Evaluation of Pressure-Stiffness Coupling in Brush Seals

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Brush seals are comprised of fine diameter fibers densely packed between retaining and backing plates. To achieve seal compliance bristles are arranged to contact rotor with some lay angle. When axial pressure load is applied, bristles interlock and get stuck at the backing plate, and seal stiffness varies under operating conditions. Operating stiffness is critical to determine seal-rotor contact pressure and wear life. Typically, seal stiffness is measured by pressing a curved shoe to brush bore as reported in open literature. Due to the complex nature of pressure-stiffness bristle behavior, static and unpressurized measurements cannot represent actual working seal stiffness. This work presents a brush seal stiffness measurement system that is capable of measuring seal stiffness under working pressure and speed conditions. Rotor speed is achieved by an integrated spindle drive, while contact forces are measured via sensitive load cells. Rotor excursions are applied through lateral motions of the seal housing that is actuated by a motorized linear slide. Stiffness testing methodology and calibration procedure are discussed. Comparative experimental data are presented for both static pressurized and dynamic-pressurized stiffness tests.

Nomenclature

BH	=	bristle height
F	=	seal force applied to the rotor surface
FH	=	fence height
K_b	=	bristle stiffness
N	=	rotor speed
mil	=	1/1000 of an inch
P_H	=	upstream pressure
P_L	=	downstream pressure
P_H	=	upstream pressure
rpm	=	revolution per minute
ΔP	=	pressure difference
ΔR	=	radial excursion

I. Introduction

To meet the increasing demand for higher efficiency, conventional labyrinth seals are almost fully optimized. The brush seal has arised as an alternative seal technology which is taking the place of the present labyrinth seal design to achieve even higher performance levels. A brush seal is comprised of a set of fine diameter metallic bristles that are densely packed between retaining and backing plates. To prevent bristles from buckling, they touch the rotor with an angle in the direction of the shaft rotation. 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 operating life and efficiency of the seal.

The brush seal offers a dramatic improvement in sealing compared to the labyrinth seal. The compliance of the bristles enables this improvement to be conserved when there are differential movements between shaft and seal housing during engine transients. During engine operation rotor excursions can take place due to maneuver loads, thermal expansion and elastic deformation of the rotor. Frictional forces are induced in between the bristles and

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between the bristle pack and the backing plate by means of the axial pressure difference. Frictional forces lead to stiffer bristle behavior when they are pushed in radial direction during a rotor excursion. Bristle stiffening results in elevated contact loads at the rotor surface, which cause high wear rates. Pressure stiffness coupling in brush seals should be studied in detail for more accurate estimation of contact loads and frictional heat generation at the bristle tips.

Bristle stiffness is one of the major seal characteristics that defines average bristle tip pressure at the rotor surface contact. Stiffness can be defined as the change in bristle tip pressure 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 given by $K_b = F/\Delta R$ where F is the seal reaction force developed at the bristle tips by imposing the radial interference ΔR.

Bristle 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 experimental studies[3,4] which intend to determine seal contact forces, however the tests are highly complicated, and prone to measurement errors if not conducted carefully. Others[1,2] produced methods to characterize an overall seal stiffness. Hence, measurement of bristle stiffness may get complicated. In one of the reported measurement procedures Basu et al.[1] and Short et al.[2] tried measuring overall seal stiffness, and converting it to the bristle stiffness. They tested 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, the experimental data reported by Basu et al.[1] and Short et al.[2] did not include any information on the test samples. The details of seal-to-bristle stiffness conversion procedure were not presented. Therefore, their results could not be used for the verification of the experiments presented here.

Typical seal stiffness without pressure load may vary from 54.3 kPa/mm (0.2 psi/mil)[2] to 230.7 kPa/mm (0.85 psi/mil) according to the experimental data[5]. However, these stiffness values grow multifold when pressure applied because of the strong friction coupling and blow-down with radial leakage flow. Chupp et al.[6] 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[1,2] demonstrate that seal contact force increase depends on the pressure load increase.

In this work, a test system capable of measuring seal stiffness under pressure load and rotor movement will be introduced. The results reveal that seal stiffness is affected by the applied pressure and rotor motion. As noted by Modi[7], dynamic impacts can crucially affect the behavior of the bristles during rotation. This work offers further insight to stiffness characteristics of brush seals. The experimental results are also compared with those in literature.

II. Experimental Setup

Housing is the base part for the seal assembly. Test air is supplied through an air inlet hole to the cavity formed by the rotor and brush pack surfaces in between two test seals (Figure 1). Three probe holes with 3/8-24UNJF-3B are provided. Two of which are for pressure gauges, and the third one is for the thermocouple. ETM-375-7BAR-A type pressure transducers are used, which have 707.927mV/BAR sensitivity and -18°C to 100°C compensated temperature range. The reaction force applied by the rotor on the seals and the housing assembly is measured via sensitive load cells. The housing assembly is supported by two load cells on each side. The housing assembly can move laterally on a motorized linear slide. To introduce precise radial interference between brush seal and rotor surface, a high precision zero backlash motorized slide is adopted to bear the assembly. By a precision slide motor interference can be applied with accurate small increments. Since the housing is connected to this slide assembly only by two load cells total reaction force is measured by the load cells. Measured pressure and load values are transferred to the 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 leakage. 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 sealing applications. Use of through bolts provides easy assembly and disassembly. Removable seal rings allow testing of different brush seal geometries and facilitate testing of other rotary seals[8].

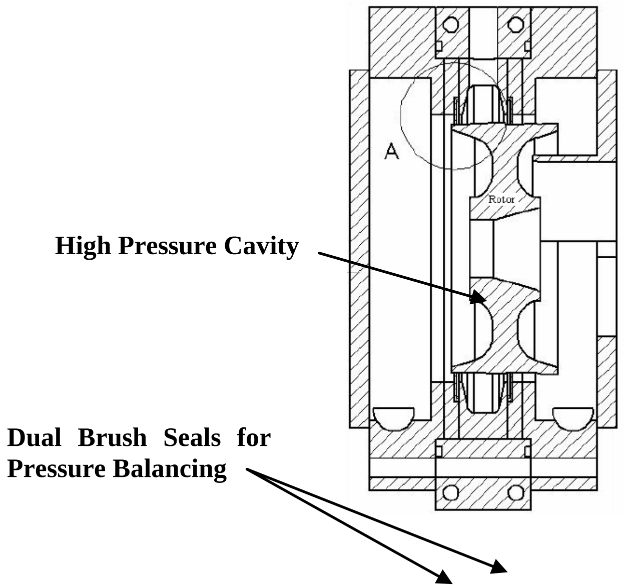
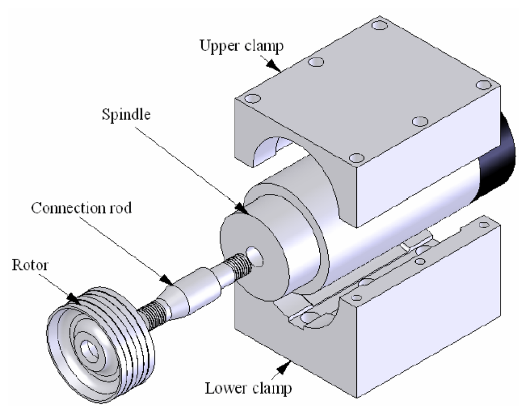


Figure 1. Cross-sectional view of the seal housing assembly

Shaft rotation is powered by a high frequency spindle. The spindle is able to reaching 45000 rpm as maximum rotational speed with 25 kW output rating. Power of the spindle is transmitted to the rotor by a connection rod, which is produced from the run out test rod supplied with the spindle.

Testing starts with a calibration procedure where housing is laterally pushed by a sensitive load cell representing rotor interference. Piezoelectric load cells carrying the housing assembly measures the reaction force which includes bias from connection hoses etc. Bias load rates are characterized by comparing actual load applied at different radial interference levels and the measured load curves from two load cells supporting seal housing. To test operating seal stiffness, seals are pressurized and rotor is brought to full speed before any lateral seal-rotor excursion is applied. Seal stiffness is measured as load per radial incursion. Experiments are conducted for both static and dynamic conditions. Pressure is introduced in the range of 1 to 3 bars for all cases. Three interference stages are considered which are 100 μm (4 mils), 200 μm (8 mils),and 300 μm (12 mils) where 1 mil corresponds to 25 μm. The measured load values are mean values for 10 seconds measurements. After correcting for the bias loads, the average of the data measured from two different load cells that support the seal housing is reported as the seal-rotor contact force for that interference.

All of the data is measured for a 130.81 mm (5.150 in) rotor diameter. The seal force data reflects the average force reading from two different load cells measured for two seals at a time. Standard density brush seals with 45 deg cant angle and 1.5 mm backing plate clearance is used in all tests. The reported seal contact forces are average force per seal.



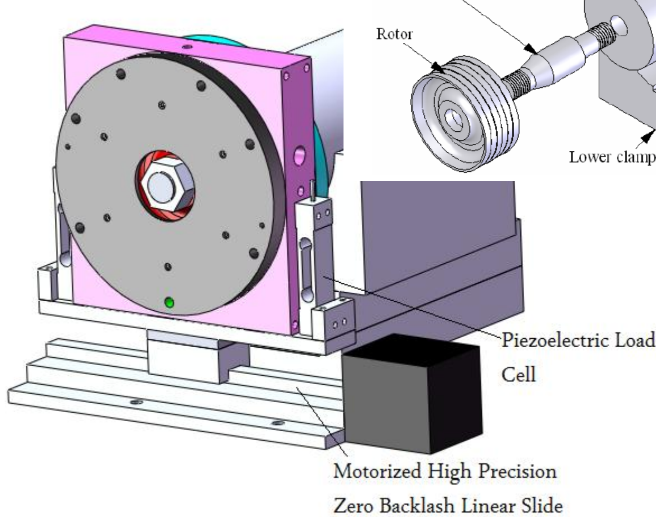


Figure 2. Housing and the spindle assembly

III. Experimental Procedure

In the following sections, the experimental arrangements to apply radial excursions and to measure the bristle stiffness under static and dynamic test conditions are described, and related data are presented in figures for several operation conditions.

A. Stiffness Measurements Under Static Conditions

The cross sectional view of the seal housing assembly is shown in Figure 2. In this set-up, two identical seals are mounted on a rotor with a face-to-face arrangement. First, a desired differential pressure is applied by pressurizing the cavity between the seals. Then, a 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 to both sides of the housing. A typical test is started with seal housing concentric with respect to the runner such that no net force is coming from bristle-runner 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 bias loads are calibrated for various radial excursion levels, and the data is corrected accordingly. As the housing is arranged eccentric with respect to the rotor, the force reading on one of the load cells increases while force on the other load cell decreases with the application of the radial seal interference. The radial eccentricity ΔR is measured by the encoder of the motorized slide as well as dail gauge attached to the housing wall. The forces at different radial eccentricity levels are measured during the horizontal motion of the housing. Two different force readings are recorded for the same radial eccentricity from the load cells supporting the housing at teo ends. Average of these readings are reported as the seal-rotor contact force thereby reducing the reading errors. The data obtained during the static experiments are shown in Figure 3.

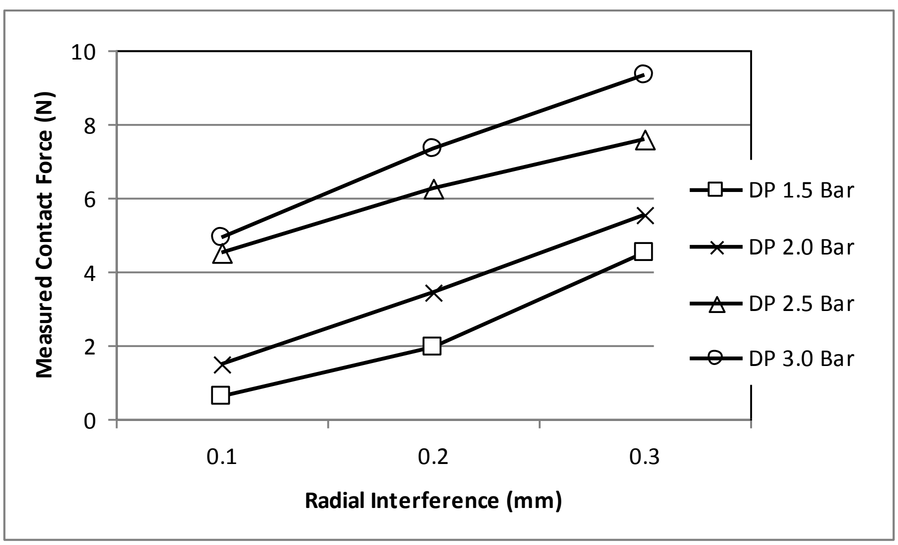


Figure 3. Seal force vs. radial interference for various differential pressure loading in static experiments

The results indicate that seal-rotor contact force increases almost linearly with increasing radial interference. The effect of radial excursion on seal force is almost linear for all cases. It is also observed that increasing differential pressure loading on the seal dramatically increases the seal contact force. Such behaiour is expected as brush seals are known to exhibit strong pressure-stiffness coupling behaviour. These measurements also is in line with the observations that seal stiffness increases multifolds with the upplication of pressure loading.

B. Stiffness Measurement Under Dynamic Conditions

Experimental procedure described in the static tests section is repeated for the dynamic tests with addition of the rotor speed introduction. The results from static and dynamic tests are later compared for evaluation. Dynamic tests are conducted for two different speeds, i.e. 3000 rpm and 10000 rpm. It is expected that through the use of two different rotor speed speeds, the impact of surface velocity on seal behaviour and contact stiffness will be better undestood.

The experiments start with the adjustment of lateral displacement of housing to define the concentric starting position. Then differential pressure load is applied before any eccentricity/radial interference is applied.Once a set of testing is completed under a specified pressure load for various interference levels, the seal is depressurized to ambient toavoid any possible hysteresis effects. The measured seal-rotor contact force data are presented in Figure 4 for 3000 rpm, and in Figure 5 for 10000 rpm rotor speeds.

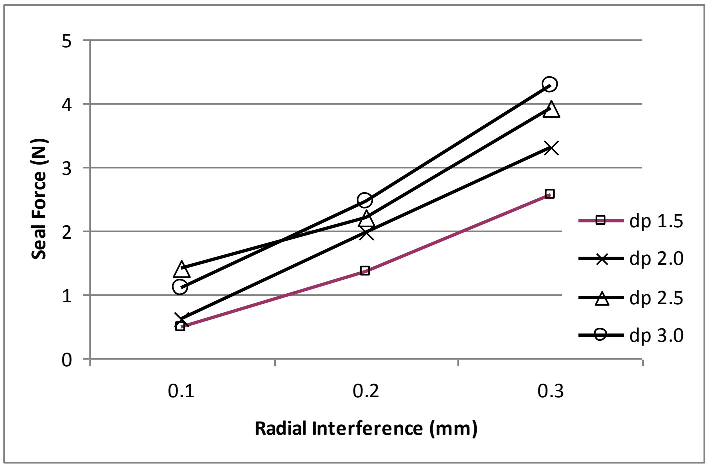


Figure 4. Seal force vs. radial interference for various differential pressure levels at rotor speed of 3000 rpm.

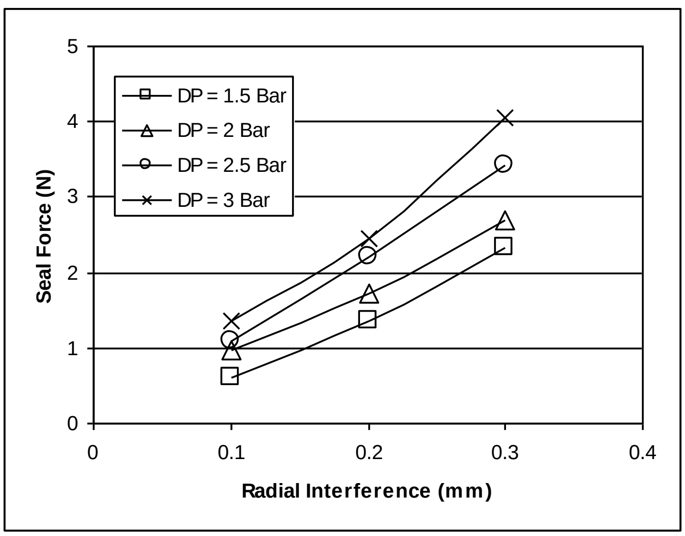


Figure 5. Seal force vs. radial interference for various differential pressure levels at rotor speed of 10000 rpm.

Similar to static test results seal contact force increases almost linearly when eccentricity/radial interference is increased. This behavior is somehow consistent with the analystical result reported by Aksit et al.[9]. In their finite element model, stiffness of bristles linearly increases until the excursion reaches 0.3 mm (12 mils). As in the static case, application of pressure load dramatically increases seal contact forces, thereby stiffness. As illustrated in Figure 6, when seal force measurements are compared for static and dynamic cases, it is observed that seal contact force drastically decreases with the application of rotor speed. However, once a rotor speed is applied, change in contact force with the increase in rotor speed remains very small.

The decrease in seal contact force with the application of the rotor speed can be attributed shaking loose of the brush pack that is compacted under axial pressure load by the disturbances introduced through vibrations and one per rev rotor motion. Such small vibrations may have reduce bristle interlocking, thereby reducing seal stiffness/contact force. Additional reduction of the seal contact force can be attributed to aerodynamic lift force generated between the inclined bristles and the high speed rotor. However, aerodynamic lift force contribution is expected to be very small –if not negligible- due to the fact that air viscousity and bearing area under each bristle are very small. However small, the minute decrease in seal contact force that is obsereved when rotor speed is increased from 3000 rpm to 10000 rpm can be attributed to the increase in the aerodynamic lift force when speed is increased.

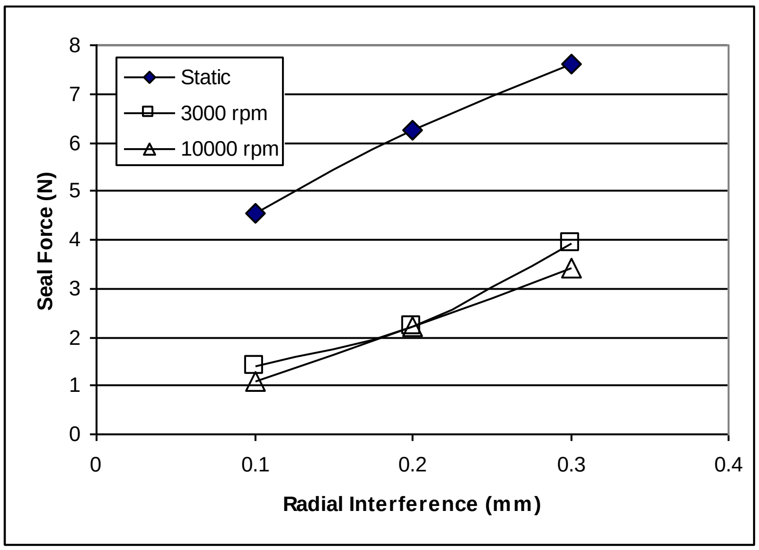


Figure 6. Comparison of seal contact forces for different speed conditions at ΔP = 2.5 bar

C. Seal Hysteresis Behaviour

Brush seals are known to exhibit load path dependent behaviour. In order to capture path dependent contact force behaviour two separate tests are conducted for both static and dynamic (3000 rpm) conditions. These hysteresis tests are condacted in such a way that first a prescribed test pressure is applied, then radial interference is introduced. The interference is gradually increased and decreased without removing the axial pressure load. The results are presented as seal hysteresis curves in Figure 7 for static case, and Figure 8 for 3000 rpm case.

The result for static case indicate that a sudden multifolds increase in contact force occurs with the application of pressure load. Then, seal contact force –although at much lower rate- linearly increases with the application of radial interference. When radial interference is reduced without removal of pressure, once again, there is a sudden drop in the contact force. This indicates that seal remains hung up under pressure load being tightly packed against the backing plate. These finding are in line with the similar trend predicted by Aksit et al.[9]

When same loading curve is followed under 3000 rpm, hystesis curve is changed. First of all, instead of a sudden jump, there is a gradual increase in contact force upon application of pressure load. The increase in the contact force with the increase in radial interference is more steep. However, maximum contact force under maximum interference remains lower for th dynamic case than the static case. As before, these is attributed to shaking loose of the brush pack through vibrations that reduce bristle interlocking, thereby decrease seal stiffness/contact force. Thisbehaviour is more evident on the return path. As interference is gradually removed under pressure, seal does not remain hung up, and contact force is also gradually reduced in dynamic case as opposed to the sudden drop observed in the static case that represents brush pack hang up at the backing plate.

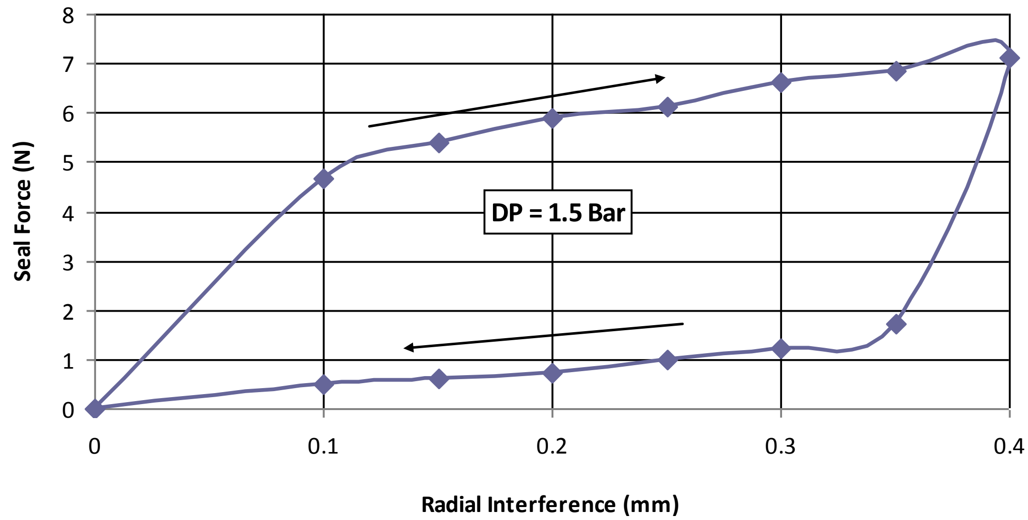


Figure 7. Seal force hysteresis curve in static experiment

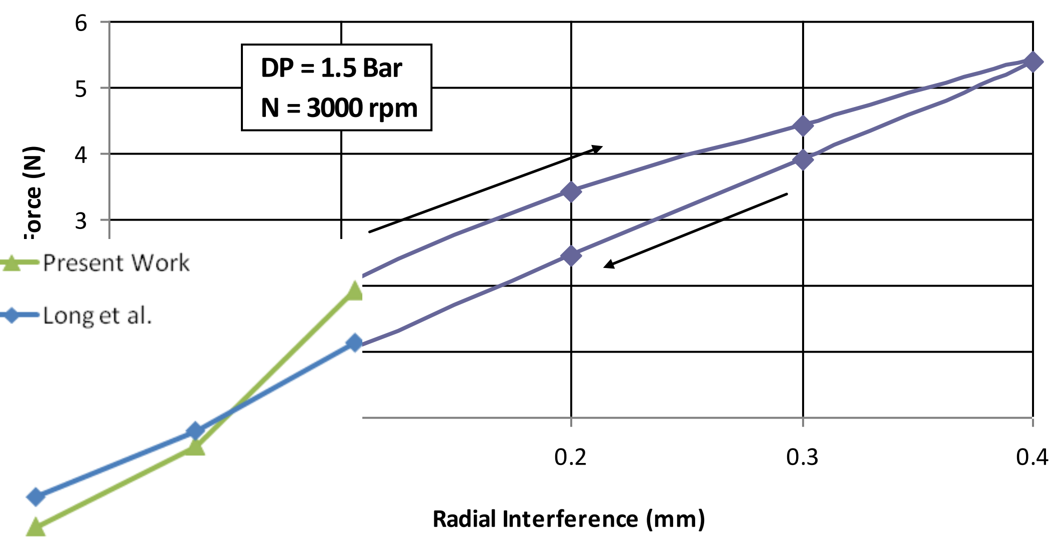
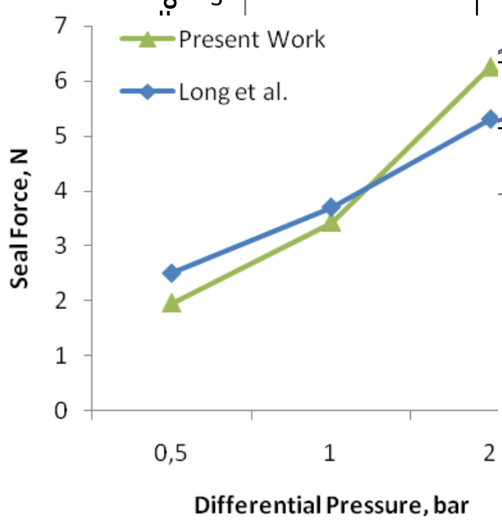


Figure 8. Seal force hysteresis curve for 3000 rpm dynamic condition.



D. Comparison to the Literature

Although there are works that study static and unpressurized stiffness of brush seals, Measuring dynamic stiffness of brush seals is rather complicated, and there is on ly one published work by Long et al.[3] They used a torque arm to measure the contact force of the bristles on the rotor. The torque arm is attached to the test piece that 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 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, torque is generated when the test piece is turned. However, the contact force of the bristles on the test piece is not the unique force and friction in the bearings and mating surfaces also contribute to torque generation. The force contribution due to the friction in the bearings are calibrated 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. They estimated those values through some calibration runs.

They reported test results with radial excursion levels to 0.2 mm or 8 mil. The results are compared in Figure 9. There is a very good agreement between the results of the present work and the one from Long et al.[3]. Eventhough measurement techniques are very different, similar outputs reveal the consistency of the presented work.

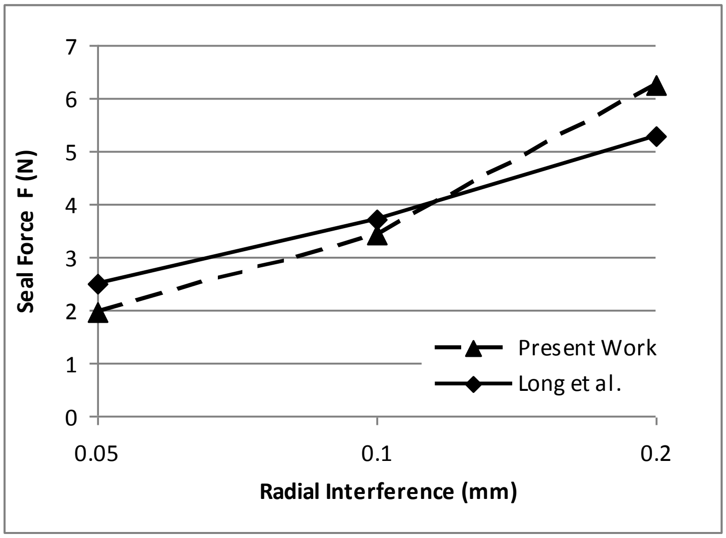


Figure 12. Comparison with Long et. al.[3] data.

IV. Conclusion

Brush seals provide a good alternative to conventional labyrinth seals with drastical improvements in sealing performance. They can also withstand large rotor excursions without permanent sealing loss or failure. Due to rubbing at the rotor surface, wear is a crucial problem that limits seal life, and reduces its effectiveness. Friction and contact loads are the main factors which determine seal wear rate. The key to better seal life and performance lies in management of the contact forces. Inherent flexibility of brush seals allows fibers to compact under pressure load. Due to the frictional interaction between the fibers and the backing plate as well as within the fibers themselves, brush seals are known to exhibit pressure stiffening and hysteresis behavior[7]. While hysteresis affects seal performance after a rotor excursion, pressure stiffening is critical in determining heat generation and seal wear during hard rubs. Although there are experimental studies which aim to determine seal contact forces, these are conducted by pressing a shoe over an unpressurized seal segment. As seal stiffens under pressure, such tests do not represent actual stiffness. This work presents some experimental data that reveals seal behaviour under pressure and speed conditions. Comparison of the tests results show good agreement with the limited data reported earlier by Aksit et al.[9] and Long et al.[3] Seal hysteresis tests indicate strong pressure stiffening behaviour under static condition. When radial interference is removed under pressure load, seal hang up at the backing plate is evident with sudden drop in the contact force. However, seal stiffness and amount of seal hang up are reduced when shaft rotation is introduced. This is attributed to shaking loose of the brush pack through rotor induced vibrations that reduce bristle interlocking, thereby decrease seal stiffness/contact force. Overall, proposed test system is proven to be affective in capturing and studying brush seal operating behaviour.

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