Aspects of Brush Seal Design

Tracey Kirk[1], Aaron Bowsher[2] and Peter Crudgington[3]
Cross Manufacturing Co. (1938) Ltd., Devizes, Wiltshire, SN10 2EU

Raymond Chupp[4]
REC Consulting,LLC., Simpsonville, South Carolina, 29681

The use of Brush Seals continues to expand in both the Power Generation and the Aerospace markets; being applied to more demanding locations and operating environments. Over the last 35 years, much has been learned about brush seal functionality and detailed design procedures and codes have been developed and refined. This information is used in the feasibility of brush seal applications and in establishing an optimized design for the given application. This paper is an introduction to elements of the design process, engineering principles, and the current state of the art used at Cross Manufacturing Co (1938) Ltd.

Nomenclature

BTP	=	bristle tip pressure (psi/.001”)
H	=	hardness (psi)
k	=	Archard wear coefficient
S	=	total sliding distance (inches)
V	=	sliding wear volume (inches^3)
W	=	normal load (lbs.f)

I. Introduction

Cross have been designing, developing and manufacturing brush seals since the 1970’s for both the Aerospace and Power Generation industries. Many successful applications have been achieved over the years yielding an extensive understanding of the behavior of brush seals in service.

An ever increasing demand for turbine efficiency has led to brush seals being applied to more challenging locations and operating environments. To properly design brush seals for these increasingly challenging applications, a cohesive, well defined design process has been implemented at Cross. This process is grounded in rigorous testing and analytical modelling, both of which have been refined and validated over the years. The process follows various industry standards to insure a high quality design approach.

This paper introduces the current state of the art of brush seal design at Cross. Included is an overview of the quality design seal process, bristle pack design, effects on leakage and wear characteristics.

[1] Test and Development Engineer, South Site, Hopton Road, England.
[2] Engineering Manager, South Site, Hopton Road, England, AIAA Senior Member.
[3] Director, South Site, Hopton Road, England, AIAA Senior Member.
[4] Consultant to Cross Manufacturing, 10 Asterbrook Ct., Simpsonville.

II. Brush Seal Design Process

Brush seal technology provides an advanced sealing solution for heavily regulated industries. A fundamental quality approach to the design of a brush seal is followed as outlined in SAE Aerospace Standard AS9100C [1]. Cross is accredited to this standard and works to this process for all brush seal designs, not just those destined for an aerospace application. An overview of the design and development process flow outlined in AS9100C is summarized in Fig. 1.

Figure 2 illustrates some of the key inputs that are required to design an effective brush seal for conditions specific to an application or location. All these inputs have a fundamental effect on the design and performance and must be considered and evaluated in the design phase. A selection of the design inputs will be discussed in the following sections detailing the effects on bristle pack design, leakage and wear.

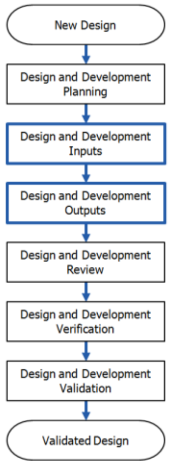

Figure 1. Product Realization for Design and Development

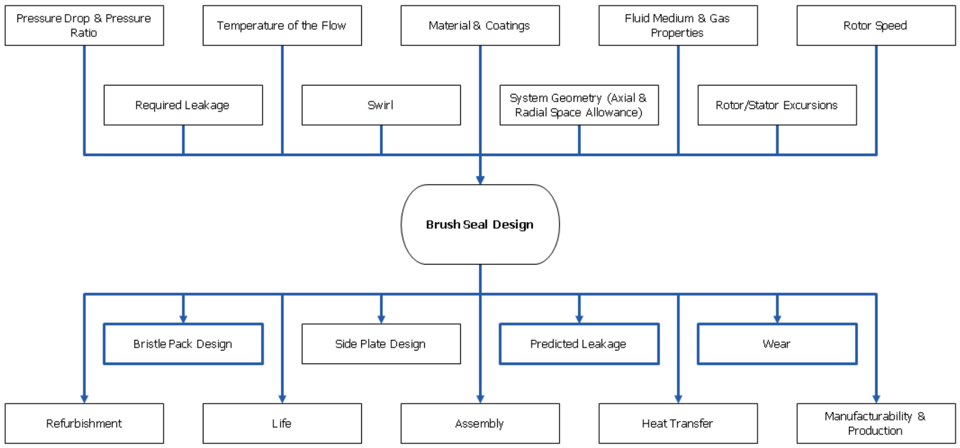

Figure 2. Brush Seal Typical Design Inputs and Outputs

III. Bristle Pack Design

The basic design of a brush seal begins with the bristle pack. The design of a bristle pack does not have an isolated solution but is a compromise between various characteristics. The basic design can be achieved by following the iterative process outlined in Fig. 3.

The fence height is the radial distance from the back plate bore to the bristle bore. It should be no less than the relative maximum radial closure of the system plus an appropriate factor of safety to ensure no contact between the shaft and back plate bore. Radial excursion uncertainty can be accommodated by increasing the factor of safety or by careful design of the back plate to ensure minimal impact to system functionality during a potential hard rub condition. The pressure drop and leakage capabilities are a function of the fence height and can be assessed by calculating the axial deflection of the bristles around the back plate due to the pressure drop [2]. This is illustrated by the effect of pressure drop on effective clearance presented in 1998 [3] and reproduced in Fig. 4. As a brush seal operates in regions 1 and 2, the leakage increases progressively with pressure drop and the fence height has little effect on leakage. As the pressure increases passed the pressure limit of the seal into region 3, the bristles are stressed passed their elastic region and leakage increases dramatically. The fence height directly influences the pressure limit of a brush seal where a larger fence height results in a lower pressure limit due to the stress on the bristles. This stress can be calculated for a given design and the fence height can be evaluated (Fig. 5) to ensure the seals maximum pressure drop remains within regions 1 or 2 (Fig.4).

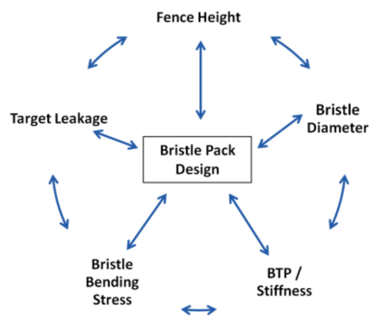

Figure 3. Bristle Pack Design Loop

A large pressure drop requirement with large radial excursions may result in the need for a larger diameter bristle. Typical bristle diameters range from 0.002” to 0.0066” with some exceptions for extreme cases. The selection of the bristle diameter is a tradeoff between pressure capability and leakage. A small diameter bristle will pass less leakage while a larger diameter bristle will have higher load capabilities and wear properties [4]. The effects of bristle diameter on the bending stress is illustrated in Fig 6. The stress is evaluated using the same principles as discussed for the fence height. The Aerospace and Power Generation industries currently favor Haynes®25 as the standard bristle material. It demonstrates good wear and oxidation characteristics but creep resistance restricts it to operating temperatures below 620°C. Lower temperature alloys are available and higher temperature alloys are currently being developed and verified for ultra super critical steam application, among others.

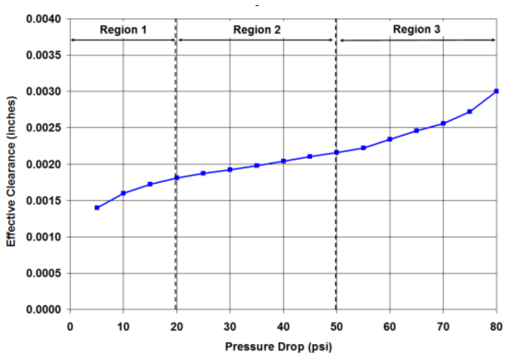

Figure 4. Typical Brush Seal Characteristic Curve [3]

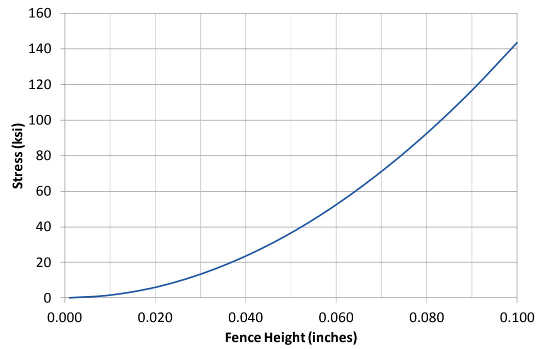

Figure 5. The effect of Fence Height on Bristle Stress using in-house Predictive Design Codes

BTP is key for establishing wear and ensuring the life of the brush seal. It is the bristle tip pressure exerted on the rotor due to the stiffness of the bristle at zero pressure drop and is a function of bristle geometry allowing direct comparison between designs with various geometry. BTP is calculated using cantilever bending theory as published by Flower in 1990 [5]. This formula dictates that wire diameter, free length and bristle angle are geometrical functions of BTP. Reducing the bristle diameter, increasing the bristle free length and increasing the bristle angle will all result in a reduction in contact pressure by varying degrees of magnitude. A reduction in BTP will reduce the contact temperature and rate of wear of both rotor and bristle, however this will also reduce the stability of the brush seal. Each sealing application has an optimum BTP based on the applications operating conditions.

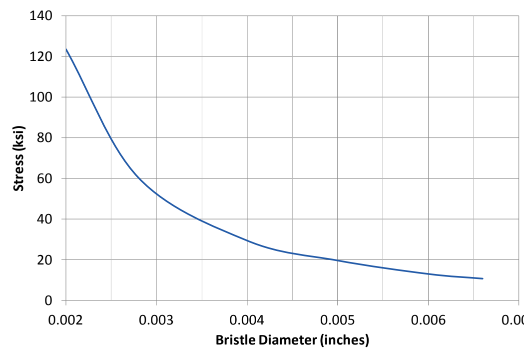

Figure 6. The effect of Bristle Diameter on Bristle Stress using Predictive Design Codes

The design output of a brush seal inevitably has to meet the input requirements from the customer. A well designed brush seal must meet fit, form and function; therefore the performance requirement must always be assessed during the design iterations. Although this section has independently discussed the effects of a selection of characteristics, the actual design of a bristle pack takes all these characteristics into account simultaneously along with many other criteria including, but not limited to: bristle density, bristle frequency relative to rotor frequency, bristle stability, heat generation and rotor dynamics.

IV. Predicted Leakage

Leakage through a single stage brush seal is calculated using the formula for effective clearance as published in 1998 [6]. The effective clearance of brush seals has been established through testing and can be applied to estimate the leakage of any given fluid medium.

In some cases a brush seal is retrofitted into an existing labyrinth system or designed as part of a sealing system rather than a standalone seal. In this instance a 1D CFD code is used to calculate the flow through multi stages of labyrinth and brush seals based on basic principles of conservation of mass and verified against labyrinth seal leakage as presented by Vermes in 1961 [7]. Figures 7 to 14 illustrate the effects of retrofitting a brush seal to an existing labyrinth gland using the predictive design code. A generic 15 tooth system was created with a tooth clearance of .001" per inch of rotor diameter (Fig 7). The pressure drop across each tooth as a percentage of the total pressure drop across the system can be seen to be non linear (Fig 8), with the pressure drop per stage increasing with each stage.

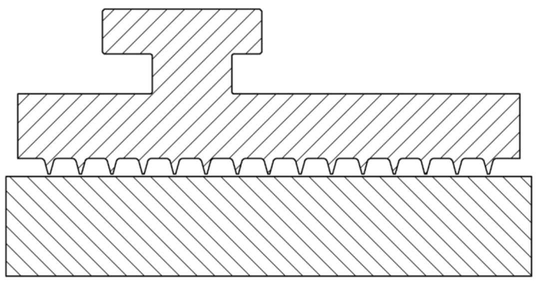

Figure 7. Generic Labyrinth Sealing Gland with 15 Teeth

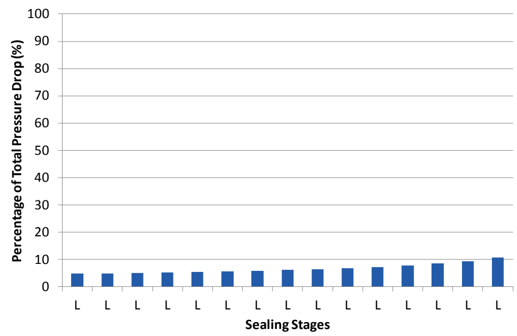

Figure 8. Pressure Distribution across the Sealing Gland using In-house 1D Predictive Code

Figure 9 illustrates a brush seal with a mid-sized wire replacing the 1st labyrinth tooth stage. All design characteristics remain identical to those used in Fig 7 with the only changes made being the effective clearance of the brush seal as established by test data. The predictive 1D code also has the capability to take into account various discharge coefficients and carry over function for the labyrinth teeth, if applicable. Figure 10 clearly illustrates that the brush seal encounters the majority of the pressure drop, in this case over 80% of the total system pressure drop, with the downstream labyrinth teeth seeing just over 1% each. It is essential that this pressure drop is taken into account when designing the seal to ensure pressure capability.

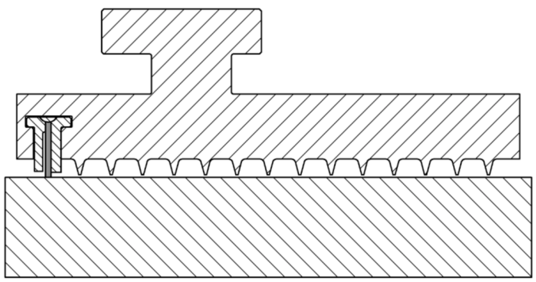

Figure 9. Generic Brush Seal Design Installed at the First Stage of a Gland followed by 14 Labyrinth Teeth

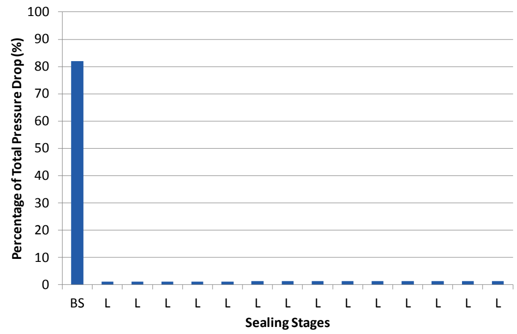

Figure 10. Pressure Distribution across the Sealing Gland with a Brush Seal Installed at the First Stage using In-house 1D predictive Code

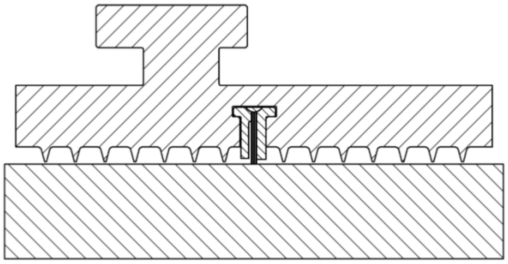

Figure 11. Generic Brush Seal Design Installed at the Mid Stage of a Gland preceded by 7 Labyrinth Teeth and followed by 7 Labyrinth Teeth

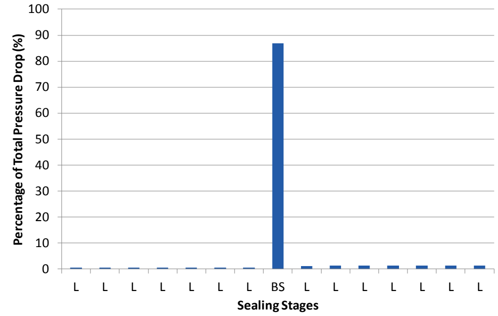

Figure 12. Pressure Distribution across the Sealing Gland with a Brush Seal Installed at the Mid Stage using In-house 1D predictive Code

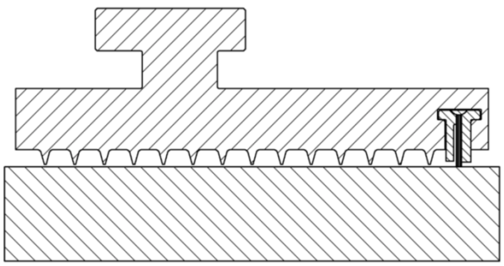

Figure 13. Generic Brush Seal Design Installed at the Final Stage of a Gland preceded by 14 Labyrinth Teeth

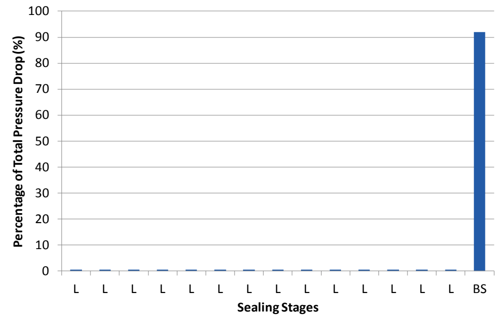

Figure 14. Pressure Distribution across the Sealing Gland with a Brush Seal Installed at the Final Stage using In-house 1D predictive Code

The location of the brush seal along the gland also has an effect on the pressure sharing between the brush seal and remaining labyrinth teeth. This in turn effects the leakage to a small extent. As the brush seal location moves from front to back (Figs. 9 to14), the pressure drop across the seal increases and the effect of the labyrinth teeth reduces. This in turn reduces the leakage as the relative system effective clearance reduces (Fig 15). The location of the brush seal within the sealing system is dependent on the application conditions. A brush seal located at the final stage will result in lower leakage than fitted in the first stage, however will see a greater stress on the bristles.

Brush seals can also be used in series to achieve leakage requirements and large pressure drops. Again, the pressure sharing between the brush seals is non linear but it is possible to use different sized wires in different locations to help balance the pressure more evenly. The addition of brush seals in sequence has a diminishing return on leakage (Fig 16), therefore inevitably resulting in a compromise between efficiency gain and cost. It is advised that brush seals placed in sequence have a method of flow disruption incorporated between each bristle pack due to the jetting effect of the fluid flow through the upstream bristle pack. This phenomenon may damage the bristle pack of the downstream seal. One such solution for this phenomena is to have a radial step in the rotor with the sequential brush seal bores at different diameters [8].

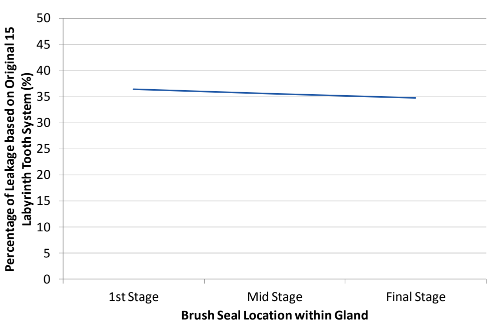

Figure 15. The Effects on Leakage of a Brush Seal Location within a Gland

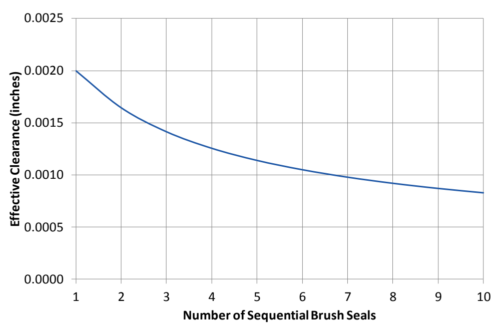

Figure 16. The Diminishing Effects on Effective Clearance of Sequential Brush Seals

V. Wear

The Archard wear law looks at the relationship between wear, load, sliding distance and material and is commonly used in the industry as a value for wear [9]. Archards wear coefficient, k, can be found by testing bristle materials against rotor material. Once the coefficient is established for the material pair, it is possible to calculate the cyclic wear characteristics for a brush seal and subsequent effect on effective clearance (Eq. 1).

$$\mathrm{V} = \frac{\mathrm{kWS}}{3H} \tag{1}$$

Haynes®25, bristles have proven wear characteristics at temperatures below 620°C. Above this temperature the creep resistance becomes undesirable. In-house testing has been carried out to investigate the potential for alternative materials to be used at temperatures above 620°C, but that are still comparable to Haynes®25 at lower temperatures. A custom test housing was designed and built around the existing high speed spindle (Fig. 17) with an interchangeable wear plate system (Fig. 18). Wear pins known as 'Pin-on-discs' (Fig.19) are manufactured from various bristle materials. Ongoing test programs have established Archard wear coefficients for bristle and wear plate material pairs currently at speeds up to 244 ft/s and temperatures up to 720°C.

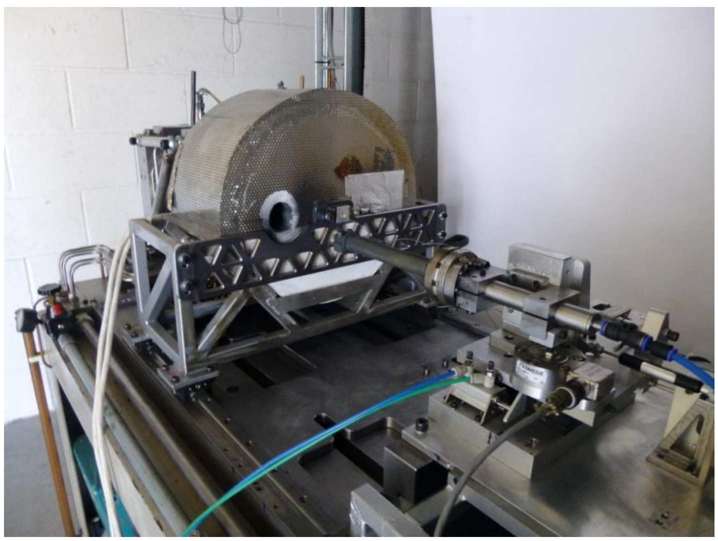

Figure 17. Overview of High Speed Spindle and Customer Wear Test Housing

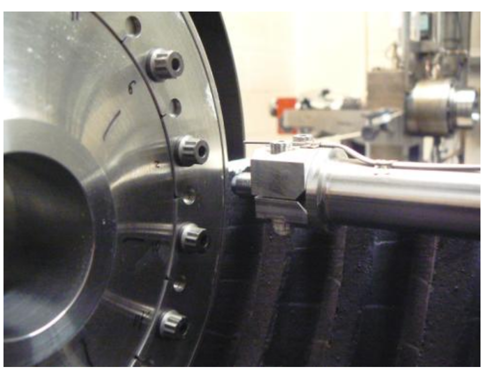

Figure 18. Close up of Pin on Disc, Rotor and Interchangeable Wear Plate

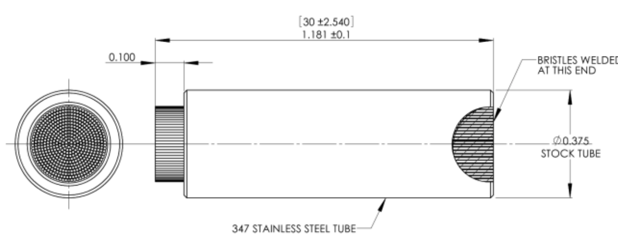

Figure 19. Pin-on-Disc Design

An in-house cyclic wear code was first developed in the early 2000's to estimate the wear per cycle of a brush seal in conditions specific to a customer's application. The purpose was to take into account all aspects that effect bristle wear including run-out, eccentricity and blow-down, as well as time dependent functions such as relative closure, pressure drop and surface speed. Over the past 15 years, this code has been refined as test data and turbine service data have given a better understanding of wear and specific conditions. The code can predict the cumulative wear over a specified number of cycles (Fig. 20) or the wear per single cycle (Fig. 21).

The latest in-house wear testing provides a catalogue of wear factors for various material pairing and operating conditions, thereby increasing the accuracy of the existing in-house predictive tools. In particular the testing is aimed at the wear characteristics in the temperature region up to and exceeding 620°C and the investigation of a new alloy. Figure 22 shows test results for this development alloy (Material B) rubbing against a chrome carbide coated wear plate at 623°C with 3N of axial force as well as a second alloy (Material A). Material A did not perform well under these test conditions and wore out in less than 2250 seconds resulting in unsuitability as a bristle material at these operating conditions. Material B shows superior wear characteristics at this temperature and the test was stopped long before the material had worn out. Figure 23 shows the same material B rubbing against a nickel based alloy wear plate at 720°C with 10N of axial force. Material B once again showed outstanding wear capability. A third alloy, Material C, was also investigated, but like Material A, proved unsuitable at these operating conditions.

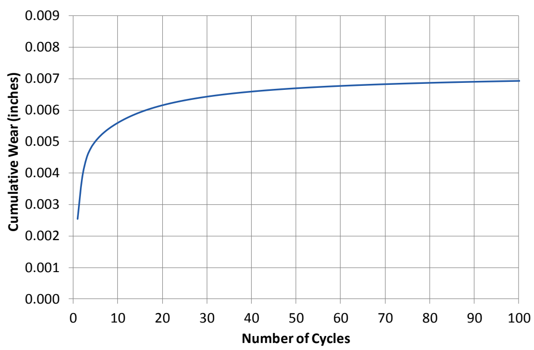

Figure 20. An Example of Cyclic Wear Predications using In-house Code

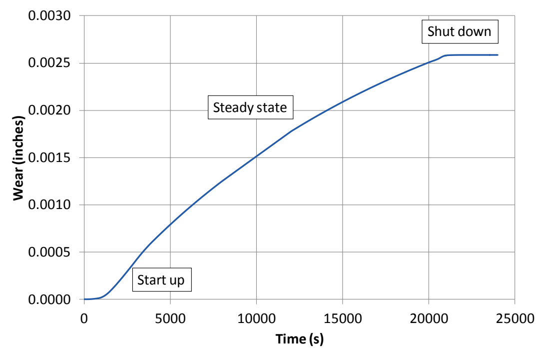

Figure 21. An Example of a Single Cycle Wear Curve Prediction

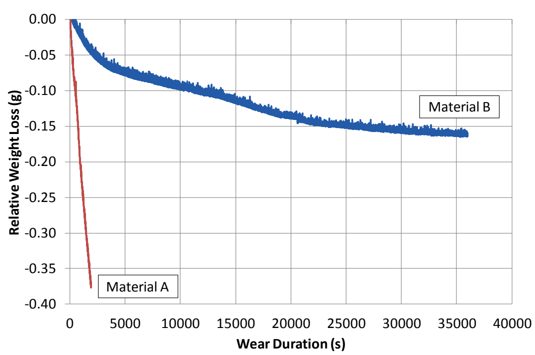

Figure 22. Pin on Disc Wear on a Chrome Carbide Coated Wear Plate under 3N of Axial Loading

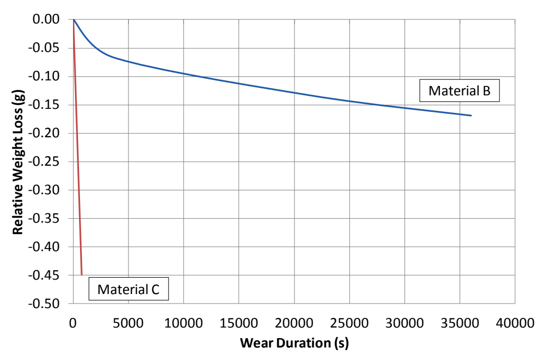

Figure 23. Pin on Disc Wear on a Nickel Based Alloy Wear Plate under 10N of Axial Loading

VI. Conclusion

Cross Manufacturing has been designing and developing brush seals for over 35 years. Over this time a large amount of test and service data have been cumulated and incorporated into design codes to ensure brush seals designed by Cross are to the highest standards possible. There is a large amount of factors that need to be considered when designing a successful and efficient brush seal for industrial applications. All these factors need to be continually reviewed through the design phase and assessed when validation results are either made available by the customer or examined internally by testing.

Efficiency and life are becoming more important to the turbine industries and brush seal designs need to support turbine design objectives. Understanding leakage performance is essential for improving secondary flow systems and understanding accurate heat transfer of turbine components. Wear testing has provided positive results for a suitable material for new demanding locations, securing the future of brush seals in the next generation of turbines.

Cross continually improves their advanced predictive codes and are willing to share their test results with customers. However, when specific requirements are needed by OEM customers, Cross can perform extensive, confidential testing beyond that discussed in this paper.

Acknowledgments

Cross Manufacturing has funded all work described in this paper and would like to thank the Directors for allowing this data to be published. Thanks also go to the dedicated team of engineers and manufacturing staff who work together to produce products with excellent quality. Matthew Jenkins at Cross is also thanked for the gathering of test data and his work in relation to wear and wear rig design.

References

[1] SAE Standard AS9100C, *Quality Management Systems – Requirements for Aviation, Space and Defense Organizations*, SAE International, 2009

[2] Modi. V., "Modeling Bristle Lift-Off in Idealized Brush Seal Configurations", ISROMAC-4, Honolulu, April 1992

[3] Crudgington, P., "Brush Seal Performance Evaluation," AIAA 98-3172, 1998

[4] Chupp, R., Hendricks, R., Lattime, S., Steinetz, B., "Sealing in Turbomachinery," NASA/TM-214341, 2006

[5] Flower, R., "Brush Seal Development System" AIAA 90-2143, July 1990

[6] Dinc, S., et al., "Brush Seals in Industrial Gas Turbines – Turbine Section Interstage Sealing" AIAA 98-3175, 1998

[7] Vermes, G., "A Fluid Mechanics Approach to the Labyrinth Seal Leakage Problem, Journal of Basic Engineering," ASME J. Eng. Power 83(2), 161-169 (Apr 01, 1961)

[8] Bagepalli, B., Chiu, R., Comer, R., Crum, G., Dinc, O., Furman, A., Marks, P., Markytan, R., Skinner, D., Turnquist, N., Wolfe, C., GE Electric Co., Schenectady, N.Y., U.S. Patent "Brush Seals and Combined Labyrinth and Brush Seals for Rotary Machines" Patent Number: 6,131,910, Oct. 17, 2000

[9] ASM International Handbook Committee, *ASM Handbook, Volume 18, Friction, Lubrication and Wear Technology.* United State of America : ASM International, 1992. ISBN 0-87170-380-7