

Bristle Angle Effects on Brush Seal Contact Pressures

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This paper examines, analytically and experimentally, how the bristle angle affects the operating contact pressure of typical brush seals at pressure drops up to 50psi. Experimental data has been gathered on two seals, one with a bristle angle low in the typical range and the other high. The test data is compared to a full 3D FEA bristle pack model. An equation is presented to represent the contact pressure of a seal with interference and pressure drop, this will allow more accurate heat generation numbers than traditional single bristle beam theory methods.

Nomenclature

A_t	=	Bristle tip area
C_p	=	Tip force correction factor
d	=	Bristle wire diameter
D_b	=	Bristle bore diameter
D_f	=	Pinch point diameter
E	=	Young's modulus
L	=	Bristle free length
P_s	=	Bristle to disk contact pressure (psi)
BTP	=	Bristle Tip Pressure (psi/0.001" of radial deflection)
δ	=	Radial bristle to disk interference
ΔP	=	Pressure drop (psi)
θ	=	Bristle angle (Measured at the bristle bore)
μ	=	Coefficient of friction

I. Introduction

Brush seals have now been used for many years in Aerospace Gas Turbines, Power Generation Gas Turbines and Steam Turbines, they offer significantly improved sealing performance when compared to traditional labyrinth seals¹. Since their original application in Aerospace Gas Turbines in the 1980's their use has now spread widely and they are fitted on many units made by a number of different manufacturers. They have demonstrated improved sealing performance in numerous applications but since their first use there have been concerns about the heat they generate in rub situations.

This paper investigates how the contact pressure that a brush seal exerts on its disk is controlled by the bristle angle of the seal. Simple beam bending relationships are first examined, and then test results from two brush seals with different bristle angles are compared. The two seals are leakage tested, stiffness tested and torque tested using three different methods. Test results are then compared to results from a 3D Finite Element Method model of a bristle pack of identical geometry.

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II. Simple Beam Theory Calculations

Many people have presented simple beam theory calculations, the first being Flower² and more recently Demiroglu et al³. The details that follow show the Cross method of calculating the free bristle length, the bristle tip stiffness, the bristle tip area and finally the bristle tip pressure. It has been the Cross convention for the past 20 years to express bristle tip stiffness and bristle tip pressure per 0.001" radial displacement of the bristle tip.

Firstly we have to accurately calculate the free bristle length, Figure 1 shows the seal geometry, note how the bristle angle must be measured at the bristle bore as this is the only place on all brush seals that you can always see the bristles. The method we use is a combination of the sin and cosine rule to produce Equation (1).

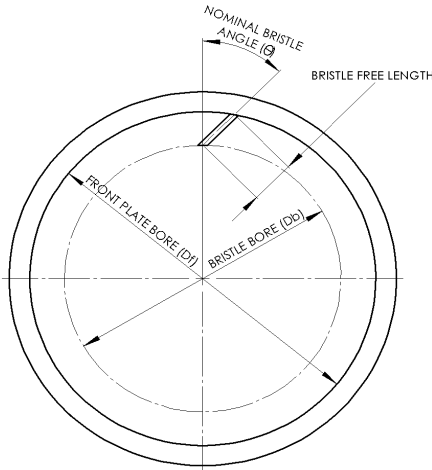


Figure 1. Brush Seal Geometry.

$$L = \sqrt{\frac{D_b^2}{4} + \frac{D_f^2}{4} - \frac{D_b \times D_f}{2} \times \cos\left[\theta - \arcsin\left(\frac{D_b}{D_f} \times \sin \theta\right)\right]} \quad (1)$$

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

$$Wire_Tip_Area = \frac{\pi d^2}{4 \cos \theta} \quad (3)$$

$$BTP = \frac{Ed^2 \cos \theta}{5333L^3 \sin^2 \theta} \quad (4)$$

The equation used for calculating the bristle tip stiffness is shown in Equation (2), the results from this are typically quoted in lbs/0.001" radial displacement. About 15 years ago we realized that it would be better to quote Bristle Tip Pressure (BTP) rather than stiffness as this would then allow direct comparison of seals with different size wires. The bristle tip area of a single inclined bristle is shown in Equation (3), Equation (4) for the BTP is then obtained by dividing Eq. (2) by Eq. (3).

Figure 2 shows the results of Eq. 1 and 4, the data shown is for a 5.1" bore test rig real with 0.0056" diameter bristles. The design BTP of this seal is 1psi/0.001" when the bristle angle is at 45°, with all other geometry remaining unchanged the bristle angle has been varied from 20° to 60°, it is very clear how much these changes affect both the Free Length and also the BTP. The typical bristle angle tolerance is +/-5° so for our seal the BTP would actually vary from 1.57psi/0.001" to 0.63psi/0.001".

The simple beam bending calculation gives an insight into the BTP of a seal with no pressure drop applied across the seal, but how does it compare to the measured BTP of brush seals? To do that we need to investigate some results from stiffness checks carried out on brush seals.

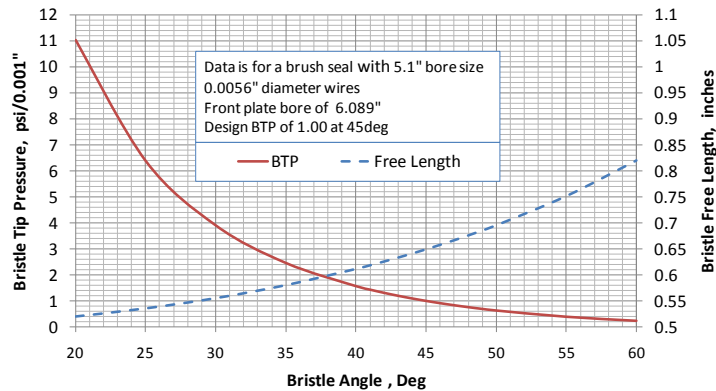


Figure 2. Effect of Bristle Angle on Free Length and BTP.

III. Stiffness Checking

Cross first stiffness checked a brush seal in 1984. Figure 3 shows our current stiffness testing machine, this is an adaption of a Lloyd standard tensile test machine. The procedure for testing a seal is quite lengthy to get an accurate answer and is as follows. Firstly the seal is tested in 4 places with a 1" wide test shoe, at each place it is tested twice, the test is then repeated in the same locations using a 0.5" wide shoe. The results from the 0.5" wide shoe are then taken away from those of the 1" wide shoe and the BTP calculated.

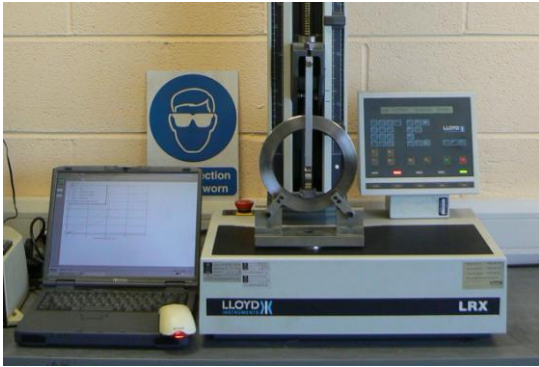


Figure 3 Stiffness Checking Machine.

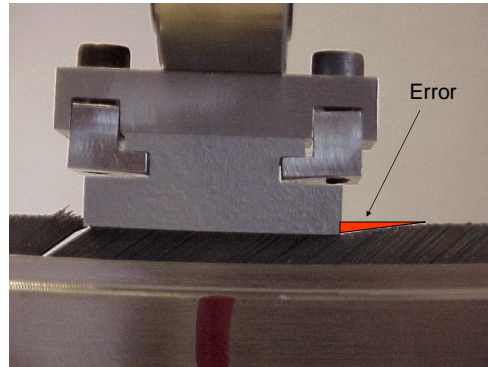


Figure 4 Additional Bristles that Move.

Figure 4 show the reason for testing with the two shoes, failure to do this results is a higher than predicted BTP being measures as the bristle adjacent to the shoe are also being deflected. By testing with two different shoe widths and taking the 0.5" data away from the 1" data you are left with accurate data for the remaining 0.5" width of shoe with no additional bristles being deflected.

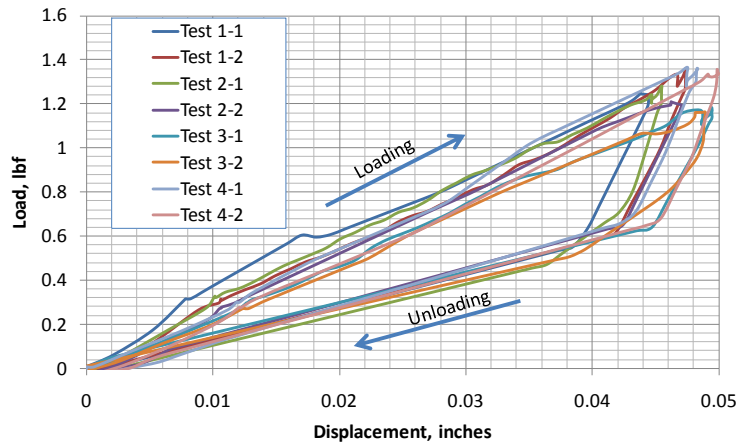


Figure 5 Raw Data from Stiffness Checking Machine

Figure 5 shows the typical raw data gathered when using a 1" wide shoe, you will note the large hysteresis loop, this is very typical of a brush seal.

Lets now compare some measured stiffness values with those calculated using the previously shown equations, the details shown in Table 1 compare the results for two test seals, Seal A and Seal B. These seals were designed to have very similar free bristle lengths with Seal A having a bristle angle towards the top of the 40-50° typical range and Seal B at the lower end.

Seal No.	Db	d	Bristle Density (Wires per inch Circumference)	BRISTLE ANGLE			Df	L	Stiffness (lbs/.001")	B.T.P. (psi/.001")			Stiffness Test BTP (Loading)		
				MEAN	MAX	MIN				Mean	Max	Min	Mean	Max	Min
Seal A	5.0989	0.0056	1398	47.25	49	46	6.48	0.914	1.16E-05	0.32	0.36	0.27	0.45	0.52	0.28
Seal B	5.0978	0.0056	1442	41	42	40	6.544	0.889	1.58E-05	0.485	0.53	0.45	0.71	0.81	0.65

Table 1. Seal A and Seal B Geometry and BTP.

From the details above in Table 1 it is evident that the measured bristle tip pressure is greater than the calculated value, this is to be expected because the simple beam theory ignores friction. In the real seal there is friction between adjacent bristles, between the last row of bristles and the back plate and between the bristle tips and the test

shoe. In our test Seal A measures some 40% higher and Seal B 46%, these values are typical of what is achieved using this method and not dissimilar to those achieved by others such as Demiroglu et. al.³

Having looked at some BTP values with no pressure drop across the seal we now need to see how we can evaluate the BTP with a pressure drop applied but before we do that we need to look at the effect pressure drop across the seal has on the sealing performance and blow-down of our two test seals.

IV. Static Leakage Testing

Cross has over the years performed many static leakage tests on brush seals as a quick and easy way of characterizing their sealing performance. All our test facilities have been described in depth in previous papers^{2,4}. The test rig shown in Figure 6 is one of many that Cross use, the test disk is not shown in this photo but we have many interchangeable disks that we can use, these typically increment in 0.010" or less.



Figure 6. Static Test Fixture

To characterize a seal for blow-down we typically test the seal at two interference cases and at least three clearance cases. Test data from Seal A and Seal B are shown in Figures 7 and 8 respectively, these plots are of effective clearance against pressure drop. The effective clearance function has been used for many years now and is fully detailed in a previous paper⁵. An effective clearance of 0.002" would be the same leakage as through a single labyrinth tooth with a radial gap of 0.002", at the same diameter as the test seal and with a discharge coefficient of 1.0.

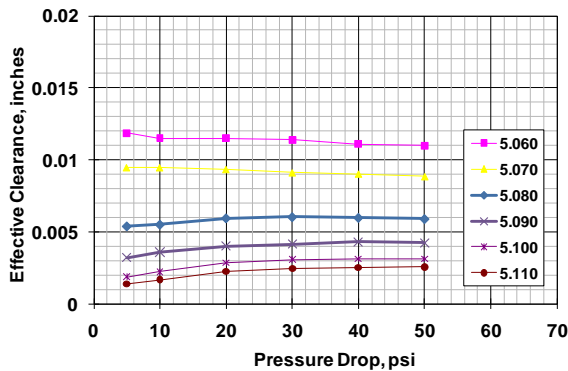


Figure 7. Static Sealing Performance of Seal A

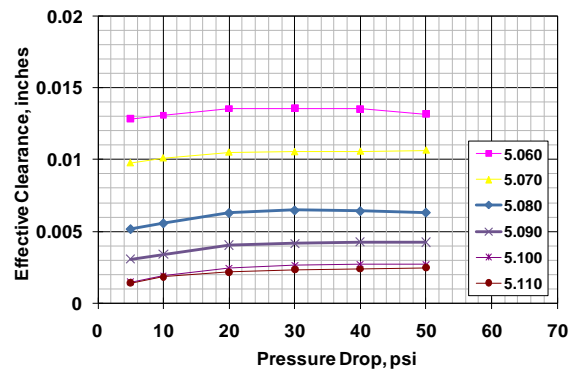


Figure 8. Static Sealing Performance of Seal B

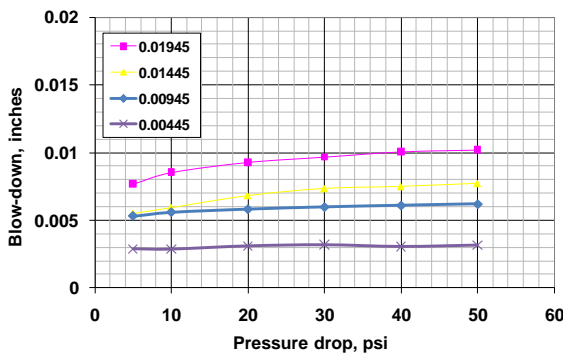


Figure 9. Static Blow-down of Seal A

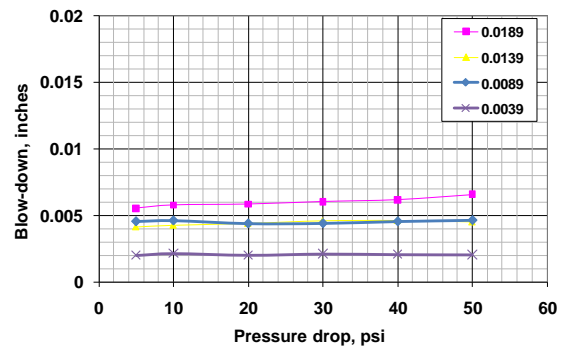


Figure 10. Static Blow-down of Seal B

The data above shows the sealing performance of both seals when tested against the 6 disks as shown in the legend, it is apparent that although the seals have similar performance Seal A has higher blow-down as indicated by the lower leakage at the smaller (higher clearance) disk sizes. Seal A has a higher bristle angle than Seal B and thus a slightly longer free length and lower BTP. The calculated blow-down for each seal is shown in Figures 9 & 10.

The blow-down calculation method has previously been discussed in depth in an earlier paper⁶ but it is clear that the two seals do have significantly different characteristics. The legend on each graph shows the cold build radial clearance of each line. Blow-down is clearly related to seal geometry, clearance and pressure drop, but lets now take a look to see if it has any affect on the contact pressure of the seal when pressure drops are applied.

V. Concentric Torque Testing

One of the many documented methods of obtaining the operating contact pressures of a brush seal is torque testing. Cross favors this method over others as described by Long⁷ as it gives a direct measurement of the torque that the seal has applied to the disk. The method described by Long would work well if the seal in question was of fixed geometry but by its very nature the brush seal is highly flexible and the bristle pack deflects axially significantly with pressure making any force balance calculations hard to reconcile. The tests performed at Cross are on a modification of the original ambient test rig as shown below in Figure 11.



Figure 11. Torque Test Rig

The modified rig shown above was fully described in an earlier paper². The modifications have involved removing the standard drive belt and replacing it with a 120W, 6rpm drive motor that drives the spindle through a torque transducer (the black box in the photo on the right). The torque transducer has a range of +/-50lbs.in and an accuracy of +/-0.19lbs.in. The first test we ran was with a 0.010” clearance 4 tooth labyrinth seal, this was performed to obtain the correction curve for the inherent friction in the spindle. As we test 1 seal at a time there is a significant axial load on the spindle so the compensation curve developed is proportional to the area of the rotor used and the pressure drop of the test.

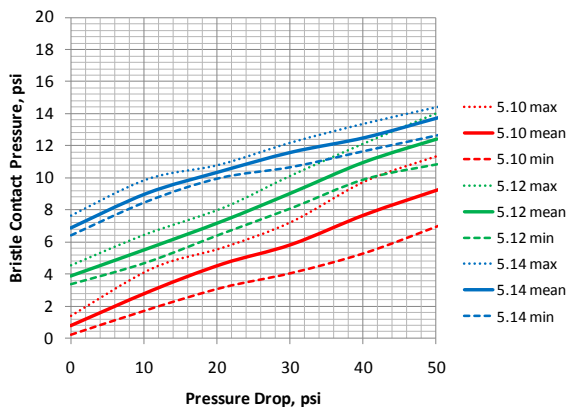


Figure 12. Contact Pressure Chart for Seal A

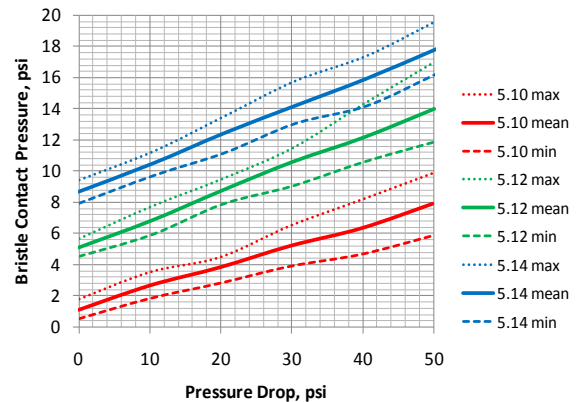


Figure 13. Contact Pressure Chart for Seal B

Seals A and B were then tested with three different disk sizes at pressures from 0 to 50psi, the data was reduced using a coefficient of friction of 0.2 between the bristle tips and the disk and then converted to bristle to rotor contact pressure using the bristle tip area and total number of bristles in each seal. As there was some variation in the torque values obtained all the data now shown will have a max, mean and min value to clearly illustrate the range of data.

The data shown in the graphs in Figures 12 and 13 clearly illustrate that the seals do indeed have different contact force characteristics with and without pressure. Figure 13 shows up some interesting characteristics, the three solid lines for the different disk sizes are all basically parallel, this indicates that the stiffness of the seal is unaffected by the pressure drop across it, lets try plotting this up a different way to illustrate this better. If we

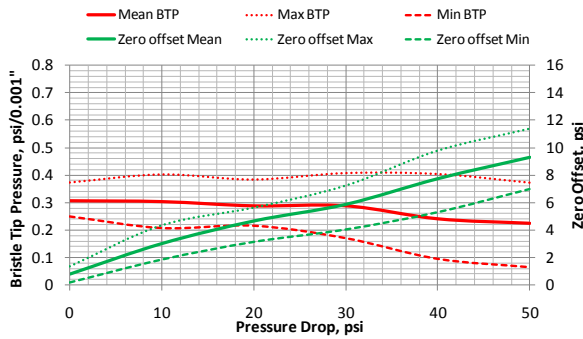


Figure 14. BTP and Zero offset for Seal A

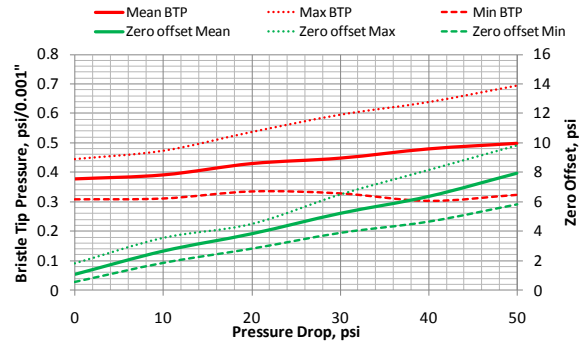


Figure 15 BTP and Zero offset for Seal B

subtract the 5.100” data away from the 5.140” data and divide the result by 20 we can obtain the instantaneous BTP of the seals, to then calculate the actual contact pressure you also need the zero interference contact pressure. This is essentially the contact pressure with the 5.100” disk and we have termed it the zero offset in the graphs shown in Figures 14 and 15.

The data shown in Figure 14 and 15 does tend to indicate that the instantaneous BTP is fairly constant in the pressures that we have looked at here. The zero offset as we have termed it is also pretty linear with pressure drop. This allows us to develop a simple equation for the contact pressure P_s that a seal will exert on a disk at an interference level of δ and a pressure drop of ΔP as shown in Equation (5).

$$P_s = (BTP\delta) + (C_p\Delta P) \quad (5)$$

C_p is a tip force correction factor for the blow-down of the seal. We can extract the BTP and C_p for each seal from the graphs above, for Seal A the BTP is 0.32 and the C_p is 0.19 and for Seal B. the BTP is 0.38 and the C_p is 0.16. Bearing in mind the large error bands visible on the graphs this form of equation is considered accurate enough for most estimations of Bristle to Rotor Contact Force, they could however be modified so BTP is a function of ΔP and C_p a function of δ to gain more accuracy.

One concern we have with this data, knowing that the performance of the brush seals typically shows large hysteresis loops due to friction, have we gathered this data in a way compatible with how a brush seals sees interference in service? We feel that how the test is run by applying an interference and then adjusting the pressure drop is not typical of seals in service where pressure drops and interferences are transiently changing as the engine goes through its operating cycle, especially on aerospace units. We thus decided to perform some torque testing introducing radial interferences on the rig in a similar manner to Jahn et.al⁸.

VI. Torque Testing with Radial Offsets

The Cross test facility has the capability of moving the vertical centerline of the seal housing horizontally relative to the vertical centre line of the disk during testing. The off-setting as we term it can be varied in magnitude and applied and removed remotely via pneumatic actuators. We set the rig up to perform 0.010” and 0.020” radial offsets, it was built with the 5.100” disk and we decided that we would run two different tests on each seal to see if there was a significant difference. Method one was simply to introduce the offset at 0psi and then increase the pressure drop in stages up to 50psi. Method two was more complex, the pressure would be set to the required level

and then the offset introduced and held for 30 seconds, the offset was then removed and the pressure reset to zero prior to moving onto the next pressure drop and offset. The resetting of the pressure to zero between each offset was to try to remove any hysteresis from the seals. The two methods are shown graphically in Figure 16.

The tests were run on both Seal A and Seal B, we then reduced the data back to bristle contact pressure, again using a coefficient of friction of 0.2. However we then need to estimate how many bristles are in contact with the rotor. We decided that the most realistic approach was as follows, at a radial offset of δ there are an equivalent of $D_b \times \text{Bristle Density}$ of bristles all with an interference of δ . With the concentric testing performed we had used

$D_b \times \text{Bristle Density} \times \pi$ again all with an interference of δ . The results obtained are shown in Figures 17 and 18 for Seal A and B respectively, this data also contains the concentric data for comparison.

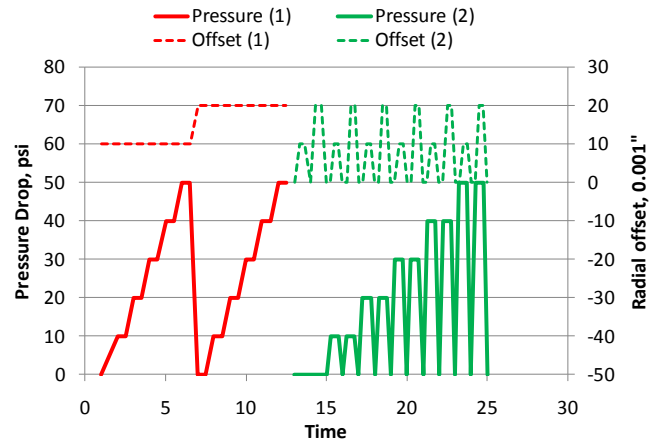


Figure 16 Offset Test Methods

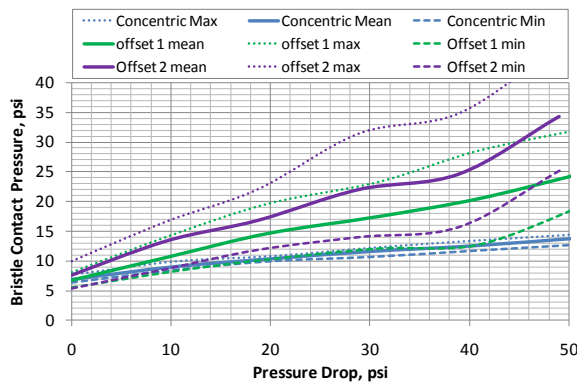


Figure 17. Contact Pressures with Offsets Seal A

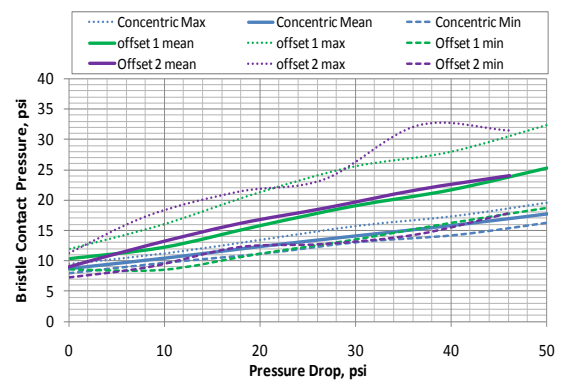


Figure 18. Contact Pressures with Offsets Seal B

From the data shown in Figures 17 and 18 we can learn a number of things, if we look first at seal A we can see that the offset method 2 gave the highest contact pressure values. At 0psi all 3 methods give very similar values, this is reassuring as it confirms our approach to estimating the number of bristles in contact with the disk is correct, but as pressure is applied so the lines diverge. The offset method 1 line, the green one, should be the same as the concentric data (blue), the difference we are seeing can be attributed to the blow-down of the seal with more bristles coming into contact with the disk as pressure is applied. This now clearly indicates that extracting meaningful contact pressure data from offset tests is not an easy task with too many unknowns data obtained can be used for comparisons but little else. The purple line on figure 17 is offset method 2 and the fact that this has a steeper gradient than that for method 1 indicates that this seal may be sensitive to the order in which the offset and pressure are applied with the seal appearing stiffer if the pressure is applied first. Seal B as seen in figure 18 does not show the same characteristic with there being very little difference between Method 1 and Method 2 data, both are however offset from the concentric data due to the blow-down effects, but to a lesser extent on this low blow-down seal.

Having looked at gathering data with offsets we now conclude that due to the unknowns on how many bristles are in contact with the disk at any time it is not the easiest or most accurate method to use. It has however indicated that some seals may be sensitive to the order in which the offset and pressure are applied and as such can be used for comparison purposes only.

VII. FEA Bristle Pack Model

Over the years we have presented details of our Finite Element Method model of the bristle pack using the commercial code Adina ^{4,6,9}, others have over the years also modeled brush seals in this way such as Aksit ^{10,11} and Guardino ¹². Our model has continued to evolve, the latest version shown here has been put together by Jas Walia who is an Adina specialist in England. Jas has taken our original 3D models and modified them in the following ways, firstly the model is now totally parametric in that you just input Bristle free length, wire diameter, bristle angle, friction and pressure values as well as the axial and radial flare parameters. The model also can now contain 9, 12 or 15 layers of bristles, the bristle tips are also now angled to align with the simulated disk. Figure 19 shows some pictures of the mesh for a 12 layer model in both the deflected and un-deflected conditions.

One of the concerns we currently have with this model is the boundary conditions that we have applied, because we only have 4 or 5 bristles in the circumferential direction it is necessary to apply boundary conditions to simulate a much larger seal. We have done this by applying rigid links between the first and last bristles in each layer, as we had done in our earlier versions of the model. At the time of writing this paper the issue of boundary conditions has not been fully rectified and because of this we are only able to show data with no pressure drop across the seal.

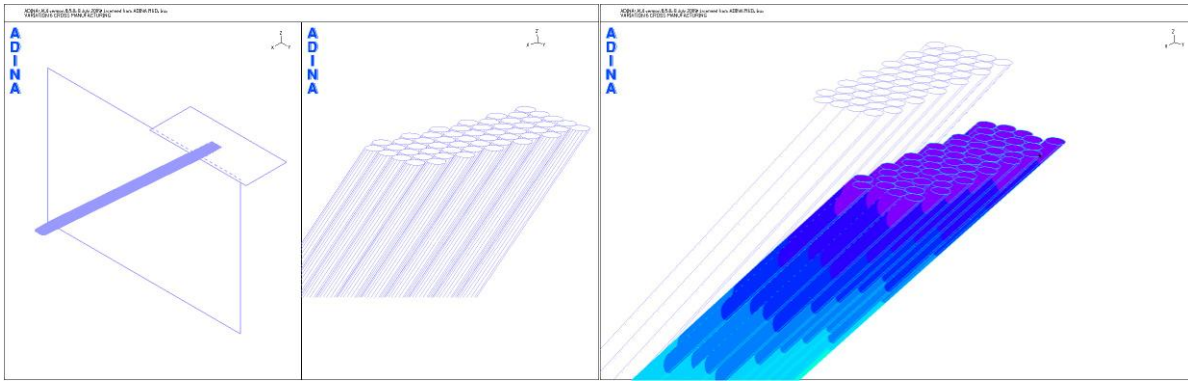


Figure 19. FEA Model of Bristle Pack

The model was run to simulate the geometry of Seal A and Seal B. The was run with no pressure drop and with the simulated disk moving radially into the bristle pack by 0.025” and then away again. The results from these two runs is shown graphically in Figures 20 and 21.

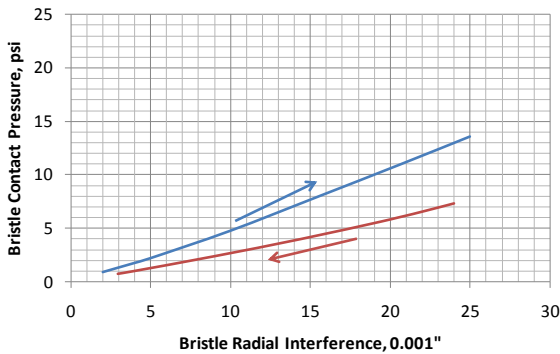


Figure 20. FEA Contact Pressures Seal A

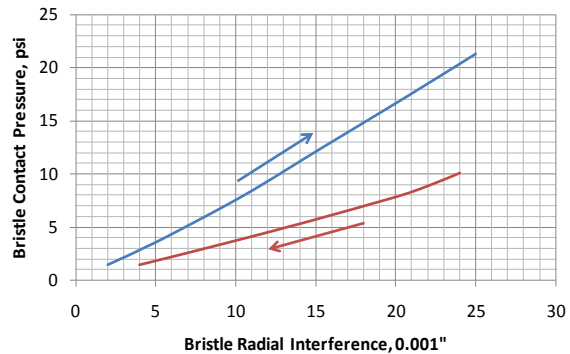


Figure 21. FEA Contact Pressures Seal B

The FEA data in both graphs does appear to show a similar hysteresis loop to the stiffness checker data shown in Figure 5. Backing out the BTP from this data yields a value of 0.55psi/0.001” for seal A and 0.87psi/0.001” for seal B. These values are about 75% higher than the simple beam theory calculation and 22% higher than the data obtained by the stiffness checking machine. There is clearly room for improvement of the model with further refinement of the coefficient of friction values being used and further investigation into the most suitable boundary conditions to use. Further development will also investigate ways of improving solution speed.

VIII. Discussion

We have seen here how the Bristle Tip Stiffness can be calculated using basic beam theory, how seals blow-down when there is a pressure drop across them. We have then looked at stiffness checking without any pressure drop across the seal and then torque testing using three different methods and finally a FEA model of the pack. The need for this work is to establish the heat generated by a brush seal when the bristles rub the disk, the torque testing is seen as a good method for doing this as it can be carried out with and without pressure drops and gives a direct reading on the torque that the bristles exert on the disk. We looked at two additional methods of torque testing by introducing radial offsets, the first by applying the offset and then the pressure with the second the other way round. This last torque test method gave the highest values but due to the uncertainty of knowing how many bristles are in contact with the disk due to the blow down of the seal it has been decided that this method can only be used for comparison purposes and not for extracting meaningful of bristle contact pressures.

The data obtained from all the tests and the FEA model is summarized in Table 2 and also graphically for Seal A and Seal B in Figures 22 and 23. All this data is however at 0 psi and not many seals operate in that condition so what interests most people is how to calculate the operating bristle tip pressure of the seal under pressure drop conditions. It has been previously suggested that the pressure drop stiffens the bristle pack, we have not found that to be the case, instead we have found that in general the Bristle Tip Pressure does not change with pressure drop but there is a correction factor that needs adding to the BTP and this factor is a function of pressure drop across the seal and is related to the blow-down of the seal.

Seal No.	B.T.P. (psi/.001")			Stiffness Test BTP (Loading)			BTP, Opsi Torque Concentric			BTP, Opsi Torque Offset			BTP, 0 psi FEA Data
	Mean	Max	Min	Mean	Max	Min	Max	Mean	Min	Max	Mean	Min	
Seal A	0.32	0.36	0.27	0.45	0.52	0.28	0.37	0.31	0.25	0.40	0.33	0.28	0.55
Seal B	0.49	0.53	0.45	0.71	0.81	0.65	0.44	0.38	0.31	0.60	0.52	0.43	0.87

Table 2 Summary of Bristle Tip Pressure Data from Different Methods

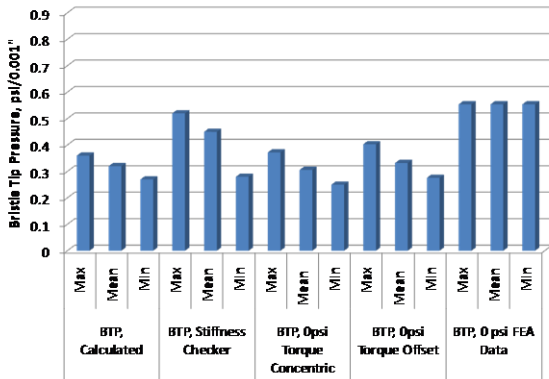


Figure 22 BTP Comparison Seal A

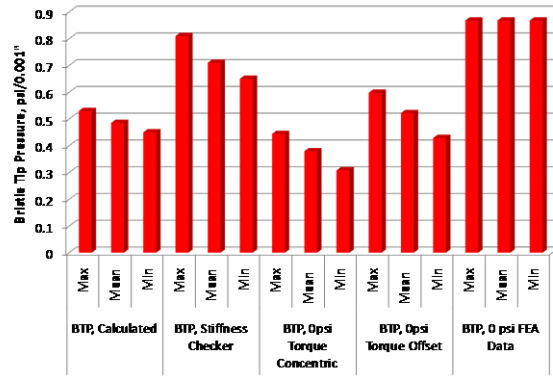


Figure 22 BTP Comparison Seal B

There is clearly quite a bit of difference in the results obtained by the different methods, the base line value of the simple beam bending equation tends to be lower than that obtained by stiffness checking, this is mainly due to friction between adjacent bristles and between the test shoe and the bristle tips. The concentric torque test data is generally pretty close to the base line data, the rotation of the disk in this test will tend to add a straightening force down the bristle, this is the closest to how a seal functions in service so it is refreshing that this data is close to the base line values. The offset torque data tends to give slightly higher values, this is down to the uncertainty of knowing accurately how many of the bristles are in intimate contact with the disk. Lastly the FEA data tends to give the highest data, this is due to a combination of excessive constraint on the model imposed by the boundary conditions and to high coefficient of friction values being used, these will be corrected as the model evolves further.

We have seen that Seal A and Seal B have significantly different blow-down characteristics and also different contact pressure characteristics. The seals are nearly identical in all aspects except for their bristle angles. Seal A has a mean bristle angle of 47.25 degrees with a free bristle length of 0.914”, Seal B is a stiffer seal having a mean

bristle angle of 41 degrees and a free bristle length of 0.889”, the seals have different pinch point bores but very similar bristle free lengths (2.8% difference). Generally the greater the bristle angle the more the seal will blow down and the more the contact pressure will increase with pressure.

IX. Conclusion

The sealing and contact pressure characteristics of brush seals are clearly influenced by their bristle angle, in general the greater the bristle angle the greater the blow-down and the more the contact pressure of the seal increases with pressure drop. Beam bending calculations produce results that are comparable at 0psi to those obtained from torque testing, both concentric and radially offset. The concentric torque test method appears to yield the most reliable data when there is a pressure drop applied across the seal, the offset test method is prone to large errors when pressure drops are present due to the uncertainty of knowing how many bristles are contacting the disk.

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