Proceedings of ASME Turbo Expo 2004:
Power for Land, Sea, and Air
June 16–19, 2004, Vienna, Austria

GT2004-54296

NON-METALLIC BRUSH SEALS FOR GAS TURBINE BEARINGS

Nitin Bhate
General Electric Global Research
One Research Circle
Niskayuna, NY 12309
Phone: 518-387-4702,
Fax: 518-387-7292
Email: bhateni@crd.ge.com

Anthony C. Thermos
General Electric Energy
300 Garlington Road
Greenville, SC 29602
Phone: 864-254-2835,
Fax: 864-254-2380
Email: Anthony.thermos@ps.ge.com

Mahmut F. Aksit
Sabanci University,
Istanbul, Turkey 34956
Phone: 90-216-483-9598,
Fax:90-216-483-9550
Email: aksit@sabanciuniv.edu

Mehmet Demiroglu
Rensselaer Polytechnic Institute,
Troy, NY 12180
Phone: 518-859-2190, Fax: 518-276-6380
Email: demiroglu@prosolutionsusa.com

Huseyin Kizil
Omega Advanced Solutions, Inc.
Troy, NY 12180
Phone: 518-276-8191, Fax: 518-276-8993
Email: kizil@omegaadvanced.com

ABSTRACT

A non-metallic brush seal has been developed as an oil seal for use in turbomachinary. Traditionally labyrinth-type seals with larger clearances have been used in such applications. Labyrinth seals have higher leakage rates and can undergo excessive wear in case of rotor instability. Brush seals reduce leakage by up to an order of magnitude and provide compliance against rotor instabilities. Brush seals are compact and are much less prone to degradation associated with oil sealing. This paper describes the benefits and development of the non-metallic brush seals for oil sealing application.

Keywords: Brush seals, Non-metallic fibers, Bearing seals, Oil sealing, Turbomachinary, Gas Turbine, Compressor

INTRODUCTION

Oil sealing in turbomachinary has been an area of concern for engineers for last several decades. Oil sealing is typically required around bearings in turbomachinary. Tighter clearances are required at these locations to avoid oil contamination of the downstream turbine components.

Traditionally labyrinth seals have been used for high-speed oil vapor sealing applications. Labyrinth seals are designed with large radial clearances to avoid any rotor contact that could result in overheating and damage to the rotor. The higher clearances result in higher leakage and subsequent performance loss. Carbon circumferential seals have provided an alternative to the labyrinth seals in the past few decades [1,2]. Carbon seals are effective for oil vapor sealing. They do not commonly generate abrasive particles that could cause damage to the turbomachinary components. On the other hand, tolerance control in large diameter applications as well as the cost of carbon seals is an important challenge. Carbon deposits building up on critical surfaces may also cause the seal to hang up.

Hydrodynamic oil rings have been employed for sealing continuous oil flow. It is crucial to maintain continuous oil flow for continued seal performance. In absence of oil flow, the seal can undergo excessive wear resulting in seal damage and loss of performance.

Use of brush seals for oil sealing is relatively recent. Brush seals have been extensively used in gas sealing in turbo-machinery applications, and have demonstrated excellent leakage characteristics. They perform very well under rotor transients due to the inherent compliance of bristles and the seal structure. However, oil applications are cause for concern due to potential for oil temperature rise and coking. Coking refers to the carburization of oil particles at excessively high temperatures. The temperature at which oil starts coking depends on the composition of the oil. Coking may result in generation of carbon deposits that stick on the blades of the compressor causing performance degradation as well as increased maintenance cost. Early studies of brush seals in liquid flow were performed by Braun et al. [3-6]. Carlile et al. [7] and Hendricks et al. [8] have studied brush leakage performance with liquid helium and have reported improved performance when a lubricant is applied to the bristles. Improved leakage performance with oil was confirmed by Aksit et al. [9]. Their work focused on the feasibility of brush seals in oil applications. Details of their work can be found elsewhere [10-13]. Based on the results from Aksit et al.'s [9] feasibility study, this work presents a gas turbine bearing oil seal application using a non-metallic brush seal.

NON-METALLIC BRUSH SEALS

Due to concerns of metal particle generation near bearings, an alternative to metallic brush seals was needed. Among the non-metallic fibers, ceramic fibers were excluded due to the abrasive nature of wear debris they generate. A loose ceramic or metal fiber in oil bearing can be hazardous. Alternative non-metallic fibers were searched for the oil seal design considered for this gas turbine bearing application. Typically, organic fibers are limited in temperature capability and tend to shrink with increase in temperature. Considering the fact that oil or oil mist in bearing cavities may reach temperatures in excess of 150^0C (302^0F), bristle shrinkage may result in increased leakage. Inertness and moisture absorption rates are the other important considerations. After a study of available fibers, aramid was selected as the candidate fiber material for the gas turbine oil sealing application. Aramid fibers are organic polymers that typically exhibit high strength and low density. They can be used for applications up to 150^0C operating temperature, and show negligible amount of shrinkage and moisture absorption.

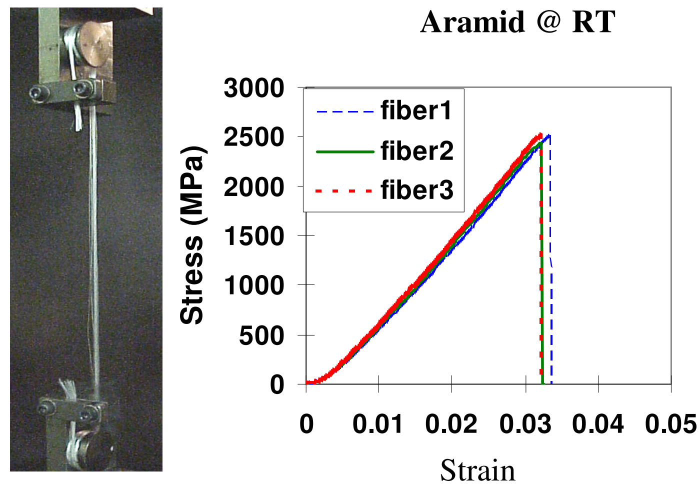
FIBER CHARACTERIZATION

Commonly available strength data for aramid fibers are generated using fully twisted yarns per ASTM standards. However, brush seal applications require use of densely packed untwisted bristles. It is known that twisting affects observed overall strength. Therefore, a set of tests was conducted to measure strength of the untwisted aramid fibers for use in brush seals. Fiber characterization involved evaluation of tensile strength and creep strength as well as wear performance. Importance of fiber properties in brush seal design and conventional design guidelines can be found elsewhere in the literature [14-16].

Tensile Tests

A special tensile test arrangement was devised to hold the bristles. As illustrated in Fig. 1a, this arrangement helped in avoiding commonly observed fiber failure at grips, and confined bristle failure within the gauge length. Tensile tests were conducted with sample fibers wrapped around pulleys before being clamped.

All the tensile tests were conducted at room temperature with 10%/min strain rate. Controlled test environment with 47% humidity was maintained. Repeated room temperature tests indicated fiber strength around 2482 MPa (360 ksi) with good repeatability (Fig. 1b). An average 3.2% strain was observed at failure.



(a) (b)

Fig. 1 Tensile tests: a) configuration b) repeatability.

High Temperature Strength Evaluations

In order to investigate the strength of fibers after long-term exposure to elevated temperatures, a second set of tests was performed. Sample fibers were exposed to selected temperatures, and tested after specific exposure periods. Fig. 2 illustrates fiber behavior after exposure to 150^0C (302^0F) for extended periods. Samples were tested at room temperature. Results indicate that after initial exposure to 150 ^{0}C strength decreases to around 2413 MPa (350 ksi). The strength loss is not significant for 21 days, after which loss is accelerated. Fiber strength decreases to 2300 MPa (334 ksi) after 35 days exposure. As illustrated in Fig. 3, strength loss is considerable and much accelerated when exposure temperature is raised to 260 ^{0}C (500 ^{0}F).

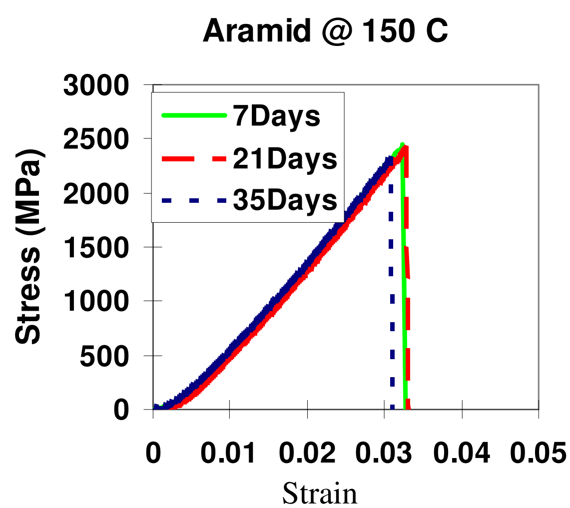


Fig. 2 Fiber strength upon exposure to 150 ^{0}C (302 ^{0}F).

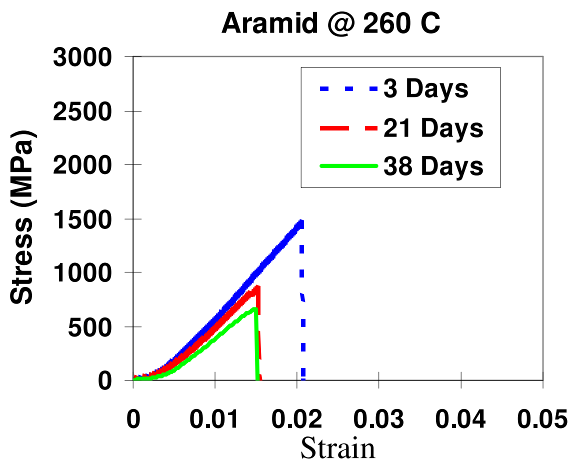


Fig. 3 Fiber strength upon exposure to 260 ^{0}C (500 ^{0}F).

Creep Tests

After evaluating the temperature limits of the selected aramid fibers, long-term temperature effects were further evaluated through creep testing. The tests were conducted at 150 ^{0}C (302 ^{0}F) under constant load (20% of the breaking load measured previously). As illustrated in Fig. 4, results indicated that aramid fibers have good creep properties at 150 ^{0}C (secondary regime). The measured rate of change in strain was 0.1%/decade.

Wear Tests

Before the selected fibers can be used for gas turbine oil sealing application, wear resistance of the fibers needs to be evaluated as well. First, individual fibers are formed into a bundle or tuft. The bundle was then substituted for the pin in an oscillating wear test setup. Flat material was selected as Ni-Cr-Mo-V, which is typical rotor material for gas turbines.

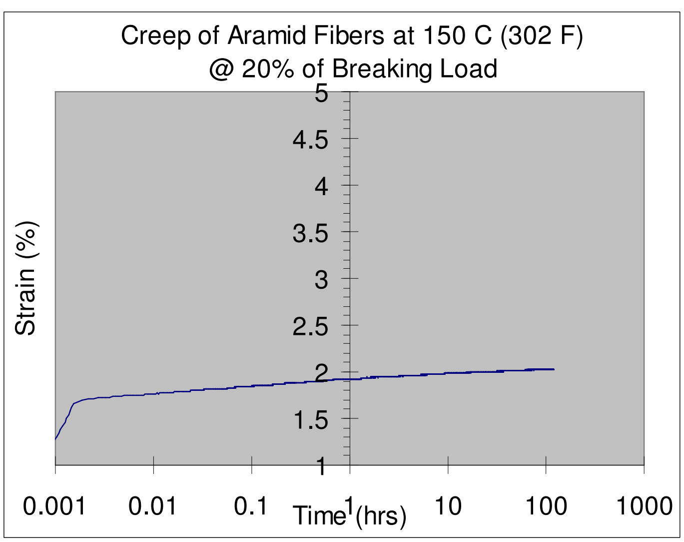


Fig. 4 Creep strength at 150 ^{0}C (302 ^{0}F).

Wear tests were performed using constant 2.22 N (0.5 lb) load with estimated contact pressure of 38 KPa (5.5 psi). Average test speed was 5.1 mm/s (12 in/min). Tests were repeated for both room temperature and 150 ^{0}C conditions, and wear measurements are illustrated Fig. 5.

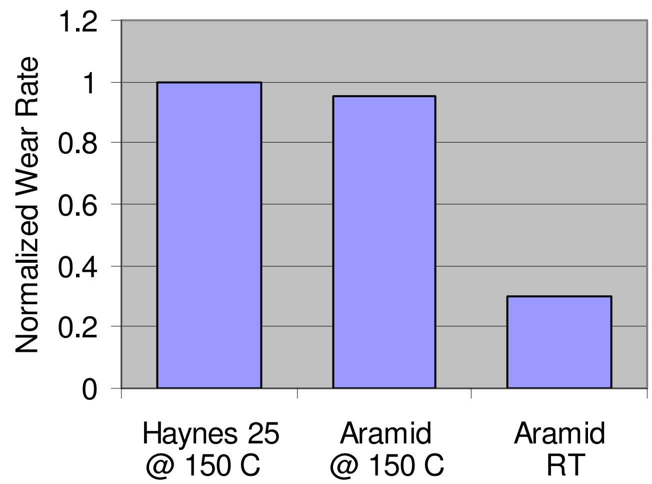


Fig. 5 Wear test results for aramid and Haynes 25 tufts against Ni-Cr-Mo-V. Data are normalized with wear rate of Haynes 25 bristles at 150 ^{0}C.

Although the tests have been conducted at much slower speeds than those encountered in a gas turbine, the tests can be used to gauge relative performance improvement of aramid fibers over metallic fibers. The results indicate a better wear performance than the typical wear resistant cobalt based superalloy bristles (Haynes 25$^©$) running on the same rotor material. Results also indicate comparable friction coefficient, which was measured at both room temperature and 150 ^{0}C tests.

SUBSCALE SEALS AND PERFORMANCE TESTS

In order to quantify the performance of non-metallic brush seals, a set of sub-scale seals were manufactured (Fig. 6).

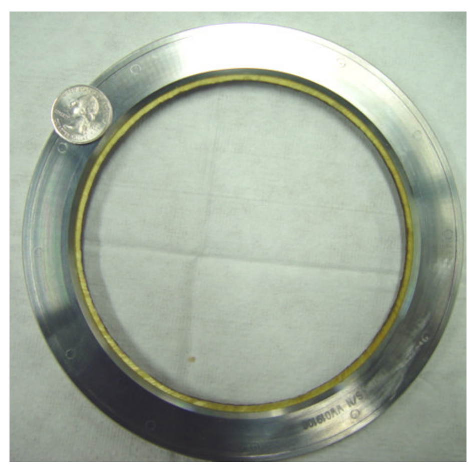


Fig. 6 Subscale aramid fiber brush seal.

The design parameters for these sub-scale seals are tabulated in Table 1.

Table 1: Subscale Test Seal Parameters

Design Parameter	Seal #1	Seal #2	Seal #3
Bristle material	Aramid	Aramid	Metal
Test rotor dia. Nom. (m)	0.131	0.131	0.129
Bristle dia (µm)	11.7	11.7	102
Cant angle (degrees)	35-45	14-25	45-50
# of bristle rows across	~90	~100	12-14
Fence height (mm)	0.76	0.76	1.02

Manufacturing of non-metallic brush seals posed some unique challenges. For example, handling of aramid fibers was more difficult compared with metallic bristles due to their smaller size and lower modulus of elasticity. Maintaining brush seal cant angle, seal assembly, as well as maintaining tight tolerances on seal dimensions also posed major challenges.

In order to establish a baseline seal performance for non-metallic brush seals, seal samples were evaluated under various operating conditions. A series of tests were performed to benchmark static and dynamic leakage and investigate important phenomena such as oil coking.

The tests were carried out on a high-speed brush seal test rig. Fig. 7 shows a schematic of the rig. It consists of a frequency modulated, high speed, motorized spindle, capable of rotating up to 40000 rpm, and a housing to mount the test seals. Spindle and housing mounted on two slides perpendicular to each other. The motorized spindle can be moved parallel to its axis allowing application of interference or clearance across the entire seal (360^0) when used with stepped rotor/shaft. The housing can also be moved normal to the rotation axis in order to apply eccentric interference or clearance on the seal. Each slide can be moved during the operation in order to simulate different interference or clearance conditions.

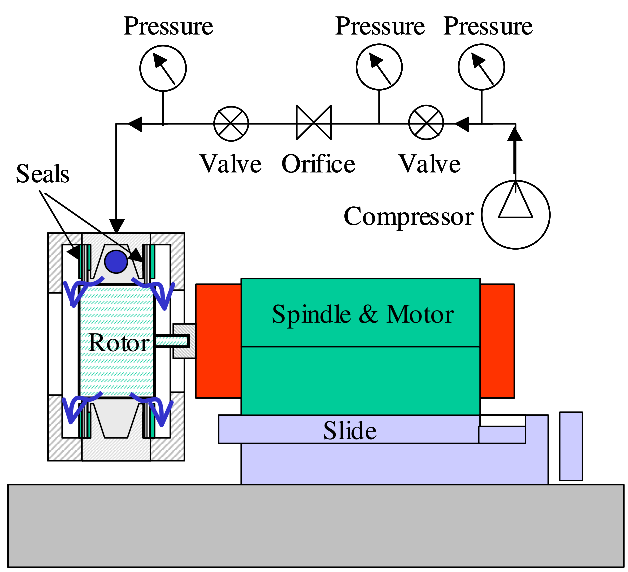


Fig. 7 Subscale dynamic seal test rig

Although the bearings for the spindle are capable of carrying up to 1000 kg axial load, a dual brush seal test configuration was utilized to balance axial loads in dynamic testing. To simulate gas turbine conditions, an oil mister was installed on the upstream of the seals. The oil mister is capable of generating an air-oil mixture with varying composition and droplet sizes.

Air and Oil Leakage Performance of Aramid Seal

Leakage performance of metal brush seals has been extensively studied in the literature. Therefore, the performance of aramid seals was the focus of this study. Dry performance with air was evaluated before any oil leakage testing. Fig. 8 plots a summary of these tests, and shows a comparison with metallic brush seals. As evident from the leakage plot, aramid brush seals were found to be superior to metallic brush seals. The superior sealing capability can be attributed mainly to denser bristle pack, resulting in reduced seal porosity. Under static conditions, baseline leakage for the aramid seal can be less than half of metal seal leakage.

After extensive testing in air, aramid brush seals were evaluated for oil sealing capability. Aramid seals were observed to reduce the oil mist leakage by more than 50% compared to labyrinth seals. Moreover, there were no noticeable oil particles passing through aramid fiber bundles as long as there was a contact with the shaft.

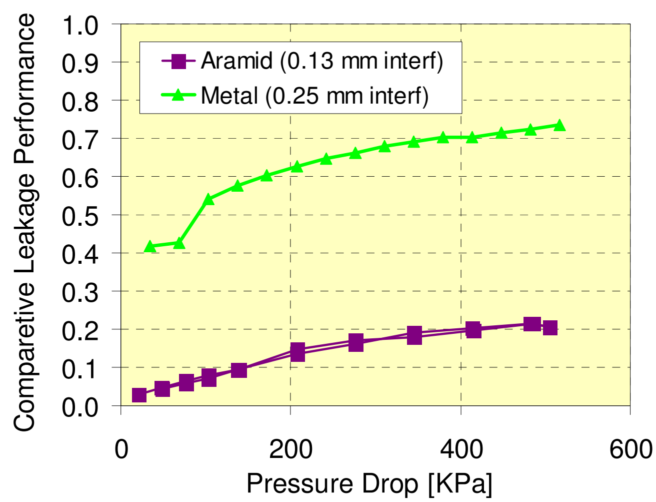


Fig. 8 Sample Aramid Seal Leakage Data

Oil Temperature Rise

During dynamic leakage tests the air/oil temperatures at the inlet cavity and the seal exit were measured at 3 places for each seal. The results indicate that heating across the seal is not excessive compared to the operating temperature limit for the fibers. It is believed that a soft seal design combined with shear thinning resulted in minimal heat generation across the seal. Close examinations of the bristle tips did not reveal any apparent oil coking. To investigate oil coking further, high-speed coking tests were performed using oil with oil temperatures around 132 ^{0}C (270 ^{0}F) at a speed of 103 m/s (337 ft/s) with 0.25 mm (0.010 in) radial interference. Downstream oil temperature reached 153 ^{0}C (307 ^{0}F). When bristle tips were examined after the coking tests, no apparent oil coking was observed.

GAS TURBINE BEARING SEALING APPLICATION

General Electric 7EA #1 bearing sealing was selected as prototype field installation for aramid brush seal. The objectives of the test included validating aramid seal leakage prediction as well as measuring the seal durability. The existing 7EA #1 (inlet) bearing contains two aluminum (labyrinth) seals, located at the forward and aft of the bearing housing. The primary purpose of the seals is to prevent the bearing oil leaking out from the bearing housing. The seals are placed circumferentially on a machined surface of the two bearing housing cavities, which are connected to the Cooling and Sealing (C&S) system of the engine (Fig. 9). The C&S system directs air from the compressor to the bearing to provide oil sealing as well as cooling of the bearing hardware.

Gas Turbine Bearing Seal Installation

Two identical 7EA gas turbine units at the same location were chosen for the validation test. One unit was installed with an aramid brush seal in bearing #1 location (Fig. 9) while the existing sealing configuration was maintained in the other unit. Fig. 9 illustrates the aramid seal arrangement along with auxiliary instrumentation.

An instrumentation plan was developed to measure the effectiveness of the aramid seal. Fig. 9 shows a schematic of the instrumentation arrangement that was employed at the aft #1 bearing location. The objective of the instrumentation was to measure the operating conditions around the seal with minimally invasive instrumentation. A detailed flow model was developed to predict the flow in the seal region. Thermocouples were mounted on either side of the aramid seal to measure the temperature profile, which was then used in the flow analysis for accurate flow prediction. Pressure probes were used to measure pressure profile of the aft bearing #1 location. Similar instrumentation plan was employed in the gas turbine without the aramid seal to quantify the aramid seal benefits.

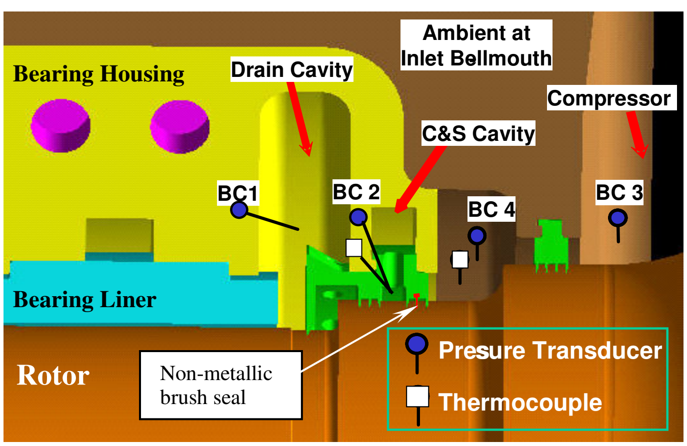


Fig. 9 The 7EA #1 (inlet) bearing sealing application

Preliminary data was gathered after restarting the units. The data shows marked difference between the pressure profiles in the two units. Fig. 10 illustrates the increase in sealing differential pressure (C&S Air - Bearing Drain) with brush seal over the baseline sealing differential pressure with conventional labyrinth seals. As much as 9 KPa higher pressure difference between sealing air and bearing drain cavity is maintained in the unit with the aramid brush seal. A bigger pressure difference is also maintained across the aramid seal as compared to the labyrinth seal based on the superior sealing capability. Based on the data, a few preliminary conclusions can be drawn related to the performance of the aramid brush seal. The aramid brush seal helps maintain a higher sealing air pressure than the drain cavity. The aramid seal provides not only a restriction to the vent flow but also an obstacle to the oil particles. In general, the brush seal significantly enhances the effectiveness of the sealing system thereby allowing less oil particles to migrate out of the bearing.

DISCUSSION AND CONCLUSIONS

Oil sealing at high surface speeds remains a challenge for most turbomachinary applications. This work attempts to investigate brush seals as a possible alternative to today's labyrinth seals, carbon seals and oil rings. Due to high velocity rubbing at the brush seal-shaft interface, oil temperature increase and coking are the most challenging issues that need to be addressed. The experimental investigation performed through this study reveals the following conclusions about the brush seal applications in high-speed oil mist sealing.

- Non-metallic brush seal leakage performance is better than metal brush seals and labyrinth seals for air and oil applications. As also observed by Carlile et al. [7] and Aksit et al. [9], leakage performance of brush seals tends to improve when wetted with a lubricant.
- The friction coefficient of non-metallic fibers is comparable or even better compared to that of metal bristles. Due to existence of oil at the seal-shaft interface, friction coefficients are expected to reduce further. However, heat generation, thus temperature rise, still remains a major risk since oil-coking temperature is within the reachable limits and needs further investigation.

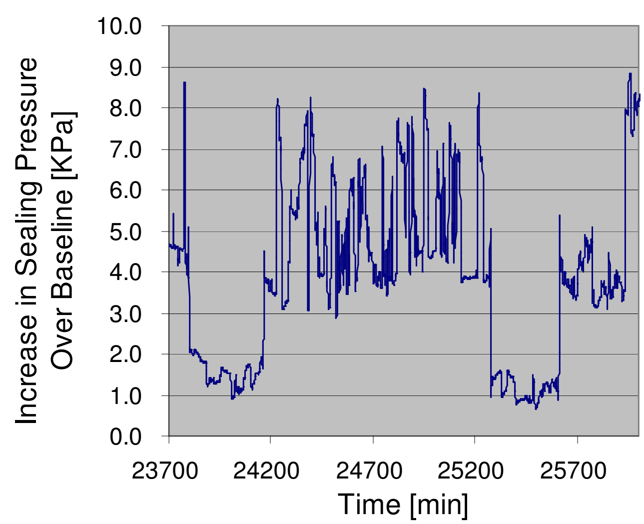


Fig. 10 Field test results for 7EA #1 (inlet) bearing sealing application.

The results from 7EA field tests also indicate advantages of aramid brush seals for this particular application. The potential benefits of aramid brush seals are:

- Less oil mist migrates out of the bearing
- Less contamination of the compressor rotor resulting in slower compressor performance degradation,
- Longer time intervals between the compressor water washes - implies less maintenance,
- Higher differential pressure between cooling & sealing cavity and drain - signals potential reduction of the C&S air flow into the bearing thus enhancement of the compressor performance.

REFERENCES

[1] Menendez, R.P. and Cunningham M.D., 1999, “Development of lift off seal technology for air/oil axial sealing applications,” AIAA/SAE/ASME/ASEE 35th Joint Propulsion Conference Paper AIAA 99-2822.

[2] Giesler, W., D. Mathis, and Hager J., 1998, “High Reliability Oil-air high-speed gearbox clearance seal,” AIAA/SAE/ASME/ASEE 34th Joint Propulsion Conference Paper AIAA 98-3287.

[3] Braun, M., and Canacci, V., 1990, “Flow Visualization and Motion Analysis for a Series of Four Sequential Brush Seals,” AIAA/SAE/ASME/ASEE 26th Joint Propulsion Conference Paper AIAA 90-2482.

[4] Braun, M., Hendricks, R.C. and Yang, Y., 1991, “Effects of Brush Seal Morphology on Leakage and Pressure Drops,” AIAA/SAE/ASME/ASEE 27th Joint Propulsion Conference Paper AIAA 91-2106.

[5] Braun, M.J., Hendricks, R.C. and Canacci, V., 1990, “Flow Visualization in a Simulated Brush Seal,” ASME Gas Turbine and Aeroengine Congress Paper ASME 90-GT-217.

[6] Braun, M.J., Canacci, V.A., and Hendricks, R.C., 1991, “Flow Visualization and Quantitative Velocity and Pressure Measurements in Simulated Single and Double Brush Seals,” Trib. Trans., pp 70-80.

[7] Carlile, J.A., Hendricks, R.C. and Yoder, D.A., 1993, “Brush Seal Leakage Performance with Gaseous Working Fluids at Static and Low Rotor Speed Conditions,” ASME J. Eng. for Gas Turbines and Power, 115, pp 397-403.

[8] Hendricks, R.C., Carlile, J.A., and Liang, D.A., 1992, “Some sealing concepts - a review, part b: brush seal systems,” Proc. of the 4th Int. Symp. on Transport Phenomena and Dynamics of Rotating Machinery (ISROMAC-4), Honolulu, HI, pp 222-227.

[9] Aksit, M. F., Bhate, N., Bouchard, C., Demiroglu, and Dogu, Y., “Evaluation of Brush Seal Performance for Oil Sealing Applications,” 2003, AIAA/SAE/ASME/ ASEE 39th Joint Propulsion Conference Paper AIAA 03-4695.

[10] Mayer R.R., Aksit M.F., and Bagepalli B.S., 2003, “Brush seal for a bearing cavity,” US Patent No. US6502824B2.

[11] Aksit M.F., Dinc O.S., and Mayer R.R., 2002, “Brush seal and machine having a brush seal,” US Patent No. US 6406027B1.

[12] Mayer R.R., Bagepalli B.S., and Aksit M.F., 2002, “Low flow fluid film seal for hydrogen cooled generators,” US Patent No. US 6378873B1.

[13] Bagepalli B.S., Aksit M.F., and Mayer R.R., 2001, “Brush seal and rotary machine including such brush seal,” US Patent No. US 6257588B1.

[14] Aksit, M. F., Chupp, R. E., Dinc, O. S., and Demiroglu, M., 2002, “Advanced Seals for Industrial Turbine Applications: Design Approach and “Static Seal Development,” AIAA Journal of Propulsion and Power, 18, pp. 1254-1259.

[15] Chupp, R. E., Ghasripoor, F., Turnquist, N. A., Demiroglu, M., and Aksit, M. F., 2002, “Advanced Seals for Industrial Turbine Applications: Dynamic Seal Development,” AIAA Journal of Propulsion and Power, 18, pp. 1260-1266.

[16] Dinc, S., Demiroglu, M., Turnquist, N., Mortzheim, J., Goetze, G., Maupin, J., Hopkins, J., Wolfe, C., and Florin, M., 2002, “Fundamental Design Issues of Brush Seals for Industrial Applications,” ASME Journal of Turbomachinary, 124, pp. 293-300.