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## Brush Seal Pack Hysteresis

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### Abstract

The hysteresis loop that brush seals produce when they are stiffness checked has proved hard to evaluate using simple beam theory.

The development of a Finite Element Analysis model of a brush seal pack is tracked through from simple beam element models through to complex multi layer non-linear models. The simple beam element models show a good match to simple beam theory but not such a good match to test data. The more complex models exhibit a good match to the test data but an increased stiffness when compared to simple beam theory. The increased stiffness is due mainly to bristle-bristle interaction.

Test data is shown on seal blow down and pressure driven stiffening that will be used to further evaluate the model of the brush seal pack

### Introduction

Cross has been producing brush seals since the 1970's for the Aerospace Industry and was instrumental in their introduction to the Power Generation Industry in the 1990's. Cross brush seals continue to be applied to a wide range of industrial and aerospace applications, with particularly strong growth in industrial applications. We continue to offer a high quality product with a consistent bristle pack that is designed for each specific application, as shown in Figure 1.

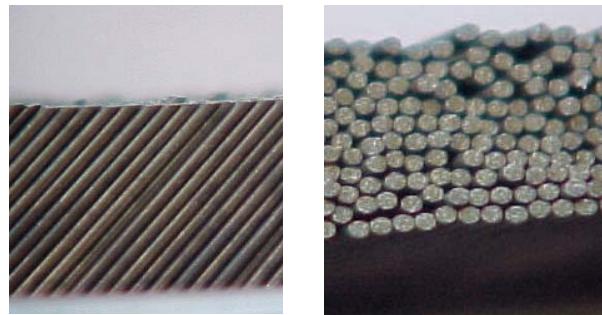


Figure 1. Typical Brush Seal Pack Quality

### Nomenclature

d	Wire diameter (inches)
L	Free wire length (inches)
E	Young's modulus (psi)
$\theta$	Bristle angle (degrees)
$\mu$	Coefficient of Friction
Wire_Tip_Area	Tip area of single bristle
Stiffness	Theoretical stiffness of a single bristle during a 0.001" radial offset (lbs)
BTP	Theoretical Bristle Tip Pressure of a single bristle during a 0.001" radial offset. (psi)

This paper describes the typical hysteresis loop produced when stiffness checking a brush seal and then goes on to reproduce these characteristics by using a Finite Element Analysis model of the bristle pack. This paper describes the initial stages of development of the Finite Element Analysis model from simple single beam element models, through to non-linear multi layer multi bristle 3D models with bristle to bristle and bristle to back plate contact. All models have contact elements at the tips of the bristles; comparisons are drawn to simple beam theory and to test data. Some coefficients of friction are experimentally determined and then used in one of the models.

Further test data is also shown that will be used to validate the model as it evolves. This test data shows some typical blow down characteristics and pressure stiffening effects of brush seals.

## Simple Beam Theory

Simple beam theory has been used for many years to characterise the contact forces exerted by brush seals under idealized conditions. Flower<sup>1</sup> presented the following equations in 1990. These equations have been in use since then by many people.

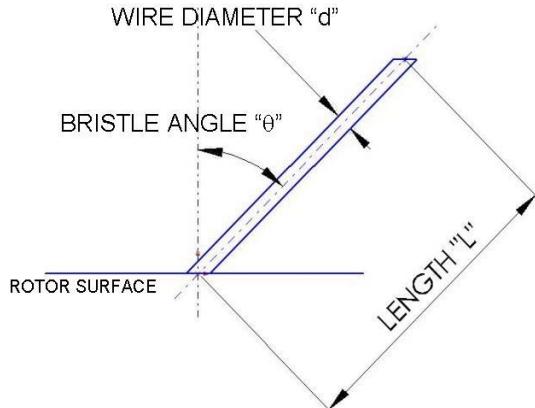


Figure 2. Bristle Geometry.

$$\text{Wire\_Tip\_Area} = \frac{\pi d^2}{4 \cos \theta} \quad (1)$$

$$\text{Stiffness} = \frac{Ed^4}{6790L^3 \sin^2 \theta} \quad (2)$$

$$BTP = \frac{Ed^2 \cos \theta}{5333L^3 \sin^2 \theta} \quad (3)$$

Both of equations (2) and (3) express load or pressure as a result of a 0.001" radial deflection of a single bristle.

Note how the bristle angle is struck from the bore of the seal, this is the only place on the seal that you can always see the bristles and thus measure the angle. The difference between the angle at the bore and measured at the outside diameter can be considerable on small diameter seals.

Some people tend to get confused between the Bristle Tip Pressure (BTP) and the stiffness. The BTP is a good way of comparing seals with differing wire sizes.

## Stiffness Measurement

As we presented in 2001<sup>2</sup> we have looked at stiffness checking brush seals since 1984. Figure 3 shows our adaptation of a standard tensile test machine, this allows for a flexible machine that can be used on round and segment seals.



Figure 3. Stiffness Testing machine

The typical hysteresis curve obtained from a test is shown in Figure 4. The test is 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.

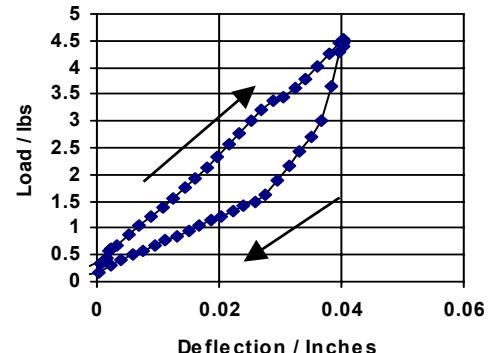


Figure 4. Typical seal stiffness curve

It is clear from the hysteresis curve above that friction is an important factor in the stiffness of a brush seal. 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.

In order to reduce the end errors we use two different lengths of shoe and the stiffness calculated by subtracting the data from the shorter one from the longer one.

## Finite Element Bristle Models

From our initial finite element data that we presented in 2001 it was clear that friction is a major influence on the brush seal. We reviewed the published data by Aksit<sup>3</sup> and Aksit and Tichy<sup>4</sup> and decided that we should approach the problem in the following 4 stages.

- Stage 1 Single bristle beam element problem.
- Stage 2 Ten Bristle single layer beam element problem.
- Stage 3 Five bristle 3D single layer 27 node element problem.
- Stage 4 twenty-five bristle 3D 5 layer 27 node element problem.

In all cases contact would be introduced between the bristle tip and the shoe with friction varying from 0 up to 0.4. For each stage 5 different bristle cases were run, these all had the same theoretical stiffness but differing angles and free lengths as shown in Table 1. All bristles were 0.0056" in diameter and a Youngs Modulus of 30E6 psi was used for all calculations.

Free Length inches	Bristle Angle Degrees	Stiffness Lbs/0.001"	BTP Psi/0.001"
1.1496	35	8.693E-6	0.289
1.0656	40	8.693E-6	0.270
1	45	8.693E-6	0.249
0.9480	50	8.693E-6	0.227
0.9066	55	8.693E-6	0.202

Table 1 Basic Bristle Data.

Stages 2, 3 and 4 would also have contact between the bristles and the back plate and between adjacent bristles. All models were run so as to compress the bristles 0.050" and then release them, typically this was accomplished with 10 equal compression steps and then 10 equal steps to release them. We continued to use the commercial FEA package Adina for all this work. Adina is very good at solving contact problems and is used to solve many metal forming problems in industry. All problems were solved on a single processor P.C. with 394Mb of RAM.

## Single Bristle Beam Element Model

This model was constructed using 20 2d beam elements, with contact at the bristle tips. This model proved very fast to run, typically solving 20 steps in less than 1 second. A view of the model is shown in Figure 5.

Results from this model are summarized in Figure 6 by comparing the data at 0.010" deflection with the simple beam theory calculation.

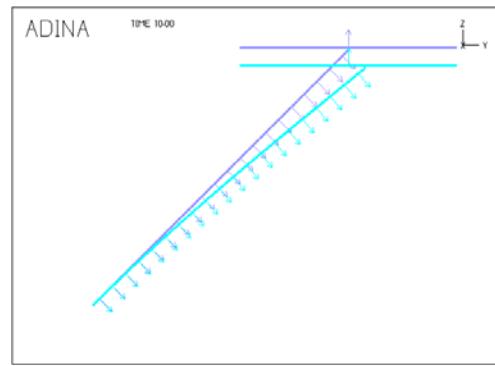


Figure 5. Single Bristle Beam Element Model

Note how with no friction present the FEA data is very close to the beam theory, however as friction is introduced this has a much greater effect on the lower bristle angles.

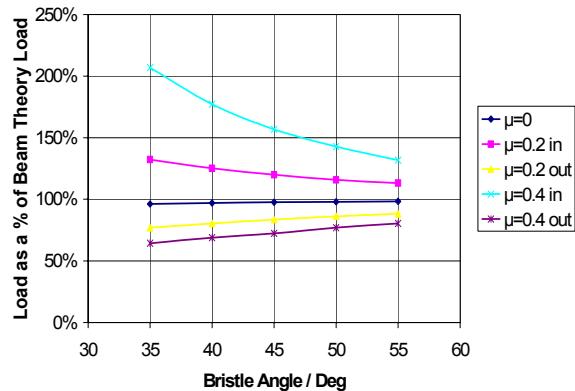


Figure 6. Single Beam FEA Data

This simple model was encouraging in the close match to the simple beam bending theory and opened our eyes to the effect bristle angle can have on the hysteresis.

## 10 Bristle Beam Element Model

Based on the encouraging data from the single bristle model we put together a parametric 10 bristle model using beam elements. Again 20 elements were used per bristle and contact was introduced between the bristle tips and the shoe and between adjacent bristles. This was achieved by using offset contact surfaces. The model is shown in Figure 7. Comparisons with the simple beam theory are made in Figure 8. Clearly this is similar to the single beam

model except that there is some additional stiffening of the pack caused by the bristle to bristle interaction.

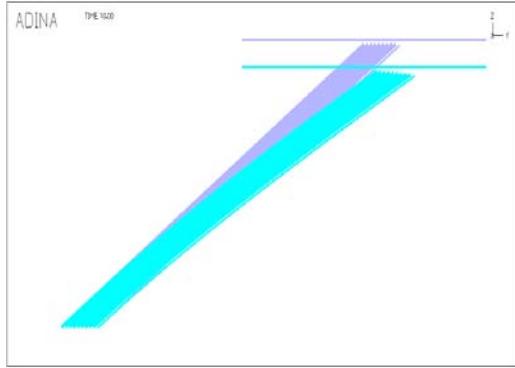


Figure 7. Ten Beam 2D Model

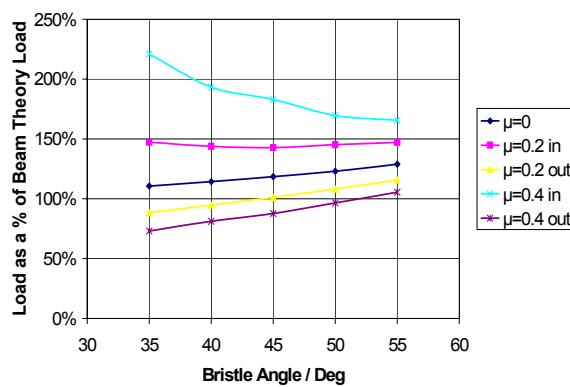


Figure 8 Data from 10 Beam FEA Model

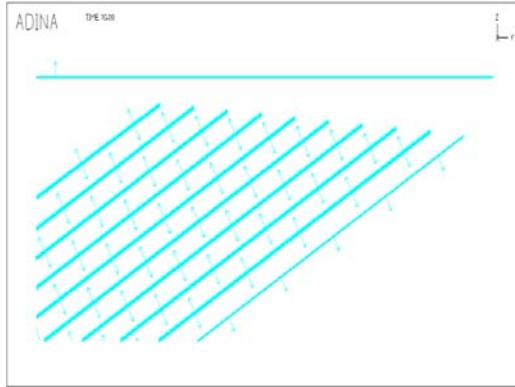


Figure 9 Deformed Shape of 10 Beam Model

Study of the deformed bristle shape as shown in Figure 9 leads to a problem with this model. The pack has not compressed evenly and the last bristle has compressed further than the first one. The beam elements do not allow for motion in the axial plane and it is not possible to transfer deflections of the first beam onto

the last beam to achieve symmetry. Clearly a 3D model would be required to fully model the bristle pack.

### 5 Bristle 3D Element Model

Based on the experience from the earlier studies a 3D parametric model was built. This model used 400 27 node solid elements and was constructed with contact at the bristle tips, back plate interface and bristle to bristle. The contact used is full 3D surface to surface contact, this allows any part of a bristle to touch any part of an adjacent bristle. The boundary conditions were also set so that the motion of the first and last bristles acted in a master-slave relationship so as to achieve symmetry. An isometric and close up view of the model is shown in Figure 10.

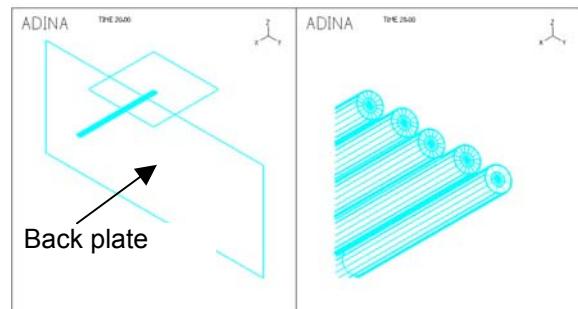


Figure 10 Five Bristle 3D Model

Friction values can be set independently for the bristle to rotor interface, the bristle to back plate interface and for bristle to bristle contact. The geometry for the model is created in an Excel spreadsheet and this data is simply cut and pasted into the input file to change bristle geometries. Bristle spacing was idealized with adjacent bristles just touching each other. Typical run times were less than 1 hour to solve 40 time steps, this time increased slightly as friction was increased. Convergence of the model proved more problematic with friction and at higher levels of friction the model would stop converging as the load was being reduced, lack of time has prevented us from rectifying this problem to date.

Data from this model is compared with the simple beam theory in Figure 11. There is now clearly a discrepancy between the beam theory and the FEA data, this is mainly due to the stiffening effect of the adjacent bristles. Note how the bristle angle still has a large effect, as the bristle angle reduces the hysteresis loop becomes larger and as the angle increases the basic stiffness with no friction increases.

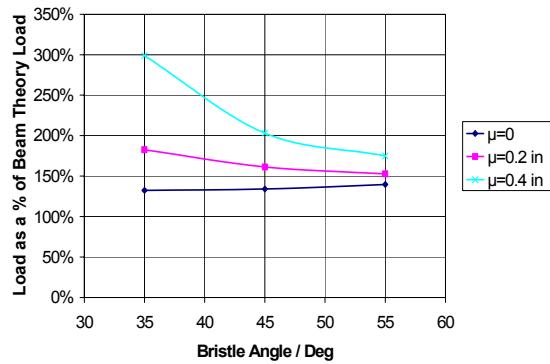


Figure 11 Data from 5 Bristle 3D FEA Model

The deflected bristle pack shape, as shown in Figure 12, now clearly shows axial movement of the bristle pack and also all bristles contacting the shoe surface. The deflection characteristics are clearly representative of actual brush seal bristle packs.

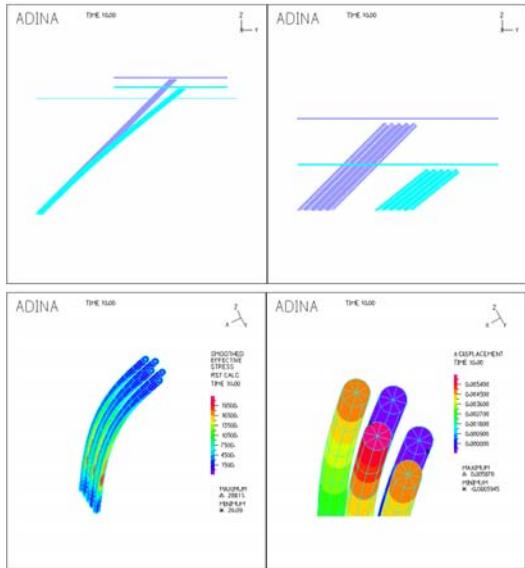


Figure 12 Deflected Shape of 5 Bristle Model

Examination of the output file revealed significant contact forces between the bristles and the back plate. This leads to increased hysteresis as can be seen in the graph in Figure 11. This additional load and resulting hysteresis is due in main to the boundary conditions that have linked the motion of the first and last bristles together. This is important because in an actual brush seal there are many thousands of wires and all the wires must move together during transient closures. As the vast majority of brush seals are circular there is no additional circumferential space available for the bristles to move into, they have to move in the axial direction as indicated by this model. As one layer of bristles had shown significant axial movement it was

considered that a multi layer would have to show significantly more axial movement and resulting back plate contact forces.

## 25 Bristle 5 Layer 3D Element Model

As a direct result of the data obtained from the single layer model a parametric 5-layer model was constructed. This model has 2000 3D solid elements. Again this had contact at the bristle tips, back plate interface and bristle to bristle, with friction values independently settable. Bristle spacing was again idealized with all adjacent bristles just in contact with each other; the layers were staggered to give the densest packing possible. Boundary conditions were set on layers 1,3 and 5 so the first and last bristles of each layer acted in a master-slave relationship for symmetry. Views of this model are shown in Figure 13.

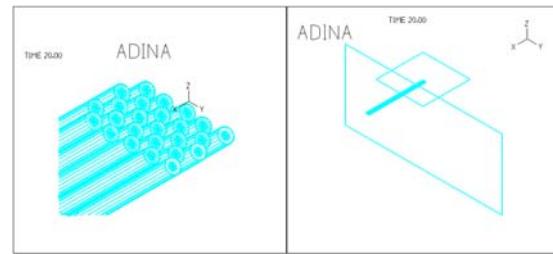


Figure 13. 5 Layer Bristle Model

To date we have only run this model without friction, the deflected bristle shape appears to be representative of the true bristle shapes as shown below in Figure 14.

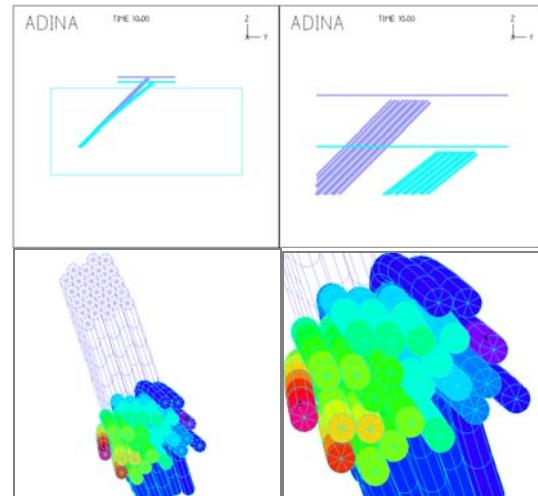


Figure 14. Deflected Shape of 5 Layer Model

The axial width of the pack thickens up from 0.025" to 0.0443" an increase of 77% during the 0.050" radial compression. Measurements on actual brush seals with 10 layers of bristles and similar geometry reveal that the pack thickens up by 10 to 25% during a 0.050" radial closure with no pressure drop. The initial thickness of the pack is however 15% thicker per layer than our model; this indicates that there is more space in the seal to start with so the thickness of the pack will increase less. The FEA model as it stands is increasing the axial thickness too much, this is clearly due to the excessively tight bristle packing we have and possibly due to the small number of bristles we have in the circumferential direction. Further work needs to be carried out to validate this model. We need to look at the effects of increase the number of bristles per layer and look at the effects of initial bristle spacing. This initial spacing is easy to change in the Excel spreadsheet used for data input. We will probably evaluate this firstly using the single layer model, as it is much faster to run. Initial data from this model has been encouraging.

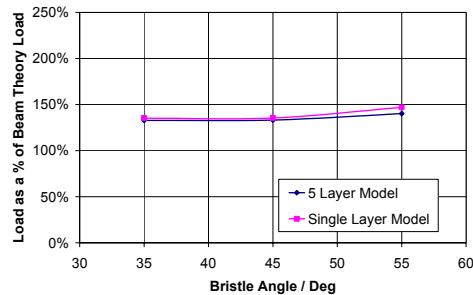


Figure 15. FEA Load data for Multi-Layer Model

The graph in Figure 15 shows how the load averaged per bristle now relates to the simple beam theory. This data is all for a coefficient of friction of zero and at a deflection of 0.0045". The multi layer data is slightly lower than the single layer model; this may be due to the different boundary conditions present on layers 2 and 4 where we do not have master-slave end bristles. At higher deflections there is typically less than 1% difference between the single layer and multi layer data at coefficients of friction of zero.

### Coefficients of Friction

Data on the coefficient of friction of Haynes 25, the most common brush seal bristle material, against it self and other alloys is not easy to track down. Fellenstein and Delecorte<sup>5</sup> quote values of 0.25 to 0.47 for Haynes 25 against Inco 718 and then in a later paper<sup>6</sup> quote 0.25 to 0.32 for Haynes 25 against Chrome Carbide coatings. Because of this lack of published

information we set about measuring the coefficient of friction of Haynes 25 against it self and typical back plate materials so these could be used in our models. We set up some simple bench tests at contact pressures of 1,10 and 100psi against sheet material. We obtained values in the range of 0.18 to 0.35 for Haynes 25 against it self, with most steady state readings at about 0.2. Against some of the popular stainless steels used for back plate materials we had a range of 0.26 to 0.46 with most of the steady state readings at 0.28. Most of the extreme high values were obtained at the lightest contact pressure where experimental errors were at their greatest. Thus in our further work we will be using values of 0.2 for bristle to bristle contact and 0.28 for bristle to back plate contact. The bristle to rotor contact will clearly depend on the rotor material chosen.

We have used these numbers in the single layer 3D model and obtained the hysteresis loop shown in Figure 16. The FEA data is compared to some actual test data

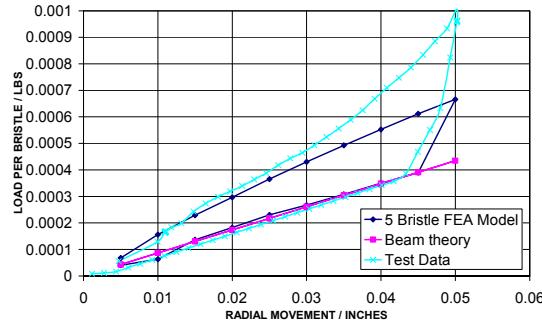


Figure 16. FEA Hysteresis Loop Comparison

The match with the test data is pretty good for displacements up to 0.030", above this displacement the axial displacement of the pack becomes more of a factor and the load increases accordingly. As the FEA data is from a single layer model there is little allowance for this effect and thus the mismatch with the test data.

### Effect of FEA Data On Stiffness Checking

From the work performed to date on the FEA models it has become apparent why we have always had some difficulty relating the measured stiffness of a brush seal to the simple beam theory. The bristle angle has a major effect on the magnitude of the hysteresis loop as does the coefficient of friction. When bristle to bristle interaction is taken into account the basic, no friction, stiffness is 30 to 40% greater than that given by simple beam theory. Stiffness checking can thus only prove accurate for a seal with a consistent bristle angle and even then allowances must be made for stiffening due to bristle to bristle interaction.

## Further Development of the FEA Model

Initial data from the FEA models has proved encouraging; we will continue to refine the model to get a better match to the experimental stiffness data. Once this is complete we will have a basic model that accurately matches the physical characteristics of a brush seal pack, this will then allow us to apply pressure drops to the model. We have measured the pressure distribution down many styles of back plate and at the bristle rotor interface, this information will be used in the model to assess some of the real brush seal operating characteristics such as blow down and transient hysteresis. Some of these characteristics are shown experimentally in the next sections of this paper.

## Brush Seal Test Data

In order to validate future FEA models we set up a test program using 12 brush seals. These seals were of two different diameters, used one wire size, and had 4 different values of theoretical Bristle Tip Pressure and two different styles of back plate designs. We had also varied the bristle angle on some of the designs.

We first looked at the blow down characteristics of each seal. We do this by statically testing the seal with 5 different rotor sizes, one being interference, one being the same size as the bristle bore and the other three being clearance sizes. Typical data obtained is shown in Figure 17.

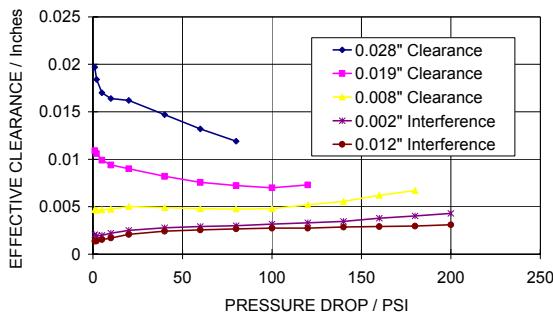


Figure 17. Typical Blow Down Data

Figure 18 shows how the blow down is influenced by the bristle angle and also the bristle tip pressure. All the data is plotted against pressure drop with ambient down stream conditions. We have looked at the effect of pressure ratio by adding back pressure to the down stream side of the seal and find that the pressure drop

the controlling function. The pressure ratio does change the pressure distribution through the bristle pack but it appears to have little effect on the blow down.

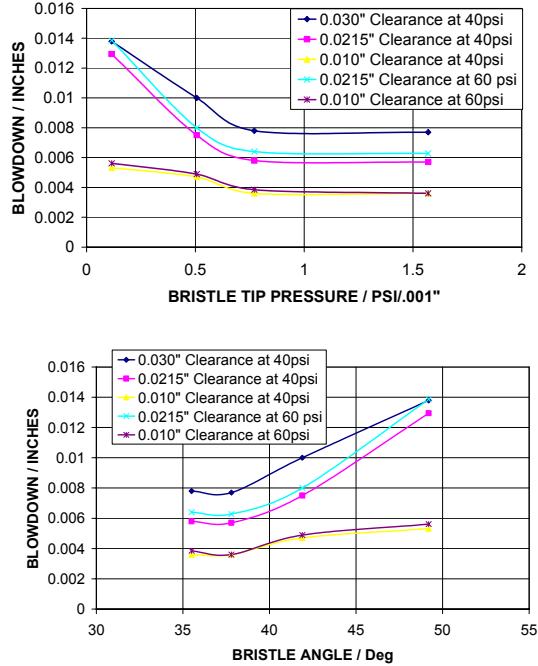


Figure 18 Blow Down against BTP and Bristle Angle

We have also been looking at how the pressure drop effects the torque generated by the seal, this gives an indication of the pressure stiffening of the seal. Some typical data is shown in Figure 19.

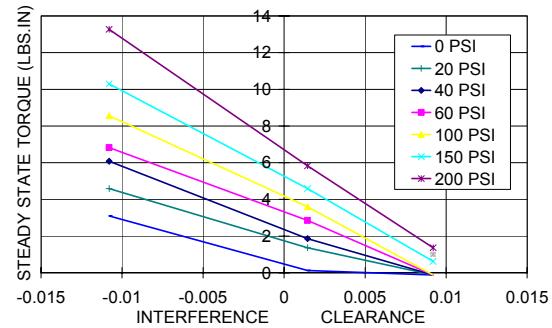


Figure 19 Typical Torque Characteristics

Again once the FEA model is fully complete we will try to match this type of data to confirm that we have correctly modelled the brush seal. It is interesting to note the degree in which the torque and thus the stiffness of the seal increases with pressure drop.

## Conclusions

The hysteresis loop produced when stiffness checking a brush seal is examined and its reproduction simulated using Finite Element Analysis models. Single beam element models match well the data given by simple beam theory but underestimate the test loads. Multi beam element models match test data better but show inconsistencies with regard to actual geometric deflections. Single layer 3D models show a good match to test data up to 0.030" deflection but underestimate the loads at higher deflections. Initial indications from a multi layer model are encouraging and the model is being developed further with more realistic bristle spacing.

## Acknowledgements

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## References

1. Flower, R., “*Brush Seal Development System*”, AIAA Paper No 90-2143.
2. Crudgington, P., “*Recent Brush Seal and Testing Developments at Cross*”, AIAA Paper No 2001-3480.
3. Aksit M. F. “*A Computational Study of Brush Seal Contact Loads with Friction*”, PhD Thesis Rensselaer Polytechnic Institute (1998)
4. Aksit M.F. and Tichy J.A. “*A Computational Model of Brush Seal Bristle Deflection*”, AIAA Paper No 96-2909
5. Fellenstein J.A. and DellaCorte C. “*A New Tribological Test for Candidate Brush Seal Materials Evaluation*”, NASA TM-10675 (1995)
6. Fellenstein J.A., DellaCorte C, Moore K.D, Boyes E. “*High Temperature Brush Seal Tuft Testing of Selected Nickle-Chrome and Cobalt-Chrome Superalloys*”, AIAA Paper No 97-2634.