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Recent Brush Seal and Testing Developments at Cross

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RECENT BRUSH SEAL AND TESTING DEVELOPMENTS AT CROSS

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Abstract

The development of brush seals continues at an increased rate at Cross. New test facilities have been installed over the past 3 years that can more accurately reproduce the environment that exists within industrial and aerospace compressors and turbines.

Facilities now exist to give a maximum surface speed of 1260ft/s, a maximum temperature of 1150^{o}F and a maximum pressure of 300psig. A static back pressure rig has also been developed that allows up to 300 psig to be applied either side of the seal. This allows low pressure drops at high line pressures to be achieved.

Stiffness checking of brush seals is now being validated with non linear, contact finite element models and friction is shown to have a large influence on the accuracy of the data produced.

Nomenclature

Df	Front plate bore (inches)
Dr	Rotor diameter (inches)
Fh	Fence height (inches)
g	Acceleration due to gravity (ft/s^2)
P1	Absolute pressure upstream of seal (psia)
Pu	Pressure just upstream of seal (psi)
Pd	Pressure just downstream of seal (psi)
PR	Pressure ratio
R	Specific gas constant (ft lbf/lb R)
q	Mass flow rate (lbs/s)
T1	Gas temperature upstream of seal (^{o}F)
w	Length or circumference of seal (inches)
γ	Ratio of specific heats
μ	Flow function

Introduction

Cross have been producing brush seals since the 1970's for the Aerospace and Power Generation Industries. Cross brush seals continue to be applied to a wide range of industrial and aerospace applications, with particularly strong growth in industrial applications. Over the past 3 years we have invested considerable time and money into upgrading all our test facilities.

The upgrades were dictated by the more arduous applications that Brush Seals are now being selected for. The facility now has a maximum pressure capability of 300psi (20.7bar), maximum temperature of 1150^{o}F (620^{o}C) and a maximum surface speed of 1260ft/s (384m/s).

This paper describes in some detail the new facilities at Cross. Starting with the air supply system and associated heater, the data acquisition system, then focusing on the new back pressure facility and the new hot rig. Some test data is also shown for typical Cross Brush Seals tested in these rigs

Stiffness checking of brush seals has long been considered desirable to validate the manufacturer's control of key characteristics; this paper describes the Cross method of stiffness checking and then compares the test data to a three dimensional non-linear finite element model with varying levels of friction at the bristle rotor interface.

Air Delivery System

The air delivery system is critical and fundamental in the design of any seal test facility. The simplified schematic in Fig 1 shows the layout of the system for the test facilities at Cross. This system was installed in late 1999 and has seen a tremendous level of use ever since.

Air at 100psig is taken from the two works screw compressors through the filters and dryer into the storage tank of $1.5m^3$ capacity, the output of the compressors totals 0.6lbs/sec and 66% of that is available for rig use. From the first storage vessel the air is then fed into the pressure booster, increasing the pressure from 100psig up to 300psig. The pressure booster is a single piston double acting unit that is fitted with pulsation dampers at the inlet and outlet. The capacity of this unit is currently set at 0.4lbs/sec but can be increased to 0.8lbs/sec by a simple pulley change. This air is then fed into the second storage vessel that has $1.5m^3$ capacity.
This air is then taken by the main supply feed to the test cells after passing through the pressure regulator. This device is electrically controlled via a closed loop feed back to the control PLC and enables the pressure in the test cell to be controlled from 1psig up to 300psig. The compressed air then passes through the flow meter, we use a Spirax Sarco Gilflow unit as this has a far greater turndown ratio then conventional orifice plates. The temperature and pressure of the air is also measured at this point to enable the mass flow rate to be accurately calculated. The tee valve in the system now allows air to either be directed down into the ambient rig or through the heater system into the hot rig.

If the air is fed down into the ambient rig it can either be used in the dynamic or static test set-ups by simply changing over the flexible pipes.

If the air is directed into the heater system, it first passes a branch that acts as a heater bypass. A flow control valve sits in the branch and this regulates the volume of air that bypasses the heater and mixes with the hot air just prior to entering the hot rig. If the air has not gone down the branch it carries on into the 108kW heater via another flow control valve. Air that exits the heater mixes with the bypass flow and then enters the rig via insulated flexible pipes. The maximum temperature of the heater is 700^{o}C and this has enabled 620^{o}C to be achieved at the test seal.

The heater system is controlled via a closed loop feedback system by a PLC. The system has two control temperatures, the first is the heater outlet temperature and the second is the temperature after the bypass flow has entered the flow path. We usually run the heater just above the hottest test temperature that we require. This enables fine tuning of the temperature by using the bypass control and also enables rapid temperature changes at the manifold as shown in Fig 2.

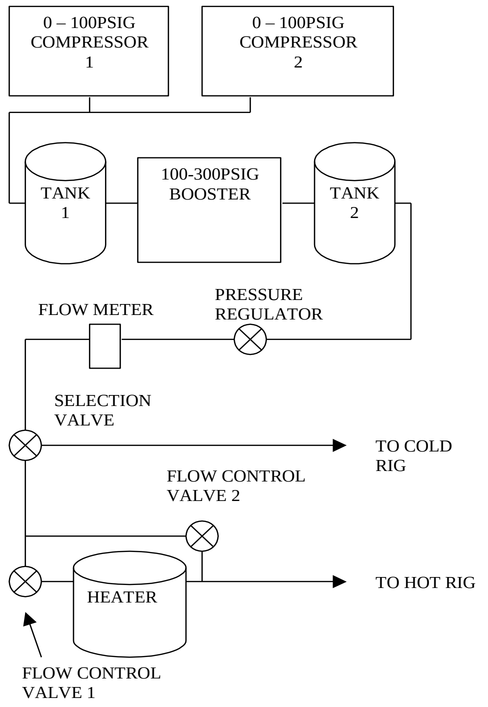

Figure 1. Schematic of Air Delivery System

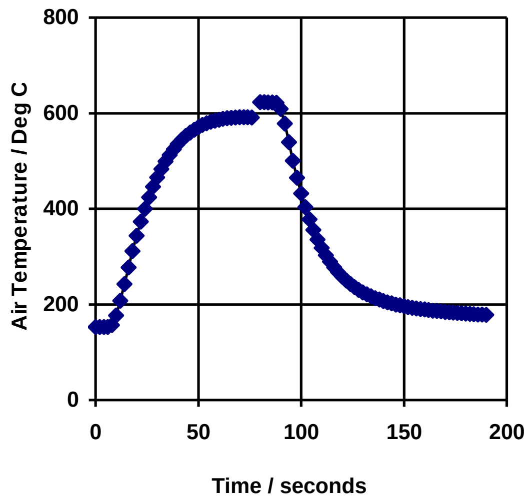

Figure 2. Temperature Response at Manifold
At 0.4 lbs/s flow rate

The temperature response at the seal is somewhat slower due to the thermal mass of the rig, obviously the higher the leakage the quicker the response is.

Data Acquisition System

The data acquisition system built into the rig allows for 85 channels of data to be stored directly into an excel spreadsheet every two seconds. The system utilises a Hewlett Packard Hp3852A unit fitted with a 6.5 digit integrating voltmeter and a high accuracy 5 channel counter. At present we are set up to log 47 type K thermocouples, 22 high accuracy pressure transducers and 16 other items such as vibration levels and mass flow. Patch panels are mounted under the hot rig to enable easy routing of mineral insulated thermocouples.

All software is written in house in the Hp VEE environment, this enables all data reduction to be performed in real time and displayed for the operator on a custom interface as shown in Fig 3. Data can either all be output into Excel or selectively stored as required.

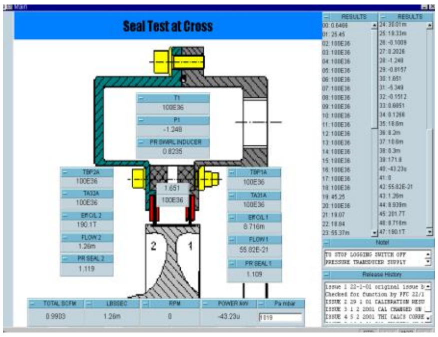

Figure 3. Typical Rig Operator Interface

Test Rigs

Cross has many static and dynamic rigs available as detailed in Table 1

Static Rigs	5.1” Dia with interchangeable disks
	7.5” Dia with interchangeable disks
	5.1” Back pressure
	Production test equipment (Segments)
Dynamic Rigs	5.1” Dia Ambient, 0 – 20000rpm with radial offsets. Many different housings and test setups available.
	7.5” Dia Hot Rig, 0- 38500rpm Ambient to 620°C. Different housings available with differing thermal masses.

Table 1. Cross Test Rigs

We are now going to focus in more detail on some of these rigs, namely the back pressure rig and the 7.5” hot rig. The original 5.1” dynamic rig, as described by Flower[1] is still in regular use and has undergone many modifications over the years to simulate gas and steam turbine seal locations. This unit is also now fitted with a video borescope unit with a stroboscopic light source to enable bristle movements to be captured accurately.

5.1” Back Pressure Rig

This rig was designed to enable a 5.1” dia seal to be tested at pressures of up to 300psig, with down stream pressures of up to 300psig also. The down stream pressure is controlled by a remotely controlled back pressure maintaining valve that is tolerant to variations in flow rate. A photo of the rig is shown in Fig 4, the rig also has an axially movable rotor with a pressure tap in it and taps for back plate pressures.

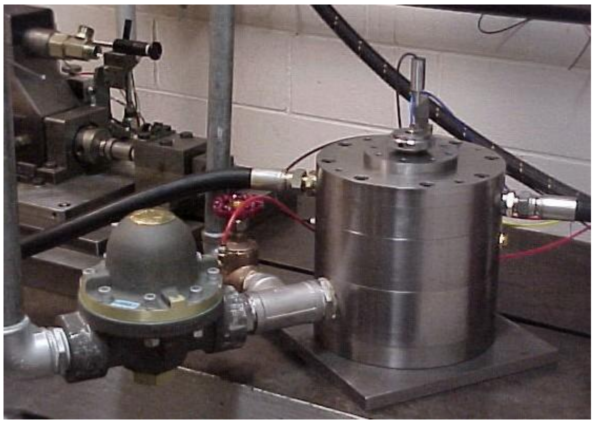

Figure 4. Back Pressure Rig

The data presented here uses the effective clearance function as defined below:-

$$\text{clearance\_eff} = \frac{q\sqrt{(T1+460)}}{P1.\,w.\,\mu}$$

μ the flow function is defined as follows:-

$$\mu = \sqrt{g.\gamma / R}.\sqrt{((2/(\gamma-1)).PR^{-(\gamma+1)/\gamma}.(PR^{(\gamma-1)/\gamma}-1))}$$

For choked flow (where PR>1.89) the flow function is held at the value for a Mach number of 1 at the seal.

The two graphs shown in Figures 5 and 6 are different representations of the same data from the same Cross brush seal . Figure 5 shows effective clearance plotted against pressure drop for five different up stream pressures and Figure 6 shows the same data plotted against pressure ratio.

The seal tested is typical of a brush seal for use in a industrial gas turbine but has a higher fence height of 0.120" and a lower bristle density. These factors reduce the pressure capacity of the seal to about 55psi.

The tests were run by holding the upstream pressure at one of five set values and then adjusting the down stream pressure in 5psi steps to give a pressure drop from 0 to 75psi for each upstream pressure.

From the two graphs it is very apparent that the driving function is pressure drop related. Similar data was previously presented in Ref. 2. Here we have a much greater range of pressure ratios and it is clear that the pressure ratio does have an effect on the pressure capacity of the seal. As the pressure ratio across the seal changes so does the pressure distribution through the pack, this rig has been designed to further explore this effect so as to better understand the complex workings of the simple brush seal.

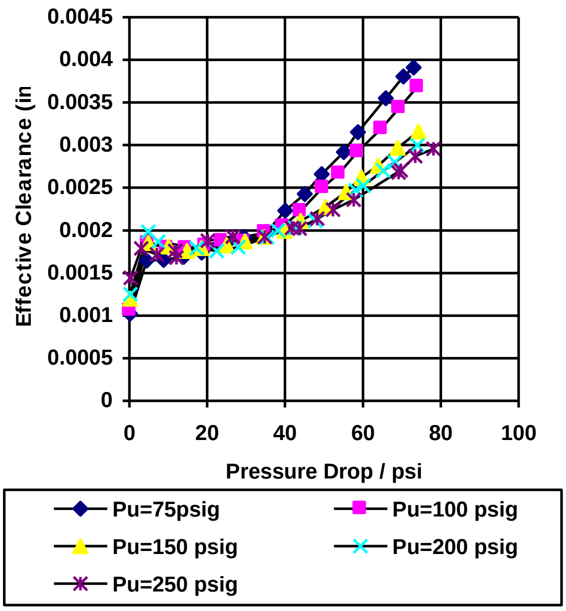

Figure 5. Graph of Effective Clearance against Pressure Drop.

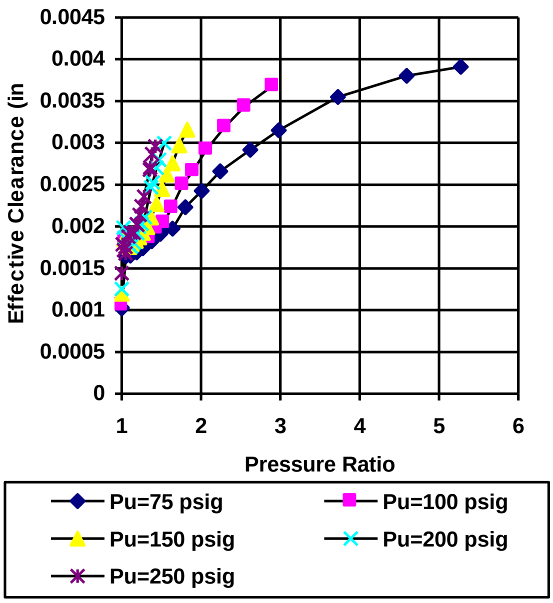

Figure 6. Graph of Effective Clearance against Pressure Ratio

The back pressure rig is now starting to yield valuable data into the behaviour of brush seals in high pressure low pressure ratio conditions typical of those found in many locations of gas and steam turbines. We are continuing to run further tests to enable better characterisation of the performance of the brush seal under varying operating conditions.

7.5" Diameter Hot Rig

The 7.5" hot rig was brought from concept in late 1998 to fully commissioned in late 1999. The concept behind the rig was to provide a test platform onto which different housing arrangements could be mounted to simulate aerospace and industrial applications with different thermal masses.

The air delivery system has already been discussed so I will now describe the spindle drive system.

The rig is powered by a 50hp (37kW) electric motor that is fitted with force fan cooling and an optical encoder. The inverter that controls this has a closed loop feed back system that is controlled via the rig control PLC. The speed control is such that the motor will run from 30 – 6000rpm giving a spindle speed range of 200- 38500rpm. This higher speed represents a surface speed of 380m/s (1250ft/s) with the 7.5" diameter disks typically used. The closed loop feed back system ensures that the speed is maintained constant under varying load conditions. The inverter is also fitted with a separate braking circuit, this allows the deceleration of the disk to match the acceleration rates. Acceleration and deceleration rates are ultimately controlled by the inertia of the disk. Typically we can accelerate from zero to full speed in 30 second and stop in the same time. This rapid speed change capability allows the rig to be driven like a gas turbine, the pressure response is as quick so it is only the temperature that tends to lag.

The spindle is driven via a flat belt drive as indicated in Figure 7.

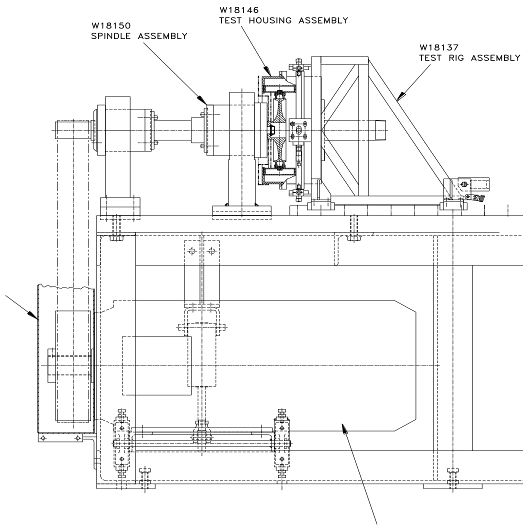

Figure 7. Drive Arrangement for Hot Rig

Both spindles utilise angular contact bearings fitted with silicon nitride balls to enable high speeds to be achieved. The bearings are oil cooled and the nose of the disk spindle is also air cooled. Cross turbo charger sealing rings are used as the primary oil seal on both spindles and they have proved very reliable under all operating conditions.

Figure 8 shows the housing arrangement used to simulate an aerospace application where rapid response was required. To further simulate the engine the disk was machined with typically 0.005" of run-out and the air was bought in between the seals through a swirl inducer. The level of swirl is governed by the pressure ratio across the swirl inducer holes, these are easily changed to allow pressure ratios from fully choked down to low numbers to be achieved.

When we are testing for industrial applications we tend to use much heavier test housings so as to give a more realistic thermal response to the application. The swirl inducer may or may not be used depending on the particular application.

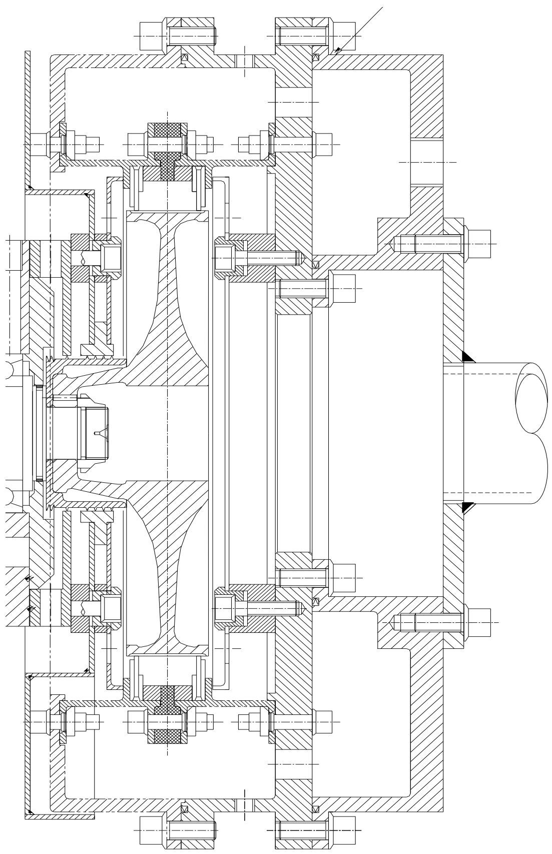

Figure 8. Section through Test Housings

Stiffness Measurement

At Cross we have over the years looked at stiffness measurements, in fact we first carried out this type of measurement in 1984. The equipment that we now use is shown in Figure 9. This adaptation of a standard tensile testing machine allows for a flexible machine that can be used on round and segment seals.

Figure 9. Stiffness Testing machine

Tests are performed by bringing a shoe, shaped to the curvature of the bristle bore, into contact with the bristles and then displacing it a further 0.040” and then retracting it slowly.

A typical response curve from this type of test is shown in Figure 10. This data was taken from a typical test rig type seal and clearly shows the characteristic hysterisis loop that all brush seals produce.

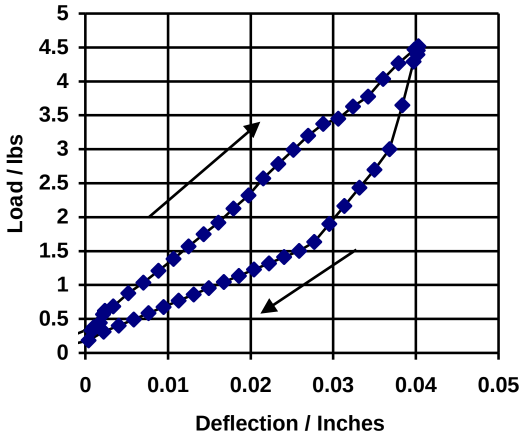

Figure 10. Typical seal stiffness curve

Trying to relate the measured data to the theoretical stiffness of the seal has always proved somewhat difficult. The loop in the experimental data leads to two different seal stiffness values, one high value taken as the bristles compress and one low value as the bristles recover. We have worked on averaging these two values but this still tends to give an error when compared to the theoretical data.

In order to reduce the error it was decided that two different lengths of contact shoe would be used and the stiffness calculated by subtracting the data from the shorter one from the longer one. This then takes out all the end errors and gives more meaningful data. This data still shows a large hysterisis loop, it was surmised that this loop was caused by friction at the bristle shoe interface and between the bristles themselves.

Finite Element Bristle Model

We were concerned at the large difference given by the hysterisis loop in the stiffness values so we set about building a finite element model of a typical bristle. The model was set up with contact elements at the tip of the bristle. We selected the Adina package to perform the analysis, this proved very easy to set up. The coefficient of friction was varied at the tip of the seal to see the extent of the hysterisis loop produced. Two views of the model are shown in Figure 11.

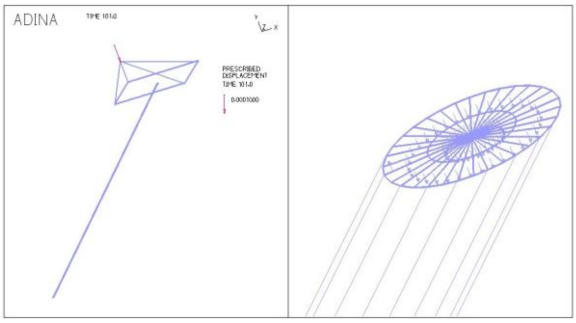

Figure 11. Bristle FEA Model

The model was built using 27 noded 3d large deflection elements. Contact was introduced at the tip using 3d contact elements, the contact surface that was brought down onto the bristle was assumed to be rigid.

Results from the model are shown in Figure 12. It is clear that as we increase the coefficient of friction the characteristic hysterisis loop appears. However even at a coefficient of friction of 0.4 the loop is still of a lower magnitude than that of the test data. Work is underway to further develop the model to include the friction between adjacent bristles and at the bristle to back plate interface.

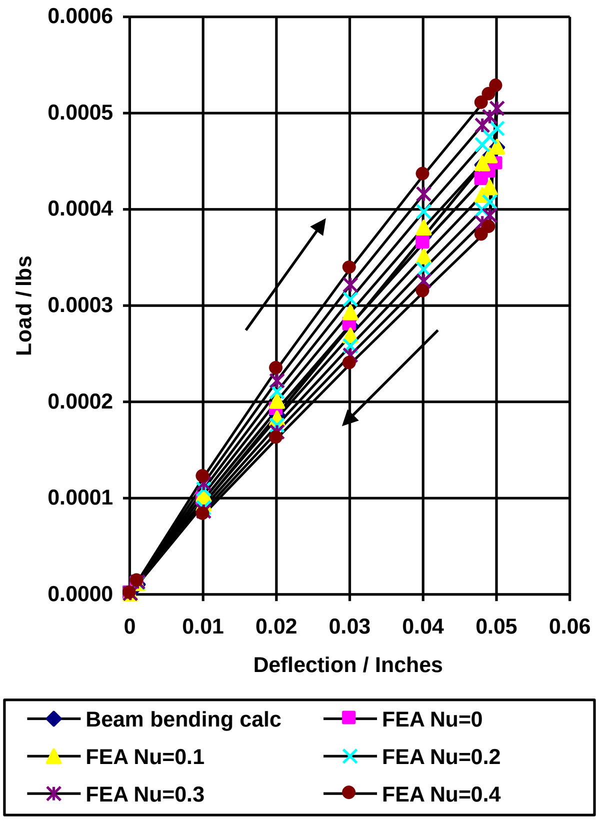

Figure 12. FEA Bristle Response Curve

Cross Brush Seals

In order to meet the extreme demands now put on brush seals Cross continues to further develop it's pressure balanced design. This design that was originally developed in the summer of 1993 was first made available commercially in February 94 after extensive testing. This cavity back plate design continues to be refined, with variations on the number and depth of pockets in order to give the optimum balance of seal flexibility and wear. A typical Cross pressure balanced design is shown in Figure 13. The deep front plate design has continued to prove effective in minimising the disruption of the bristle pack caused by high swirl ratios.

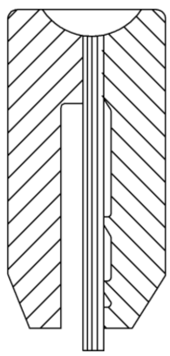

Figure 13. Cross Brush Seal Section

These seals have now been successfully applied to industrial and aerospace gas turbines, steam turbines and other industrial compressors

Work is also continuing on the ceramic seals that we first made in 1992[3]. We are also looking again at other non-metallic fibres, similar to some of the work that we originally carried out back in 1990, for some specific applications.

Conclusions

Brush seal development continues at an increased pace at Cross. We now have a much more comprehensive range of test facilities available including the new back pressure rig and the new hot high speed facility.

We have worked hard to bring closure to a stiffness measuring program and have been developing a finite element model of the bristles to validate the experimental data.

Brush seals are now being successfully applied in a wide range of industrial and aerospace applications.

Acknowledgements

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Special thanks go to Aaron Bowsher for all his work on many aspects of this paper and his assistance over the past three years. Mark Bolwell at Cross is also thanked for gathering all the test data shown. I would also like to thank all the brush seal production staff at Cross for performing a magnificent job under all circumstances.

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