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STABILITY ANALYSIS AND TESTING OF A TRAIN OF CENTRIFUGAL COMPRESSORS FOR HIGH PRESSURE GAS INJECTION

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ABSTRACT

This paper describes the rotor dynamic stability analysis and the PTC-10 Class 1 test of a three body centrifugal compressor train for high pressure natural gas injection service. This train had a full load full pressure string test on hydrocarbon gasses to a final discharge pressure of 500 BAR (7250 PSIA). Each compressor is of the back to back configuration, and is equipped with tilting pad seals, damper bearings, and a honeycomb labyrinth at the division wall with shunt holes. The driver is a gas turbine.

1. INTRODUCTION

In 1995 a string of high pressure barrel compressors was given a full load full pressure hydrocarbon test at the manufacturer's facility. The final discharge pressure of the compressor train was very high, 500 BAR (7250 PSIA). Because of successes with a honeycomb seal with shunt holes at the division wall (e.g., Memmott, 1994), these were applied at the design stage to each of the three compressors.

The rotor dynamic analysis of the compressors with the honeycomb seals predicted that the compressors would be very well damped. This was confirmed by the testing, which showed no evidence of subsynchronous vibration, in fact the first natural frequency was so well damped that it could not be found during the full load test, for every compressor in the train.

The component rotor dynamic designs applied to these compressors consisted of four specific items, tilting pad seals, damper bearings, shunt holes at the division wall, and a honeycomb seal at the division wall. A significant amount of operating experience has been accumulated in high pressure applications, such as natural gas reinjection, with compressors using the first three items. (Coletti and Crane, 1981; Shemeld, 1986; Memmott, 1987, 1990, 1992, 1994, 1996; Marshall, et al., 1993). Honeycomb seals have been used for years and were applied recently for improved rotor dynamic characteristics (Kuzdzal, et al, 1994; Sandberg, et. al., 1994; Memmott, 1994; Gelin, et. al., 1996).

The full load full pressure hydrocarbon test results will be shown, as well as the lateral rotor dynamic stability analysis of the high pressure compressor. A full analysis is made of all the labys, toothed and honeycomb, and their coefficients are included in the stability analysis. The large amount of damping in the honeycomb laby will be seen in the analysis and from the test results.

2. DESCRIPTION OF EQUIPMENT

Each turbomachinery train consists of a gas turbine driving three centrifugal compressors through a speed increasing gear as shown in Fig. 1. The train is designed for natural gas injection service. The PTC-10 Class 1 full load full pressure performance testing on hydrocarbon gas was completed in 1995 and the compressors were shipped in late 1995 and early 1996. They were started up in 1997.

Each compressor is of the back-to-back configuration. The low pressure compressor, discharging at 179 BAR (2596 PSIA), has eight stages of compression. The medium pressure compressor, discharging at 308 BAR

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(4460 PSIA), has six stages of compression. The high pressure compressor, discharging at 500 BAR (7253 PSIA), has six stages of compression. There are four identical trains designed for natural gas injection service. A sketch of the train is shown in Fig. 1. The MW is 19.0, the maximum continuous speed is 11502 rpm, and the power at the normal speed is 17500 KW (23500 BHP) for the train.

This paper will focus on the high pressure compressor, a 191B3/3, with the cross section depicted in Fig. 2. The seal span is 1006 mm (39.62 in), the impeller bore is 114.3 mm (4.5 in), and the bearing diameter is 95.25 mm (3.75 in). The compressor power is 4280 KW (5750 BHP).

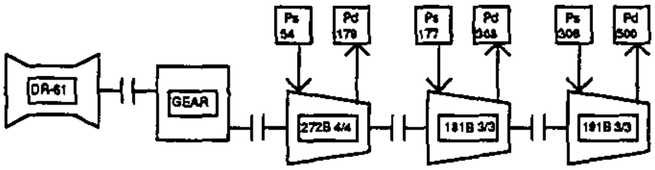


Fig. 1 - Train Sketch

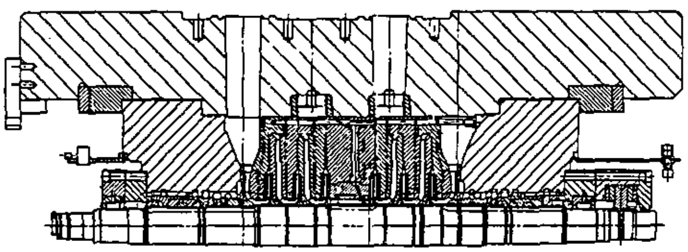


Fig. 2 - Compressor Cross Section

3. COMPONENT DESCRIPTION

The tilting pad seals, damper bearings, shunt holes at the division wall, and honeycomb seal at the division wall are described below.

3.1 Tilting Pad Seals

The tilting pad seal is shown in Fig. 3. The tilting pads are mounted concentrically with the outer oil ring and minimize any cross coupling coefficients created by the ring. The tilting pads also provide substantial load carrying capability. The tilting pad seals create a four bearing system, with beneficial spring and damping coefficients at the seal location.

The tilting pad seals effectively shorten the bearing span of the compressor. The first critical of the compressor is raised by a large amount, and thus the flexibility ratio, or ratio of the maximum continuous speed over the first critical speed, is lowered. This effect of the tilt pad seals has been discussed many times (Memmott, 1990, 1992, 1994, 1996; Marshall, et al., 1993). A low flexibility ratio is seen as beneficial (Kirk and Donald, 1983; Fulton, 1984ab; Memmott, 1990, 1992, 1994, 1996; Marshall, et al., 1993; Bromham, et. al., 1996; Gelin, et. al, 1996).

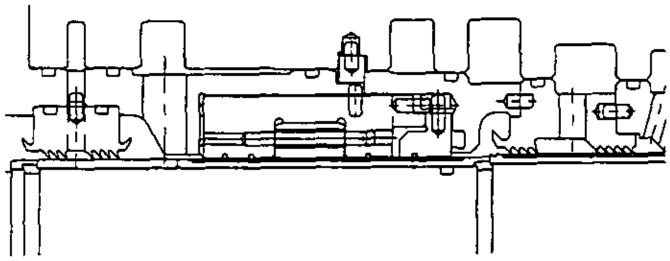


Fig. 3 - Tilt Pad Seal

Over 610 compressors with tilting pad seals have been installed around the world, in many different services. The design has been used at sealing pressures up to 21475 kpa (3100 PSIG), and for speeds up to 22800 rpm. The first installation was in 1972, and the consistent reliability of units using the design has led to it becoming the standard in many types of compressors. Damper bearings are installed in 37% of these compressors.

3.2 Damper Bearings

The specific damper bearing design uses the lubricating oil supply as the damping medium, and o-rings to both seal and center the damper. The damper, in series with the hydrodynamic bearing oil film, provides much improved stiffness and damping characteristics. A cross-section of the damper bearing is shown in Fig. 4.

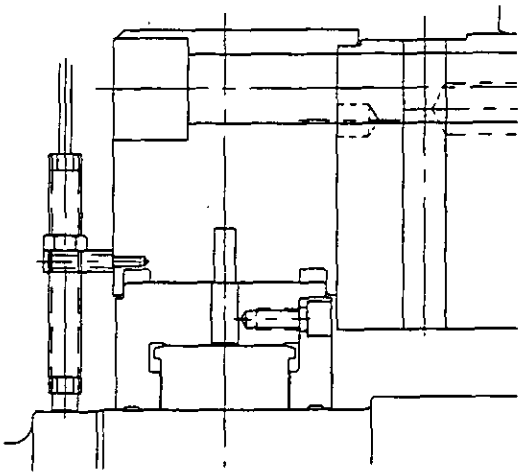


Fig. 4 - Damper Bearing

Dampers of this type have been used in over 460 compressors since their first field installation in 1973 to a 207 BAR (3000 PSIA) discharge syn gas compressor (Memmott, 1992). Their typical application has been in high pressure compressors to improve the resistance to subsynchronous vibration problems. The rotor-support system is tuned by softening the support to provide more optimum rotor dynamic characteristics.

The compressors discussed in this paper have tilting pad seals and damper bearings. Over 230 compressors with gas or laby seals have damper bearings. Applications of damper bearings to these compressors are discussed in (Memmott, 1987,1990,1992).

3.3 Shunt Holes

The application of shunt holes from the final stage diffuser to the division wall of this back-to-back compressor is shown in Fig. 5. The pressure of the gas in the diffuser is higher than that at the exit of the impeller, due to the static pressure rise in the diffuser. The gas supplied by the shunt holes to the labyrinth comes in radially to the laby. The shunt holes eliminate prerotation of the flow into the division wall labyrinth and provide radially outward flow along the back side of the last impeller. Minimizing prerotation into labyrinths decreases the cross coupled stiffness coefficients generated by the labyrinths (Childs, 1993). This in turn increases the stability of the rotor system.

The gas behind the last stage impeller should flow out radially and not inward. If it flows inward, as it would without shunt holes from the diffuser, an unstable flow field can be created in the region between the rotating disc and the stationary wall (Marott, et. al., 1959). There is evidence that a shunt hole system from the discharge volute of a straight-though compressor is not the best application (Kuzdzal, et. al., 1994; Sandberg, et. al., 1994; Gelin, et. al., 1996).

Without the shunt hole from the last stage diffuser, a flow down the back of the impeller would fight the natural pumping action of the back side of the impeller. A tight gap between the back side of the impeller and the stationary wall increases the number of unsteady flow regions (Marott, et. al., 1959). There is test experience which shows that the lack of shunt holes at the division wall honeycomb laby of a back-to-back compressor contributed to instability (Memmott, 1994). This could be due either to the action going on in the back of the impeller or due to the high pre-swirl coming into the honeycomb.

Shunt holes were first applied in 1974 to a 145 BAR (2100 PSIA) CO_2 compressor (Memmott, 1987), and over 410 compressors have been built with shunt holes to date. Most of these applications have been to back to back compressors at the division wall, where they have become a standard (Coletti and Crane, 1981; Shemeld, 1986; Memmott, 1990, 1992, 1994). Shunt holes have been applied to straight-through compressors at the balance piston. (Memmott, 1990, 1992, 1996; Marshall, et. al., 1993; Gelin, et. al., 1996). Their primary usage has been in high pressure compressors, since the cross coupled stiffness coefficients caused by labyrinths increase with increasing pressure differential.

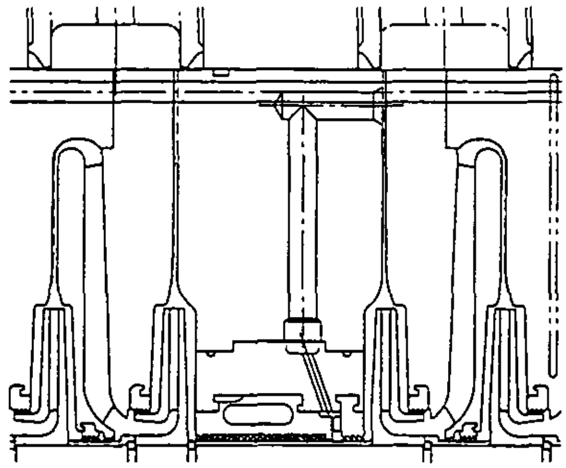


Fig. 5 - Shunt Holes & Honeycomb Seal At The Division Wall

3.4 Honeycomb Division Wall Laby

The honeycomb seal at the division wall with the shunt holes is shown in Fig. 5. Honeycomb seals supply large amounts of damping compared to the negligible amount available from toothed laby seals (Memmott, 1994; Kuzdzal, et. al., 1994; Sandberg, et. al., 1994; Gelin, et. al., 1996). This has been taken advantage of in assuring stable operation in several recent compressor applications. It is important to note that they do not decrease the cross-coupled stiffness over that of a comparable toothed laby, it is the large increase in direct damping that is crucial. Honeycomb seals also have a much larger direct stiffness than a toothed laby.

The damping available from the honeycomb seals is not so high that it can always control subsynchronous vibration without a shunt hole system from the diffuser. This was shown by the full load full pressure testing of the 400 BAR (5800 PSIA) compressor in 1994 (Memmott, 1994). This is also implied by (Gelin, et. al., 1996).

Honeycomb seals were initially used in centrifugal compressors for strength reasons, where differential pressures were in excess of what could be handled by conventional knife edge seals. They have been used since the late 1960s for that purpose, mainly in high pressure syn gas compressors. More recently they have been applied for the benefits of damping available from the honeycombs. This followed research in the 1980s (Childs, 1993, pp. 331-341) and the development of design codes for stiffness and damping calculations for honeycomb seals (Scharrer and Pelletti, 1994-5).

Over 110 compressors with honeycomb seals have recently been shipped, not counting the numerous previous applications to syn gas service. The honeycomb seal has been especially helpful in application to high pressure gas injection compressors. See (Memmott, 1994) for discussion of an application to a 400 BAR (5800 PSIA) natural gas injection compressor. See (Gelin, et. al., 1996), for discussion of applications to a 205 BAR (2970 PSIA) reinjection compressor and a 141 BAR gas lift compressor.

4. STABILITY EXPERIENCE GUIDELINES

Various experience guidelines have been used to evaluate the stability of high pressure compressors. These are used as a preliminary guide in deciding on the feasibility of building a compressor, to see that the compressor is within the range of experience with the components that are being used. The guidelines are not used for the detailed design of the components and analysis of the system, as sophisticated rotor dynamic codes have become available. Application of two of these guidelines to the compressors for this installation are shown below.

4.1 Flexibility Ratio vs. Average Gas Density

A widely used experience guideline is the plot of flexibility ratio, maximum continuous speed divided by the first critical speed, vs. average gas density, commonly called the Fulton plot, as shown in (Fulton, 1984ab; Memmott, 1992, 1994, 1996). A similar plot is where discharge, instead of average, gas density is used (Bromham, et. al., 1996). Tilting pad seals raise the first critical speed significantly and thus lower the flexibility ratio (Memmott, 1990, 1992).

The Fulton type plot is shown in Fig. 6, for a representative sample of compressors with tilting pad seals. The compressors discussed in this paper are marked. They are comfortably within the range of experience and well below Fulton's typical threshold line. Without the tilting pad seals, the flexibility ratio would put them above Fultons acceptable line, where he predicts instability is likely to occur.

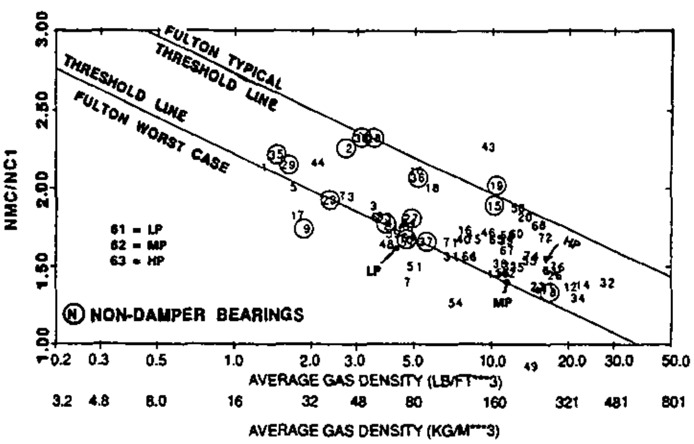


Fig. 6 - Flexibility Ratio With Tilt Pad Seals Vs. Average Gas Density

If the gas is less dense and the first critical is higher then it is harder to excite the first critical speed.

4.2. Flexibility Ratio vs. Discharge Pressure x (Discharge Pressure - Inlet Pressure)

Another experience guideline is the plot of flexibility ratio vs. discharge pressure x (discharge pressure - inlet pressure) as shown in (Kirk and Donald, 1983; Fulton, 1984b). This is commonly called the Kirk-Donald plot.

The Kirk-Donald type plot is shown in Fig. 7, for a representative sample of compressors with tilting pad seals. The compressors discussed in this paper are marked. They are comfortably within the range of experience. Without the tilting pad seals, the flexibility ratio would put the low pressure compressor just above and the other compressors just below Kirk-Donalds acceptable line.

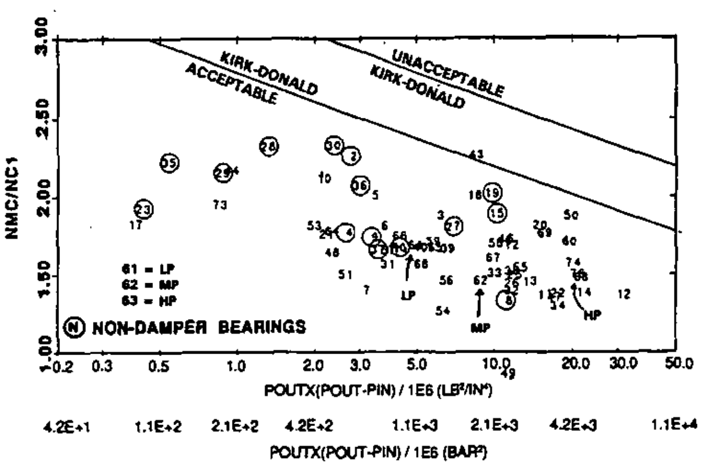


Fig. 7 - Flexibility Ratio With Tilt Pad Seals Vs. Discharge Pressure X (Discharge - Inlet Pressure)

If the discharge pressure is lower, the differential pressure through the compressor is lower, and the first critical is higher then it is harder to excite the first critical speed.

5. STABILITY ANALYSIS

Rotor dynamic stability analyses were made of the low, medium, and high pressure compressors. The analytical results are presented for the high pressure compressor.

The damped natural frequency program used to calculate the log dec after the input of the rotor model and bearing, oil-film seals, and toothed laby and honeycomb seal stiffness and damping coefficients was developed from the original paper by Lund (1974).

The stability program is used to produce a plot of the log dec of the first critical frequency vs. aerodynamic excitation at the mid span. The use of this plot as a tool in stability analysis of compressors is shown in many papers (Memmott, 1987, 1990, 1992, 1994, 1996; Marshall, et. al., 1993; Kuzdzal, et. al., 1994; Sandberg, et. al., 1994; Gelin, et. al., 1996).

5.1 Oil Film Seal Analysis

The cross coupled stiffness coefficients of the outer seal ring with the tilt pads in the seal is shown in Table 1. The cross coupled stiffness coefficients are considerably smaller than they would be without the tilt pads (Memmott, 1990, 1992, 1994, 1996), and thus the compressor will be more stable.

Tab. 1 - Outer Oil Ring Cross Coupled Stiff. Coeff.

CLEARANCE	KXY	KYX
	(N/M)	(N/M)
MINIMUM	2.26 E+6	-2.56 E+6
MAXIMUM	2.00 E+6	-3.10 E+6

The majority of the load on a tilt pad seal ring is carried by the pads (Memmott, 1990, 1992, 1996). In this case the pads carry 9O to 95% of the load. The load carrying capacity of the tilt pads is such that the tilt pad seal assembly acts like a tilt pad bearing, and not like a sleeve bearing.

The log decrement of the first critical as it responds to aerodynamic excitation (cross-coupled stiffness) at the mid span is plotted in Fig. 8. A study is made of the sensitivity of the compressor to arbitrary added amounts of cross-coupling at the mid span. The speed of the rotor is the maximum continuous speed. In Fig. 8 the effects of the toothed labys and honeycomb seal are not included.

Fig. 8 shows that that the effect of the outer ring is negligible, due to the tilt pads, and it customarily may be ignored in the analysis. This is true for both minimum and maximum clearance bearings and seals. For the rest of the analyses the outer ring is included.

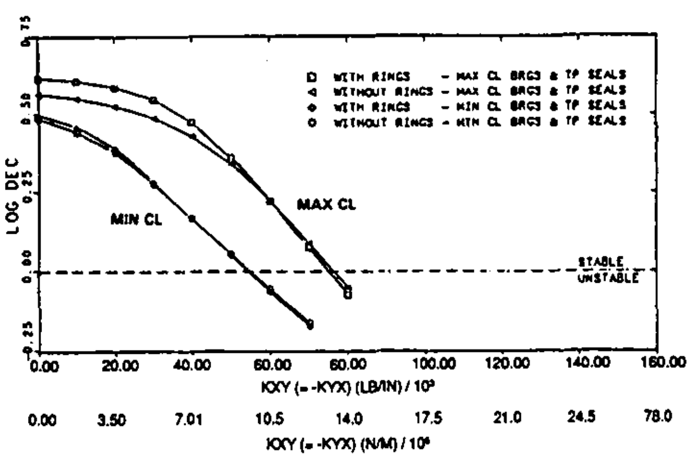


Fig. 8 - Log Dec Vs. Aero. Excitation With and Without Outer Ring Seal - No Labys

5.2 Toothed Laby & Honeycomb Seal Analysis

Toothed laby seals can provide significant stiffness and damping coefficients. They are more significant as pressures, differential pressures, and length of labys rises. The code (Kirk, 1990a) used to analyze the toothed laby seals is described in a series of papers (Kirk, 1985, 1990b).

The code used to calculate the coefficients of honeycomb seals was developed by Scharrer and Pelletti (1994-5). According to them, it matches the test data as taken at Texas A&M (Elrod, et . al., 1988).

Negative direct stiffness terms from the toothed labys are ignored. Compressor natural frequencies calculated using negative direct stiffness coefficients have not matched test data (Memmott, 1994; Gelin, et. al., 1996). A full pressure test done on a 400 BAR (5800 PSIA) discharge compressor yielded data which would say tha direct stiffness coefficients are positive for toothed labys (Memmott, 1994).

The direct stiffness coefficients have been measured to ba positiva in a test rig of a division wall toothed laby for pressures in the range 8 to 19 bar, with the higher the pressure and the higher the pressure differential the more positive is tha direct stiffness. (Private test for a contract). More tasting needs to be done to refine toothed laby codes. The direct damping terms from the toothed labys are small, but raise the log dec of tha system much in relation to their size, as is noted in (Childs, 1994). It would be conservative to not use them for toothed labys.

The calculated stiffness and damping coefficients for the toothed labys and honeycomb division wall seal are shown in Table 2. It would be advantageous to deswirl the impeller eyes, but this was not done. Due to the large amount of direct damping available from the honaycomb laby it was used at the division wall and not a toothed laby. There is test data (Childs, 1994, p. 339) which shows that honeycomb labys have much higher direct damping than toothed labys.

The division wall seal was analyzed as deswirled, because of the shunt holes. If the honeycomb seal at the division wall had not been deswirled the cross-coupled stiffness would be about double that shown, with not much change in the other coefficients.

Tab. 2 - Laby. Coefficients

	KXX	KXY	BXX
	(N/M)	(N/M)	(N-S/M)
IMP EYE *	7.29 E+4	2.57 E+6	2.45 E+3
EYE DESWIR *	-1.65 E+5	2.85 E+4	2.42 E+3
INTST SEAL *	-1.21 E+6	3.26 E+5	6.36 E+2
DW - TOOTH	-1.20 E+6	-9.26 E+5	5.31 E+2
DW - HONEY	1.60 E+7	2.03 E+7	9.02 E+4
2ND SEC IN	-5.13 E+6	-3.85 E+5	4.99 E+2

* AVERAGE

The four stability analyses made with the toothed labys and the honeycomb division wall seal and the one without the labys or honeycomb seal are shown in Table 3. The stability threshold is the amount of additional arbitrary cross-coupling needed at the mid-span to produce a zero log dec. The stability analysis with the inclusion of the toothed labys and toothed division wall laby, but without their small amount of calculated damping, shows instability without any added safety factors. This corresponds with the experience shown for a similar compressor (Memmott, 1994). If the damping from the toothed labys is included and the impeller eyes are not deswirled, stability is predicted with the toothed division wall laby, which does not correspond with recent experience. Deswirling the impeller eyes is a definite advantage, but the honeycomb division wall has so much damping that this is not necessary, which fits recent experience.

Tab. 3 - Log Dec & Stability Threshold

For Minimum to Maximum Clearance Bearings and Oil Seals	LOG DEC.	STABILITY THRESHOLD (N/M)
Without Labys & W/O Honeycomb Seal	.50/.62	9.51 E+6/ 1.31 E+7
Tooth DW, No Laby Damp, No Eye Deswirl	-.16/-.03	
Tooth DW, Laby Damp, No Eye Deswirl	.51/.78	9.32 E+6/ 1.13 E+7
Tooth DW, No Laby Damp, Eye Deswirl	.50/.71	9.49 E+6/ 1.21 E+7
Honey DW, No Laby Damp, No Eye Deswirl	3.0/3.8	6.41 E+7/ 6.23 E+7

In Fig. 9, a plot of log dec vs. arbitrary aero excitation at the midspan is made, comparing the system with a honeycomb division wall laby and no impeller eye deswirl to the system with a toothed division wall laby and impeller eye deswirl.

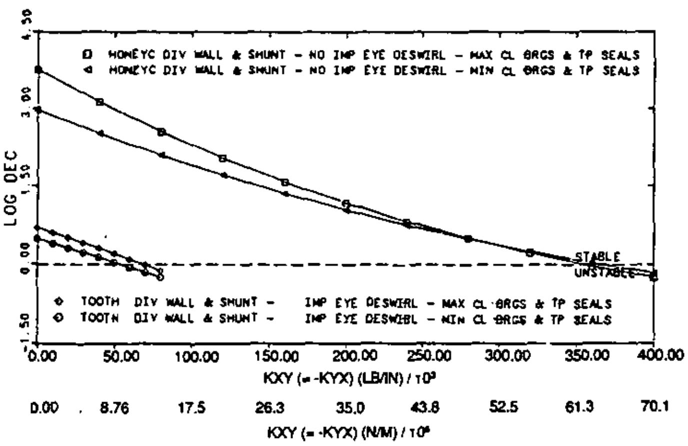


Fig. 9 - Log Dec Vs. Aero. Excitation with Laby Honeycomb Division Wall Laby & No Impeller Eye Deswirl vs. Toothed Division Wall Laby & Impeller Eye Deswirl

Since the shunt holes are included, the division wall is deswirled. No damping is assumed at the toothed labys. The log decs plotted vs. the horizontal axis are due to extra additional cross-coupled stiffness at the midspan besides that of the labys or honeycomb seal. The large log dec and resistance to aerodynamic excitation by use of the honeycomb seal is evident.

6. TESTING OF THE COMPRESSOR

Each compressor had an API-617 Mechanical Test with a ASME PTC 10 Class III Performance Test. There was a String Test with Full Load Full Pressure on Hydrocarbon Gas. This included the contract gas turbine. The full load, full pressure hydrocarbon test was completed in less than a week.

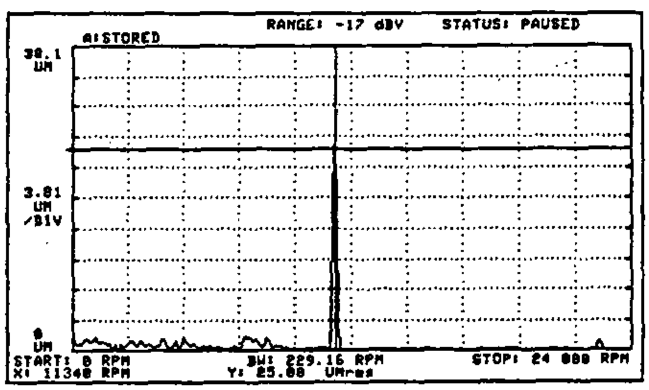
6.1 Mechanical and Performance Test

There were minor problems with the mechanical and performance tests. Post test inspection of the XX1Bs early on before the full load test showed evidence of rubbing of the honeycomb division wall laby at the tight end of the tapered bore. The clearance was increased slightly at the tight end (taper angle was changed) and no more rubs occurred. The XX1Bs performance tested slightly low in capacity. This was fixed by opening diffusers and return channel areas.

6.2 Full Load Full Pressure Hydrocarbon Test

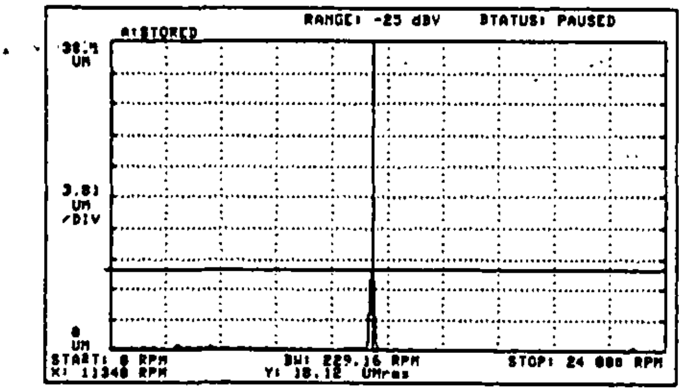
No evidence of subsynchronous vibration is seen at the first critical frequency or any other kind of subsynchronous. The decell plots will show that the first critical is so well damped it did not show up as a peak in response. This was true for all three compressors.

Figures 10-14 are from the four hour full load full pressure test. There are spectrum plots from near the end of the test for each compressor and a steady state plot for the four hours and a final decell plot for the high pressure compressor. The low and medium pressure compressors looked the same. The decell took 22 minutes, so the damped first critical is from the honeycomb seal.



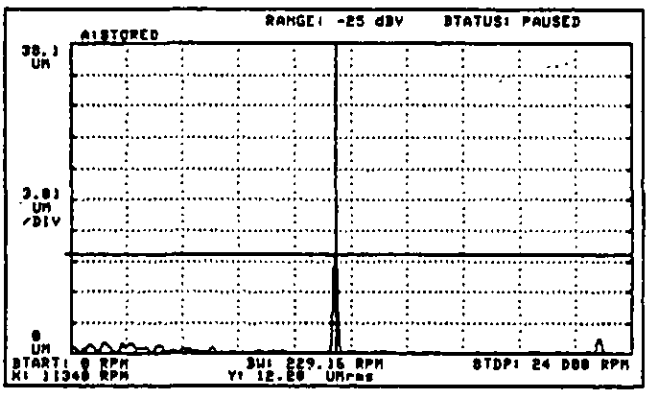
272B 181 End "B" Probe 16:00

Fig. 10 - Spectrum Plot - 272B4/4 Compressor



181B 191 End "B" Probe 16:00

Fig. 11 - Spectrum Plot - 181B3/3 Compressor



191 NDE "B" Probe 16:00

Fig. 12 - Spectrum Plot - 191B3/3 Compressor

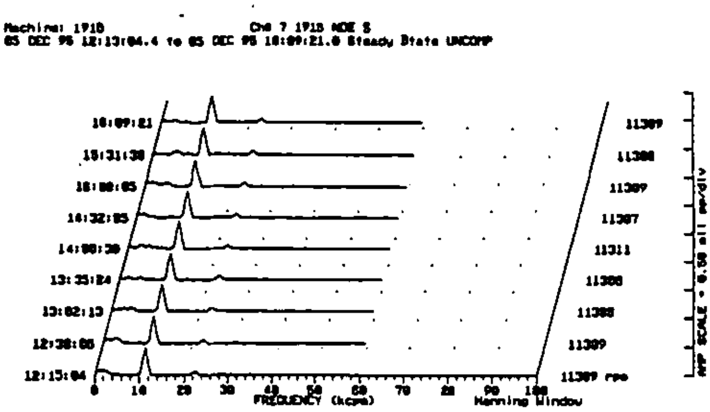


Fig. 13 – Steady State Plot – 191B3/3 Compressor

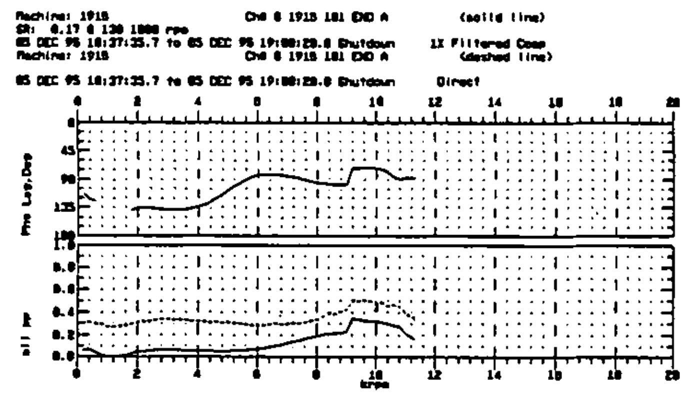


Fig. 14 - Final Decell - 191B3/3 Compressor

7. OTHER HIGH PRESSURE COMPRESSORS

There has been experience with the application of honeycomb division wall seals to other high pressure back to back compressors. These compressors also had shunt holes at the division wall, tilt pad seals, and damper bearings. They were of the same frame size and impeller bore, and all had longer seal spans, which is the important comparison with the tilt pad seals (Memmott, 1900, 1992). Besides the compressor described in (Memmott, 1994) they are:

In 1991 a ten stage back to back compressor with 285 BAR (4135 PSIA) discharge was shipped without full load full pressure testing. In the field there was subsynchronous at the first critical and the division wall was changed to honeycomb (still with shunt holes) in 1994 and the subsynchronous vibration disappeared.

In 1995 six trains of back to back compressors were shipped without full load full pressure testing. Honeycomb seals with shunt holes at the division wall were part of the initial design. They have since started, with discharge pressures from 243 to 312 BAR (3523 to 4524 PSIA) and all are free of subsynchronous vibration and performing well both aerodynamically and mechanically.

In 1997 a train of back to back compressors was shipped after a full load full pressure test on hydrocarbon gas of the high pressure compressor, with a discharge pressure of 381 BAR (5520 PSIA). There was no subsynchronous vibration.

In 1997 a back to back compressor was shipped after a full load full pressure Class 1 Hydrocarbon test, with a discharge pressure of 405 BAR (5875 PSIA). Again, there was no subsynchronous vibration.

8. CONCLUSION

Full load full pressure hydrocarbon testing of the high pressure compressors discussed in this paper, along with the same testing and field experience for other high pressure compressors, confirms the rotor dynamic stability analyses that shows that a honeycomb seal at the division wall supplies sufficient damping to ensure stable operation. Thus another weapon has been added to the time proven tools of tilt pad seals, damper bearings, and shunt holes to insure satisfactory operation of high pressure compressors.

9. ACKNOWLEDGMENTS

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