Brush Seal Dynamic Stiffness Behavior

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Brush seals are emerging as alternative to labyrinth applications in turbomachinery. Seal stiffness determines wear rate and long-term leakage performance. Stiffness increases under pressure as bristles get stuck at the backing plate. Complex brush structure does not allow simple modeling of operating conditions. Common static-unpressurized measurements do not represent operating stiffness due to changes in bristle behavior under pressure load. This work presents brush seal stiffness under combined pressure and speed conditions. Contact forces are measured via load cells. Static and combined dynamic stiffness results are presented for various operation conditions. Testing methodology, calibration procedure, and stiffness results are discussed.

Keywords: Brush seal, gas turbine, rotor wear, stiffness measurement.

Nomenclature

\( BH \quad = \quad \text{bristle height} \)
\( F \quad = \quad \text{seal force applied to the rotor surface} \)
\( FH \quad = \quad \text{fence height} \)
\( K_b \quad = \quad \text{bristle stiffness} \)
\( N \quad = \quad \text{rotor speed} \)
\( mil \quad = \quad 1/1000 \text{ of an inch} \)
\( P_H \quad = \quad \text{upstream pressure} \)
\( P_L \quad = \quad \text{downstream pressure} \)
\( P_H \quad = \quad \text{upstream pressure} \)
\( rpm \quad = \quad \text{revolution per minute} \)
\( \Delta P \quad = \quad \text{pressure difference} \)
\( \Delta R \quad = \quad \text{radial excursion} \)
\( \Theta \quad = \quad \text{lay angle} \)

1. Introduction

Conventional turbine seals are improved substantially and pushed to their limits to meet the demand for greater efficiency in competitive energy market. A more advanced brush seal technology is steadily taking the place of the present labyrinth seals. Brush seals offer higher sealing effectiveness thanks to the compliant bristles that enable reduced seal-rotor clearance. A brush seal is a set of fine diameter metallic bristles that is densely packed between front and backing plate (Figure 1). To keep away the bristles from buckling during rotor interference, they are aligned with an angle compatible with the rotational direction of the rotor. The angle allows bending to reduce the contact loads that impacts wear rate significantly. Wear at the rotor surface due to the rotor-bristle contact becomes the determining factor for the operation life and performance of the seal.
In comparison to the conventional labyrinth seal, the brush seal offers a dramatic improvement in leakage rate that contributes to the efficiency and the power of the turbomachinery. The compliance of the bristles also enables this improvement to be conserved when there are differential movements between shaft and seal housing during engine transients. During operation rotor excursions may take place due to maneuver loads, transient thermal expansion and elastic deformation of the rotor. Frictional forces are generated between bristles themselves and between the bristle pack and the backing plate by means of the axial pressure difference. Frictional forces induce much stiffer bristle behavior when seal is pushed in radial direction during a rotor excursion. This pressure stiffening causes a dramatic increase in contact loads at the rotor surface, which results in high wear rates. Pressure-stiffening coupling in brush seals needs to be studied in detail for accurate estimation of contact loads and frictional heat generation at the bristle tips.

Stiffness is one of the critical seal characteristics, which defines average bristle tip pressure at the rotor surface contact. It can be defined as the change in bristle tip force per unit change in radial interference. Some authors prefer to define it as the pressure required at the bristle tips to displace them radially by a unit magnitude [1, 2]. Bristle stiffness $K_b$ is calculated by dividing the seal reaction force $F$, developed at the bristle tips, to the imposed radial interference $ΔR$.

Stiffness is a measure of the brush seal bristle resistance to deflection. For a given seal, its value changes with differential pressure. In general, due to the pressure-friction coupling, seal stiffness increases with pressure increase. The very complex nature of brush deflection and inter-bristle interactions does not allow a well-posed analytical formulation. In the literature, there are some experimental studies [3, 4] which aim to directly measure seal-rotor contact forces. However, these tests are highly complicated, and may involve measurement errors if not conducted carefully. Others [1, 2] produced methods to characterize an overall seal stiffness. Basu et al. [1] and Short et al. [2] tried to measure overall seal stiffness through testing of different seal designs under various differential pressure levels. In their measurement procedure, fully circular seals are pressurized and radial interference is applied. The normal force required for the displacement is collected through a load cell. Dividing this force by the employed interference gives the so-called ‘seal stiffness’. However, all these experiments conducted statically, i.e. in the absence of any shaft rotation. In addition, the experimental data reported by Basu et al. [1] and Short et al. [2] did not include any information on the test samples. Franceschini et al. [5] utilized a slow speed rotating test facility to measure the stiffness of the brush seals. They simulated the eccentric operation and supplied pressure to the system. Although the facility could capture the stiffness behavior of the seals under pressure, it cannot fully demonstrate the effect of real rotor speed and radial growth of the shaft.

Typical brush seal stiffness without pressure load may vary from 54.3 kPa/mm (0.2 psi/mil) to 230.7 kPa/mm (0.85 psi/mil) according to the experimental data [6]. However, these stiffness values grow multifold when pressure is applied because of the strong friction coupling and blow-down with radial leakage flow. Chupp et al. [7] reported 2-3 folds increase, while Short et al. [2] reported up to six folds increase in the overall seal stiffness with growing pressure loads. In fact, various experimental observations demonstrate that seal contact force increase depends on the pressure load increase. Due to the complex behavior of brush seals, stiffness values markedly change based on operating conditions. The above mentioned studies presented estimated seal stiffness under pressure load conditions when seal is tested without rotation. There is no reported work in open literature that presents measured brush seal stiffness under combined pressure and rotor speed conditions.
In this work, a special test system that is capable of measuring brush seal stiffness under combined pressure application and rotor movement is utilized to measure brush seal dynamic stiffness under pressure. The results reveal that seal stiffness is affected both by the applied pressure and the rotor motion. As noted by Modi [8], dynamic impacts affect the behavior of the bristles during rotation. This work offers a deeper insight to the dynamic stiffness characteristics of brush seals. Results of the experiments are discussed by comparing to the limited data in open literature.

2. Dynamic Stiffness Test System

The components of the custom designed brush seal test rig is demonstrated in Figure 2. Housing is the base part for the seal housing assembly. Test air is supplied through an air inlet hole to the cavity formed between rotor and two test brush seals [9]. Three probe holes are threaded two of which are for pressure gauges, and the third one is for the thermocouple. The reaction force applied by seals to the housing assembly during lateral displacement is measured via sensitive load cells. The seal housing assembly is attached to a linear slide through the load cells. There is a very small gap, i.e. 0.4 mm, between housing and linear slide that prevents direct contact. The housing assembly can move laterally on the linear slide. To introduce precise interference between brush seal and rotor surface a high precision zero backlash motorized slide is adopted to bear the assembly. By rotating the motor with small increments a desired radial interference can be applied. Since the housing is hanged on load cells, total reaction force is measured without frictional effects. System is calibrated by subtracting any bias loads that are measured with a load cell pushing on the housing without any seal-rotor contact. The measured pressure and load values are transferred to a computer by means of 8-channel data acquisition system. Since the testing medium is pressurized air, O-rings have to be used between mating housing parts to avoid bias leakages. For this purpose, O-ring grooves are opened on both sides of the housing. Seal housing components are assembled using bolt-nut combination.

The seal housing assembly is designed to test two seals at the same time. Symmetric location of two brush seals eliminates any axial loading on the rotor. This brings the advantage of high speed testing at elevated pressures which are typically experienced in gas turbine brush seal applications. Bolt holes are drilled through the side surfaces of the housing to attach seal holders. Use of through bolts provides easy assembly and disassembly. Removable seal holder rings allow testing of different brush seal geometries and facilitate testing of other rotary seals.
Shaft rotation is powered by a high frequency spindle. The spindle is capable of reaching 45000 rpm as maximum rotational speed at 25 kW output rating. Power of the spindle is directly transmitted to the rotor by a connection rod, which is produced from the run out test rod that is supplied with the spindle. The test rod gives 8 µm run out at 10 cm from spindle end.

Testing starts with calibration procedure where housing is laterally pushed by a sensitive load cell representing seal-rotor interference. Piezoelectric load cells carrying the housing assembly measures the reaction force which includes bias from hoses etc. Bias load rates are characterized by measured load curves. During tests, typically the seals are pressurized, and rotor is brought to full speed before any lateral seal-rotor excursion is applied. However, load history depends on the desired test conditions. Seal stiffness is measured as load per radial incursion. Experiments are conducted for both static and dynamic conditions. Three interference stages are considered which are 4 mils, 8 mils and 12 mils where 1 mil corresponds to 25.4 µm. The measured load values are average values for a specific time range i.e. 10 s after system stabilizes around the test condition or reaches steady state. The data obtained from two load cells are also averaged to reduce measurement errors. Two different brush seal types are used to compare the effect of the retaining/front plate length. Type 2 seal design is similar to that illustrated in Figure 1. Retaining/Front plate of Type 1 seal is extended to the same level as the backing plate. However, front plate that has a relief to allow bristles to move. The test seals are revealed in Figure 3.

2.1. Stiffness Measurement Under Static Conditions

In this set-up, two identical seals are mounted on a rotor with a face-to-face alignment. The desired differential pressure is applied by pressurizing the cavity between the seals. The simulated radial excursion is introduced by moving the housing laterally with respect to the fixed runner via a linear slide that carries the assembly. The load applied to the rotor by bristles due to the interference is measured through load cells mounted on both sides of the housing. The tailored design of the housing eliminates frictional effects. The original position of seal housing is concentric with the runner such that no net force is coming from bristle-runner interface interaction. The only force read from the sensors is coming from the hose and other assembly components due to the initial arrangement. The effects of these disturbances are calibrated and the data is corrected accordingly. When the eccentricity of the housing with respect to the rotor is increased, the force on the load cells also increases. The radial eccentricity ΔR is measured by a displacement sensor. The forces at different radial eccentricities are measured during the motion of the housing towards both (left and right) directions. The forces read for the same radial eccentricity in left and right interferences are averaged to minimize the effects of disturbances. The data obtained during the static experiments are shown in Figure 4. It should be noted that all presented data points represent an average for minimum of 4 measurements.
For the first type seal, as the applied differential pressure increases seal contact force also increases. The effect of radial excursion on seal force is almost linear for all cases. Behavior of type 2 seal is noticeably different from that of type 1. For type 2 seal, the reaction force markedly increases upon application of even a small differential pressure load. However, reaction forces for different pressure loads remain similar with changing differential pressure. This behavior can be attributed to bristles experiencing maximum differential pressure load across the seal radial height due to the fact that type 2 seal has no retaining/front plate. Even a small pressure load application may lock the bristles against across the backing plate. Whereas in type 1 seal has retaining plate that is extended up to the backing plate. It is expected that retaining plate may help reduce penetration of pressure load on the brush pack in the retaining plate region above the fence height. Therefore, type 1 seal does not suffer sudden stiffness increase upon pressure application. However, its reaction force increases with increasing pressure load. However, the forward bow of the brush in the retaining plate region (illustrated in Figure 5) may cause bristles to rub against the retaining plate on the upstream side as well as the backing plate on the downstream side, as pressure load increases.
When a radial interference loading and unloading sequence is applied, both seal types yield different behavior. Figure 6 presents the hysteresis behavior for both brush seals under static conditions. The seal force is measured by increasing eccentricity incrementally, and comparing to the values obtained during the return of the rotor to the concentric position. Unlike the test case presented in Figure 4, hysteresis tests are conducted under constant 1.5 bar pressure load while varying radial rotor interference. The area between two measurements is quite high for type 1 seal that indicates the sticking of the bristles to the backing plate (and possibly the retaining/front plate due to forward bow beyond the fence height region) under pressure application that does not allow them to return to the original position. On the other hand, the area is very small for type 2 seal, because bristles can deflect and recover their original shape.
2.2. Stiffness Measurement Under Dynamic Conditions

The same set-up is utilized for dynamic tests to facilitate a better comparison between the static and dynamic behavior of seals. The experimental procedure in dynamic analysis is almost identical to the static case except rotation of the runner is added. The tests are conducted for two speed levels i.e. 3000 rpm and 10000 rpm. Two different test speeds are used to evaluate effect of speed on brush seal stiffness, if any. Dynamic experiments start with the adjustment of lateral displacement of housing to imitate rotor excursion. Then a desired differential pressure is applied to the cavity between conjugate seals in the housing. The lateral position of housing is finely tuned when rotor reaches to the test speed. To avoid hysteresis effects, pressure is decreased to ambient at the beginning of each step, and increased to the desired pressure.
Figure 7. Seal force vs. radial displacement for various differential pressure at rotor speed of 3000 rpm for a) type 1 b) type 2

When the eccentricity increased the seal force almost linearly increases for both cases as observed in Figure 7. This behavior is somehow consistent with the result found by Aksit et al.[10]. In their finite element model, stiffness of bristles linearly increases until the excursion reaches 12 mils and then stabilizes when the eccentricity further increased. The pressure causes a dramatic growth for seal force as in the static case.

Figure 8. Seal force vs. radial displacement for various differential pressure at rotor speed of 10000 rpm

The trend in 10000 rpm tests displayed in Figure 8 is similar to the previous case but the force values are slightly lower. It may be said that there is some role of increasing swirl and the aerodynamic lift on the bristle tips as the surface velocity increases and the bristles are pushed away due to this small pressure. Hence, some reduction in the reaction force on the rotor surface is observed.
Effect of rotor/surface speed is better illustrated when static, 3000 rpm and 10000 rpm cases are compared for different radial interference values. The contact force applied by the bristles to the rotor surface dramatically decrease as the rotor speed is introduced. As indicated in Figure 9, there is almost no improvement when speed is increased more than 3 folds. This is an indication of the fact that dynamic effects and vibration environment shake bristles free to reduce seal stiffness. There is very little contribution of increased speed due to limited aerodynamic lift developed between the bristles and rotor surface.

The hysteresis behavior also changes under dynamic conditions as seen in Figure 10. The area shrinks considerably because rotor motion which contributes to bristle recovery by introducing some vibration to bristles. Once bristles are shaked free from frictional lock at the backing and retaining plates under dynamic conditions, type 1 seal performs better than type 2 seal.
2.3. Comparison to the Literature

Although there are a number of studies investigating the static behavior of brush seals, there is very limited study on dynamic characterization of brush seals. Long et al. [3] used a torque arm to measure the bristle-rotor contact forces. The torque arm is attached to a test piece, and the test piece is free to rotate as it is supported on a ball bearing. By using a PTFE pressure pad located on the upstream side of the test piece, friction and leakage inside this assembly is also reduced. The bristles of the brush seal touch the test piece, and they impose a load onto it. Due to this contact and resulting friction, some torque is generated when the test piece is turned. However, the contact force of the bristles on the test piece is not the only force measured. Friction in the bearings and mating surfaces also contribute to torque generation. The force contribution due to the friction in the bearings are calibrated out by measuring the torque required to turn the test piece when the brush seal is replaced by an annular calibration plate. However, to estimate the contact force of the bristles on the test piece, friction coefficient is required to be known. This can be estimated by carrying out tests without flow.

![Figure 11. Comparison to the literature](image)

The mean contact force applied to the test piece by bristles for several pressure differences are demonstrated in Figure 11. The simulated excursion is 0.2 mm or 8 mils. Data indicate very good agreement between the results of the present work and that of Long et al. [3]. Even though measurement techniques are quite different, similar outputs reveal the consistency of the presented work.

3. Conclusion

Brush seals provide a good alternative to conventional labyrinth seals with drastically improved sealing performance. They can also withstand large rotor excursions without permanent sealing loss or failure. However, due to rubbing at rotor surface, wear is a crucial problem that limits seal life. As life improvement possibilities through experimental material selection process have been already exploited, the key to better seal life and performance lies in management of the contact forces. The presented experimental work provides an insight for brush seal contact forces under combined pressure and speed conditions.

Two different seal types (type 1 with a front plate, and type 2 without a front plate) has been studied. The results indicate the following:

- Reaction force of the seal without a front plate increases markedly even with small pressure load introduction. Whereas the reaction of the seal with front plate increases with increasing friction.
- Once locked under pressure very little changes occur in the reaction of type 2 seal with changing pressure load. Whereas the change in reaction of the type 1 is more pronounced. This may be attributed to the possible additional frictional contact of bristles at the upstream side on front plate due to forward bending of brush under pressure load.
- Type 1 seal experiences more hysteresis under static condition while it performs better under dynamic conditions.
- Introduction of rotor speed drastically reduces seal-rotor reaction force. This may be attributed to vibration that shakes bristles free from backing and front plate frictional lock.
- Increasing speed and aerodynamic lift has very little effect on reaction forces.
- Overall, brush seal with a front plate performs better under dynamic conditions.

References