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The Effect of Inlet Swirl on Brush Seal Bristle Deflections and Stability

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ABSTRACT

Brush seals can be used in the secondary air systems of gas and steam turbines to reduce parasitic leakages and engine specific fuel consumption. Application of these seal types in highly pressurised and highly swirling environments is limited due to the inherent risk of aerodynamic instability and seal failure. This paper considers coupled 3D CFD and structural modelling of brush seals, and applies the technique to investigate the effects of inlet swirl on the bristle pack. The model applies aerodynamic forces generated by CFD to a bristle pack model that includes interaction between bristles. Iteration between CFD and structural models is used to ensure consistency between the fluid and structural solutions. Inlet swirl is shown to increase bristle circumferential aerodynamic forces. At a critical value of inlet swirl, the aerodynamic force on the upstream bristles is sufficient to cause circumferential slip of the upstream bristle row, despite axial compression of the bristle pack under its operating differential pressure. In practice this may lead to instability of the bristle pack and is consistent with anecdotal reports of seal behavior. The critical swirl velocity was reduced when the downstream pressure level was raised, keeping the same upstream total to downstream static pressure difference. This is caused by the increased dynamic head associated with the inlet swirl. Inclusion of a front plate in the seal design does not offer the intended protection to the bristle pack in highly swirling environments, and in fact further lowers the critical swirl velocity where bristle slip occurs. This is associated with highly swirling flow impinging on the bristle tips. Increasing the bristle diameter and bristle stiffness does not necessarily prevent slip at higher inlet swirl velocities, but reduces the magnitude of slip of the upstream bristles.

KEY WORDS

Brush seals, Coupled model, Bristle bending, Inlet swirl, Bristle stability

NOMENCLATURE

D	Bristle diameter /m
E	Young's modulus
F	Aerodynamic force /N
H	Height of bristle length /m
L	Bristle length /m
$\dot{m}$	Mass flow rate / kg/s
P_{in}	Total pressure at inlet /Pa
p_{out}	Static pressure at outlet/Pa
R	Reaction force /N
SV	Swirl velocity / m/s
T	Total temperature / K
UDF	User defined function
x, y, z	Circumferential, axial, radial direction respectively
Z_{rotor}	Initial rotor interference /m
μ_f	friction factor
δ	Minimum clearance between bristles /m
ϕ	Lay angle /°
Δp	Pressure difference across seal /Pa

Subscripts

ax	axial
circ, θ	circumferential
n	normal
rad, r	radial

1 Introduction

Advances in sealing technology for gas turbine engines could have significant impact on engine performance and efficiency. Brush seals [1-3] are established as an option for engine sealing, but their use is limited by issues related to wear, heat generation and overall predictability. Remarkably improved leakage performance can be achieved by brush seals compared with labyrinth seals [3-6]. However, the range of operating conditions in which they are applied is limited, and each application demands a bespoke seal design.

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Typical brush seals consist of a dense pack of bristles, front plate and backing plate, as shown in Fig. 1. The flexible bristle pack is inclined in the direction of rotor rotation with a lay angle typically between 30 degrees and 60 degrees to the radial direction. The flexible bristle pack is therefore able to tolerate rotor excursions, allowing the seals to run with low clearance or even contact with the rotor.

Fig. 1 Schematic of a brush seal

Several numerical and experimental investigations [7-13] have been conducted on the leakage performance and mechanical characteristics of brush seals. These indicate that the leakage flow of brush seals can be significantly less than that of labyrinth seals at the same operation conditions.

The high internal friction and damping characteristic of brush seals [14] helps to make these seal types resistant to dynamic excitation. However, as reported by Kirk et al. [15] bristle packing density, bristle frequency (relative to engine-order frequencies), bristle stability and rotor dynamics are still important aspects of the brush seal design process. Schwarz et al. [16] carried out flow tests of an MTU-style brush seal on a static test rig, and captured a rotating flutter mechanism (in this case of the entire bristle pack) with a high speed camera.

Disturbances such as jets, recirculations and swirl in the upstream cavity can result in bristle flutter and uneven wear. The former could result in fatigue failure of the flexible elements, and the latter has consequences for seal performance and overall machine efficiency. However, there have been very limited studies that investigate the potential for aeroelastic instability of the bristle pack. Helm and Pugachev [17] measured and modelled swirl velocity in the upstream and downstream seal cavities, demonstrating that brush seals significantly reduce swirl across the sealing stage, which can improve rotor stability. Sharatchandra and Rhode [18] used CFD to show that the steady state bristle forces are substantially modified by the circumferential flow momentum, so as to deflect the bristles radially outward. This can result in increased seal leakage and lower efficiency. Aksit and Dogu [19] modelled the effect of the presence of varying geometries of front plate, using a bulk porous medium approximation for the bristle pack. Insight is gained into the effect on the upstream pressure field, and the

inclusion of holes in the upstream front plate with clearance is shown to suppress the radially outward flow within the upstream gap. However, the bulk porous modelling approach alone does not permit analysis of bristle forces and stability.

In an effort to predict brush seal bristle pack behavior, there have been numerous studies on bristle bending. A commercial finite element analysis package was used by Aksit et al. [20] to establish a three-dimensional mechanical model of a brush seal. However, bristle-to-bristle contact was not included in their model. Crudgington and Bowsher et al. [21, 22] modelled the bristle pack at steady state by adding bristle-to-bristle contact and friction into a commercial FE code, demonstrating that hysteresis (friction) and aerodynamic loading (blow-down and pressure-stiffening) effects are the dominant factors influencing brush seal behavior. Stango et al. [23] considered bristle-rotor contact and examined the brush seal forces arising along the surface of the rotor due to the radial interference between the bristle and the rotor.

Guardino and Chew [24] presented a new 3D bristle-bending mode, denoted SUBSIS, incorporating bristle-to-bristle, backing ring, and shaft contacts. The iterative method was based on linear beam-bending theory, and the so-called 3D-splay and inclined prop effects were also taken into account. The effects of rotor incursions into the bristle pack, increase of the pressure load, and changes in the lay-angle and Young's modulus were investigated. A combined 3D CFD and 3D bristle bending model for brush seals was developed by Lelli et al. [25] to further investigate brush seal behavior for an increased number of circumferential rows, although the SUBSIS method also allows the use of the bulk porous medium CFD modelling approach. The developments included automation of the CFD meshing after bristle deformation, (and improved communication between the CFD and FE solvers). An interesting outcome of this study of a larger bristle pack was the formation of slip-planes between groups of bristles.

Despite previous research in the area of inlet swirl, this is one area of brush seals which is not fully understood, and for which design rules, tools and methods are somewhat limited. In particular, there is a poor understanding of aerodynamic forces acting on the upstream face for brush seals operating with a high inlet swirl, and of the potential aerodynamic instabilities that could arise. This paper further develops the methodologies outlined in ref. [25] to investigate the effect of inlet swirl on bristle stability under different brush seal operating conditions. Results from simulations with a front plate and larger diameter bristles are also reported. The aim of this paper is to provide physical insight into the problem of swirl-induced bristle instability, and direction for future research. The simulations provide detailed information that would be very challenging to acquire from experiments, and may not be captured by more simplified models.

2 The coupled CFD/SUBSIS model

In this research, bristle bending behavior was predicted using a coupled 3D CFD and mechanical model, as described

by Lelli et al. [25]. The commercial CFD code FLUENT [26] was used with bristle bending calculated using the SUBSIS code described by Guardino et al. [24]. Further details are given in the following subsections. Iterations of CFD and SUBSIS solutions were implemented to obtain convergence of the coupled solution. The specific steps in each iteration are shown in Fig. 2.

Fig. 2 Iterative Solution of Coupled Fluid-Structural Model

2.1 Geometry and CFD mesh generation

The first step in the model generation is the definition of the initial brush seal geometry. The seal geometry under investigation is shown in figure 3 and summarised in Table 1. The base seal has no front plate and hexagonal packing is assumed. The model assumes periodicity in the circumferential direction. The CFD model considers 15 complete bristles and 10 half bristles. As shown in the figure, x, y, z coordinates in the CFD model correspond to the circumferential, axial and radial directions for the seal. Shaft curvature is neglected in this study.

For the initial geometry, the periodic planes are parallel to the axial direction. However, for the deformed geometry, the bristles are deflected in the axial and circumferential directions. In order to accommodate this, the periodic planes have to move slightly in the circumferential and radial directions. Bristle contact is modelled in the mechanical model, as bristles are forced into contact under axial and radial pressure loads. The bristle diameter is reduced slightly in the CFD model to permit a representative flow through the seal and to avoid meshing problems associated with vanishing volumes at contact points. This corresponds to the minimum clearance between bristles given in Table 1.

2.2 CFD solution

CFD simulations were carried out using Fluent [26]. The $k-\epsilon$ turbulence model and standard wall functions were used in the simulation. All of the 3D steady solutions were obtained using a first order upwind scheme. The SIMPLE algorithm was applied to solve the discrete algebraic form of the governing equations. Density was calculated based on the ideal gas law. No-slip boundary conditions were applied on the upper z-plane boundaries and on the bristle and rotor surfaces. Total pressure, total temperature and velocity angle were specified at inlet. The static pressure was specified at the

outlet plane. For the baseline brush seal model, operating conditions are shown in Table 2. Calculations were considered converged when normalised residuals of continuity, energy and turbulence equations reached 10^{-5} , and the representative flow variables, such as pressure, velocity and mass flow rate did not change with further iterations.

Table 1 Brush Seal Geometry Parameters

Packing Type		Hexagonal
Number of Axial Rows		10
Lay Angle to Radial Direction	ϕ	40°
Bristle Length	L	13.35 mm
Height of Bristle Overhang	H	1.00 mm
Bristle Diameter	D	0.100 mm
Young's Modulus	E	$2.25 \times 10^{11} \text{ N/m}^2$
Minimum clearance between bristles	δ	0.004 mm
Initial rotor Interference	Z_{rotor}	0.00 mm

2.3 Extraction of bristle forces

Aerodynamic forces on each of the bristles were extracted from CFD solutions with a UDF (user defined function). In order to calculate the bending behavior in SUBSIS, the bristle was discretized into a number of elements along the length of bristle. The number of elements in this paper (K) was set to 100. The forces on each element were calculated by integrating all the forces exerted on each element area. This distribution of discrete elemental forces along the bristles provided the boundary conditions for the mechanical model.

2.4 Mechanical model

SUBSIS (Surrey University Brush Seal Iterative Simulator) was used to predict the 3D bristle bending behavior. This code uses linear beam-bending theory and follows earlier model development [9, 24]. This mechanical model considers bristle-bristle, bristle-rotor and bristle-backing ring contact. Friction between the bristles and between bristles and backing ring was not modelled, and (while not an insignificant contribution) this is secondary to the reaction forces with regard to bristle stability and response to inlet swirl in these studies. The mechanical model used consisted of 20 complete bristles. As mentioned above, the bristles were discretized into a number of elements. The elemental aerodynamic forces extracted from the CFD solutions were applied to the bristles in the mechanical model to calculate bristle deformation. Following this coupling, SUBSIS outputs the new deformed geometry for CFD meshing, and the process is repeated. The bristle deflections were considered converged when the change in deflections between iterations was less than 1% of bristle diameter, as shown in Fig.2. In this study, three complete iterations were carried out between the CFD and mechanical model, starting from the idealized hexagonal packed arrangement. The deformed bristle pack for the baseline model was used as the initial geometry in both the

Fig.3 CFD domain and brush seal geometry

(a) CFD mesh

(b) Section of root surface mesh for initial geometry

(c) Root surface mesh for deformed geometry

Fig. 4 Typical CFD Mesh for the brush seal model

Table 2 Operating conditions

	Baseline	Inlet swirl study	Front plate study	Diameter study
inlet total pressure (P_{in})	0.5 MPa	0.5 MPa / 1.4 MPa	1.4 MPa	1.4 MPa
outlet static pressure (p_{out})	0.1 MPa	0.1 MPa / 1 MPa	1 MPa	1 MPa
inlet total temperature (T_{in})	300K	300K	300K	300K
Inlet swirl velocity (SV)	0m/s	0-280 m/s	0-280 m/s	150 m/s

the inlet swirl study and front plate study, and to confirm mesh-independence of the solutions.

3 Solution without inlet swirl

The aerodynamic forces causing bristle bending are the axial force (F_{ax}) due to the pressure load and the force normal to the bristle (F_n) in the radial-circumferential plane. Note that a negative F_n signifies a blow-down force which pushes the bristle down onto the rotor. Forces parallel to the bristle are assumed to have no effect. The diagram for calculating F_n , from the radial and circumferential forces given by CFD, is shown in Fig.5, where F_{rad} represents the radial force and F_{circ} is the circumferential force exerted on the bristle. Therefore, the force normal to the bristle can be expressed as,

$$F_n = F_{circ} \times \cos \phi + F_{rad} \times \sin \phi \quad (1)$$

Fig.5 F_n calculation diagram

For the baseline model, aerodynamic forces per unit length extracted from CFD solutions are plotted in Fig.6. It can be seen from the force distributions that the axial force gradually increases from bristle root toward the backing ring, reaching a maximum near the backing ring inner radius. Moreover, the axial force rises significantly in the streamwise direction, and the local force at the backing ring inner radius position of the last row is observed to be the highest. The normal force is substantially smaller in magnitude. The normal force is associated with radial pressure gradients that form in the bristle pack due to the presence of the backing ring and the turning of the flow. Strong inflow near the backing ring inner edge produces a blow down effect. The variation in normal force on the last bristle below the backing ring inner radius is due to recirculating flow just downstream of the bristle pack and an associated change in the circumferential pressure distribution on the last bristle.

The bristle deflections of the baseline model are presented in Fig.7. Fig.7a shows the axial deflection of the bristle centerlines, and a radial view of the bristle tip displacements is provided in Fig.7(b). The bristle cross-sections appear elliptical in this and subsequent plots due to their angle of inclination to the radial direction of 40° . The numbering of rows starts with the upstream bristles (positioned at $Y = 0$ in both Figs.7a and 7b and the backing ring location is shown. Some axial (Y) bending is clear with minimal circumferential (X) deflection.

(a) F_{ax}

(b) F_n

Fig.6 Aerodynamic forces from CFD solution

The bristle axial deflections along the bristle length obtained from three iterative steps (see Fig. 2) are plotted in Fig. 8. Step_01 (or iteration 1) represents the bristle deflections due to the aerodynamic forcing from CFD with an undisturbed bristle pack. For the baseline seal, the bristle tip deflection changed by less than 1×10^{-6} m between step_02 and step_03. In the mid-bristle region the deflection difference was only slightly above 1×10^{-6} m, approximately meeting the convergence criterion given earlier. This indicates that the convergence of bristle deflection has been achieved after two iterations.

A CFD mesh dependency study was conducted on the deformed geometry (obtained at the third iteration). Meshes with 100 nodes, 200 nodes and 300 nodes along the bristle length, with the same mesh in the x-y plane, were tested. Aerodynamic forces on the first row (which are of particular interest in this study), and in the bristle tip region, are shown in Fig.9. It is evident that there is no discernible difference in the magnitude of the force distributions along bristle length at 200 mesh nodes and above, hence this length-wise mesh density was selected for subsequent simulations.

(a) Bristle centerlines on the axial- radial plane

(b) Bristle tips on the axial-circumferential plane

Fig. 7 Bristle Deflections of the base model

Fig.8 Bristle deflections in the iterative steps

In addition, the effect of varying mesh density in the bristle gaps with 200 nodes along the bristle length was investigated. Meshes with 1.8 million, 2.4 million and 3.6 million cells were applied. The leakage mass flow rates obtained are shown in Table 3 and bristle deflections are shown in Fig.10. Results indicated that the mass flow rate as well as the deflection changed less than 1% for these three meshes, as listed in Table 3. It was concluded that, despite the use of a first order difference scheme, the level of mesh independence was

sufficient for the current investigation. The mesh with 2.4 million cells was used for further studies reported here.

Fig. 9 Aerodynamic forces on first row with different radial mesh numbers

Table 3 Mesh dependency study

Mesh number (million)	1.8	2.4	3.6
Mass flow rate ($\times 10^{-3} \text{ kg/s}$)	1.408	1.402	1.401
Mass flow rate deviation	< 0.5%		
Bristle deflection deviation	< 1%

(a) 1.8 million cells

(b) 2.4 million cells

Fig. 10 Bristle deflections obtained with different meshes

4. The Effect of Inlet Swirl

4.1 Results at low downstream pressure

The effect of inlet swirl flow on the bristle deflections and stability is reported in this section. Inlet swirl was applied in the direction of the rotor rotation. Results were obtained for a range of inlet swirl velocities.

Figure 11 shows the axial aerodynamic forces on bristles in all ten rows for the case with inlet swirl velocity of 250 m/s and the normal aerodynamic forces on upstream bristle. Apart from the inlet swirl, all the boundary conditions are the same as for the baseline case, given in Section 2. The geometry used also corresponds to the baseline case. Comparing to the case without inlet swirl in Figs 6 and 9, it can be observed that the inlet swirl velocity component results in an increase in the normal force on upstream bristles. The axial force is reduced compared to Fig. 6, since inlet total pressure is unchanged and the dynamic head associated with the inlet swirl is significant.

The predicted bristle deflections for the case with inlet swirl velocity of 250 m/s are given in Fig.12. Figures 12a and 12b show large axial and circumferential deflections of the upstream bristle, while the rest of the pack remains relatively

Fig.11 Aerodynamic forces at SV=250 m/s

Fig.12 Bristle deflections at SV=250 m/s

unaffected by the inlet swirl. The magnitudes of the axial and circumferential deflections exceed the valid region for linear beam bending, which forms the basis of these calculations in SUBSIS. However, the results indicate that the inlet swirl has caused slip and possible instability of the upstream bristle. The axial deflection of the bristles in the downstream direction is merely a consequence of these large deflections and the finite circumferential extent of the domain. In the real brush seal, the presence of bristles would prevent this deflection, but would not prevent circumferential displacement and instability of the upstream bristle row. It should also be noted that further SUBSIS iterations were not run on this deformed domain, therefore the extent to which this instability could penetrate the bristle pack is not known.

To understand the conditions for slip in more detail, Fig.13 illustrates a simple 2D force analysis of the upstream bristle contact region, in a plane perpendicular to the bristle axis. $\mathbf{R}$ represents the reaction force on an upstream bristle against a second row bristle. $\mathbf{F}_{ax}$ and $\mathbf{F}_n$ are the axial force and normal force acting on the upstream bristles respectively. Since the bristles are packed hexagonally (and friction is neglected), $\mathbf{R}$ is at an angle of 60° to the bristle normal direction, as shown.

Fig.13 Idealized Upstream Bristle Row Contact Force Analysis

The maximum possible reaction force opposing a positive normal to bristle aerodynamic force occurs when the upstream bristle is entirely supported by one second row bristle. For frictionless equilibrium at this condition, axial and bristle normal components of $\mathbf{R}$ can then be expressed as,

$$\mathbf{F}_{ax} = \mathbf{R} \cdot \sin 60^\circ \quad (2)$$

$$\mathbf{F}_n = \mathbf{R} \cdot \cos 60^\circ \quad (3)$$

Hence,
$$\mathbf{F}_n = \mathbf{F}_{ax} / \tan 60^\circ = 0.577 \mathbf{F}_{ax} \quad (4)$$

The ratio of the axial to the normal force is of interest in this case, because the axial force acts to compress the bristle pack into tighter packing, aiding the support of upstream trailing bristles, while the normal force acts to displace the bristle from its position within the hexagonal pack. Therefore, for this idealized bristle pack geometry, slip of the upstream bristle is expected if the ratio of normal to axial aerodynamic forces is greater than 0.58. For comparison, integrating forces $\mathbf{F}_n$ and $\mathbf{F}_{ax}$ in Fig.11 (b) over the length of the bristle gives $\mathbf{F}_n/\mathbf{F}_{ax}=1.25$. Note that following fluid-structure coupling, non-idealized geometries of the bristle pack result. This can

change the angle between the reaction force and normal force (shown as 60° in Fig 2). For example, a plausible change in the angle from 60° to 65° changes the threshold force ratio to 0.47. Note also that aerodynamic forces and contact conditions vary along the bristle length, challenging the validity of the 2D contact force model and possibly resulting in a substantially different slip criterion. Nevertheless, the above model provides a useful preliminary analysis and understanding.

The simplified analysis above is readily extended to include static friction. For a friction factor μ_f , the angle of the reaction force to the bristle normal direction in the critical condition reduces to $60^\circ - \tan^{-1}\mu_f$. The critical ratio $\mathbf{F}_n/\mathbf{F}_{ax}$ is then $1/\tan(60^\circ - \tan^{-1}\mu_f)$. For a friction factor of 0.3, this gives $\mathbf{F}_n/\mathbf{F}_{ax}=1.06$.

Calculations were conducted for a range of inlet swirl velocities to find the point at which the CFD-SUBSIS model predicted slip of the upstream bristles. The results are summarized in Table 4. It can be seen that for inlet swirl velocities up to 200 m/s, the upstream bristle did not slip. For the above geometry, slip occurs when the swirl velocity, and therefore the ratio between axial and circumferential forces, reaches a critical value (between 200 m/s and 250 m/s). In addition, the leakage mass flow rate of the brush seal decreases gradually as the inlet swirl increases, since the inlet total pressure is held constant and a greater component of the flow velocity is attributed to circumferential flow rather than axial through-flow. The integrated force ratio $\mathbf{F}_n/\mathbf{F}_{ax}$ is strongly dependent on inlet swirl. At lower inlet swirl variation of $\mathbf{F}_n/\mathbf{F}_{ax}$ with SV could be approximated as quadratic, but there is more than a threefold increase in $\mathbf{F}_n/\mathbf{F}_{ax}$ between SV = 250 m/s and 280 m/s. At SV = 200 m/s, the bristles did not slip, despite the calculated force ratio exceeding the theoretical slip threshold from the 2D model. This shows that the 2D model approximation may not capture important 3D effects, as discussed above, which can substantially change the threshold force ratio.

Table 4 Slip results for inlet swirl flow ($p_{out}=0.1\text{MPa}$)

SV m/s	0	100	150	200	250	280
Slip or not	✗	✗	✗	✗	✓	✓
$\mathbf{F}_n/\mathbf{F}_{ax}$	0.014	0.18	0.359	0.76	1.25	4.01
$\dot{m}$ (10^{-5} kg/s)	1.402	1.283	1.246	1.169	1.132	1.083

4.2 Results at elevated downstream pressure

Calculations were also conducted at elevated downstream pressure, a commonly occurring operating condition for seals within gas and steam turbines. The downstream static pressure was raised to 1MPa, and the pressure difference from upstream total to downstream static pressure was kept the same ($\Delta p=0.4\text{MPa}$). At this pressure, the upstream bristles were found to slip at swirl velocity SV=150 m/s (but not for

(a) $p_{out}=0.1\text{MPa}$, $p_{in}=0.5\text{MPa}$

(b) $p_{out}=1\text{MPa}$, $p_{in}=1.4\text{MPa}$

Fig. 14 Static pressure on selected bristles (SV=150 m/s)

(a) $p_{out}=0.1\text{MPa}$

(b) $p_{out}=1\text{MPa}$

Fig.15 Aerodynamic forces on upstream row at different downstream pressures (SV=150 m/s)

SV=100 m/s). This case was then examined in some detail.

While the pressure difference has been maintained, the elevated downstream pressure case has a reduced inlet to outlet pressure ratio compared to the ambient outlet pressure case, and this is expected to affect the pressure distribution in the bristle pack. Figure 14 shows the circumferentially averaged static pressure distribution along the bristle, for selected bristle rows. The elevated pressure/lower pressure ratio case in Fig. 14b shows lower pressure drop between rows 9 and 10, and higher pressure drop between rows 2 and 5, compared to the low pressure case. This is as expected for the change in pressure ratio. The pressure drop from row 1 to 2 is similar in the two cases, and is possibly affected by the inlet swirl. The dynamic head associated with the inlet swirl is considerably higher for the elevated pressure case, and this

results in a higher loss of total pressure across the first bristle row.

Aerodynamic forces on upstream bristles for SV= 150 m/s at $p_{out}=0.1\text{MPa}$ and $p_{out}=1\text{MPa}$ are shown in Fig. 15. This illustrates that for the elevated pressure case, the positive normal force rises significantly because of the increased dynamic head. The axial force results show negative values over the outer two thirds of the bristle length, indicating that stagnation of the swirl flow as it enters the bristle pack tends to lift the bristles axially upstream.

The bristle tip deflections for SV=150 m/s in Fig.16 show slip for the higher pressure case. Table 5 summarises results for the high pressure case and shows that for $p_{out}=1\text{MPa}$, the minimum swirl velocity for the upstream bristle slip is in the region 100 m/s to 150 m/s, compared to 200 m/s to 250 m/s for $p_{out}=0.1\text{MPa}$.

(a) $p_{out}=0.1\text{MPa}$

(b) $p_{out}=1\text{MPa}$

Fig.16 Bristle tip deflections at SV=150 m/s

Table 5 Slip results for inlet swirl flow ($p_{out}=1\text{MPa}$)

SV m/s	0	100	150	200	250	280
Slip or not	✗	✗	✓	✓	✓	✓
F_n/F_{ax}	0.01	0.46	4.35			

Figure 17 shows the swirl dynamic head, calculated from the compressible isentropic flow relation, over the swirl velocity range of interest and at ambient and elevated downstream pressure. It is interesting to note that the inlet swirl dynamic heads for $SV = 150 \text{ m/s}$ at 1.4MPa inlet total pressure and for $SV = 250 \text{ m/s}$ at 0.5MPa inlet total pressure are quite similar. As these represent the lowest swirl velocities at which we showed slip for the two cases, this suggests that swirl dynamic head may be a useful correlating parameter for bristle slip, and is consistent with the observation of aerodynamic forces above.

5. The Effect of a Front Plate and Bristle Diameter

5.1 Front plate

The baseline model was modified to include a front plate, to investigate its effect on bristle forcing and stability with a swirling inlet flow. The front plate length from the seal outer radius was set equal to that of the backing ring, and the gap

Fig.17 Variation of inlet swirl dynamic head with swirl velocity for the two pressure levels considered

between the front plate and the first bristle row was fixed at 0.2 mm , as shown in Fig.18. In brush design, a larger starting gap would be normally allowed to accommodate bristle pack compression and widening under conditions of rotor incursion into the bristle, so as to avoid any contact and excessive seal stiffness. However, in this analysis the pack width and a constant clearance of the front plate from the upstream were maintained, and the axial gap is considered to be reasonably representative at the higher duty conditions. Simulations were carried out with a range of inlet swirl velocities at the elevated pressure level ($p_{out}=1\text{MPa}$, $P_{in} = 1.4\text{MPa}$), and the SUBSIS procedure described earlier was implemented to obtain bristle forcing and deflections.

Figures 19 and 20 show the tangential velocity and radial velocity contours for the case $SV=150 \text{ m/s}$. It appears from Fig. 19 that the front plate significantly reduces the area of bristle exposed to the high swirl velocities, thereby potentially offering some protection to the upstream bristles. However, it is also evident from Fig 19b (with front plate) that the tangential velocity in

Fig.18 Baseline geometry with a front plate

Fig. 19 Circumferential velocity contours (SV=150 m/s)

Fig. 20 Radial velocity contours (SV=150 m/s)

the overhanging region is much higher compared to the case without the front plate. Figure 20 shows that an outward radial flow is present in the region of the gap entrance between the front plate and first bristle row, although this is significantly smaller in magnitude than the swirl (tangential) velocity in this region.

The aerodynamic forces and bristle deflections are presented in Fig. 21. For comparison, the integrated forces over the length of the bristle in the case with front plate give $F_n/F_{ax}=3.99$, while the value in the baseline geometry without front plate is around 4.35. These values of F_n/F_{ax} are much higher than the critical value given by the simple 2D analysis, and are consistent with the solutions showing slip. Comparing figures 21 and 15b it is also noted that the form of the force distribution along the bristle is quite different in the two cases. The normal force distribution shows a distinct peak of higher magnitude at the front plate inner radius that is not present for the cases without a front plate, and the axial force acts over a much-reduced radial extent, primarily within the overhang region. It is also evident that the negative force on the upper two thirds of the bristle, which was visible in the case with no front plate (Fig. 15b), is not present in the bristle pack with a front plate. However, this does not prevent the bristle from slipping, and this is primarily due to a

substantially reduced axial force in the bristle tip region, so that the normal force now exceeds the axial force over the lower half of the bristle (Fig. 21a).

(a) aerodynamic forces

(b) deflections at bristle tip

Fig. 21 Aerodynamic forces and bristle deflections with a front plate ($p_{out}=1\text{MPa}$, $SV=150\text{m/s}$)

The front plate may be expected to improve bristle stability by offering some protection to the upstream row from the inlet swirl flow. Nevertheless the bristle tip positions in Fig 21b show that the bristles have slipped, albeit that the displacements have been reduced compared to those without the front plate in Fig.16b.

Simulations were additionally run for the case of $SV = 100$ m/s, with and without a front plate. In contrast to the $SV = 150$ m/s case (Fig. 21a), the force distribution plot in Fig. 22 shows a much reduced ratio of the normal to axial forces, which is more similar in form to the previous cases without the front plate. The presence of the front plate again suppresses the negative axial force on the outer part of the bristle observed for the baseline case. However, this configuration shows slippage of the upstream bristles (Fig. 23b), while the seal with no cover plate is stable at these operating conditions (Fig. 23a). The integrated force ratios for the cases with and without front plate were 0.52 and 0.46 respectively. As discussed above, the simple 2D model does

not capture the full complexity of the problem. These results show that force distributions along the bristles are important, confirming the need for 3D models for both fluid flow and bristle deflections. The results also suggest that inclusion of a front plate in brush seal design does not offer the intended protection to the bristle pack in highly swirling environments. In fact, this has the opposite effect, lowering the swirl velocity threshold at which the upstream bristles become unstable, potentially having a detrimental effect on seal performance and integrity. This can be attributed to the high level of swirl velocity in the flow impinging on the bristle tips in the overhang region, as shown in Fig. 19b, which is in contrast to the case without front plate (Fig. 19a).

Fig.22 Aerodynamic forces on upstream row for SV=100m/s with front plate ($p_{out}=1\text{MPa}$)

5.2 Bristle Diameter Effect

In this subsection, results are reported from simulations of a bristle pack with 0.15mm wire bristles and no front plate, to understand the effect of this change on bristle stability. One simulation was conducted at 1MPa downstream static pressure with the inlet swirl velocity of 150 m/s. As before, the pressure difference across the bristles was 0.4MPa. The results for bristle deflection are shown in Fig. 24 and may be compared to those for the 0.1mm diameter bristles in Fig. 16b. It is evident that the upstream bristles still slip circumferentially, but that the larger diameter (and significantly increased bristle stiffness) has significantly reduced the circumferential deflections.

Larger bristle diameters are expected to provide more resistance to slippage and hence stability margin, although this would need to be traded off against overall seal stiffness, contact forces and wear, and frictional heat generation. Assuming aerodynamic forces F_n and F_{ax} to be proportional to bristle diameter, linear bending theory indicates deflections will be inversely proportional to D^3 . The ratio of D^3 for the two cases is 3.375. This is consistent with the results obtained, noting that the thicker bristles slip one diameter and are then supported by the next set of bristles.

(a) Without front plate

(b) With front plate

Fig.23 Bristle tips for SV=100m/s ($p_{out}=1\text{MPa}$)

Fig. 24 Bristle tip deflections, $D = 0.15\text{mm}$, $SV = 150\text{ m/s}$

6 Conclusion

Brush seals are pressure-activated devices, and aeroelastic stability is an important design consideration. Prior work has shown that instabilities can occur in bristle packs with high levels of inherent friction. In environments with high inlet swirl and with an elevated downstream pressure, the risk of aerodynamic excitation of the bristles is higher. There is a poor fundamental understanding of the conditions under which such instabilities might occur.

The Surrey University Brush Seal Iterative Simulator (SUBSIS) has been used to couple a 3D CFD model with a 3D mechanical model of a brush seal bristle pack, so as to investigate the effect of geometry and operating conditions on

bristle deflections and stability. This method provides a detailed resolution of the bristle pack, allowing individual bristle aerodynamic loads and deflections to be inspected. A converged solution was obtained for bristle deflection in the mechanical model. For the purposes of this study, and as a conservative design philosophy, an instability was regarded as a circumferential slip of the upstream bristle row from its original position, without consideration for further possible flutter behavior, or oscillation of the bristle aerodynamic loads and bristle position.

A baseline case was established with no inlet swirl, confirming observed trends of bristle aerodynamic load distributions and deflections from prior work. Inlet swirl was then introduced, showing that this primarily affects the upstream bristles, but does not significantly impact on the deflections of downstream bristle rows. A 2-D analysis of an idealized contacting bristle pack at equilibrium shows that bristle instability, or slip, can occur when the ratio of the normal to axial forces exceeds a threshold of 0.4 - 0.6 in the absence of friction, and substantially higher (greater than 1) when friction is included. This force ratio is sensitive to the initial bristle pack configuration and 3D effects, which are not captured by the 2D approximation. The ratio is independent of bristle diameter, although diameter and bristle stiffness do have a significant effect on bristle deflection once slipped. For the selected bristle pack geometry, and with an ambient downstream pressure, bristle slip was observed to occur at inlet swirl velocity between 200 m/s and 250 m/s and above. For an elevated pressure case, the bristles were shown to slip at a lower swirl velocity between 100 m/s and 150 m/s, due to the higher flow density and momentum forces. For conventional brush seal designs with no front plate, this work indicates that inlet swirl dynamic head can be correlated with instability (slip) of the upstream bristle row, hence useful design rules can be formed using the outlined methodology or empirical approaches. An inspection of the pressure distribution in the bristle pack revealed that the change in pressure ratio (from 4 to 1.4) did not strongly affect upstream row aerodynamic loading and bristle deflections, but did change the axial pressure gradient in downstream rows approaching the seal exit.

A front plate of equal length to the backing ring was included in the geometry. This significantly altered the upstream pressure and velocity field, resulting in substantially different bristle force distributions. It may be expected that a front plate can offer some protection and swirl resistance to the upstream bristles, but this work has shown that the bristles still displace, and in fact that the threshold swirl velocity for this slippage was further reduced to 50 m/s to 100 m/s. These results confirm the need for 3D flow and structural models that are able to resolve these subtle changes for more complex seal designs.

As mentioned above, increasing the bristle diameter by 50% did not affect the initial slippage, suggesting that aeroelastic instability and fretting of adjacent rows of bristles could still ensue, but deflections were considerably reduced. A

slippage of one bristle diameter was observed before the bristles were supported by the next circumferential row.

This work has demonstrated that a coupled CFD-mechanical model can give considerable insight into brush seal bristle pack behavior, and that the presented method and results open up a number of possibilities for future work. The SUBSIS model itself can be improved to include friction, and this can be used to investigate different geometries of front plate and backing ring, and their impact on bristle pack stability in highly swirling and highly pressurized environments. This approach might also be developed further to investigate the effect of time-varying bristle loading on bristle stability.

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