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DYNAMIC CHARACTERISTICS OF THE DIVERGING TAPER HONEYCOMB-STATOR SEAL

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ABSTRACT

This paper introduces a variant on the honeycomb-stator seal, which can extend the already strong stabilizing influence of this seal geometry for centrifugal compressors. The paper presents predicted and measured dynamic characteristics, demonstrating how a clearance, which diverges axially from inlet to outlet, increases the maximum effective damping of a honeycomb-stator seal, even though the average clearance is increased. The results also show a strong negative direct stiffness at zero and low frequency for this seal geometry (termed the "Diverging Taper Honeycomb Seal (DTHCS)"). The predictions are made with ISOTSEAL[1], software developed at The Texas A&M Turbomachinery Laboratory. The test data, also obtained at the Turbomachinery Laboratory, confirms the nature and magnitude of both these dynamic characteristics with close fidelity. However, measured leakage falls significantly below predicted leakage. The frequency dependence of the seal dynamic characteristics and the strong negative static stiffness require careful attention in rotor dynamics analysis.

KEYWORDS: *Honeycomb Seal; Centrifugal Compressor; Rotor Stability*

INTRODUCTION

Centrifugal compressors have demonstrated time and again their potential for subsynchronous vibrations excited by

[1] This code was developed within the Turbomachinery Laboratory, and is not a commercial product (not for sale).

working fluid forces within the gas flow path. Conditions which make this phenomenon more likely to occur include: high head, high density, high speed, high power (size), and high ratio of running speed to first natural frequency. While high discharge pressure characterizes some of the most well known and severe examples (e.g., Ekofisk, 530 bar), some large propane compressors in refrigeration service with discharge pressures of only 15 to 25 bar have raised concerns with evidence of self-excited subsynchronous vibrations.

A substantial amount of research has established that labyrinth seals within the compressor are a significant source of destabilizing excitation. Their potential for negative effective damping at the first natural frequency increases with high inlet swirl, with high ratio of running speed to first natural frequency, with flow length, and with high pressure differential across the seal. Either the labyrinth balance piston or the labyrinth center seal in back-to-back designs can make a particularly significant destabilizing contribution [1, 2]. Reducing inlet swirl to a labyrinth balance piston by means of swirl brakes or shunt holes has proved effective in reducing this destabilizing effect [3-5].

Alternatives to the labyrinth balance piston design have demonstrated their value, including the pocket damper seal or TAMSEAL [6, 7] and the honeycomb-stator seal [1]. The characteristic demonstrated by tests and analysis for the honeycomb seal has high effective damping that remains positive for a wider range of subsynchronous frequencies than the labyrinth seal.

The honeycomb-stator seal employs a pattern of closely spaced holes in the stator surface, of hexagonal shape (when viewed from directly above the surface). Derived from aircraft engine technology, the honeycomb-stator seal has some disadvantages from a manufacturing and procurement point of view, and the material hardness of the aircraft technology seals creates concerns regarding the result of a rub. The use of aluminum for construction and/or the use of closely spaced circular holes [8] (which has demonstrated comparable stabilizing capability) represent options to reduce these concerns.

The present paper addresses the honeycomb-stator seal and, in particular, a variant whose radial clearance increases linearly from inlet to outlet. It illustrates the geometry of the Diverging Taper Honeycomb Seal (DTHCS), and shows that adding a diverging taper in the flow direction to a honeycomb-stator seal can significantly enhance its positive damping characteristics. It also shows how the taper exaggerates the tendency for this seal to exhibit a negative stiffness at low and zero frequencies, and discusses the implications. The paper compares predictions and measurements of effective stiffness and damping for a DTHCS operating in air at different clearances, pressures, and rotational speeds with encouragingly close agreement.

NOMENCLATURE

$C_{effective}$	=	$C_{xx} - K_{xy}/w$.
C_{xx}	=	direct damping coefficient.
C_{xy}	=	cross-coupled damping coefficient.
D_i, h_i	=	diameter and flow passage width for wheel *i* for wheel *I*
r_D / r_S	=	the ratio of discharge to suction density for wheel *i*.
$(Horsepower)_{i,j}$	=	power for wheel *i* to compress stream *j*.
$K_{effective}$	=	$K_{xx} + w\,C_{xy}$.
K_{xx}	=	direct stiffness coefficient.
K_{xy}	=	cross-coupled stiffness coefficient.
$(K_{XY})_i$	=	cross-coupling for wheel *i.*
N_S	=	number of streams passing through.
RPM	=	speed, revolutions per minute.
w	=	angular velocity, rad/sec.

CONFIGURATION

Figure 1 shows a honeycomb-stator balance piston located behind the last stage in a large, straight through, centrifugal compressor designed for propane refrigeration service. The balance piston sets the radius at which pressure acting axially on the compressor transitions from discharge pressure to suction pressure. Since flow through the balance piston seal must return via an internal or external line to compressor inlet, the compressor's performance is influenced significantly by the amount of gas at discharge pressure, which flows through the balance piston as leakage.

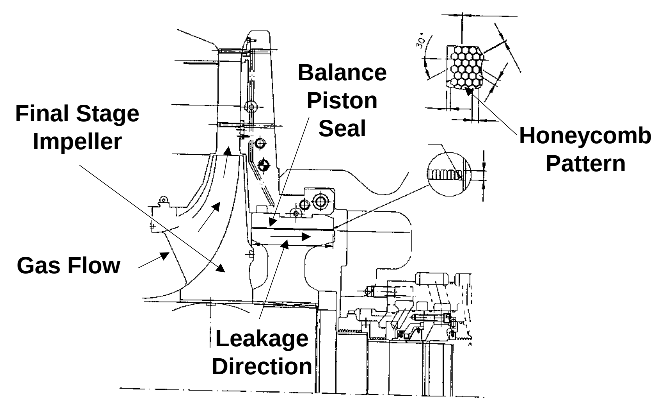

Figure 1. Side Elevation of Honeycomb-Stator Balance Piston Seal for Large Centrifugal Compressor in Propane Refrigeration Service

Figure 2 shows schematically how a honeycomb-stator seal with a diverging taper differs geometrically from a constant clearance ("straight") honeycomb-stator seal; the main difference is that the clearance at exit is higher than the clearance at inlet–typically by a factor of two or more. As will be shown, the beneficial result of the taper is to increase effective damping of the seal–even when the taper increases the seal's mean clearance. An intuitive rationalization is that the diverging taper concentrates the axial pressure drop near the inlet. Radial eccentricity of the rotating element relative to the stator causes the axial pressure distribution to vary circumferentially. Concentrating the axial pressure variation near the inlet also ensures concentrating this circumferential pressure variation near the inlet, and so the net radial pressure forces can act over a larger axial length of the seal. The circumferential acoustic response of gas film pressure to normal relative velocity of stator and rotor has a strong influence on the honeycomb-stator seal's damping. The honeycomb-stator cell depth directly affects the effective compliance of the gas volume, and thereby the acoustic response. Again, concentrating axial pressure drop near seal inlet maximizes the sensitivity of film forces to the circumferential acoustic response.

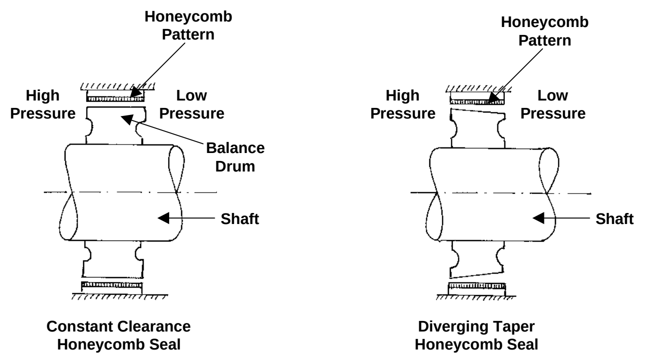

Figure 2. Schematic Comparison between Straight and Diverging Taper Honeycomb-Stator Seals

Table 1 shows the dimensions of a seal configured for evaluation in an existing test rig available at Texas A&M University's Turbomachinery Laboratory. A realistic drawing cannot illustrate such a small clearance and taper, but this table makes clear the essential dimensions. The test sought to increase confidence in predicted characteristics of a DTHCS designed as an option for the propane compressor, with the balance piston shown in Fig. 1. The option was to be used should subsynchronous vibration be excessive in the compressor. This large propane compressor had a balance piston with diameter close to 900 mm and a length of over 200 mm, with upstream pressure of over 20 bars, and an exit pressure of about 1 bar. The test seal is naturally smaller, but maintains a pressure ratio, surface speed, and ratio of inlet to outlet clearance similar to the target application. Table 2 presents the matrix of speeds and inlet pressures planned for the test.

Using the test seal as a reference, the next sections of the paper will provide quantitative data on how diverging taper influences damping and stiffness, followed by confirmatory evidence based on testing.

Table 1. Key Dimensions of Test Seals

Length	85.71 mm	85.71 mm
Test Shaft Diameters	114.295 mm	114.071 mm
Inlet Clearance	0.1198 mm	0.2316 mm
Outlet Clearance	0.2158 mm	0.3276 mm
Cell Depth	2.6 mm	2.6 mm
Cell Face Spacing	1.6 mm	1.6 mm

Table 2. Planned Test Condition Matrix

Test Case	Speed (RPM)	Upstream Pressure (Bar)	Surface Speed
1	12,000	8	71.95
2	12,000	17	71.95
3	17,000	8	101.9
4	17,000	17	101.9
5	22,000	8	131.9
6	22,000	17	131.9

DYNAMIC CHARACTERISTICS

The dynamic characteristics of the test seal have been predicted using the computer program ISOTSEAL developed by the Texas A&M Turbomachinery Laboratory [9]. This isothermal analysis uses a bulk flow solution to the momentum and continuity equations. A zero order concentric solution establishes the axial pressure distribution, and then linear perturbation provides complex derivatives of pressure with respect to radial displacement of the rotating element for a range of frequencies. The pressure derivatives appropriately integrated over the surface area provide stiffness and damping coefficients. The presence of the holes in one surface of the film adds to the effective compressibility over that which would be achieved without the holes. This essentially reduces the speed of sound in the axial and circumferential film-wise directions and, thereby, has a significant influence on the dynamic stiffness and damping characteristics and their variation with frequency.

Figure 3 shows the variation of effective damping with excitation frequency for both the test DTHCS, and for a straight seal with the same clearance at inlet. The definition of effective damping is as follows:

$$C_{effective} = C_{xx} - K_{xy}/w \qquad (1)$$

Thus, in this definition of effective damping, the direct damping is reduced by a contribution from the destabilizing cross-coupling of the seal.

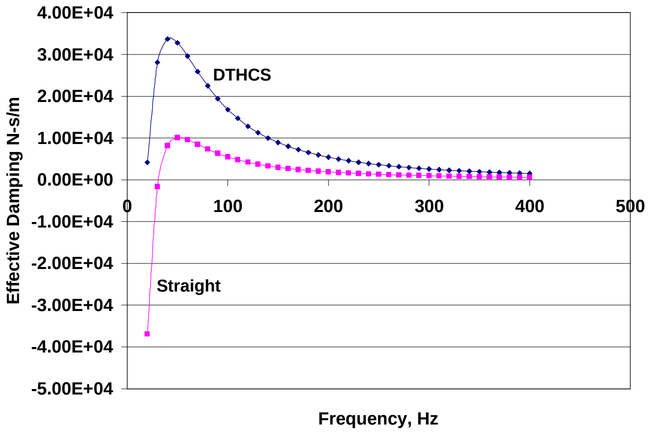

Figure 3. Predicted Effective Damping of Test Configuration Diverging Taper Honeycomb-Stator Seal compared to a Straight Seal; 22,000 RPM; 17 Bar Upstream Pressure; 1 Bar Downstream Pressure

Conditions for the analysis of Fig. 3 are 22,000 RPM, 17 bar supply pressure, and 1 bar discharge pressure. In spite of the fact that the diverging taper seal has a mean clearance, which is one and a half times that of the straight seal, its predicted damping is higher by a factor of three or more for most of the frequency range. Both straight and diverging taper seals have a reasonably broad damping peak, which should make it possible to put the optimum damping near the subsynchronous frequency where it is needed most. In addition, the frequency at which effective damping transitions from positive to negative is at least 30 percent lower for the diverging taper seal when compared to the straight seal. The target application has an operating speed of 3600 RPM, so the frequency range of highest positive damping (1500 CPM; 25 Hz) is about where the first natural frequency is likely to occur.

Figure 4 shows how the effective stiffness varies with frequency, both for the DTHCS, and for the straight seal. Effective stiffness is defined as follows:

$$K_{effective} = K_{xx} \quad + \quad w \;\; C_{xy} \qquad (2)$$

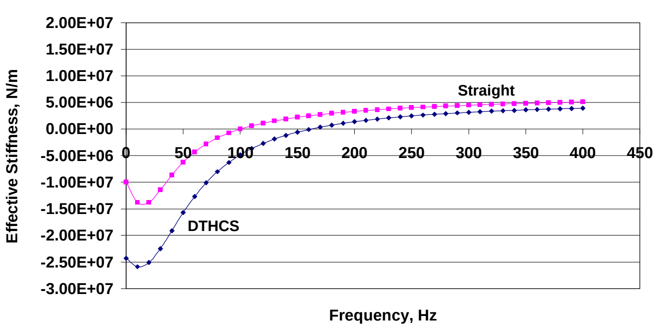

Figure 4. Predicted Effective Stiffness of Test Configuration Diverging Taper Honeycomb-Stator Seal compared to a Straight Seal

In Fig. 4, we see that both straight and tapered seals have a negative effective stiffness over a wide band of frequencies below 150 Hz. Of particular relevance is the negative stiffness at zero frequency, which has nearly 2.5 times the magnitude for the DTHCS compared to the straight seal. This negative stiffness can influence the static deflection of the rotor in it supports, and for a sufficiently soft rotor system could contribute to static instability of the system. It will be shown that the rotor bearing system for the target straight-through compressor had substantial margin relative to static instability, but this is an important question to address in any application of the honeycomb-stator seal, straight or tapered. An additional concern is a drop in the first critical speed to a sufficiently low frequency that the seal's effective damping is negative. Camatti et al [10] recently reported an instability on a back-to-back compressor caused by an unintentional divergence in the center division-wall seal that induced a strong negative stiffness leading to a reduced natural frequency and unstable compressor.

Figure 5 presents information on how the depth of the honeycomb-stator cell is predicted to influence effective damping of the DTHCS. Clearly, there is a peak damping value which increases as the cell depth is increased. However, the peak becomes narrower and the frequency of the peak becomes lower–making very clear the need to match the frequency range of increased damping to the frequency range of concern for subsynchronous vibration in the specific application. The choice of 2.6 mm cell depth for this application provides a good balance of high damping and broad frequency coverage for that high damping.

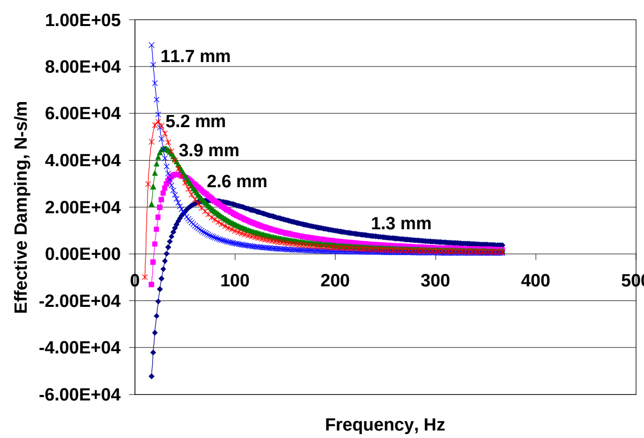

Figure 5. Damping of the Test Configuration Diverging Taper Honeycomb-Stator Seal

TESTING AND TEST RESULTS

Figure 6 shows the test rig, which has been described in detail elsewhere [11]. In summary, a stiff rotor is mounted in very stiff hydrostatic bearings. High pressure air is supplied to a central feeding groove between the sealing gap of two nominally-identical test seals. To achieve the high pressure ratio across the seal, required for this configuration, the gas exhausts to atmosphere for these tests. The seal's stator is driven in two orthogonal directions by hydraulic shakers, which apply a force measured by load cells and adjusted to achieve relative motion between stator and rotor of about one-tenth the clearance. Motion of the stator relative to the rotor is measured by eddy current displacement probes at both ends of the seal, and in the same orthogonal directions as the planes in which forces are applied. Stator motion is constrained to be parallel to the rotor by stabilizing bars that add a small amount of tare stiffness to the assembly. Accelerometers on the stator measure its acceleration along the orthogonal x and y axes. The measurements of input excitation force, relative rotor-stator displacement, and stator acceleration vectors over a range of excitation frequencies provides the information necessary to calculate complex dynamic coefficients matrix entries for a seal. Tare complex dynamic stiffness coefficients for the assembly to account for the exit labyrinth seals, hoses, etc. are obtained by testing with a seal pair with greatly-enlarged clearances. The tests produce frequency-dependent direct and cross-coupled stiffness and damping coefficients. The contributions of a single seal are simply half of the totals for the dual seal assembly.

Table 3 presents the specific characteristics of the seal as tested, including supply pressure, speed, low or high clearance, and summary predictions of how the straight and tapered configuration would perform under these conditions. The seal was tested at two clearances, three speeds, and three pressures, an almost complete combination. The highest supply pressure combined with the tightest clearance produced a negative stiffness strong enough to overcome the hydraulic shakers and pull the stator hard against the rotor. For reduced supply-pressure cases, the shaker stiffness was sufficient to overcome the seal's negative stiffness. The clearance values used in these predictions are the result of averaging diameter and bore measurements at a series of angles to account for small out of roundness in the stator. Specifically, the inlet and exit "high" radial clearances are 0.2316 mm and 0.3276 mm; the inlet and exit "low" clearances are 0.1198 and 0.2158 mm.

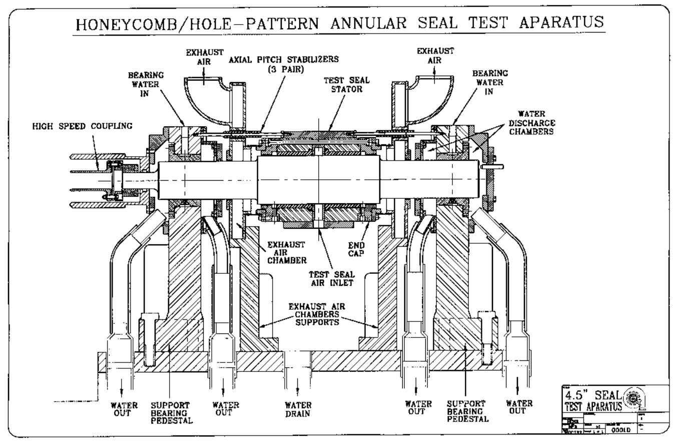

Figure 6. TAMU Turbomachinery Lab Air Seal Test Rig

Table 3. Summary of Operating Conditions and Predicted Characteristics for Seal as Tested

Psupply	Speed	Inlet Clearance	Straight Keff at 20 Hz	Diverging Keff at 20 Hz	Straight Ceff, Max	Diverging Ceff, Max	Straight Frequency of CeffMax	Diverging Frequency of CeffMax	Straight Flow Kg/Sec	Diverging Flow Kg/Sec
7.841	10200	High	-1.65E+06	-2.76E+06	2770	3260	80	80	0.0569	0.0768
7.841	17000	High	-1.63E+06	-3.43E+06	1670	2880	100	100	0.0569	0.0742
7.841	22000	High	-1.48E+06	-3.37E+06	1090	2250	130	110	0.0569	0.0742
18.27	10200	High	-2.35E+06	-6.91E+06	4650	8040	90	80	0.1423	0.1851
18.27	17000	High	-3.90E+06	-6.70E+06	3990	5310	110	110	0.1377	0.1851
18.27	22000	High	-3.69E+06	-8.04E+06	2710	5310	130	110	0.1377	0.1790
7.25	10200	Low	-2.90E+06	-7.55E+06	6110	14400	40	30	0.0202	0.0309
7.25	17000	Low	-2.83E+06	-7.79E+06	2300	10500	60	50	0.0202	0.0309
7.25	22000	Low	-4.62E+06	-7.61E+06	2910	8070	60	50	0.0191	0.0309

Figures 7, 8, 9, and 10 compare measured and predicted values for effective damping and effective stiffness under four conditions of Table 3, and show consistently good agreement both in magnitude and in characteristics. In particular, the frequency and magnitude of the peak in effective damping are well predicted, as are the trends to negative stiffness and damping at low frequency. The agreement remains good over a wide range of conditions for the different clearance values, for different inlet pressures, and for different speeds.

In addition to stiffness and damping coefficients, leakage was measured under all conditions. Table 4 compares measured leakage with predicted leakage under several test conditions, and consistently shows measured leakage below predicted by an amount in the range of 20 to 30 percent. For large compressors with high pressure ratios, since leakage can significantly impact efficiency and throughput, the low leakage discrepancy is in the preferred direction. A contributory factor to this discrepancy is the existence of choking or sonic velocity at some axial position in the seal. With a diverging taper, leakage is clearly influenced by the distance along the axial flow path at which choking sets in. While not explaining the discrepancy, leakage data in Table 3 shows that treating the seal as having a constant clearance equal to the inlet clearance produces flow predictions, which match the measured data much more closely than the Table 4 data. Using a constant clearance solution just for leakage predictions may provide a working approach to bounding the flow uncertainty for balance pistons with diverging taper and high pressure ratios across them.

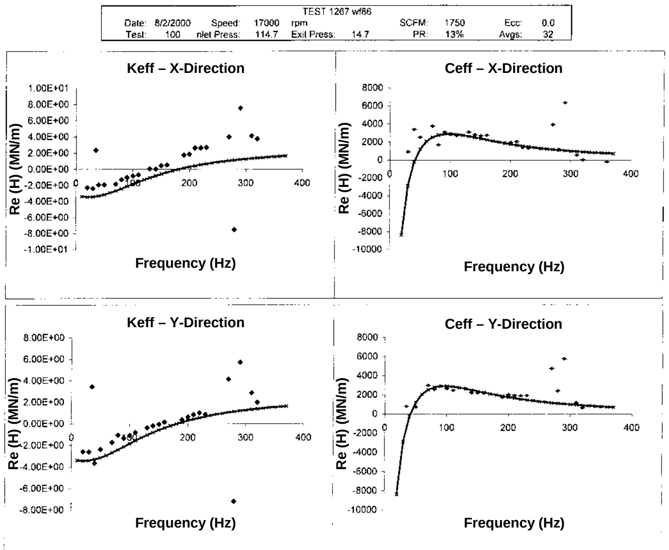

Figure 7. Comparison of Predicted and Measured Characteristics for the DTHCS at 22,000 RPM under 7.841 Bar Supply Pressure with High Clearance

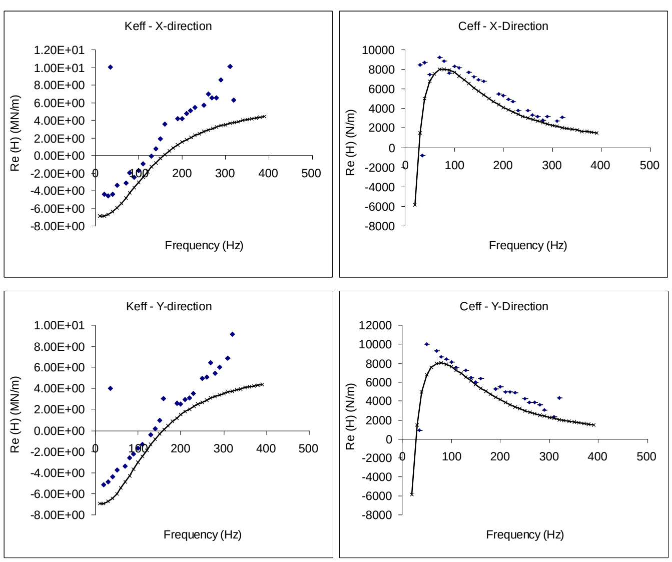

Figure 8. Comparison of Predicted and Measured Characteristics for the DTHCS at 10,200 RPM under 18.27 Bar Supply Pressure with High Clearance

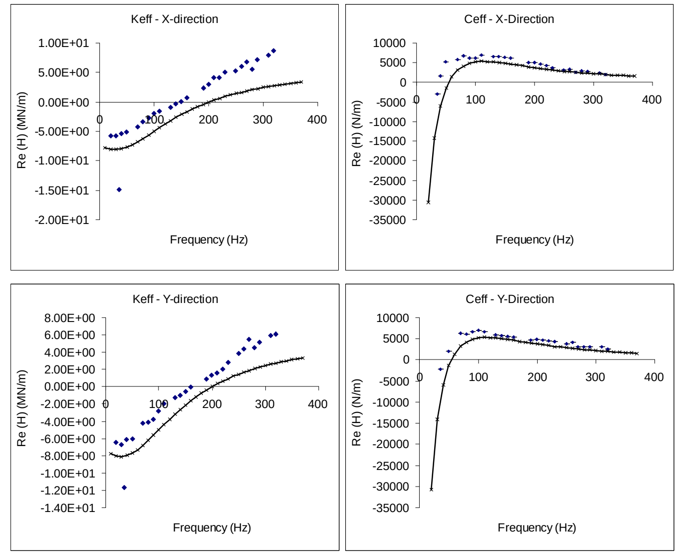

Figure 9. Comparison of Predicted and Measured Characteristics for the DTHCS at 22,000 RPM under 18.27 Bar Supply Pressure with High Clearance

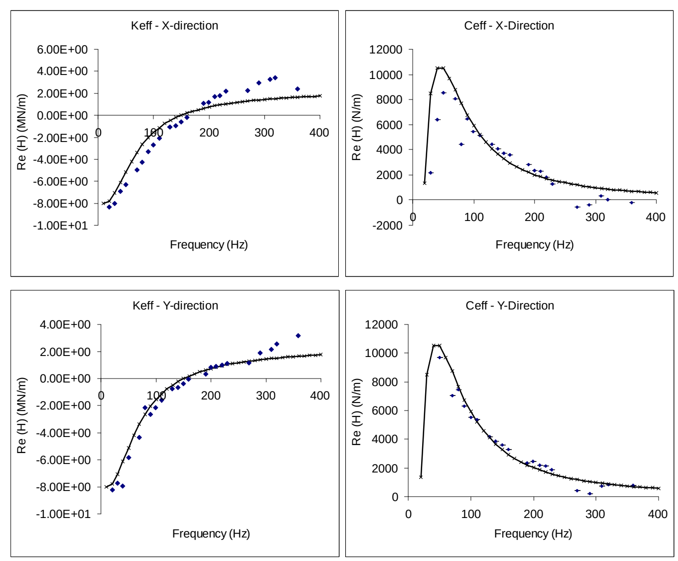

Figure 10. Comparison of Predicted and Measured Characteristics for the DTHCS at 17,000 RPM under 7.8 Bar Supply Pressure with Low Clearance

Table 4. Comparison of Measured and Predicted Leakage

Pressure, Bara	Speed, RPM	Measured, kg/sec	Predicted, kg/sec
7.841	10,200	.0626	.0768
7.841	17,000	.0620	.0742
7.841	22,000	.0606	.0742
18.27	10,200	.1409	.1851
18.27	17.000	.1378	.1851
18.27	22,000	.1332	.1789

FULL-SCALE ROTOR SYSTEM PREDICTIONS

Figure 11 compares the log decrement for the first natural frequency of the target 65 Mw compressor, predicted for several different balance piston configurations, including a labyrinth seal, a labyrinth with straight and reverse flow shunt holes, a straight honeycomb-stator seal, and a diverging taper honeycomb-stator seal (optimized for this application). Figure 11 shows that the least stable or most negative log decrement results from the conventional labyrinth. The straight honeycomb-stator design makes the predicted log decrement less negative. However, for this example, only the diverging taper honeycomb-stator seal provides a prediction of positive log decrement. This compressor in question has three side streams, which combined with the first stage suction stream, lead to gas compression through four different pressure ratios from individual stream suction to final discharge. The highest of these pressure ratios is close to 20 –between first suction and final discharge. This leads to a similar pressure ratio across the balance piston. Damped natural frequencies were predicted for this compressor, accounting for aerodynamic excitation based on the following adaptation of the Wachel [12] method to account for the side streams:

$$(K_{XY})_i = 63,000 * \frac{Mole\ Weight}{10} \sum_{j=1}^{N_S} \frac{(Horsepower)_{i,j}}{RPM\ *\ D_i\ *\ h_i} \left(\frac{r_D}{r_S}\right)_j \qquad (3)$$

for wheel i, streams $j = 1$ to $(N_S)_i$.

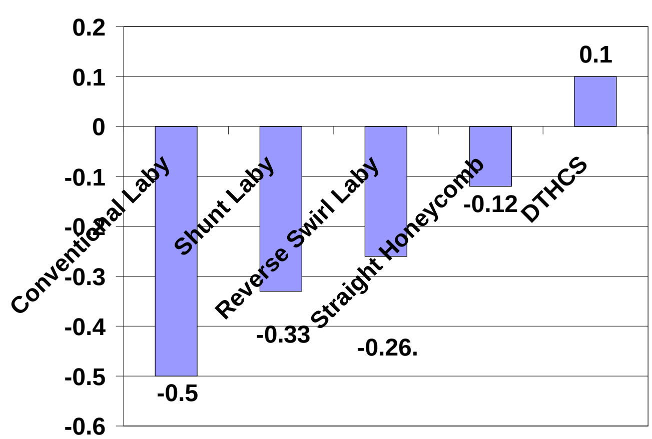

Figure 11. Comparison of Predicted Log Decrement for Various Configurations of Balance Piston Seal in Target Propane Compressor

As predicted and confirmed by test, the DTHCS also develops a strong static negative stiffness. For the target application, the static deflection of the rotor in response to an external force of 4454 Newtons, applied at the balance piston location, is 15.06 microns without the addition of this negative stiffness; with the negative stiffness included, the deflection increases to 21.84 microns. The negative stiffness of the DTHCS essentially acts in parallel with the positive bearing stiffness; the effective system stiffnesses with and without the negative stiffness contribution are 296 and 204 N/m, respectively, indicating about 30% reduction caused by the DTHCS. Thus, while the influence of the negative stiffness is apparent, the system maintains a strongly positive system stiffness when the negative stiffness contribution of the diverging taper honeycomb-stator seal is accounted for. Any application of the DTHCS should include a comparable check on its possible contribution to system static stability. In addition, the first damped natural frequency prediction must include all honeycomb dynamic coefficients, evaluated at the natural frequency. This may require an iterative process, since lowering the natural frequency makes the stiffness more negative, which in turn further lowers the natural frequency!

The diverging taper honeycomb-stator seal was maintained as an option if the full load string tests of the target propane compressor should indicate a need for it. The balance piston configuration used in the string tests was a labyrinth with reverse flow shunt holes–Case 3 in Fig. 11. The propane compressor was tested over a range of conditions and exhibited small subsynchronous vibrations, but these never exceeded 5 microns, leading to a decision that the reverse flow shunt hole labyrinth would not be replaced with the diverging taper honeycomb-stator seal, since performance and vibration characteristics were acceptable without it.

CONCLUSIONS

The preceding results lead to the following conclusions:

- The diverging taper honeycomb-stator seal offers distinct advantages in achievable, effective, damping relative to a straight or constant clearance honeycomb-stator seal with the same minimum clearance.
- The peak effective damping can be a factor of two or more higher than that for the straight seal.
- The diverging taper honeycomb-stator seal has a distinct negative static stiffness, which must be considered when designing for a particular application; other positive stiffnesses acting in parallel on the rotor must be sufficient to provide a net positive system stiffness. In addition, care must be taken that the first critical speed does not drop excessively, creating a negative effective damping for the seal at the critical-speed frequency.
- For the particular application considered, the influence of negative stiffness from the balance piston reduces positive system stiffness by about 30 percent; thus, the system preserves positive static stability and the effective damping is positive, large, and effective.
- Test results confirm that the computer program, ISOTSEAL, predicts effective stiffness and effective damping of the DTHCS with good accuracy over a

range of conditions, sufficient for use as an engineering tool.

- For the conditions tested (very high pressure ratio across the seal), measured leakage is consistently lower than predicted leakage, by an amount in the range of 20 to 30%.

- Although physically inconsistent, predicted leakage with constant clearance equal to the minimum clearance of the diverging taper seal comes much closer to the measured data; by combining this with the leakage prediction for a diverging taper seal, the uncertainty in leakage may be bounded for these high pressure ratio conditions.

- Analytic predictions, research testing, and finally an industrial full load test [10] clearly show the potential impact of tapering on honeycomb seal dynamic coefficients. In addition to intentional tapering, the possibility of unintended tapering induced by structural deformation must be considered during the design phase.

- Both seal and balance drum deformation under operating loads must be evaluated for possible undesirable tapering causing an excessive negative stiffness and associated reduction of the first natural frequency. If so, corrective action must be taken to eliminate, reduce, or compensate for the structural deformation.

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REFERENCES

[1] Childs, D. W., Jordan, L. T., and Vance, J. M., 1997, "Annular Gas Seals and Rotordynamics of Compressors and Turbines," *Proc. 26th Turbomachinery Symposium*, Houston, TX, pp. 201-220.

[2] Memmott, E. A., 1996, "Stability of an Offshore Natural Gas Centrifugal Compressor," 15th Machinery Dynamics Seminar, CMVA, Banff, Alberta, Canada.

[3] Memmott, E. A., 1994, "Stability of a High Pressure Centrifugal Compressor Through Application of Shunt Holes and a Honeycomb Labyrinth," 13th Machinery Dynamics Seminar, CMVA, Toronto, Canada.

[4] Memmott, E.A., 1998, "Stability Analysis and Testing of a Train of Centrifugal Compressors for High Pressure Gas Injection," International Gas Turbine & Aeroengine Congress & Exhibition, Stockholm, Sweden, ASME Paper No. 98-GT-378.

[5] Miller, H. F., 2002, "Centrifugal Pipeline Compressor Technology Evolution-High Pressure Pipeline Applications," GMRC Gas Machinery Conference, Nashville, Tennessee.

[6] Vance, J. M. and Schultz, R. R., 1993, "A New Damper Seal for Turbomachinery," ASME 14th Biennial Conference on Mechanical Vibration and Noise, Albuquerque, NM, DE-Vol. **60**, pp. 139-148.

[7] Vance, J. M. and Li, J., 1996, "Test Results of a New Damper Seal for Vibration Reduction in Turbomachinery," J. of Engineering for Gas Turbines and Power, **118**, pp. 843-846.

[8] Yu, Z., and Childs, D. W., 1998, "A Comparison of Experimental Rotordynamic Coefficients and Leakage Characteristics Between Hole-Pattern Gas Damper Seals and a Honeycomb Seal," ASME J. of Engineering for Gas Turbines and Power, **120**, No. 4, pp. 778-783.

[9] Kleynhans, G. F., and Childs, D. W., 1997, "The Acoustic Influence of Cell Depth on the Rotordynamic Characteristics of Smooth-Rotor/Honeycomb-Stator Annular Gas Seals," ASME J. of Engineering for Gas Turbines and Power, **119**, No. 4, pp. 949-957.

[10] Camatti, M., Vannini, G., Fulton, J. W., and Hopenwasser, F., 2003, "Instability of a High Pressure Compressor Equipped with Honeycomb Seals," *Proc. Thirty-Second Turbomachinery Symposium*, pp. 39-48.

[11] Childs, D., and Hale, K., 1994, "A Test Apparatus and Facility to Identify the Rotordynamic Coefficients of High-Speed Hydrostatic Bearings," ASME J. of Tribology, **116**, pp. 337-344.

[12] Wachel, J. C. and von Nimitz, W. W., 1980, "Assuring the Reliability of Offshore Gas Compression Systems," European Offshore Petroleum Conference and Exhibition, London, England, EUR 205.

