Influence of geometrical parameters on the performance of carbon brush seals for aero-engines bearing chambers

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Abstract
This paper presents the experimental work currently carried on by the ULB on the characterization of carbon brush seals performance. After highlighting the importance of improving sealing efficiencies on aircraft gas turbine engines, the theoretical advantages of using carbon brush seals over labyrinth seals will be displayed.

This work also provides a description of the test bench developed by the ULB, in collaboration with French engine manufacturer SNECMA, which intends to reproduce the severe working conditions brush seals will encounter in aero-engines bearing chambers, as realistically as possible. The aim is to improve the brush seal behaviour knowledge by identifying the most influential geometric parameters acting on its wear, and by evaluating both air and oil leakage performance.

So far, a preliminary analysis of the brush seals performance in function of the geometrical configuration has been performed. The analysis is yet to be extended to the effect of the rotational speed, the oil presence, and the fluid temperatures. Ultimately, this work should allow predicting the performance and endurance of an optimum design.

Introduction
Lower fuel specific consumptions for turbofans can be reached by increasing the bypass ratio and the overall pressure ratio. In one hand, a high bypass ratio is strongly linked with a high turbine inlet temperature. Advanced materials must be used to increase the maximum temperature that can be reached by the blades without melting danger. The blades are also machined with small internal cooling tunnels that facilitate the temperature increase limitation. In the other hand, a higher overall pressure ratio allows higher thermal and global efficiencies for gas turbines. Sealing is of crucial importance when it comes to guarantee the highest pressures.

Better sealing of the rotating machines (especially between the rotating and static elements of compressors and turbines) means less secondary air leakage, therefore better overall performance. Elsewhere, in bearing chambers, proper sealing reduces oil losses that contaminate the turbine components, and is expelled into the atmosphere. It also reduces the specific oil consumption, and allows longer time intervals between oil replacements in the lubrication circuit.

In each case, sealing devices are functional thanks to air pressurization. Air and oil are confined thanks to the effect of
high pressure air, which goes in the opposite direction of the leaking working fluid. Figure 1 illustrates the sealing process in a typical bearing chamber design.

![Bearing chamber schematics](image)

**Figure 1: Bearing chamber schematics**

Bleed air is actually extracted from the compressor stage of the engine for sealing purpose. The higher the seal performance, the lesser air is need for a same pressure difference across the seal, thus higher pressures can be achieved for gas turbine cycles. Air and oil sealing efficiency have a direct relationship with improvement in the engine propulsive efficiency and specific fuel consumption.

**Brush seals for oil sealing**

Over the last few decades, labyrinth seals are used for bearing chambers sealing since they proved to be reliable and low cost. But they are designed with relatively large clearances in order to avoid any interference between the static and rotating elements, which would result in a premature destruction of the labyrinth seals. Interference would occur under the effect of eccentricity, or rotor dilatation due to thermal and centrifugal effects. However, large clearances leave more room for the oil to freely flow outside the bearing chamber.

Naturally, sealing solutions were explored using contact between seals and shaft. Abradable materials are added to the stationary element of the labyrinth seals to accommodate with the shaft/rotor interference without excessive wear, especially when the rotating machine configurations require close clearances [1].

Carbon face seals were then considered. They are built with a very small clearance, and are spring loaded to apply pressure on the shaft, actually creating constant contact. Carbon is chosen because of its high thermal conductivity and oxidation resistance, and its low friction coefficient. The seal rubs on either a smooth, polished rotating metal surface that is assembled on the shaft, either on coating made of chrome carbide, silicon carbide, or tungsten carbide [2].

The other contact seals considered for oil sealing applications are brush seals, which are the subject of this work. Brush seals are annular seals of high density bristles disposed in its internal diameter. In most of the designs, the bristles are canted in the shaft rotation direction, as shown in Figure 2. Canted bristles are able to accommodate with rotor eccentricity and dilatation of centrifugal and thermal nature, without significant performance losses. However, carbon brush seals considered in this study have a lay angle of 0°. The friction torque may be higher than with canted bristles, but such brush seals are more suited in case of reverse windmilling that happens in aero-engines. Also, carbon fibers are very flexible, which limits the
rotor eccentricity-related damages. Figure 2 shows a schematic front view of a typical brush seal.

![Figure 2: Schematic front view of a brush seal](image)

Bristles materials for brush seals are chosen for their high temperature capabilities and their tribological performance. Friction torque must be reduced to limit the power losses, while maximizing the endurance.

First, metallic brush seals are widely used in turbomachinery applications. Most of the alloys used were cobalt or nickel-based. The most well known one is Haynes 25, which is a chromium-cobalt-nickel-tungsten superalloy. The advantage is that they withstand temperatures reaching 980°C, and provide excellent resistance to oxidation [3]. But drawbacks of metallic bristles include high wear properties, especially on the shaft, which have to be refinished in case of excessive degradation. Also, they present the danger of generating small metal chips resulting of wear, which would very harmful for bearing. Finally, bristles progressively lose resilience, until the point where bristles will bend out of shape.

Progressively, the trending has been to switch to ceramic brush seals. Bristles made of ceramic fibers are less likely to be subjected to plastic deformation, especially at high temperatures. Another advantage is their lower friction coefficient with steel and/or coated shaft surfaces. Ceramic bristles also provide high resistance to abrasion, high temperature limit (1000°C) and the corrosive oil present in the environment [4]. In the recent years, aramid (or Kevlar ®) fibers were used as they perform way better than the metallic brush seals in terms of leakage performance and heat generation. It is mainly because the diameter and the stiffness of the fibers are smaller: metallic bristles usually do not have a lower diameter than 0.07 mm, whereas with aramid fibers, it is possible to use bristles with a diameter as low as 0.012 mm. And aramid particles resulting from wear are less a concern for the bearing chambers than the metallic or ceramic ones. Nevertheless, the high performance for heat generation is compensated by a lower temperature capability: 250°C.

Finally, the most recent material tested for brush seals is carbon, which this research is focused on. Recent tests on carbon brush seals showed even better performance than the aramid brush seals especially in term of heat generation (66% reduction in overall [5]), due to a lower friction coefficient and higher thermal conductivity. Also, carbon bristles withstand higher temperatures: 370°C instead of 250°C. But due to the fact that carbon bristles are easier to deflect (carbon fibers diameters can be below 0.01 mm), carbon brush seals are also well suited to low pressure applications.

The bristle pack is wedged between a front plate, and a backing plate. The front plate enables protection against upstream turbulence flow,
and the back plate provides a support for the bristles when axially bent through the effect of air pressure.

Two particular zones can be distinguished when the seal is mounted, as drawn on Figure 3:

- The fence height, which is the distance between the shaft and the backing plate. Its calculation is made from a trade-off between the bristles radial deflection due to pressure loading, and the rotating shaft thermal dilatation due to friction. When designed correctly, the backing plate provides a minimum sealing capability, in case of the brush seal failure [6].

- The clearance region, being the distance between the bristles tip and the shaft. If the shaft diameter is greater than the seal internal diameter (bristles included), the bristles are bent, and the clearance takes the name of “interference” instead, and its (negative) value is given by the difference between both radii of the shaft and the internal region of the seal.

The experimental work to be carried on by the ULB and SNECMA consists in evaluating the influence of the geometrical parameters on the performance of brush seals. The performance is characterized by air consumption, friction torque, dissipated heat, and oil leakage. The working parameters are air pressure and temperature, oil flow and temperature, and rotation speed.

**Test bench**

The following gives a description of the experimental setup used to carry on this work. A general schematic on Figure 4 gives an overview of the test bench principle.

The range of parameters describing the capabilities of the test bench is listed in Table 1.

The following test bench is designed to measure the performance of the brush seals, notably oil leakage, air consumption, torque friction and wear. The use of this
bench can naturally be extended to any type of seal considered for oil/air sealing, (or air/air sealing). Table 1 summarizes the range of parameters of the installation.

<table>
<thead>
<tr>
<th>N \ (rpm)</th>
<th>( T_{\text{air}} ) \ (°C)</th>
<th>( \Delta p ) \ (bar)</th>
<th>( \dot{V}_{\text{oil}} ) \ (L/h)</th>
<th>( T_{\text{oil}} ) \ (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Min</td>
<td>0</td>
<td>Amb.</td>
<td>0</td>
<td>Amb.</td>
</tr>
<tr>
<td>Max</td>
<td>20.000</td>
<td>150</td>
<td>1.5</td>
<td>100</td>
</tr>
</tbody>
</table>

Table 1: Test bench range of parameters

The test bench is divided into three parts: An air supply, an oil circuit, and a mechanical rotating unit, which is the spot where the seal will be submitted to the working conditions encountered in aero-engines.

As shown on Figure 6, the oil circuit is constituted of two loops. The first loop begins with an oil tank divided into two parts: An internal tank (70 l) storing the working fluid used for the tests, and an external tank (145 l) where oil with a high thermal conductivity is both used as a heater and temperature holder during the tests. The lubrication oil used as the working fluid is the BP 2380, which respects the norm MIL-PRF-23699 as the one used in real aero engines.

The mass air flow is measured with a Coriolis mass flow meter. Optionally, a 10 kW heater can be introduced between the flow meter and the air chamber. This operation would highlight the density effects on the flow factor of a brush seal.

A gear pump generates the oil through the control of the rotation speed with a precision of 0.1% of the maximum value. The oil is filtered, then its flow is measured by a flow meter, which is a turbine providing a value to be corrected in function of the oil temperature.
and viscosity. The return to the oil tank closes the first stage loop of the circuit, enabled at the beginning of the test for heating purpose before directly injecting oil in the rotating unit. The second loop leads to the injector installed inside the oil chamber. Typical oil flows must be comprised between 34 l/h and 37 l/h. The scavenge port leads to a pump that sends the oil back to the oil tank through a second filter. The latter gathers brush seal bristles that may be ripped off during testing.

Figure 7 gives finally a cross section of the rotating unit. The brush seal sample is clamped between two steel lodging elements that separate oil mist and air. Oil enters the left chamber, which the shaft penetrates. The bearings are greased for life, to ensure oil is not consumed by bearing lubrication, and to avoid any additional heat increasing the oil temperature. Oil is injected as a high-pressure jet. It is important to not directly spray oil right on the brush seal to prevent any harm to the latter. Instead, it will flow on the chamber internal sides before wetting the seal. Finally, an interchangeable rotor disc is mounted. This ensures the ability to test several clearance/interference configurations, as well as different roughness. The right chamber confines pressurized air on the other side of the seal, and a zinc-selenium window is mounted, to provide via an IR camera, thermal imaging to observe the gradient temperature on the rotor disc, therefore deducing the dissipated heat due to friction.

Results

Eight carbon brush seals prototypes were tested. For confidentiality purpose, the authors were not allowed to communicate the bristles dimensions, nor the fabrication process. It will only be mentioned the bristles cant angle equals 0°, which means all the bristles converge to the center of the brush seal. Table 2 lists the geometric parameters of the tested brush seals. Also, eight rotors will be tested. Their diameters will be expressed on Table 3 as a percentage of the brush seals internal diameter, which is the same for all.
Free length | Density | Interplate distance | Bristle pack thickness
--- | --- | --- | ---
1 | Nominal | High | Nominal | Nominal
2 | Nominal | High | + 80% | Nominal
3 | + 133% | High | Nominal | Nominal
4 | + 133% | High | + 80% | Nominal
5 | Nominal | Low | Nominal | - 12%
6 | Nominal | Low | + 80% | - 12%
7 | + 133% | Low | Nominal | - 12%
8 | + 133% | Low | + 80% | - 12%

Table 2: Geometric parameters of the tested brush seals

<table>
<thead>
<tr>
<th>Rotor (% of brush seals internal diameter)</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>100.12</td>
<td>100.42</td>
<td>101.14</td>
<td>101.84</td>
<td>102.45</td>
<td>103.07</td>
<td>104.29</td>
<td>105.37</td>
<td>106.04</td>
<td></td>
</tr>
</tbody>
</table>

Table 3: Available rotor diameters

Preliminary results in static conditions give first orders of magnitude to expect as far as air consumption and friction torque are concerned. More details about the development of these equations can be found in [7]. It is assumed that the bristles are homogeneously disposed along the axial direction and the internal diameter of the seal.

A modified definition of the flow factor was taken into account to highlight the effect of density

\[
\varphi' = \frac{m \sqrt{T_{air}}}{P_{air} A_0}
\]  

\(m\) is the air mass flow, \(T_{air}\) is the average temperature, \(P_{air}\) is the upstream pressure, and \(A_0\) is the section defined by the gap between the brush seal ring and the rotor.

Figure 8 shows the evolution of the modified flow factor in function of the bristle density with no interference.

![Figure 8](image)

Figure 8: Modified flow factor as a function of the pressure ratio for the lowest fence ratio height

The green curve represents the theoretical evolution of air flow in function of the pressure ratio for a brush seal with no density. The calculation was made upon the orifice flowmeter model defined by [8]. Seals 1 and 3, and seals 5 and 7 share the same bristle axial density, but different bristle free lengths. Experimental results have shown that the curves for brush seals with the same density are confounded for small pressure ratios. The modified flow factor equation of a brush seal with a known density has been estimated by multiplying the orifice flowmeter equation \(\varphi_0\) by a discharge coefficient that is expressed in function of the bristles axial density factor \(\sigma\).

\[
\varphi' = \varphi_0 C_d^\sigma
\]  

Therefore:

And \(\sigma\) is expressed in function of the bristle axial density. The
formulas is undisclosed, but its curve is plotted in figure 9.

![Figure 9: Evolution of the density factor as a function of the axial density](image)

Unfortunately, only two different values of density were available at this time to mathematically establish its influence. More experiments should be led to obtain a general equation, but as a first approach, it is assumed the density factor $\sigma$ curve follows a logarithmic shape.

Once the porosity of the bristle back with no interference has been estimated, the next step in the static model development would be to take into account the effect of the interference. Unlike for canted brush seals, the blow down phenomenon does not exist, therefore, mounting a brush seal with a clearance configuration (a positive gap between the bristle pack and the shaft surface) would be useless because of the additional leakage created by this gap.

On Figure 10, the relative flow factor (the percentage of the flow factor that has been obtained with no interference) has been plotted in function the interference percentage, which is the ratio between the interference measured in millimetres over the bristle free length for a rotation speed of 300 rpm.

![Figure 10: Flow factor in function of the interference](image)

As expected, the flow factor decreases with an increasing interference. An interference of 20% is enough to reduce by half the air consumption of a brush seal. But the most important observation is that each curve evolves along the same pattern. The relative influence of the interference remains the same for every brush seal. Therefore, it is possible to define an equation to model the effect of the interference on the flow factor. The interference will induce a correction factor that will decrease the flow factor of the brush seal, as expressed in equations (3) and (4)

\[ \varphi' = \varphi_0 C_d' \sigma f(l/L) \]  

(3)

With:

\[ f(l/L) = e^{-0.0369(l/L)} \]  

(4)

As far as friction torque prediction is concerned, the
evolution of the friction torque with speed and pressure needs to be consolidated. However, first trends have been identified concerning the direct effect of the geometry on the starting torque in function of the interference. The starting torque of the brush seal is defined as here as the maximum torque the motor needs to develop to overcome the brush seal resistance, and to start rotating. Figure 11 shows the evolution of the starting torque of 8 brush seals in function of the interference. For this graph, specifically, the interference is defined as a percentage of the longest bristles available on the prototypes.

The highest friction torques are reached by seal 1 and 2 for an interference of 20% of the longest bristles, and for an interference of 62% for seals 5 and 6. Also, Seal 5 and seal 6 reach a friction torque of 50% for a higher interference than seal 1 and seal 2. This means the primary parameter that acts on friction is the bristle free length. The shorter the bristles free length, the higher are the risks of bristles being squeezed under the action of the rotor, which creates the additional unneeded friction. The inter-plate distance effect on the friction is the most noticeable on bristles with short lengths (at an interference 20%, seal 1 and seal 5 friction torques are twice higher than seal 2 and seal 5).

Therefore, the bristle free length and the axial density are the primary parameters that act on the brush seal air consumption, whereas the inter-plate distance would rather act on the friction torque by softening the bristle pack. This allows less friction for a limited impact on the air consumption.

Finally, a trade-off between friction torque and air consumption has been highlighted by plotting the air consumption-to-pressure load ratio in function of the friction torque on figure 12.

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Three zones can be distinguished:

Zone 1 regroups high permeability seals, with low torque friction. The sealing performance is regarded as very poor, which make these seals unsuited for the application. Zone 2 regroups low permeability seals, with high torque friction. Although the seal would definitely fill properly its sealing role, high torque friction indicates premature wear on both the brush seal and the shaft, therefore a
supposedly low endurance. Finally, zone 3 regroups the seals with the best trade-off between friction and permeability. The optimum combination brush seal/rotor diameter should be in this zone. Furthermore, rotor dynamics issues tend to increase the torque friction and improve the brush seal performance while amplifying its wear. It is then suggested to be on the left area of zone 3.

**Conclusion**

ULB, in collaboration with SNECMA, developed a new test rig reproducing the realistic working conditions the brush seals will be submitted. So far, empirical relations have been developed to predict the evolution of the air consumption as a function of the geometrical data, notably the axial density, the bristle free length and the interference. But these measures need to be consolidated, especially regarding the friction torque prediction and the characterization of bristle pack lifting. A trade-off zone between friction torque and air consumption has been defined by a Pareto-frontier shape. Some combination of carbon brush seals with some values of interferences present sensibly the same level of performance. These samples will be ultimately submitted through an endurance testing, as the evolution of the performance with time will also be implemented in the model. Experiments underwent by SNECMA showed the brush seal performance decrease and converge quickly to a steady state, corresponding to a non-existing interference. The next steps of the experimental study consist in evaluating the influence of high temperature oil, rotation speed, the combination of rotation speed and high temperature oil, and air temperature.

Ultimately, the empirical model has to be confirmed by the coupled CFD/FEM model that is currently being developed by the ATM department [9].

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**References**


