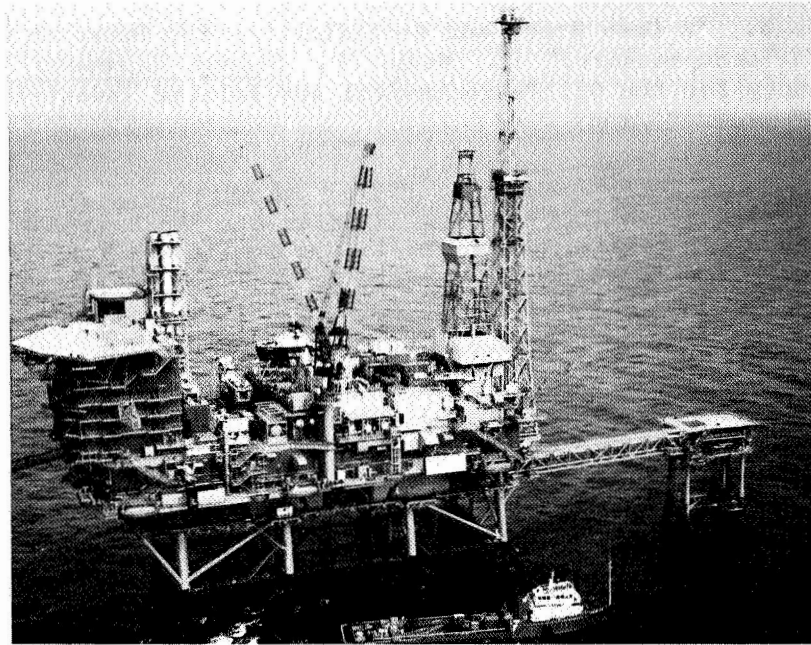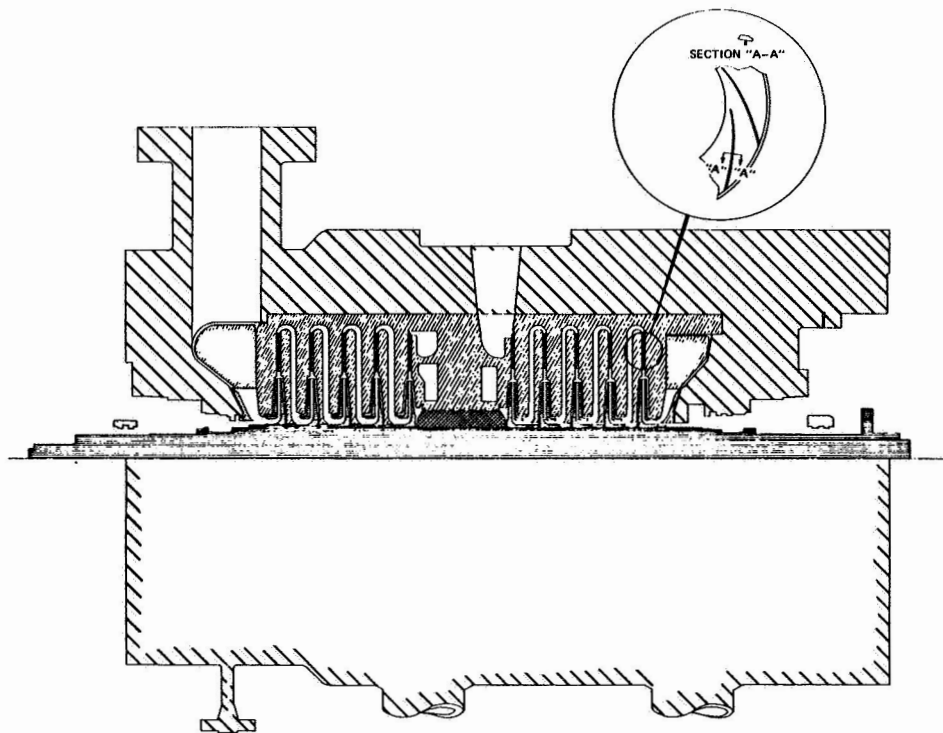
Figure 1. - North Sea petroleum production platform.

SECTION "A-A"

Figure 2. - Cross-sectional view of centrifugal compressor train.

	BASIC LOG DECREMENT	AVERAGE GAS DENSITY (Kg/m^3) (Lbm/Ft^3)	
PRESENT CASE			
BEFORE MODIFICATION	0.16	32	(2)
AFTER MODIFICATION	0.32	30	(5)
SIMILAR COMPRESSOR (SAME VENDOR)	0.14	104	(6.5)
INERT GAS COMPRESSOR			
BEFORE MODIFICATION	0.23	43	(3)
AFTER MODIFICATION	0.43	82	(5)
REFERENCE 9 COMPRESSOR	0.443	208	(13)
RECOMMENDED LOG DECREMENT FOR HIGH DENSITY	0.3	-	-

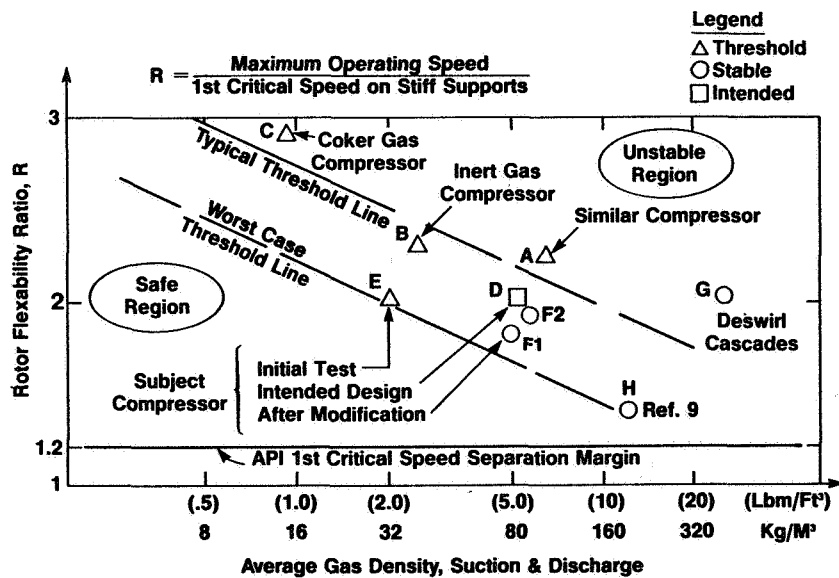Figure 3. - Log decrement values for various compressors.

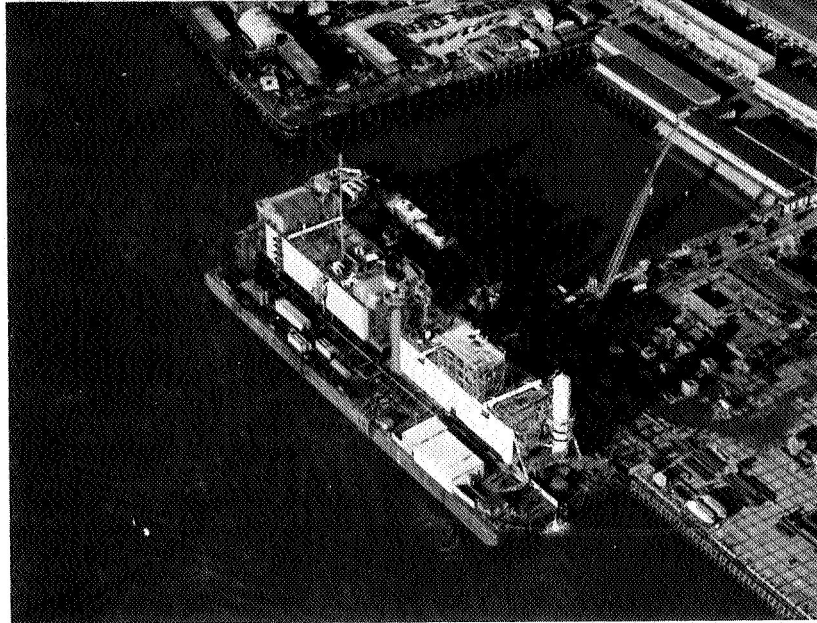Figure 4. - Sood's model for instability thresholds applied to various compressors.
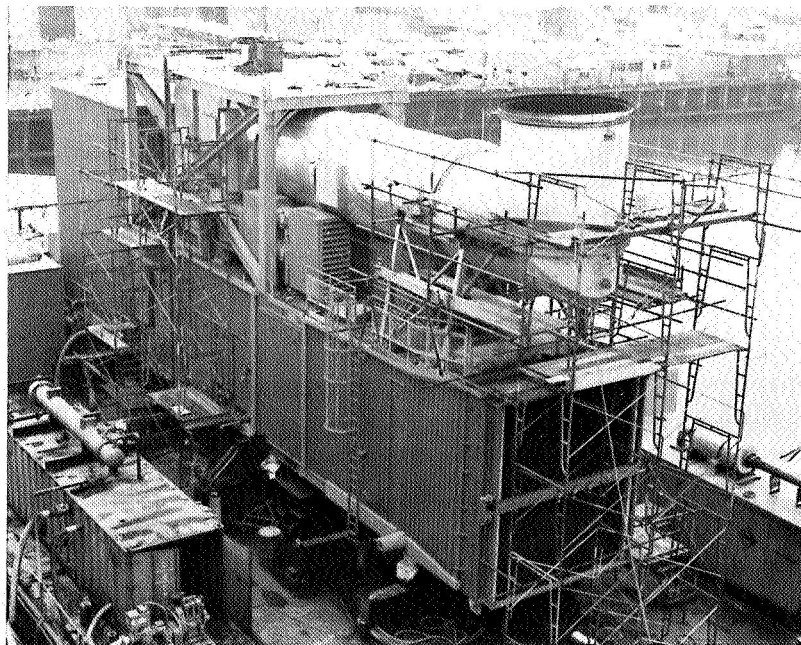
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Figure 5. – Tow-out barge test platform.

Figure 6. – Gas-turbine-driven electric generator for pump power.

TEST STEP	A	B	C	D
SUBSYNCHRONOUS FREQUENCIES (CPM)	4125	3825	4575 & 4800	4575 & 4800
SUBSYNCHRONOUS AMPLITUDES (MICRONS)	22	20	4 & 4	5 & 3
COMPRESSOR SPEED. (RPM)	7800	7725	7742	8121
SUCTION PRESSURE (kPa GAUGE)	1580	1560	2280	2840
DISCHARGE PRESSURE (kPa GAUGE)	6500	8200	14400	17200
OBSERVED FIRST CRITICAL RANGE (RPM) (APPROXIMATE)	3800-3900	3900-4000	4100-4200	4100-4200

Figure 7. - Highlights of test results.

BEARING CONFIGURATION

| TEST STEP	A	B
TYPE	TILTING PAD	TILTING PAD
ORIENTATION	LOAD BETWEEN PADS	LOAD ON PAD
LENGTH INS.	1.5	1.5
DIAMETER INS.	5.0	5.0
MACHINED DIAMETER CLEARANCE INS.	0.012	0.009
PRELOAD	0.489	0.0
ASSEMBLED CLEARANCE INS.	0.006	0.008
BASIC LOG DECREMENT	0.148	0.164
AERO CROSS-COUPLING STIFFNESS 'Q' LB/IN	4644	4644
LOG DECREMENT WITH 'Q' AERO CROSS-COUPLING	-0.284	0.122

Figure 8. - Tilt-pad-bearing modifications.
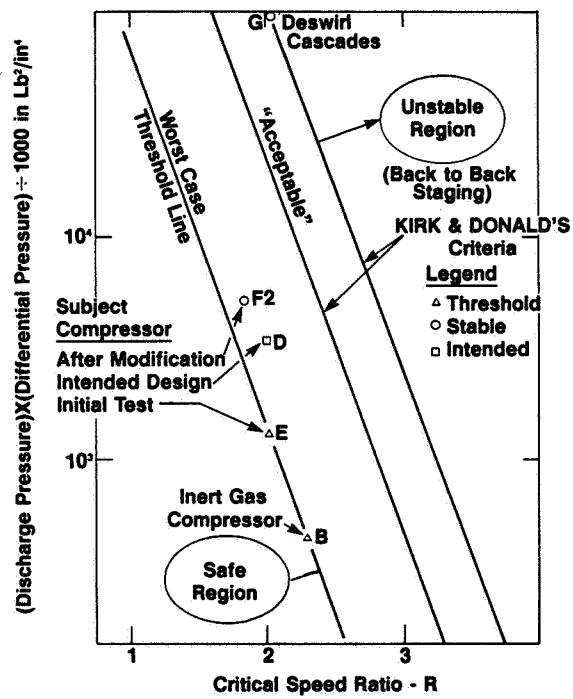
Figure 9. — Kirk and Donald's stability criteria applied to various compressors.