Testing of Carbon Brush Seals for Aero-Engine Bearing Chambers

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Abstract
Recent developments in the aeronautic domain focus strongly on the reduction of aero-engine specific oil consumption thanks to the optimization of the lubrication oil system for civil aircraft gas turbine engines. Specifically, as brush seals have shown tremendous leakage performance in sealing secondary flows compared to classic labyrinth seals over the last few decades, an increasing idea is to extent their utilization to oil bearing chamber applications.

This paper presents general points about the brush seal functioning, a description of the new test bench developed by the ULB in collaboration with SNECMA, the complete experimental study to be performed on brush seals by the second part of the year 2013, and the first air consumption performance results obtained on brush seals prototypes.

1. Introduction

For ecological and economical reasons, efforts are intensified to find new ideas to reduce the aero-engines necessary resources, while maintaining at least the same level of performances. Theses resources comprise fuel, air, and oil consumption

Bearing chambers are compartments where bearings are lubricated by oil. An accurate lubrication needs to go through accurate sealing. As brush seals showed tremendous leakage performance in sealing secondary flows, an increasing idea is to substitute the commonly used labyrinth seals by brush seals in bearing chambers [1]. However, interrogations arise on the brush seal behavior in contact with oil at high temperature, and what will be the repercussions on the long-term performance seal.

In the frame of the European FP7 E-Break project, the Aero-Thermo-Mechanics department of ULB collaborates with French aircraft engine manufacturer SNECMA in order to investigate experimentally the brush seal behaviour in an environment simulating the bearing chamber working conditions. The aim is to improve the brush seal behaviour knowledge by identifying the most influential geometric parameters acting on its wear, and by evaluating the leakage performance on both sides of the seal (oil and air leakage).

2. Brush seal

A brush seal is a clearance seal used in turbomachinery since the 80's. It is an annular seal composed of high density bristles, usually canted in the shaft rotation direction, to prevent any harm due to the shaft radial excursion from its initial position.

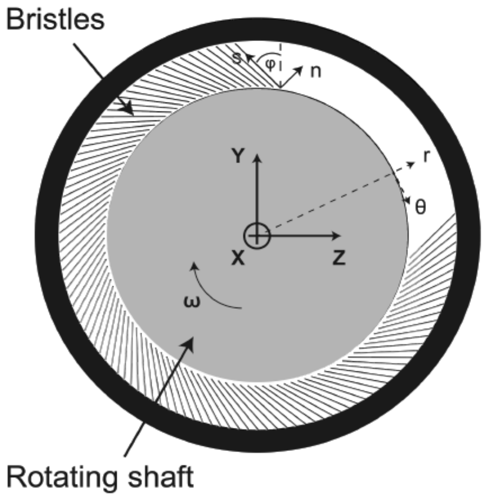

Figure 1 : Schematic front view of a brush seal

Carbon brush seals are more and more considered because of their supposedly low heat generation, especially compared to one of the last materials tested successfully for bristles, aramid fibers. The latter withstand indeed lower temperatures than the carbon ones (523 K compared to 643 K) [2-5], and is likely to present a lower friction coefficient. Meanwhile, metallic brush seals present the danger of generating small metal chips resulting of wear, which would very harmful for bearings [3].

The seal 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 [4].

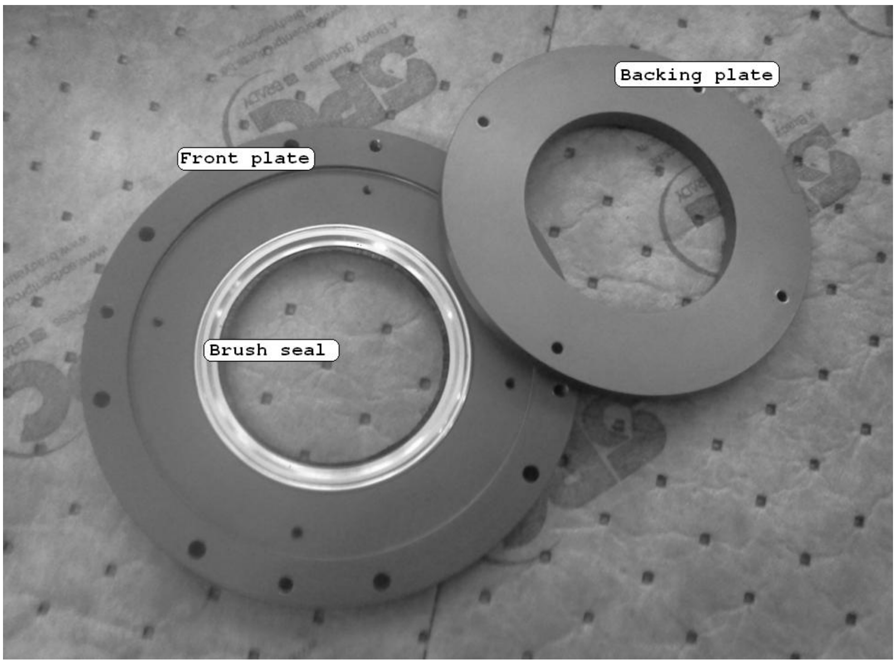

Figure 2 : Brush seal, front plate, and backing plate

Two particular zones can be distinguished when the seal is mounted:

- 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 [4].

- 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.

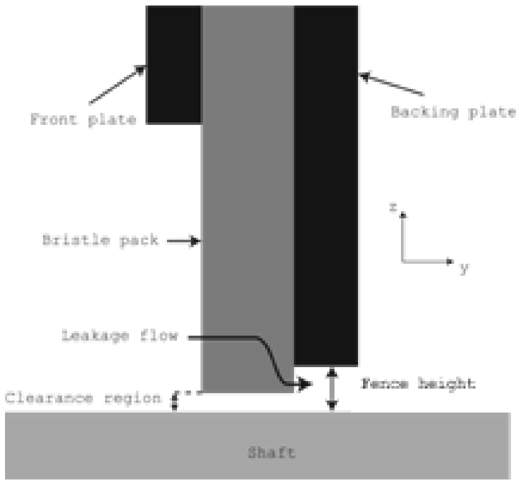

Figure 3 : Schematic side view of a brush seal

Beside the material, the main parameters influencing the brush seal design in function of the shaft rotating through are the bristles free height, diameter, density in the axial and radial direction, the pack thickness, and the lay angle. The seals must be sufficiently efficient to limit oil leakage, but a significant amount of oil have to flow through the bristles to prevent overheating, resulting in the seal destruction.

3. Test bench

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), as long as their dimensions do not exceed the oil and air chambers separation plate bulk. And of course, the backing plate and the front plate are elements which are specific to the seal, which means both of them have to be sized and fabricated adequately (bore diameter, depth, seal compression between the plates) to fit in the test rig.

Figure 4 : Brush seals test bench

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

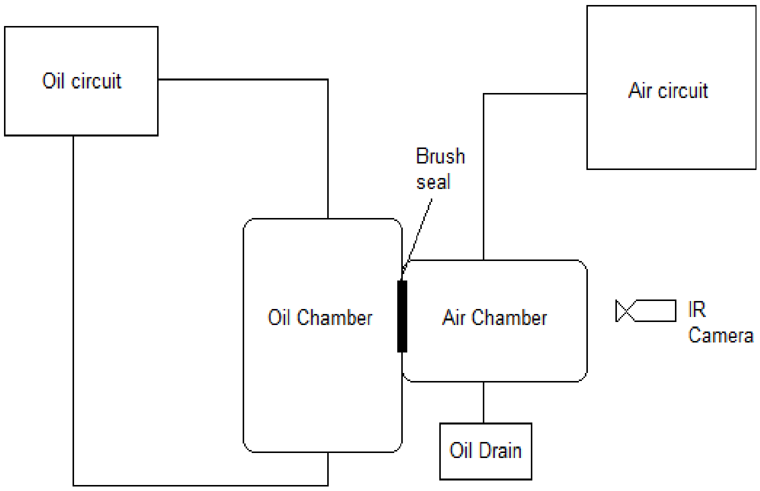

Figure 5: General test bench principle

The mechanical part is considered as the brain of the whole test bench. A recognizable characteristic is its small dimension, as all the elements to be described below hold on a 550x450 mm aluminum plate First, the seal is clamped between a steel lodging element (front plate) and its backing plate. The whole set compose the interface between two chambers.

Figure 6 : Mechanical part (with chambers)

Oil is injected in the form of a mist in the first chamber, with a wide angle spray nozzle injector. It is important not to directly spray oil right on the brush seal to prevent any harm on the latter. Instead, the oil has to flow on the chamber internal sides before wetting the seal. The second chamber confines pressurized air on the other side of the seal.

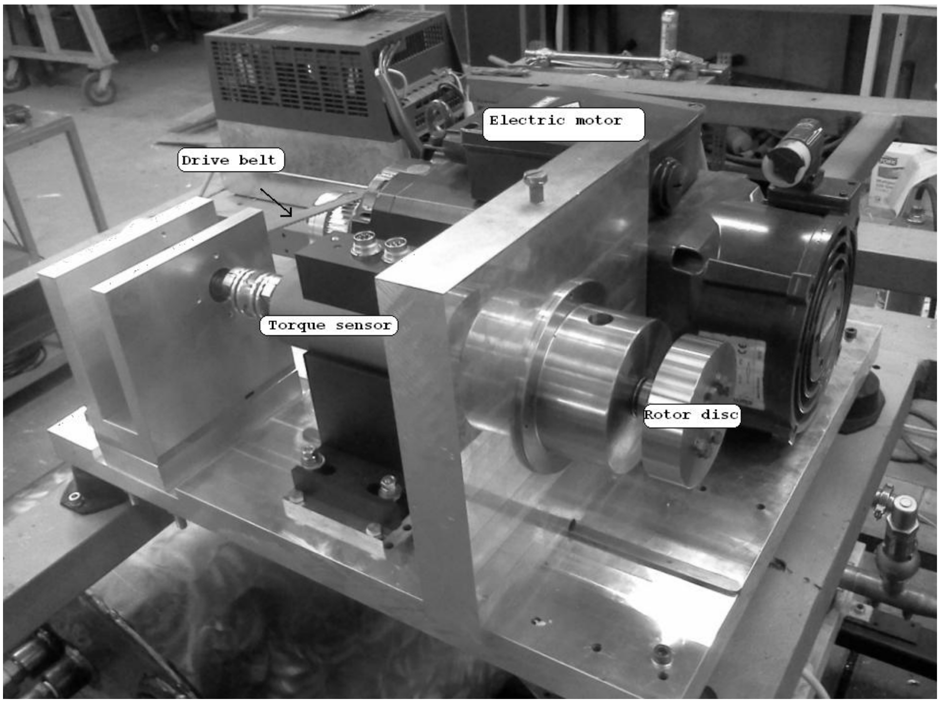

Figure 7 : Mechanical part (without chambers)

A stainless steel shaft penetrates the oil chamber. High rotation speeds to be achieved lead to decompose the shaft design into two parts. The first one is a long shaft with a very small diameter (less than 22 mm), to fit in rolling bearings that meet the rotation speed specifications of the bench. And the second part is a rotor disc, mounted on the small shaft, and which role is to rub on the brush seal. Another advantage of this configuration is the ability to test several clearance/interference values, thanks to an easy switch of the rotor discs. The latter are usually manufactured with a precision of 1/100 of a millimeter, which guarantees a balanced configuration at very high rotation speeds.

The shaft is linked with a torque sensor. This instrument will give a value of the torque friction generated by the contact between the brush seal and the rotor disc. The evolution of the torque friction with the working duration will give a quantification of the wear. The torque sensor full scale is at 3 Nm, and its precision is at 0.2%.

The torque sensor is linked to an electric motor thanks to a timing belt made of polyurethane with steel armatures. With the torque control activated, the electric motor maximum speed reaches 17000 rpm. The speed ratio of the belt transmission is 2:1, which gives then a maximum rotation speed of 34000 rpm for the shaft located in the oil chamber. Therefore, the maximum linear surface speed seen by the bristle tips would top 150 m/s.

Instrumentation is added around the seal to monitor the temperature around it during a test. It is composed of six Pt100 sensors with dimensions of 5x2 mm, and with a precision of +/- 0.1 K, and disposed around the seal each 60 degrees. Three of them are disposed on the backing plate, close to the bristles temperature, and the other three are placed on the lodging element, close to the seal annular element. Their assembly is guaranteed at temperatures reaching 180°C thanks to epoxy glue.

The temperature measurement around the seal is completed by an observation of the temperature gradient on the rotor disc thanks to an infrared camera. The imaged are provided through a zinc-selenium window, placed 20 cm in the front face of the air chamber.

The air circuit is composed of a rotary screw compressor, able to deliver a maximum continuous pressure of 10 bars. A filter is placed at the compressor outlet, retaining particles down to 5 µm. It is combined to a manual pressure regulator and a drier as air delivered by such a compressor is usually humid.

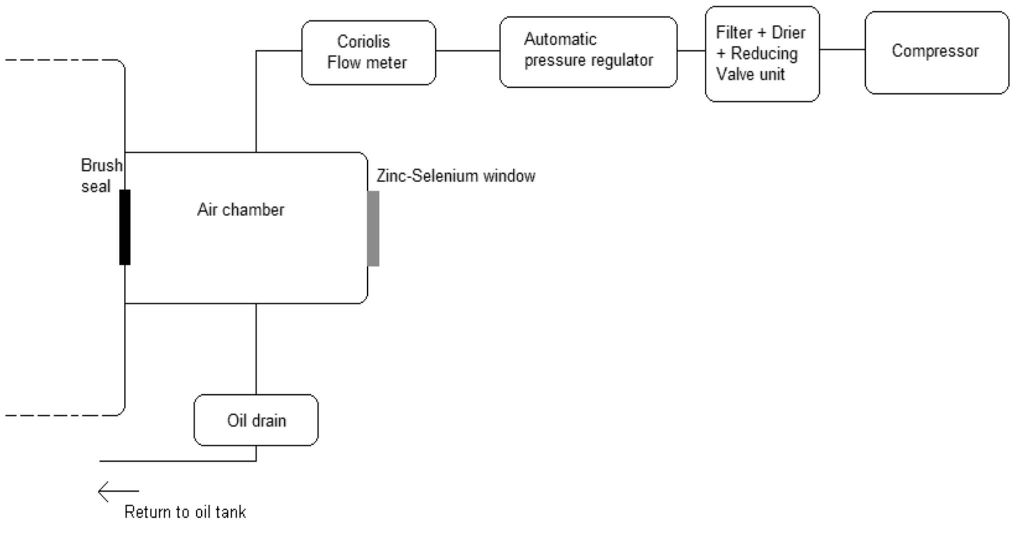

Figure 8 : Air circuit principle

The pressure is automatically regulated with a valve in function of the oil chamber static pressure, in order to maintain a constant pressure difference between the two chambers, which should reach maximum 1.5 bars

The air flow is measured thanks to a Coriolis mass flow meter. The measuring membrane of this instrument is a vibrating tube in which the fluid flows. The higher is the fluid density, the higher is the deformation due to the flow. The flow value is calculated by the flow meter thanks to the measurement of the frequency vibration of the tube, which is proportional to the flow.

The air will not be heated during the future tests to be performed on the bench. The reason is to estimate accurately the rotor disc dilatation only related to the temperature increasing due to friction. The test bench configuration will although allow to introduce a 10 kW heater between the flow meter and the air chamber if needed.

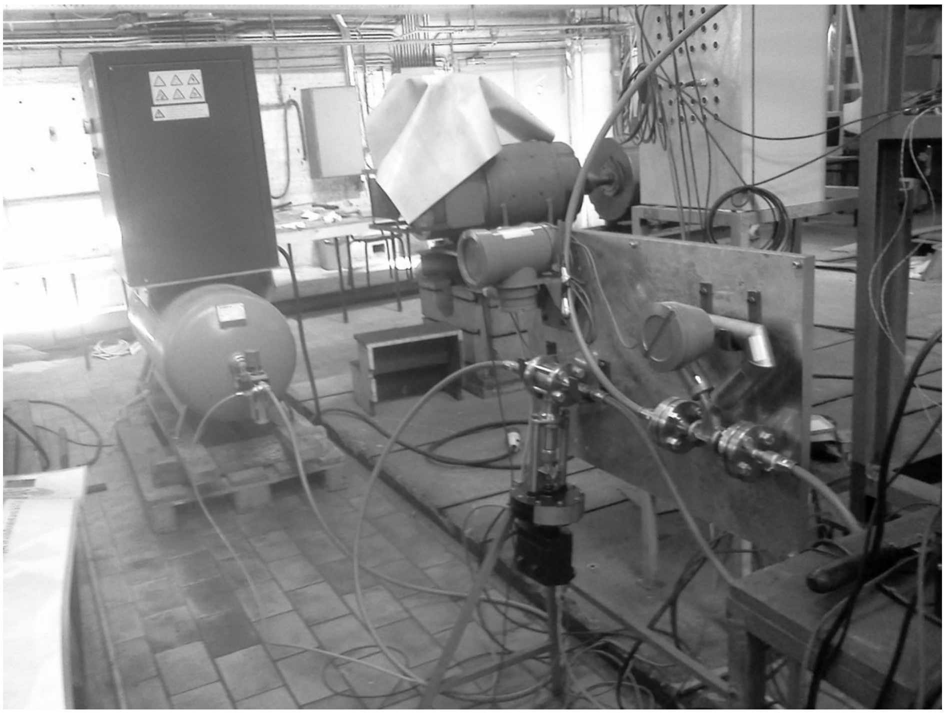

Figure 9 : Air circuit (partially mounted)

The circuit ends in the air chamber, where the air flow circulates through the seal. An oil drain is connected below the air chamber to collect, and then measure the oil leakage by regular intervals.

The oil drain is constituted of a small pipe, with two valves at the inlet and outlet. The pipe, placed vertically, contains also two contact sensors at its extremity, constituting the leakage measure module.

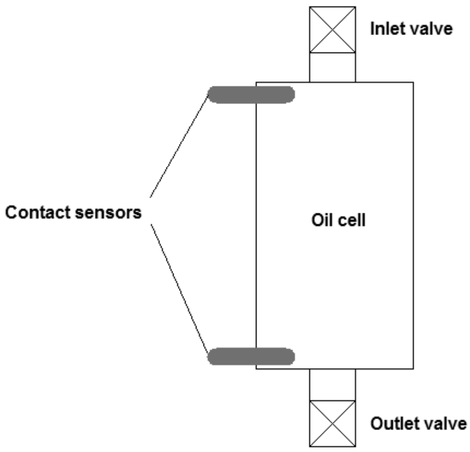

Figure 10 : Oil leakage measuring module principle

When the module is activated, the inlet valve opens, the outlet valve stays closed, and the pipe starts to be filled with oil, with the first contact sensor being immerged. When the oil reaches the second contact sensor at the top, the pipe inlet valve closes immediately before the outlet valve opens for oil evacuation, to prevent any depressurization of the air chamber. The oil leakage is deduced by the LabView program, in function of the pipe volume, the oil density, and the Δt of the immersion of both contact sensors. The process can be repeated during the whole test duration.

The oil circuit is the same utilized for testing of air cooled oil coolers breadboards at the Royal Military School of Brussels [7], and adapted to the bench requirements. The oil must respect the norm MIL-PRF-23699 as the one used in real aero engines.

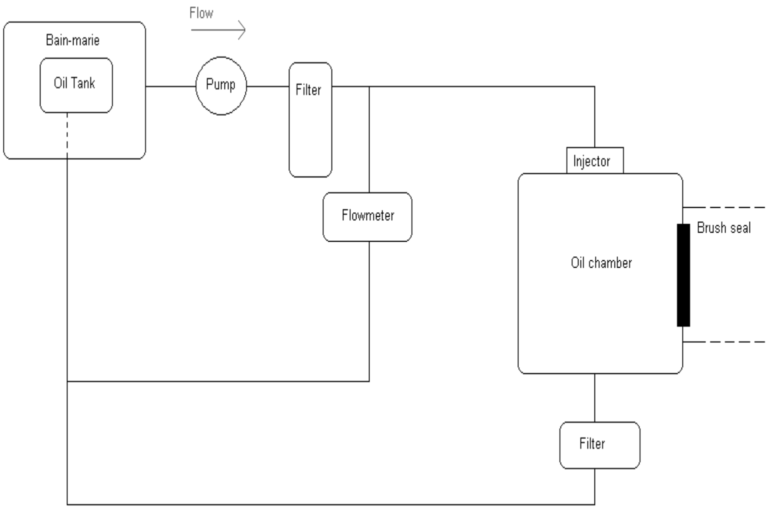

Figure 11 : Oil circuit principle

It is composed of an oil tank divided into two parts: An internal tank (70 l) storing the oil 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 maximum temperature that can be reached by the oil is 160°C, but it will be limited to 100°C only for the tests.

A gear pump generates the oil through the control of the rotation speed with a precision of 0.1% of the maximum value, enabling a flow reaching up to 1800 l/h.
The oil is filtered, then 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 oil chamber.

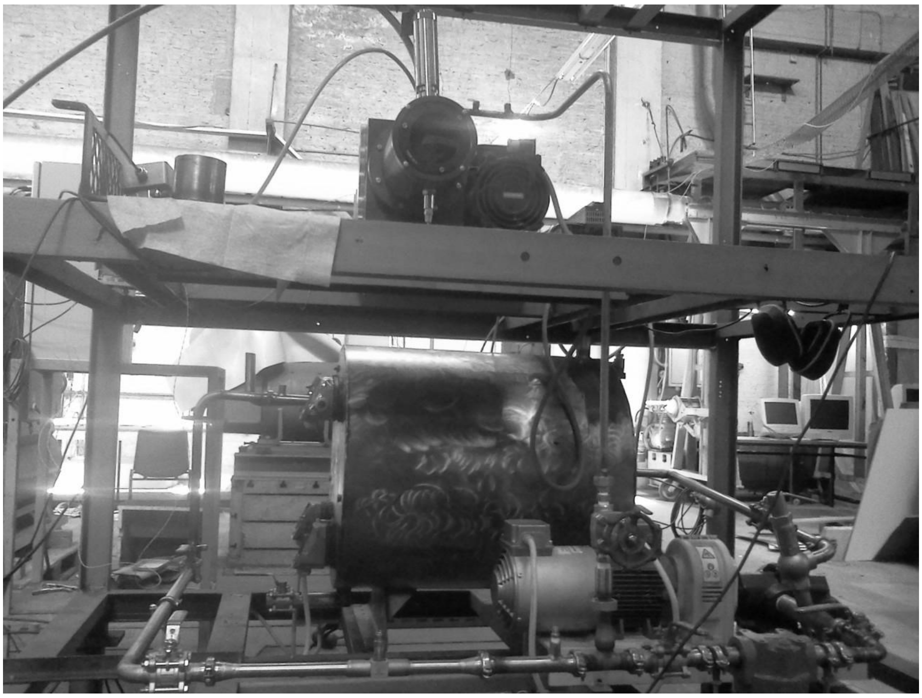

Figure 12 : Oil circuit (partially mounted)

The second loop is constituted of a manual slide gate valve, and pressure and temperature sensors are placed before the injector, which is installed inside the oil chamber, in which the oil flow must be comprised between 34 l/h and 37 l/h. The gear pump being currently used on the bench is oversized for the application, and a short term answer was to regulate the flow with a slide gate valve to be opened progressively after the oil has reach the right temperature.

The scavenge port leads to a second filter where oil (along with the air flowing through the seal) is evacuated using the gravity. Brush seal bristles that may be ripped off during the test are gathered by the filter, before returning to the oil tank. It was not necessary to add an air/oil separator downstream the oil chamber because the air/oil mixing is poured into a non pressurized tank, therefore the separation would be done naturally in the tank.

Finally, the bench is secured thanks to the equipment of software (accelerometers, pressure and temperature sensors all around the bench) and hardware (pressure relieves, kill switches) protections. Also, an air renewal system is installed to extract all the oil vapors from the bench and rejecting these to the outside, and a metallic construction will cover the whole bench with an acoustic protection.

4. First Test Campaign

The first test campaign to be led with this test bench features carbon brush seals with a squared section.

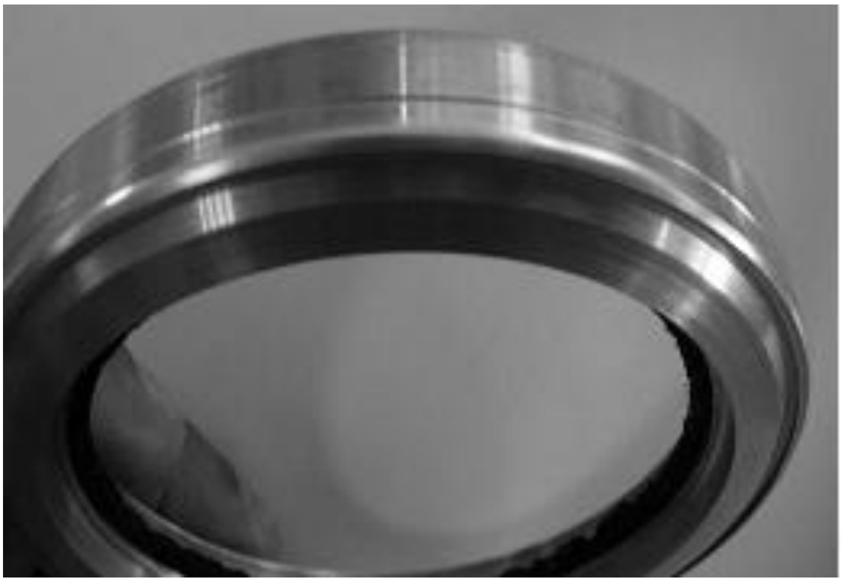

Figure 13 : Squared section carbon brush seal

The campaign will first be preceded by a reproducibility/repeatability analysis, to ensure the measurements are valid and coherent.

The most significant data to be collected are:

- Torque friction
- Rotor temperature
- Temperatures around the seal
- Air and oil temperatures
- Pressure in both chambers
- Oil leakage
- Air consumption

The main objective of this campaign is to identify the most influent parameters on their leakage performance (oil leakage and air consumption) and their wear, quantified by the torque friction (and deducing thereafter a brush friction coefficient) and the temperature reached by the bristles tip thanks to the infrared camera.

Each seal will be submitted to a static permeability measure, which means no oil will be injected, and the shaft will stay at 0 rpm. The Δp between the chambers will vary between 100 mbar and 1.5 bar. The purpose of these is to verify the simple presence of oil between the bristles reduces the amount of air needed for proper sealing.

Figure 14 shows the cycle (Rotation speed/Δp) that will have to be applied on each brush seal. The cycle time is only mentioned on an indicative basis. To preserve the integrity of the bench, especially when exceeding 20000 rpm, the measuring duration will be the shortest possible for each combination of rotation speed and Δp, as long as all the collected data converge.

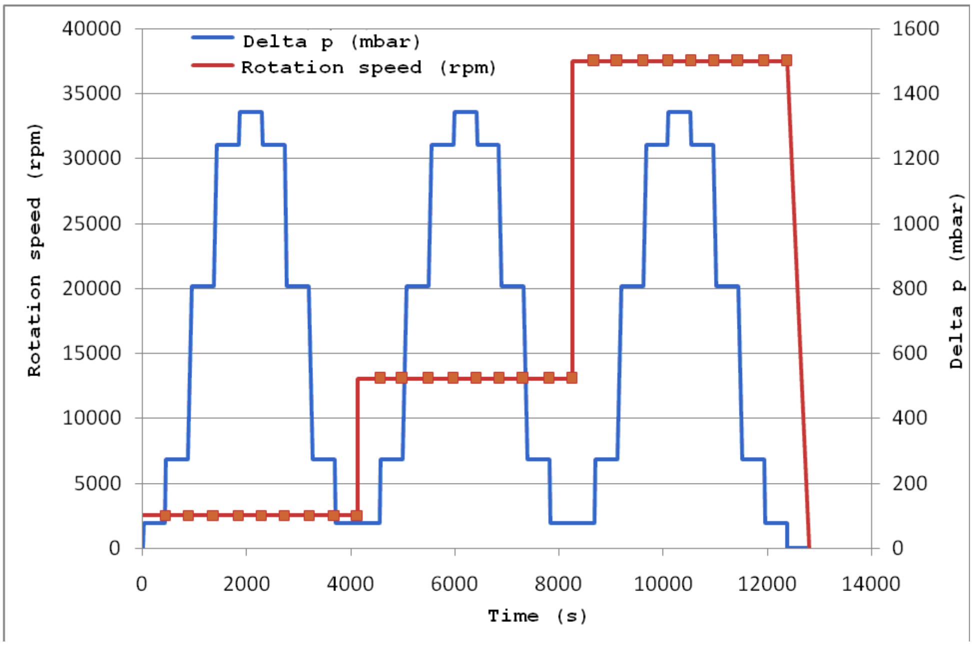

Figure 14 : Cycle to be applied on brush seals during the campaign

Ten different brush seals will be tested, as their inner diameter will be fixed at 81.5 mm, and their lay angle fixed to 0°.

The main geometric parameters to be evaluated are the bristles free length (1.5 mm, 2.5 mm and 3.5 mm), the bristle pack thickness (1 mm, 1.4 mm and 1.8 mm), and the number of bristles per strand (24000, 36000 and 48000).

Interferences of 0.17 mm, 0.46 mm and 0.75 mm will be tested, as well as the influence of the roughness (0.25 µm, and 1.6 µm).

An inspection of the seal will be provided before and after the test. The wear will be measured and characterized by evaluating the inner diameter and the brush seal ring diameter with a shadowgraph by SNECMA. The effect of coking will also be evaluated.

A finite element model will have to be developed in order to estimate the power dissipated through heat friction in function of the temperature profile measured with the infrared camera.

Finally, the results obtained with the test bench will help to validate the simulations performed in the scope of the work [8], and will provide additional data that can not be estimated precisely in the first place, such as the torque friction coefficient.

5. First results

The first results obtained with the test bench were measurement of the static air consumption (i.e. the air flowing through the brush seal) in function of the pressure difference between the chambers, meaning the shaft does not rotate at all, with the oil circuit is deactivated. The oil chamber pressure is kept close to the atmospheric pressure. These tests were the first opportunity to highlight the repeatability of the results provided by the test rig.

The following results are obtained by testing a seal with an axial density of 48.000 bristles per millimeter, a bristle pack thickness of 1 mm, and a free length of 1.5 mm.

Also, the interference between the seal and shaft equals 0.75 mm.

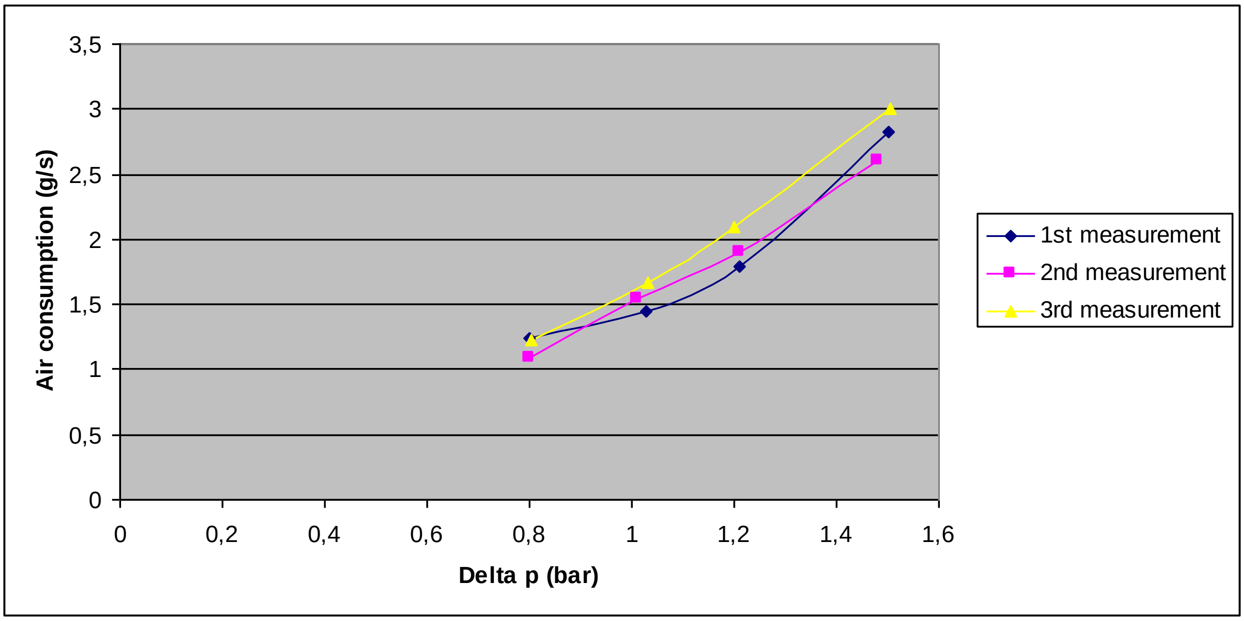

Figure 15 : Air consumption measured in function of the pressure difference

Overall, the air consumptions measured in all three tests are quite similar, taking into consideration the low precision at low flow measured. It is noted that the delta p lower limit was settled to 800 mbar due to the flow meter which can't measure flows below 1 g/s. A particular attention is paid to the values of the air consumption which appear to be very low (3 g/s for a delta p of 1.5 bars, whereas for labyrinth seals, the mass flow measured is 10 to 15 times higher [9]). A second test was made by oiling the brush seal with a paintbrush (therefore the oil is kept at ambient temperature).

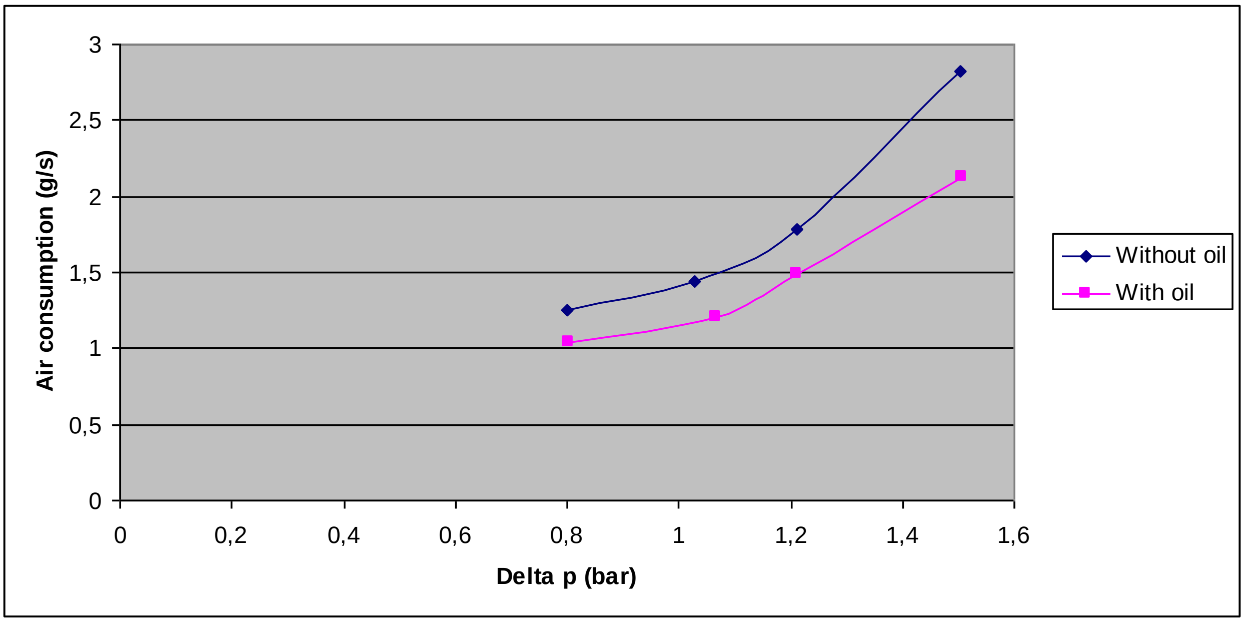

Figure 16 : Air consumption comparison between air and oil

It appears the oiling of the brush seal improves significantly the air consumption (up to 30%), as it appear the only presence of the oil stuck between the bristles reduces its porosity. The interpretation needs to be deepened, especially with the values of the friction torque, and the evolution of these data in function of the shaft rotation speed.

6. Conclusion

Brush seals are widely considered as one of the main technologies to be developed for bearing chambers in the future. However, new interrogations arise since these are in contact with oil at high temperature, especially about their performance and their endurance.

ULB, in collaboration with SNECMA, developed a new test rig reproducing the realistic working conditions the brush seals will be submitted. The test bench is a modular one. Any type of seals beside the brush ones can be tested, and parameters such as the rotor interference/clearance or roughness can easily be tested by simply switching the rotor discs.

The results of the first campaign, expected during the second part of 2013, feature a comparison of different brush seals designs. The inner diameter, the material (carbon) and the lay angle are all the same, but the influence of the bristles free height, pack thickness, and density will be highlighted. Answering the questions mentioned above allow a deeper comprehension of the brush seals functioning, and will provide date for the CFD tools developed by the ULB in parallel.

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