EXPERIMENTAL AND ANALYTICAL LEAKAGE CHARACTERIZATION OF ANNULAR GAS SEALS: HONEYCOMB, LABYRINTH AND POCKET DAMPER SEALS

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ABSTRACT

Modern day turbomachinery requires the use of annular gas seals to provide flow restriction from high pressure to low pressure regions within the machine. These flow restrictions are critical design points in the overall performance of the machine and directly impact the system-level efficiency. Consequently, understanding the leakage performance of a given seal element as a function of operating pressure, rotor speed, and rotor offset is critical to the successful design of the turbomachine. In the present work, three annular gas seals are experimentally tested on a leakage test rig at GE Global Research (Niskayuna, New York). The test rig is capable of high-speed, high-pressure flow testing and has a radial degree of freedom that enables nonconcentric leakage characterization. The leakage performances of a labyrinth, honeycomb and pocket damper seals are compared over a range of inlet pressures and pressure ratios. Analytical tools, including a CFD model and a Bulk Flow Code, are developed to provide leakage prediction and to establish understanding of underlying flow mechanisms. Predictions of the seal leakage are found to be in good agreement with experimental data.

INTRODUCTION

Annular gas seals are important turbomachinery components used to restrict leakage flow between a rotating part and a stationary housing. However, annular gas seals can also have significant impact on the rotordynamic characteristics of turbines and compressors [1]. The increasing demands for rotordynamic stability and machine reliability has led to the development of seals that augment the damping properties of the system. It is important that the designer properly evaluate the leakage performance of various sealing concepts in order to reach an optimum balance between efficiency and stability. This article aims to study the leakage characteristics of three types of annular seals: the more commonly used labyrinth seals and honeycomb seals, as well as pocket damper seals.

Labyrinth seals are made of a series of teeth and cavities. The annular restrictions formed by the seal teeth cause the working gas to throttle and then expand repeatedly, hence reducing the total pressure of the gas and the overall leakage rate. Labyrinth seals exist in many forms, from straight-toothed to angle-toothed to stepped configurations. In general, as discussed in Chupp et al. [2], the angle at which the flow usually approaches the seal throttle is 90°; however, throttles that slant into the flow are typically more effective seals. While labyrinth seals are most widely used, they have certain undesirable rotordynamics characteristics related to instability; in addition, labyrinth seals offer limited damping of rotor vibrations [3]. By definition, honeycomb seals are considered to be part of the labyrinth seal family. The rough honeycomb surface acts to increase resistance to passing flow, thereby reducing the leakage. Previous investigations of annular honeycomb seals have demonstrated improved leakage performance over straight-through labyrinth seals, as well as excellent rotordynamic characteristics [4, 5].

Pocket damper seals (PDS) have recently been used in centrifugal compressor applications to provide damping to rotor vibration and to increase rotordynamic stability [6]. A PDS consists of axially spaced blades with baffle walls inserted circumferentially between paired blades. The main mechanism for rotor-bearing stability enhancement is the maximized damping capacity stemming from the diverging clearance geometry. Ertas and Vance [6] summarized the evolution of the PDS design and discussed the advantage of a new type of damper seal, a fully partitioned damper seal (FPDS).

The objective of this work is to evaluate and compare the leakage performance of the three types of annular seals: labyrinth, honeycomb and pocket damper seals. First, leakage measurements of the seals are presented over a range of inlet pressures and pressure ratios. Seal effective clearances are then calculated for direct performance comparison among various sealing configurations. In addition, experimental data are used to validate analytical tools, including a CFD model and a Bulk Flow Code, developed for efficient seal leakage prediction.

DESCRIPTION OF TESTING APPARATUS

All testing was conducted on a leakage test rig at GE Global Research (Niskayuna, NY). The goal of the various tests conducted was to characterize the leakage behavior of the labyrinth, honeycomb and pocket damper seals as a function of differential pressure up to 27.6 bar. The following subsection details the design of the leakage test rig and its supporting infrastructure.



Figure 1. PHOTOGRAPH OF THE CSR USED TO CONDUCT THE LEAKAGE TESTS: 1) INLET SUPPLY, 2) EXHAUST TO FLOW METER, 3) MOTOR HOUSING, 4) RADIAL ACTUATOR, AND 5) SEAL BODY.



Figure 2. CROSS-SECTIONAL VIEW OF THE CSR TEST SETUP: 1) INLET SUPPLY, 2) EXHAUST TO FLOW METER, 3) MOTOR HOUSING, 4) RADIAL ACTUATOR, 5) SEAL BODY, AND 6) TEST ROTOR.



Figure 3. SCHEMATIC OF LEAKAGE PATHS PRESENT IN THECSR TEST SETUP.

Seals' Test Platform: Cold Seal Rig (CSR)

The main workhorse for testing the labyrinth and honeycomb seals was the Cold Seal Rig, also known as the CSR. The CSR is intended for leakage and wear testing of rotating seals. The rig consists of a rotor, spindle, and housing. These assembled components are mounted to a machine base with two degrees of freedom, axial and horizontal radial travel. The axial travel is adjusted manually using a precision lead screw, while the radial offset is actuated via an automated subroutine in the LabView software. A photograph of the CSR setup for the tests is provided in Figure 1. The specifications for the rig are as follows:

- Maximum air inlet pressure: 34.5 bar g at supply temperature
- Maximum air inlet temperature: 60°C
- 170mm rotor, 31.75 mm diameter shaft

• Maximum rotational speed: 20,000 rpm (150 m/sec surface speed)

In the present setup, compressor air was supplied to the inlet plenum of the housing via a manifold and a total of three 19 mm diameter lines (two of which can be seen in Figure 1). From the inlet plenum there are only two leakage paths: 1) through the test seal; and 2) through a dedicated labyrinth seal that is part of the test rig housing. A cross-sectional view of the CSR setup is shown in Figure 2, and a schematic of the two primary leakage paths is shown in Figure 3.

Figure 3 depicts the primary leakage paths for clarity. To independently identify and characterize the leakage through the test seal, a vortex flow meter was installed downstream of the test seal. A 50.8 mm diameter NPT fitting and hose, as shown in Figure 1, connected the seal leakage to the flow meter. Consequently, estimation of the bias leakage flow out the backside of the rig did not have to be taken into account throughout the tests. Also, as shown in Figure 3, the test setup enables the installation of various pre-swirl rings to condition the inlet swirl within the supply plenum to the seal. Presently, all tests have been conducted with zero pre-swirl.



Figure 4. VIRTUAL INSTRUMENT INTERFACE FOR DATA ACQUISITION AND CONTROL OF THE CSR.

Table 1. SUMMARY OF SENSORS USED TO MONITOR THE SEAL TESTS IN THE CSR.

Description	Signal Type	Input	Accuracy
		Range	
Omega Engineering Type	mV	0 to 150°C	± 0.5 °C
T thermocouple			
Scanivalve DSA3218	Ethernet	0 to 35 bar	$\pm0.05\%$ FS 1
	IEEE 802.3		
Rosemount 8800D Vortex	4-20 mA	0 to 6.55	± 1.35% of
Flow Meter		cfm	reading
Monarch Instruments	4-20mA	0 to 250,000	$\pm0.0015\%$ of
ROS-W optical encoder		rpm	reading
Sensotec LVDT Model	± 5 V	± 2.5 mm	± 0.5% FS
S7C			

¹ FS: full scale

Data Acquisition System and Sensors

Various operating conditions were monitored throughout the test sequence via a National Instruments SCXI-1000 4-slot data acquisition chassis. The two primary input modules for acquisition are the SCXI-1112 (8-channel) thermocouple input card and the SCXI-1102C card with an SCXI-1308 (32channel) analog input front plate adapter. The SCXI-1000 platform is capable of 3.33×10^5 samples per second aggregate. The system interfaces with a Labview 8.2 executable file. At a sampling rate of 0.25 Hz, the chassis takes a data point from all available inputs (thermocouples, current readings, and voltage readings), and records them in an Excel spreadsheet for postprocessing. The user interface is shown in Figure 4.

The inlet plenum temperature and pressure, housing pressure and temperature, rotor speed, and exhaust plenum pressures and temperatures were constantly monitored by the CSR data acquisition system. Table 1 outlines the main sensors, their output signal, and stated accuracies as supplied by the vendor. Using the sensed feedback from the SCXI chassis, the leakage of a given test seal could be easily quantified and the effective clearance of the seal could be computed.

DESCRIPTION OF ANALYTICAL APPROACHES (1): CFD MODEL

The performance of 'Fully Partitioned Damper Seal' (FPDS, [6]) was evaluated using CFD and compared against experiments. The objective was to validate CFD and use it as a design tool for faster conceptual evaluation.

CFD Domain and Grid

The FPDS seal has total of fifty-six pockets, a two dimensional array with eight pockets in circumferential direction and seven in axial direction. Assuming zero eccentricity and considering the periodic nature of the geometry, a 1/8th sector as shown in Figure 5, was considered for CFD computation. Hexahedral mesh was used to discretize the flow domain. The worst determinant was greater than 0.5 (determinant of 1 indicates a perfect cube and 0 or less than zero indicates a warped or negative volume cell, respectively). The worst included-angle between the cell faces was greater than 45°. Growth ratio of the elements from the wall was 1.1 and the average aspect ratio less than 30. The 1st grid point distance was maintained at y^+ of about 1. The grid size was about 2.5 million nodes. Twenty-eight elements were used across the seal - rotor clearance of 0.292mm (11.5 mils), to ensure flow field resolution.

Boundary Conditions in CFX

The working fluid was air and assumed an ideal gas. Total pressure and static temperature of 25° C was specified at the inlet. Inlet turbulent intensity and eddy viscosity ratio were at 5% and 10 respectively. For each test case, the inlet total

pressure value was iterated until a gauge static pressure of 6.89bar (100psig) was attained at the monitor points in the inlet plenum. The reference pressure was 1 atm. A no-slip rotating wall condition of 15000rpm was imposed on the rotor boundary in the direction of the pre-swirl. Domain exit was maintained at gauge pressure of 0 bar. The sector boundaries were rotationally periodic. The rest of the boundaries that include pocket/blade walls were assigned as no-slip walls. SST k- ω model was used. Continuity, Momentum and Energy equations were solved as steady state equations. RMS residuals for all transport variables were less than 1e-4, a recommended target residual value [7]. Energy and mass imbalances are less than 0.1%. Exit mass flow and inlet static pressure were monitored and attained a constant value.



Figure 5. CFD 1/8TH SECTOR DOMAIN AND HEXAHEDRAL GRID.

DESCRIPTION OF ANALYTICAL APPROACHES (2): BULK FLOW CODE

The approach to use a "bulk flow" model to predict rotordynamic characteristics was first developed by Iwatsubo in 1980 [8]. San Andrés [9] extended the model to include features such as lubricant properties and turbulent flow for the tilting pad bearings. Researchers further adapted the bulk flow model to provide theoretical predictions for the rotordynamics and leakage performance of the damper seals [10, 11]. In this study, a Bulk Flow Code (BFC) was developed for the fully partitioned damper seals.

In the Bulk Flow Code, the pockets of pressurized gas in the damper seals are modeled as individual control volumes; conservation of mass equations are solved for each control volume. Comparisons between 1-D (axial) and 2-D (axial and circumferential) model results show that while the circumferential flow component impacts the rotordynamics coefficients of the seal substantially, its effect on leakage performance is almost negligible. Hence this analysis adopted a 1-D approach, where the leakage of the damper seal was predicted according to the upstream and downstream pressures and the number of pockets in the axial direction. A discharge coefficient was introduced to account for the effect of pocket depth. The discharge coefficient can be tuned with experimental data to yield better accuracy (a default value of 0.9 was used in this study). The BFC serves as a convenient design tool for damper seal leakage prediction.

EXPERIMENTAL RESULTS

Tested Seal Configurations

Four sets of seals were tested in the CSR leakage test set up, as shown in Figure 2 and 3. The four seals include a labyrinth, a honeycomb and two Pocket Damper Seals (PDS) with different cavity depth for sensitivity study. The labyrinth seal has a standard straight-through configuration with total of 14 teeth. The two pocket damper seals have a fully partitioned configuration [6], with eight pockets in the circumferential direction and seven in the axial direction. The two PDS possess different cavity depth; subsequent discussions refer to the PDS with smaller cavity depth as PDS1 and the one with larger depth as PDS2. All four seals have a nominal radial clearance of 0.0115" (0.292 mm) from the rotor with a diameter of 6.693" (170 mm). Images of the honeycomb and the labyrinth seal test hardware are shown in Figure 6.

Each seal configuration was tested at three fixed inlet pressures: 6.89, 13.79, and 20.68 bar (100, 200 and 300 psi); for a fixed inlet pressure, the exhaust pressure was altered to achieve pressure ratios ranging from 1.1 to 2.0. The measured leakage flow rates for the four seals are plotted in Figure 7 as functions of pressure ratio.



Figure 6. HONEYCOMB AND LABYRINTH SEALS TESTED IN CSR. PHOTOS COURTESY B. H. ERTAS.

Effective Clearance Calculation

The effective clearance calculation is a means of normalizing the performance of a seal such that it can be compared to other seals of varying geometries. In essence, the calculation takes into account the properties of the process fluid upstream and downstream of the seal and computes an effective annulus that would provide an equivalent resistance to the fluid path. The equation for calculating effective clearance for a seal is given by:

$$Cl_{eff} = \frac{\dot{m}\sqrt{T_o}}{\mu D\pi P_o} , \qquad (1)$$

where m is the measured leakage through the seal, T_0 is the measured supply temperature, P_0 is the measured upstream pressure, D is the diameter of the rotor, and μ is the flow function based on theoretical leakage model for flow through an orifice with round cross-section. The flow function is dependent on the pressure ratio (P_0 / P_1) and choked / not choked status of the leakage through the seal. In a choked flow condition μ is equal to 0.532 $lbm \cdot R^{1/2} \cdot lbf^{-1} \cdot sec^{-1}$. The equation for non-choked flow μ is given by

$$\mu = \sqrt{\frac{2\gamma g_c}{(\gamma - 1)R} \cdot \left(\frac{P_0}{P_1}\right)^{-\left(\frac{\gamma + 1}{\gamma}\right)} \cdot \left(\left(\frac{P_0}{P_1}\right)^{\left(\frac{\gamma - 1}{\gamma}\right)} - 1\right)},$$
(2)

where γ is the heat capacity ratio of the working gas, *R* is the specific gas constant and g_c is the gravitational conversion constant².

Figure 7 shows that for fixed upstream pressure of 6.89 bar (100 psi), the seal leakage flow begins to saturate when the pressure ratio increases to $1.8 \sim 2.0$. This flow limiting phenomenon is consistent with the theoretical prediction that an ideal orifice flow becomes choked at pressure ratio ≥ 1.8 (calculation of the flow function given in Equation 2 reveals that μ reaches 0.532 at pressure ratio of 1.8), hence further decrease of the downstream pressure (i.e., increase of pressure ratio) does not increase the mass flow rate. The leakage data with 13.79 and 20.68 bar (200 and 300 psi) fixed inlet pressure also exhibit nonlinear trends with increasing pressure ratio; however, more data points are needed to declare a choked flow condition. It should be noted that even at choked flow, increasing the upstream pressure could increase the leakage rate.

Figure 8 depicts the linear growth of leakage flow with increasing seal upstream pressure for a fixed pressure ratio of 2.0. Overall, for the same inlet pressure and pressure ratio, the pocket damper seals tend to leak more than the labyrinth and honeycomb seals and the pocket damper seal with larger cavity depth (PDS2) tends to leak more than the one with smaller cavity depth (PDS1). Figure 9 compares the effective clearance of the four seal configurations calculated from the leakage data shown in Figure 6 at pressure ratio of 2.0. Figure 9 shows that for a given seal geometry, the effective clearance is independent of the upstream pressure; in fact, the Cl_{eff} calculated from 6.89 bar (100 psi) vs. 13.79 bar (300 psi) leakage data only differ 5-10%. Using the labyrinth seal as baseline for comparison purposes, the honeycomb seal has the best leakage performance ($Cl_{eff} \downarrow 4\%$), while the pocket damper

seals tend to leak more ($Cl_{eff} \uparrow 9\%$ for PDS1 and $Cl_{eff} \uparrow 21\%$ for PDS2). Among the pocket damper seals, the one with smaller cavity depth (PDS1) has better sealing capacity.



Figure 7. LEAKAGE TEST RESULTS AS FUNCTION OF PRESSURE RATIO, INLET PRESSURE = 6.89, 13.79 AND 20.68 BAR (100, 200 AND 300 PSI). ESTIMATED TREND LINES ARE BASED ON THE FLOW FUNCTION.



Figure 8. LEAKAGE TEST RESULTS AS FUNCTION OF INLET PRESSURE, PRESSURE RATIO ~ 2.0.

 $^{^{2}} R = 53.3533$ ft lbf lbm⁻¹ R⁻¹ for dry air, $g_{c} = 32.174$ ft lbm lbf⁻¹ sec².



Figure 9. SEALS EFFECTIVE CLEARANCE CALCULATED AT PRESSURE RATIO ~ 2.0 FOR UPSTREAM PRESSURES 6.89, 13.79, AND 20.68 BAR (100, 200 AND 300 PSI). SEAL PHYSICAL CLEARANCE = 0.3 MM (0.012 INCHES).

ANALYTICAL RESULTS

CFD model and Bulk Flow Code (BFC) are developed as design tools for pocket damper seals. Leakage predictions of these analytical models are compared against CSR experimental data for validation purposes. Here the PDS1 seal is selected as the target configuration for leakage validation.

Figure 10 depicts the BFC and CFD predictions of PDS1 leakage flow for pressure ratios ranging from 1.1 to 2.0 at fixed upstream pressures of 6.89, 13.79, and 20.68 bar (100, 200 and 300 psi), respectively; CSR test data are also plotted for comparison purpose. The analytical predictions are found to be in excellent agreement with experimental data. Both BFC and CFD models tend to slightly under-predict leakage at higher inlet pressures; however, the maximum deviation from test data (at 20.68 bar (300 psi) inlet pressure, pressure ratio = 2.0) is only 6%. Figure 11 compares the effective clearance of PDS1 calculated from the Bulk Flow Code, the CFD model and the CSR leakage test data. The seal upstream and downstream pressures are identical in all cases (13.79 / 6.21 bar). Figure 11 shows that both BFC and CFD model predictions match the test data with 95% accuracy, suggesting both analytical approaches have the potential to serve as tools for optimum pocket damper seal design.

CONCLUSIONS

The leakage performances of a labyrinth, honeycomb and two fully partitioned damper seals are experimentally evaluated over a range of inlet pressures and pressure ratios. The labyrinth seal and honeycomb seal exhibit similar leakage performances. The pocket damper seals are known to provide improved damping effects to rotor vibration and to provide increased rotordynamics stability. The two fully partitioned damper seals are found to leak 9-21% more in comparison to the labyrinth seal. However, the leakage performance of PDS can be improved through optimizing seal geometry parameters, such as the cavity depth and the clearance ratio of the seal. Further optimization is needed in the design of PDS to reach a balance between rotordynamics and leakage performance. Analytical tools, including a CFD model and a Bulk Flow Code, are validated against experimental data as future optimization tools for the design of pocket damper seals.



Figure 10. BULK FLOW CODE (BFC) AND CFD LEAKAGE PREDICTIONS FOR PDS1, IN COMPARISON WITH CSR TEST DATA.



Figure 11. CFD AND BFC PREDICTIONS OF EFFECTIVE CLEARANCE FOR PDS1, IN COMPARISON WITH CSR TEST DATA. SEAL PHYSICAL CLEARANCE = 0.3 MM (0.012 INCHES).

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NONMENCLATURE

- *Cl_{eff}* Seal effective clearance [L]
- D Rotor diameter [L]
- g_c Gravitational conversion constant [ML/Ft²]
- \dot{m} Mass flow rate [M/t]
- P_0 Upstream pressure [F/L²]
- P_1 Downstream pressure [F/L²]
- *R* Specific gas constant [FL/MT]
- *T*₀ Upstream temperature [T]
- Greek Symbols
- μ Flow function [MT^{1/2}/Ft]
- γ Specific heat ratio [-]

Dimensions

- F Force
- L Length
- M Mass
- t Time
- T Temperature

REFERENCES

[1] Childs, D. W. and Vance, J. M, 1997, "Annular Gas Seals and Rotordynamics of Compressors and Turbines," Proceedings of the 26th Turbomachinery Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 201-220.

[2] Chupp, R. E., Hendricks, R. C., Lattime, S. B., and Steinetz, B. M., 2006, "Sealing in Turbomachinery," Journal of Propulsion and Power, Vol. 22 (2), pp. 313-349.

[3] Elden, A. M. G., 2007, "Leakage and Rotordynamic Effects of Pocket Damper Seals and See-Through Labyrinth Seals," Ph. D. Thesis, Texas A&M University.

[4] Childs, D., Elrod, D., and Hale, K., 1989, "Annular Honeycomb Seals: Test Results for Leakage and Rotordynamic Coefficients; Comparison to Labyrinth and Smooth Configurations," ASME Transactions, Journal of Tribology, Vol. 111 (2), pp. 293-301.

[5] Armstrong, J. and Perricone, F., 1996, "Turbine Instability Solution-Honeycomb Seals," Proceedings of the 25th Turbomachinery Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 47-56. [6] Ertas, B. H. and Vance, J. M., 2007, "Rotordynamic Force Coefficients for a New Damper Seal Design," ASME Transactions, Journal of Tribology, Vol. 129, pp. 365-374.

[7] Ansys-CFX solver modeling guide, Release 12.0.

[8] Iwatsubo, T., 1980, "Evaluation of Instability of Forces of Labyrinth Seals in Turbines or Compressors," Workshop on Rotordynamic Instability Problems in High-Performance Turbomachinery, Texas A&M University, NASA Conference Publication No. 2133, pp. 139-167.

[9] San Andrés, L., 1996, "Turbulent Flow, Flexure-Pivot Hybrid Bearings for Cryogenic Applications," ASME Journal of Tribology, Vol. 118, pp. 190-200.

[10] Sundararajan, P. and Vance, J. M., 1993a, "A Theoretical and Experimental Investigation of a Gas Operated Bearing Damper for Turbomachinery – Part I: Theoretical Model and Predictions," Proceedings of the 14th Vibration and Noise Conference, Albuquerque, NM, ASME DE-Vol. 60, pp. 67-83.

[11] Ertas, B. H., 2005, "Rotordynamic Force Coefficients of Pocket Damper Seals", Ph.D. Thesis, Texas A&M University.