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RADIAL PRESSURE DISTRIBUTIONS IN AN AIR-RIDING FACE SEAL

Amir Ibrahim1*, Prof. David Gillespie2, Dr. John Garratt3

Abstract
Non-contacting face seals rely on high pressures induced in a thin air-film between stationary and rotating faces. They offer ultra-low leakage and very low wear compared to contacting seals in aircraft engines. Large axial and radial movements and high temperature gradients can cause excessive distortion of the sealing faces which may become amplified at large radii, high differential pressures and rotational speeds. Such distortions alter the geometry of the gap thereby affecting the seal’s performance. This paper presents an extensive investigation into the air-film behaviour of a face seal under convergent and divergent engine representative coning distortions = 0.5 – 2 degrees, gap = 50 – 300 μm, and operating pressure differentials =70 – 350 kPa. The investigation approach is both numerical and experimental. Experimental tests allowed the introduction of a known distortion onto the static face of the seal. Arrays of static pressure tappings in the primary sealing gap were used to measure the radial and circumferential variations. The experimental data are used to validate a 3D CFD model of the primary leakage path. The CFD model was generated using ANSYS ICEM and solved using ANSYS FLUENT. The models were run at the full range of operating pressures and geometries. Results show that converging coning provides the largest air-film pressures and hence the largest opening force while a diverging coning provided the least. At higher pressure ratios divergent gaps exhibited expanding supersonic flow but with unexpected levels of pressure recovery within the diverging duct. The pressure loss at the entrance to the gap was observed to be significant, particularly where entry gaps are larger. This effect was partially captured by CFD. The most significant discrepancies between CFD predictions and experiments were for the converging gap cases where the increased air-film pressure causes the disc to deform under pressure resulting in the CFD model over-predicting the pressure in the gap.

Keywords
Seals — Turbomachinery — CFD

INTRODUCTION
Shaft seals are found wherever rotating and stationary components are in close proximity and flow must be restricted in the gas turbine. There may be as many as 50 installed seals in a jet engine [1]. Two main categories of seals are typically used: air-to-oil seals and air-to-air seals. The former type are bearing sump seals that restrict leakage from a high pressure region into the bearing compartments. The compressor and turbine interstage seals are classified as air-to-air seals. They are the primary means of metering cooling airflow at a range of pressures to provide the necessary cooling air to the turbine blades and vanes and also preventing gas recirculation [2].

In the secondary air system, conventional face seals have been considered as potential replacements for labyrinth seals in an attempt to reduce leakage rates and reduce wear through non-contact operation [3]. Although they exhibit significantly lower leakage rates than labyrinth seals their operation is limited to lower pressure differentials (0.69 MPa) and temperatures (480 °C) [4]. In locations where high temperatures and high rubbing speeds are prevalent, the sealing faces tend to distort. Distortion of the sealing faces changes the pressure distribution in the sealing gap which in turn alters the force required to arrive at a balanced seal. To achieve non-contacting operation, Johnson and Ludwig [4] described the ‘Self-acting Face Seal’. The seal was to incorporate lift-generating features, influenced by the design of gas lubricated self-acting thrust bearings where the stationary part included Rayleigh step pads for lift generation. The self-acting face seal was tested by Ludwig and Johnson [5] in a rig which simulated a gas turbine engine bearing sump at pressure differentials ranging from 0.35 to 2.07 MPa, temperatures up to 650 °C. Endurance runs amounting to 320 hours under a sliding speed of 122 m/s were performed. Leakage rates were found to be a tenth of that of a labyrinth seal. Lynwander [6,7], ran a series of tests on a self-acting seal with circumferential Rayleigh steps including a 500-hour endurance test and the outcome agreed well with what Ludwig presented before, minimal wear was incurred by the sealing faces implying that lift off had been successfully achieved and maintained.

Later developments were presented by Munson et.al. [2], where an attempt was made to modify an existing film-riding seal used for pipeline compressors to one that would operate successfully in gas turbine environment. Twelve spiral grooves were used for hydrodynamic lift generation. Initial runs reported that the seal failed due to mechanical distortion caused by the divergent coning of the mating ring. Munson [8] also considered a design for a self-acting face seal for use at the compressor discharge location in a gas turbine engine. The analysis considered radial coning deformations and misalignment. However, only converging coning was considered as the author ignored a diverging coning possibility due to its undesirable...
effects. The prototype seal underwent testing by Munson [9], at pressure differentials from 3.45 to 41 bar, temperatures from 316°C to 649°C and rotational speeds up to 25,000 RPM. It was reported that a radial divergent coning has decreased the opening force compared to what had been predicted in a previous analysis that did not account for divergent coning.

Although the concept of the ‘film riding face seal’ has been proved to be successful for industrial compressors and in the pipeline industry, it has not found the same success in aerospace gas turbine engines. Large thermal gradients and shaft movements, particularly during engine transients, in the aerospace engine increase the severity of sealing face distortions. As a result, keeping the surface flatness within confined limits to maintain non-contacting operation cannot be guaranteed. To resolve this issue, research was conducted where the rigid front face was replaced by compliant foils. The purpose of the compliant face was to enhance the seal’s ability to track an out-of-flat runner. Munson et al [10] reports the development and testing of a circumferential foil seal with the intention of using it in turbine rim locations. The leakage rate was reported to be slightly higher than a self-acting seal but was considerably lower than of a brush seal. However, the seal concept was abandoned as it eventually failed due to issues related to curvature mismatch between the segments and the runner at high operating temperatures. Salehi et al [11] tested the performance of a compliant foil seal experimentally in a small gas turbine simulator at high temperature (up to 600°C) and rotational speeds up to 56 kRPM. The leakage rate of the foil seal was reported to be four time lower than the brush seal. Heshmat et al [12] tested a compliant foil seal for application in a high-speed hydrogen compressor. Static and dynamic tests were performed at elevated temperatures (260 °C) and rotational speeds up to 60 kRPM. The study focused on leakage rates which were reported to be very low. While for seal installations it is desirable to run close to engine temperature, for the current series of tests, the purpose was to investigate the pressure and inertia induced forces on faces of known geometry and so tests are run at ambient conditions to avoid thermal distortion.

Munson et al [10] considered implementing a foil face seal by combining the thrust-bearing part found on the stationary face of a compliant foil gas thrust bearing with a conventional face seal concept instead of the circumferential design. The process focused on optimizing the compliant foil thrust bearing part having a high film stiffness, load capacity and maximum flexibility of the foil elements. The primary seal ring design consisted of segmented top foil thrust bearings supported by corrugated bump foils. Coning distortions and circumferential-out-of-flatness (waviness) of the rotor were assessed. The range of distortions covered was reported to be over two orders of magnitude larger than a conventional face seal would be capable of handling. Munson et al [13] further tested another proof of concept foil face seal. For the circumferential out-of-flatness, three and five waves were machined on the surface on the runner with an amplitude of 76 microns. It was reported that introducing waviness decreased the load capacity of the seal.

The research reported in the paper is part of a wider programme with the University of Nottingham to develop an air-riding flexible face seal. The seal has many of the elements of a conventional face seal as shown in Figure 1. The seal is designed to track large axial movements of the runner using a hydrostatically balanced sprung carrier, with an additional flexible front face, of lower stiffness to maintain non-contacting, low leakage, conditions for an out-of-flat runner and maintain face separation. This paper reports a series of experiments conducted to generate pressure distributions in the primary sealing gap of the seal, with and without lift enhancing features and compares the results to a 3D CFD model of the primary leakage path for different coning configurations and operating conditions. The work at Nottingham has identified that the front face may be constructed using elements considerably more robust than the thin foils typical of those used in a thrust bearing. The route to understanding the seal operation is threefold. First to understand real pressure distributions induced by the faces under static and rotating conditions in close proximity to the runner, then to verify that high axial movements can be accommodated and finally to understand the properties of the flexible element needed to follow high frequency disturbances. Figure 3 represents the geometry tested in the current study. It uses a solid front face to isolate the flow behaviour in radially coning gaps which are of primary interest.

![Figure 1 Air-Riding Flexible Face seal Concept.](image)

### 1. EXPERIMENTAL SETUP

Figure 2 shows a schematic of the experimental facility outlining its main components. The rig consists of a baseplate on which is installed a rotating disc driven by an electric motor through a surface acoustic wave torque meter. The disc can be rotated at speeds of up to 15,000 rpm. The remaining key component of the test rig is the seal housing, which is connected to the baseplate via a load cell and electrically operated actuator. This allows displacement of the housing relative to the rotor of up to 4.1 mm to be applied in increments of 10 μm. The load cell capacity is 100 kN for the tests conducted. The facility is installed in a pressure vessel allowing upstream pressurisation of the test rig. The seal housing is surrounded by high pressure air, with exhaust to atmospheric conditions through a secondary seal formed by a plane close fitting ring. While the load cell provides an overall measure of the load on the system, it is subject to pressure loadings unique to the test rig, rather than the sealing faces, and thus, while useful in determining differences between geometric cases, the determination of surface forces by integration of the surface pressures is felt to be the most representative of those required for design of a real seal. At high loads and very small closures the load cell was replaced by a solid block to increase the stiffness of the entire system.

Figure 3 shows a cross-sectional view of the sealing face installed within the seal housing. The primary leakage path of
the flow is indicated with arrows. Note that the primary flow path is now left to right.

![Figure 2 Non-Contacting Seals Rig Schematic.](image)

Figure 2 Non-Contacting Seals Rig Schematic.

The sealing face geometry is machined onto a removable part which is an annulus made from PVC. This allows different seal face gap profiles to be tested. The surface finish is nominally smooth in all cases with a measured Ra of 1.4663 μm using an Alicona optical microscope. Thirteen radial pressure tappings are placed at each of ten circumferential locations (0°, 5°, 10°, 15°, 40°, 49°, 68°, 90°, 180°, and 270°) around the annulus. In total, 130 pressure tappings provide sufficient resolution to assess the pressure field within the sealing gap, and assess circumferential variation in the results.

Once circumferential consistency had been assessed, four circumferential locations were used to populate the data sets reported here.

Both the upstream housing and downstream housing are made of mild steel. A set of radial pressure tappings on the downstream housing allows the exit pressure to be measured. Figure 4 shows a labelled photograph of the test rig, outside the pressure vessel.

![Figure 4 Non-contacting Seals Rig.](image)

1.1 Experimental Procedure

Regarding operation, the rig is first inserted into the pressure vessel and the cover plate fitted. The pressure is then increased to the required level monitored by a tapping placed in the upstream cavity. To set the running clearance, the position of contact between upstream face and the disc is first determined at the operating pressure. This is accurately determined by bringing the rotor and housing into contact at a low operating speed and using the change in torque signal to determine where an initial touch down occurs. This eliminates any error caused by the flexibility of the load cell and the pressurisation of the rig. The housing is then displaced to achieve the required minimum operating gap. Two dial gauges are used to read the axial displacement of the housing relative to the disc. Once the required inlet pressure and gap height are reached, the DAQ system is armed to capture subsequent pressure distributions.

![Figure 3 Seal Leakage Path.](image)

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![Figure 4 Non-contacting Seals Rig.](image)

1.2 Uncertainty Analysis

To assess the level of confidence in the measured data, an analysis was performed on the equipment being usedestimated 95% confidence intervals for the measured results. The procedures used to perform the uncertainty analysis are outlined in Moffat [14], Abernathy et al. [15], Rolls-Royce guidelines [16] and Performance Test Codes [17]. Table 1 shows the systematic and random errors associated with the transducers used on the rig.

<table>
<thead>
<tr>
<th>Table 1 Test rig transducer uncertainties.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measurement</td>
</tr>
<tr>
<td>Systematic Uncertainty</td>
</tr>
<tr>
<td>--------------------------------------------</td>
</tr>
<tr>
<td>Pressure Transducers (70mbar)</td>
</tr>
<tr>
<td>Pressure Transducers (350mbar)</td>
</tr>
<tr>
<td>Pressure Transducers (1 bar)</td>
</tr>
<tr>
<td>Pressure Transducers (5bar)</td>
</tr>
<tr>
<td>Displacement</td>
</tr>
<tr>
<td>Temperature</td>
</tr>
</tbody>
</table>

2. EXPERIMENTAL RESULTS

2.1 Parallel Gaps

The first set of experiments were performed on nominally parallel sealing faces. The parallel gaps are used as a benchmark to which other coning configurations will be compared. Care was taken to account for bending of the disc under pressure loading, as described below. Table 2 shows the range of the upstream $h_u$ and downstream gap height $h_d$. Figure 5 shows a schematic outlining the notation used. Note that, the upstream gap height is set to be the minimum gap height such that $h_{min} = h_u$. This is at the lowest radial position on the disc as the disc undergoes slight bending under pressure. Table 2 lists the range of minimum gaps and downstream gaps tested. In all subsequent datasets, the data are plotted against a normalised radial
displacement. This is represented by the red line on Fig. 5. The tappings are aligned such that the first tapping is coincident with the inner radius of the upstream housing and the 13th tapping is aligned with the rotor outer diameter. The overall extent is represented by \( |R^*| \) and \( r \) is the local radial distance from the housing inner radius.

**Figure 5** Geometry notation for parallel gaps.

**Table 2** Axial upstream and downstream gap heights for parallel gaps.

<table>
<thead>
<tr>
<th>Minimum Gap (( h_{\text{min}} = h_u ))</th>
<th>Downstream Gap ( h_d ) (( \mu \text{m} ))</th>
</tr>
</thead>
<tbody>
<tr>
<td>50</td>
<td>4050</td>
</tr>
<tr>
<td>100</td>
<td>4000</td>
</tr>
<tr>
<td>200</td>
<td>3900</td>
</tr>
<tr>
<td>300</td>
<td>3800</td>
</tr>
</tbody>
</table>

Figures 6 and 7 show the average radial pressure (gauge) distributions for four different minimum gap heights (50 \( \mu \text{m} \), 100 \( \mu \text{m} \), 200 \( \mu \text{m} \) and 300 \( \mu \text{m} \)) at two different inlet pressures (1.03 barg and 3.45 barg). In each case the exit pressure from the rig was atmospheric pressure. The pressure was averaged at four different locations around the circumference, and the data are reported as normalised pressure (gauge pressure / gauge inlet pressure). The maximum circumferential out-of-flatness (peak to peak) on the disk was in the order of 25 \( \mu \text{m} \). Note that, most test cases shown in the current study were performed statically. Rotating the disc at the slow speed available during this stage of the test campaign and repeating tests showed no systematic change in the pressure distribution. As there are no changes in gap height in the circumferential direction, rotating the disc will have minimal effect on the air-film pressure distribution. All tests were run at ambient temperature conditions. The effect of higher rotational speed is discussed further in the CFD section of the paper.

The pressure distributions show that at smaller gaps a higher film pressure is maintained at a given inlet pressure. A pressure drop is evident at the entrance to the primary sealing gap. The entry loss is driven by inertial forces as the flow accelerates around the sharp corner at the inlet, with possible flow separation. As the mass flow increases non-linearly with increasing gap height the inertia-driven entry loss increases. With higher gaps it is noted that the exit pressure rises. This occurs as downstream of the seal there are inertial losses prior to the flow exiting to atmosphere. Notably, at 1.03 barg, the pressure distribution appears nearly linear with increasing radius, whereas for compressible Reynolds flow a parabolic profile might be expected. Moving radially outwards causes a more rapid decrease in pressure compared to a linear case because of simple continuity requirements. At higher inlet pressure the expected characteristic is restored, particularly in the exit region of the seal.

A cross-over was observed between the pressure distribution curves for the 50 \( \mu \text{m} \) and 100 \( \mu \text{m} \) gap heights at 1.03 barg, Figure 6. This may be due to the very slight bending under pressure loading of the disc which is more significant at smaller gap size and becomes a dominant feature in parallel gaps. This characteristic disappears at larger gap heights as the effect of surface flatness becomes less prominent compared to the magnitude of the gap height. For each of the stationary cases shown, the corresponding rotating case is represented by a dashed line. Little difference is observed between stationary and rotating cases due to the absence of a significant change in gap height in the circumferential direction apart from the inherent out-of-flatness of the disc. At 3.45 barg, Figure 7, the pressure drop in the exit passage is significant causing the exit pressure to rise and a cross-over of pressure distributions at \( r/|R^*| \sim 0.5 - 0.6 \). This is an artefact of the experimental test rig, and for validation purposes the pressure drop from inlet conditions to the exit of the face seal should be considered as the driving pressure in the system. The pressure normalised on that basis, \( p_{\text{in}} - p_f \) is shown in Figure 8. Here the normalisation is by the local driving pressure difference across the seal, however the inlet pressure is the same in all cases. The change in gradient in the pressure drop reflects the increased load capacity at low gaps caused by the lower leakage mass flow rate.
2.2 Converging Gaps

The second set of tests considered a converging gap (a converging path from gap inlet to gap outlet). The coning angle is denoted by \( \alpha \) and takes a negative sign for converging cases, (refer to Figure 9 for a schematic of a converging gap, note that the downstream gap geometry is unchanged from the parallel cases). The converging angle was machined on the static sealing face (removable face) by gradually removing material off the front face to give the desired angle. The same arrangement of pressure tappings was used to read the static pressures in the gap. Three converging angles were considered, \( \alpha = -0.5^\circ, -1.0^\circ \) and \(-1.5^\circ\). Table 3 shows the range of coning angles, their corresponding coning heights and the range of minimum gaps considered. Note that, the total upstream gap will be the sum of the minimum gap height \( h_{\text{min}} \) and the coning height \( h_{\text{coning}} \). The minimum gap will be at the outer radius of the sealing faces while the total gap will be at the inner radius. The total axial space remained at 4.1mm. The touch-down location was determined as previously, however, for the converging gap case, touch down occurs first at the outer radius of the sealing face (the closest point to the rotating disc).

![Figure 9 Geometry notation for converging gaps, upstream gap.](image)

### Table 3 Range of minimum gap heights, converging coning angles and coning heights.

<table>
<thead>
<tr>
<th>Coning angle (deg.)</th>
<th>Coning Height ( h_{\text{min}} ) (( \mu )m)</th>
<th>Minimum Gap ( h_{\text{min}} ) (( \mu )m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>-0.5</td>
<td>-135</td>
<td>50,100, 200, 300</td>
</tr>
<tr>
<td>-1</td>
<td>-270</td>
<td>50,100, 200, 300</td>
</tr>
<tr>
<td>-1.5</td>
<td>-405</td>
<td>50,100, 200, 300</td>
</tr>
</tbody>
</table>

Figures 10 to 13 show the circumferentially-averaged radial pressure distributions for all three converging angles considered. Two minimum gap heights, 50\( \mu \)m and 100\( \mu \)m, are considered for a given converging angle. The blue curve denoted by \( \alpha = 0^\circ \) represents the corresponding parallel case for a given minimum gap height. In addition to experimental data, two dashed curves show the prediction obtained from solving a 1D compressible form of the Reynolds equation (Eq.1) for the parallel case and the largest converging angle case, \( \alpha = -1.5^\circ \). The solution takes into account the entry loss by introducing a loss factor that is iteratively applied to the initially prescribed inlet pressure until the best match between both data sets is found. This was necessary as the Reynolds equation does not account for inertial losses which become quite significant at geometric discontinuities such as the sharp corner leading into the gap. Note that not all of this loss is caused by flow irreversibilities and even the isentropic acceleration of the flow into the entrance of the seal would cause a significant drop in pressure. The numerical simulations are not adjusted in this manner.

\[
\frac{\partial}{\partial x} \left( h^3 \frac{\partial p}{\partial x} \right) = 0
\]  

(1)

A general trend is immediately evident from the range of pressure distributions shown below whereby a converging gap consistently maintained a higher film pressure for a given minimum gap and inlet pressure compared to a parallel gap. Moreover, increasing the converging angle resulted in an increase in film pressure as the air-film gets squeezed into the converging gap.

As with the parallel gaps, an inlet pressure loss was evident. Interestingly, the entry loss appeared to be more a function of the minimum gap rather than the converging angle as minor changes were observed in the magnitude of the entry loss for a range of converging angles at a given minimum gap. The pressure distributions for both minimum gap heights at both inlet pressures (0.69 barg and 2.07 barg), figures 10 and 11, show that the entry loss for converging cases was consistently higher than the parallel configuration regardless of the gap size.
This effect is consistent with a near constant loss coefficient at the inlet of to the seal. This can be seen by reference to Figure 14 which shows the mass flow rate through the converging gaps normalised by the minimum mass flow rate for all tests (50 μm, parallel gap, P_{in} = 0.69 barg). Here, at a given inlet pressure the mass flow rate is seen to rise approximately with α^{0.5}, whereas the inlet gap increases linearly with α. Thus the dynamic head at inlet is approximately constant.

Also notable are the gap exit pressures, which are higher than the parallel cases and increased steadily with the converging angle. This is a direct result of the increased mass flow rate for all converging gaps compared to the parallel cases.

Additionally, immediately downstream of the inlet a region of pressure recovery exists as the flow which separates around the entrance corner re-attaches. The recovery region was more pronounced in cases featuring larger angles (α = -1° and -1.5°) and a minimum gap height of 100 μm particularly at higher inlet pressures, Figures 11 and 13.

In general, the behaviour seen from having a converging coning angle in the primary sealing gap was favourable as it remarkably increased the air-film pressure. The 1D Reynolds flow predictions showed reasonable agreement with experiments. Although the entry loss does not perfectly match with experiments, the applied entry loss factor aims at minimising the difference in the overall lift compared to a case where no entry losses are accounted for. This suggests that the flow remains laminar in the converging gaps – possibly because of the strong flow acceleration in this region and this is confirmed by the CFD results below.
2.3 Diverging Gaps

The final set of tests were conducted with a diverging angle machined on the face of the removable part. The diverging coning angle is also denoted by $\alpha$ but given a positive sign. Figure 15 shows a schematic outlining the geometry and notation for a diverging gap. The definition of the coning angle and height remain the same as the converging case. In the case of a diverging gap the minimum gap will be at the inner radius of seal face while the total gap height ($h_{coning} + h_{min}$) will be at the outer radius. Table 4 outlines the range of coning heights, angles and minimum gaps considered in this study. Two diverging angles were considered $\alpha = +1.0^\circ$ and $\alpha = +2.0^\circ$. The experimental procedure remained the same as before but contrary to the converging case, the touch-down occurred at the inlet to the gap where the primary face was in closest proximity to the rotor. Figures 16 to 17 show the circumferentially averaged radial pressure distributions for both diverging angles considered at two minimum gap heights (50 $\mu$m and 100 $\mu$m) and two inlet pressures (0.69 barg and 3.45 barg). The corresponding parallel case with the same minimum gap height is also plotted in blue and denoted by $\alpha = 0^\circ$.

Upon inspection of the radial pressure profiles in Figures 16 to 17, it is evident that a diverging gap causes the air-film to behave very differently compared to both previous configurations. The flow loses most of its pressure coming into the gap and pressure continues dropping below the exit pressure before suddenly rising again prior to exiting the gap. One might then expect that the normalised pressure is fixed and independent of the divergent angle at the geometric throat of the system where sonic conditions should be expected. However, as also observed by Al-Bender and van Brussel [18] as the flow accelerates into the entrance, should the flow become choked it will do so some distance downstream of the entrance, meaning that the pressure ratio at the inlet need not be 0.52. In fact provided there is sufficient pressure drop, at small angles the flow will continue to accelerate until a pressure rise is required to satisfy the exit pressure conditions. In this highly viscous flow regime, this occurs through a series of weak shock waves leading to gradual pressure recovery in the channel.

The only case where no abrupt pressure recovery was present was for the $h_{min} = 50$ $\mu$m gap at $\alpha = 2.0^\circ$ and $P_{in}=0.69$ barg, Figure 16, where the flow loses pressure at the inlet before immediately dropping to atmospheric levels. The smaller angle ($\alpha = +1.0^\circ$) for the same configuration showed a pressure drop followed by an abrupt recovery to atmospheric levels, similar to all other cases. The inlet ‘loss’ which is associated with the precise flow acceleration and size of any separation at the entrance of the channel appears lower for the larger diverging angle for most cases and increased with larger minimum gap heights.

Furthermore, the pressure drop in cases with the higher diverging angle was more severe compared to that of the smaller angle for a given minimum gap and operating pressure. However, the radial location of the abrupt pressure recovery showed little variation between both coning angles tested.
Figure 17 Experiments, Diverging coning, Radial pressure distribution, \( h_{\text{min}} = 50 \) and \( 100 \mu \text{m} \), \( P_{\text{in}} = 3.45 \text{ barg} \).

Figures 18 to 19 illustrate the effect of changing the inlet pressure for a given minimum gap and coning angle. The main observation was that increasing the inlet pressure and the minimum gap extended the low pressure region further into the gap and moved the location of abrupt pressure recovery further downstream.

Contrary to the previous two configurations where increasing the inlet pressure resulted in a desirable higher film pressure, it has proven to be detrimental to the film pressure where a diverging gap was present. Moreover, the drop in air-film pressure was further worsened at larger gap heights and coning angles.

The abrupt pressure recovery characteristic is believed to be due to choked flow conditions at the gap inlet, i.e. where the minimum gap exists (throat). At the inlet pressures considered, the flow reached supersonic speeds within the gap. Hence, as the flow chokes at the inlet a region of supersonic flow will exist where the flow accelerates as the area increases further in the radial direction. The supersonic region is eventually terminated by a series of weak shocks that causes progressive deceleration of the flow towards subsonic conditions at the gap exit, marked by the sudden pressure rise. As the inlet pressure increases, the maximum Mach number also increases as the supersonic region extends further in the radial direction. This explains the downstream-shifting of the recovery region at higher pressures. A more detailed insight is gained from the 3D CFD results in the following section.

Figure 18 Experiments, Diverging coning, Radial pressure distribution, \( h_{\text{min}} = 50 \mu \text{m} \), \( \alpha = +1.0^\circ \) and \( \alpha = +2.0^\circ \).

3. 3D CFD MODEL VALIDATION

3.1 Geometry and Mesh

A 3D CFD model of a 36-degree sector of the primary leakage path is shown in Figure 20. In addition to cyclic boundary conditions, the pressure inlet and outlet in the domain are also highlighted in Figure 20. A multi-block structured mesh was made using ANSYS ICEM, the total number of elements was 1.4 million.

3.2 Turbulence Modelling

The realizable \( k-\varepsilon \) model was applied in the turbulent zones. The initial values prescribed for the solution were based on the default settings in the solver, ANSYS Fluent. Since the seal gap is very small, the \( y^+ \) values are inherently small and a \( y^+ \) value of less than 1 was maintained within the seal gap. Enhanced wall treatment was used to allow for the laminar sub-layer to be adequately resolved.

3.3 Material Properties

The flow was treated as an ideal gas (air) making it necessary to solve the energy equation. The viscosity was set to be calculated using Sutherland’s Law. Furthermore, thermal conductivity, specific heat and molecular weight values were set to be constant.

3.4 Boundary Conditions

The inlet total and outlet static pressures were used as the flow boundary conditions, prescribed relative to atmospheric reference values. Only the inlet total pressure was varied, while the outlet pressure static was atmospheric pressure for all cases.
(in line with experimental conditions). The direction of the bulk flow was assigned to be along the axial direction. The walls were set to be adiabatic walls and stationary, unless otherwise stated. The rotational/cyclic periodic boundary conditions set the periodicity of the model to be about the $x$-axis (axial direction).

3.5 CFD Results & Discussion

3.5.1 Parallel Gaps

Figures 21 and 22 show the pressure distributions for 50 and 100 $\mu$m gaps at two different inlet pressures (1.03 barg and 3.45 barg). Results show that the CFD overpredicts the pressure within the primary gap for the 50 $\mu$m gap case. However, the entry loss is captured accurately by the CFD model as it is the exit gap pressure for the lower inlet pressure (1.03 barg). For the higher inlet pressure (3.45 barg), Figure 22, the CFD underestimates the entry loss and maintains a higher film pressure compared to the experimental case. The discrepancy for the higher pressure case is believed to be mainly attributed to the coning of the disc under the pressure applied from the air-film, particularly at small gaps where the air-film pressure is highest. Figure 21 also shows the effect of varying the rotational speed and the surface roughness (SR) for the 50 $\mu$m gap case. The alternative rotational speed essentially lie on top of each other. Minor differences were observed between the stationary CFD pressure distribution and both rotating cases (500 RPM and 20 kRPM), mainly due to the absence of a change in gap height circumferentially. Similarly, varying the surface roughness height did not improve the CFD predictions. This is perhaps unsurprising. Whereas in a flow system the effect of pumping would be to increase the pressure moving radially outwards, here the pressure at the exit is prescribed in the model and so the shape of the distribution will be affected by the pumping of the flow, but the overall pressure drop is not.

The 100 $\mu$m minimum gap cases, show strong agreement between CFD predictions and experimental data at both operating pressures (1.03 barg and 3.45 barg), apart from a slight variation in the magnitude of the entry loss. Note that, for the 50 and 100 $\mu$m gaps, the fluid block representing the flow in the primary gap was assigned as a laminar zone. The rest of the blocks were defined as turbulent zones and the realizable $k$-$\epsilon$ turbulence model was employed.

Figure 23 shows the resulting coning angle of the disc under a range of operating pressures at a range of initial gaps. An FEA model was set up where the pressure load from the air-film at different gap heights and inlet pressures was applied over the range $r = 0 - |R^*|$. The remainder of the disc was under the load from the inlet pressure. Whereas the loading is high and at small gaps may cause displacement of the outer radius of the disc by as much as 50% of the gap height, because the touching clearance is actively controlled, the nominal seal clearance can be restored. There is, however, always a slight divergence in the range of 0.005 – 0.029 degrees.

3.5.2 Converging Gaps

This section compares the radial pressure distributions from the 3D CFD model against those obtained experimentally for the converging configuration. Each plot in Figure 24 and Figure 25 shows the pressure profile at a given inlet pressure (0.69 barg and 3.45 barg) for two minimum gap heights (50 $\mu$m and 100 $\mu$m).
µm) at two converging coning angles (α = -0.5° and -1.0°). Refer to Table 3 for corresponding coning heights.

Considering the full range of data shown, it is evident that the film pressure predicted by CFD is significantly higher than that obtained experimentally. However, the trend from both data sets matches fairly well. Unlike the inlet pressure loss shown experimentally, which was more sensitive to the minimum gap height rather than the coning angle, CFD predictions suggest that the larger coning angle (α = -1.0°) resulted in a lower entry loss which increased with increasing gap heights.

The air-film pressure distribution predicted by the CFD for the smaller gap height (50 µm), illustrated in Figures 24 and 25, shows that the air-film maintains most of the upstream pressure in the gap before suddenly dropping in close proximity to the gap exit. The effect is clearly illustrated at higher inlet pressures. Moreover, the discrepancy between both data sets significantly increased at higher inlet pressures, Figure 25. Although the trend from the experimental curve is similar, the air-film pressure was maintained over a smaller region and a considerable drop was already conceived before exiting the gap.

Furthermore, the smaller angle case (α = -0.5°) from CFD showed very little signs of pressure recovery downstream of the entry loss location which agrees with what has been shown experimentally. On the other hand, the larger angle showed a recovery region in experiments but not in the CFD. Although the magnitude of entry loss was markedly different between CFD and experiments they showed a similar trend whereby a very subtle difference in entrance loss was seen in for both angles at the smaller gap height (50 µm). Similar to experiments, increasing the coning angle increased the air-film pressure.

For the larger gap height case (100 µm), the agreement between CFD predictions and experiments showed a slight improvement. Nevertheless, CFD predictions were still larger in magnitude compared to experiments and the discrepancy continued increasing at higher inlet pressures. Additionally, the inlet pressure loss was higher than smaller gap cases. However, and contrary to what experiments suggest, CFD predicted a lower entry loss in cases featuring the larger angle. The trend of pressure recovery downstream of entrance loss location was captured by the CFD.

The main cause of discrepancy between the results is mainly due to the structural deformation of the disc under the significantly increased air-film pressure in a converging configuration, as also suggested by the results from the previously shown FEA analysis. This suggests that it would be necessary to include the structural element of the disc deformation into the numerical model. A two-way fluid-structure interaction model is a possible solution to obtain better predictions and is currently being developed.

3.5.3 Diverging Gaps

Figures 26 to 29 show predictions from CFD against experimental results for both diverging angles at two different minimum gaps (50 µm and 100 µm) and two inlet pressures: 1.03 barg and 3.45 barg. The CFD predictions are represented by the dotted lines while the solid lines correspond to experimental data.

Across all the data presented, the entry loss from the CFD was consistently more severe than the corresponding experimental case. As expected, the entry loss increased with increasing the minimum gap. Also, the entry loss predicted by the CFD was persistently lower for the largest angle.

The case for the lowest inlet pressure (1.03 barg), figures 26 and 27, showed good agreement between the CFD and experimental results in terms of predicting the radial position of the abrupt pressure recovery. As seen previously, the pressure recovery radial location moved further downstream as the pressure increased. The pressure levels predicted by the CFD were lower than experiments which is mainly due to the higher entry loss predicted by the CFD. The match between both data sets converged towards the gap exit.

At higher the inlet pressure (3.45 barg), Figures 28 and 29, the CFD prediction of the abrupt pressure recovery occurred at a slightly higher radial location for the smaller angle (α = +1.0°). Although both data sets showed good agreement at locations towards the gap exit, pressure recovery from experiments was quicker. For the larger angle (α = +2.0°) case, recovery was predicted to occur earlier in CFD than experiment.

Furthermore, the larger angle cases from the CFD model showed the abrupt pressure recovery region to occur at a more upstream radial location compared to the smaller angle cases and recovery to gap exit pressure was achieved earlier. Both trends point towards the presence of a stronger shock wave. Experiments only showed the same trend at the lower inlet pressure (1.03 barg), and the radial location of the abrupt
pressure recovery showed little variation with coning angle for a given minimum gap. As the inlet pressure increased, experiments showed the smaller angle cases to exhibit a less severe pressure drop.

In general, predictions from the 3D CFD model were in reasonably good agreement with experiments, particularly in terms of capturing the correct trends. Although the magnitudes were not a complete match they were in a much better agreement compared to CFD predictions for the converging cases. This supports the idea that the due to the very low the air-film pressures in a diverging configuration, the load on the disc will not be significant and the structural aspect can be ignored. The main mismatch between the CFD predictions and experiments for diverging cases was the shifting of the location of the abrupt pressure recovery. This is thought to be due to machining inaccuracies which will become amplified at higher pressures.

Figures 26, 27, 28, and 29 show the radial average Mach number distribution within the gap obtained from CFD at three inlet pressures (0.69 barg to 3.45 barg) for two minimum gaps (50 μm and 100 μm) and both diverging coning angles considered. As illustrated in figure 30, at the lowest inlet pressure (0.69 barg) the Mach number initially increases due to the inlet pressure loss but does not fully reach sonic conditions. Subsequently, the Mach number starts dropping as the pressure recovers. However, no shockwaves were evident. On the other hand, as the inlet pressure increased the Mach number approached sonic conditions close to the inlet and continued increasing to supersonic speeds further downstream. The supersonic region was then terminated by a shockwave marked by the sudden drop in Mach number and abrupt pressure recovery. As the coning angle increased the shockwave was significantly more pronounced and occurred earlier than for the smaller angle which agrees well with what has been seen from the pressure distributions.

As the minimum gap height increased, Figure 31, the flow reached sonic conditions close to the gap entrance, including the low inlet pressure cases. The same trends described before apply to the rest of the cases shown. Additionally, the shock occurred at a more downstream radial location compared to the smaller gap height cases shown in figure 30.
To confirm that choked flow conditions exist in the diverging configurations considered, the non-dimensional mass flow rates through the primary gap from the CFD were plotted against inlet pressures in Figure 32. The red markers represent the $\alpha = +1.0^\circ$ case while the green marker denotes the $\alpha = +2.0^\circ$ case. Different marker shapes correspond to different gap heights. The results confirm that the flow chokes as long as the same minimum gap height (throat area) is maintained at a given inlet pressure, regardless of the diverging angle. This ties in well with the behaviours seen in the radial pressure and average Mach number distributions shown earlier.

5. SUMMARY AND CONCLUSION

An overview of the non-contacting face seal test facility along with the experimental procedure have been outlined. An extensive investigation has been carried out on the behaviour of the air-film under three different gap configurations: parallel, converging and diverging. Radial pressure distributions have been assessed for all three configurations at multiple operating conditions. All cases showed an inertia-driven entry pressure loss which decreased with smaller gaps. For converging gaps, entry loss was more sensitive to the minimum gap height rather than the coning angle. Radial pressure distributions for converging configurations showed the highest air-film pressures, which increased with an increase in converging angle. On the other hand, diverging gaps maintained the lowest air-film pressure out of all configurations. Additionally, radial pressure distributions from diverging cases uncovered a highly compressible behaviour involving choked flow conditions at the gap inlet and a supersonic flow region that was terminated by a series of shockwaves. The radial location of the minimum pressure moved further downstream at higher inlet pressures as higher Mach numbers were reached. This location showed little variation over the diverging angles tested for the same minimum gap. It was found to be more a function of inlet pressure and minimum gap height. For diverging gaps, entry loss increased as the minimum gap height increased, however, larger angles incurred a less severe loss in most cases.

The 3D CFD model compared well with parallel cases at larger minimum gaps and lower inlet pressures. The mismatch observed at high inlet pressures and small angles was due to coning of the disc under pressure. Comparison between CFD predictions and experiments for converging cases showed large discrepancies which increased with an increased air-film force at higher inlet pressures and larger coning angles which points towards significant structural deformation of the disc under higher loads.

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