Effects of Purge Flow Configuration on Sealing Effectiveness in a Rotor–Stator Cavity

Secondary air is bled from the compressor in a gas turbine engine to cool turbine components and seal the cavities between stages. Unsealed cavities can lead to hot gas ingestion, which can degrade critical components or, in extreme cases, be catastrophic to engines. For this study, a 1.5 stage turbine with an engine-realistic rim seal was operated at an engine-relevant axial Reynolds number, rotational Reynolds number, and Mach number. Purge flow was introduced into the interstage cavity through distinct purge holes for two different configurations. This paper compares the two configurations over a range of purge flow rates. Sealing effectiveness measurements, deduced from the use of CO2 as a flow tracer, indicated that the sealing characteristics were improved by increasing the number of uniformly distributed purge holes and improved by increasing levels of purge flow. For the larger number of purge holes, a fully sealed cavity was possible, while for the smaller number of purge holes, a fully sealed cavity was not possible. For this representative cavity model, sealing effectiveness measurements were compared with a well-accepted orifice model derived from simplified cavity models. Sealing effectiveness levels at some locations within the cavity were well-predicted by the orifice model, but due to the complexity of the realistic rim seal and the purge flow delivery, the effectiveness levels at other locations were not well-predicted. [DO: 10.1115/1.4040308]

Introduction

High efficiencies and power density in gas turbines are increasingly important for both power generation and aviation propulsion. To increase thermodynamic cycle efficiencies, manufacturers are continually increasing overall pressure ratios, which also increase temperatures in the engine. Increasing temperatures can lead to durability concerns for hot section components so secondary air, which is bled from the compressor, is used to provide the cooling and sealing flows necessary to maintain durability of the hot section components. Excessive use of the secondary air results in a parasitic loss to the thermodynamic cycle of the engine so efficient use of the air is necessary to maintain efficiency. Sealing technologies, in particular, are needed to protect the cavity regions inboard of the airfoil platform. Rim seals are located at the platform, where rotating blades and stationary vanes meet. Rim seals are purposefully complex, using a combination of axial and radial clearances to minimize the hot gas ingestion into the cavities inboard of the airfoil platform. Seal clearances, however, must be large enough to allow for thermal expansion and engine transients. Propulsive efficiencies of aircraft engines are improving through increasing bypass ratios, which result in reduced engine core sizes. In these engines, the seal clearances do not always shrink as much as the engine cores so these compact engine cores may exhibit increased relative clearances resulting in more ingestion, and thus, a more difficult task for the secondary air system to minimize the flow while maintaining sealing performance. To supplement rim seals, sealing flow, also referred to as purge flow, is used to purge the cavity of hot gas. Johnson et al. [1] reported that a 50% reduction in the sealing flow in a two stage turbine would increase the turbine efficiency by 0.5% and decrease fuel consumption by 0.9%. There is a need to develop more advanced sealing technologies that minimize flow requirements while maintaining sealing performance.

This paper describes the sealing effectiveness in an engine-realistic rim seal and rim cavity at engine-relevant conditions for two distinct purge flow configurations. A review of previous literature will be given that illustrates the uniqueness of the work presented in this paper, followed by a description of the facility, turbine, and benchmarking. The sealing effectiveness was determined through concentration measurements whereby CO2 was used as a tracer gas in the secondary air supply. For the data presented in this paper, sealing effectiveness measurements were made in the front vane-blade cavity for a 1.5 stage test turbine for two engine-realistic purge flow configurations. The effectiveness data will also be compared to the orifice model reported by Owen et al. [2].

Literature Review

Our understanding of the complex topic of hot gas ingestion has increased over the past several decades due to several
fundamental studies. This section will briefly review some of these fundamental studies that correspond to the influencing parameters identified by Johnson et al. [1] including rotational effects, such as disk pumping; external effects, such as periodic vane and blade pressure fields; and geometry effects, such as the rim seal design. Additionally, the influence of the purge flow delivery system and the influence of operating conditions on sealing effectiveness will be briefly discussed.

Early ingestion experiments used simplified rotor–stator cavities, where one side was a rotating plane disk and the other side was a stationary plane disk, to study the effects of rotation. Bayley and Owen [3] developed a simple correlation to predict the minimum flow rate required to seal a plane rotor–stator cavity from ingestion. Phadke and Owen [4] studied a similar plane rotor–stator cavity with several rim seal geometries. Through flow visualization, they found that the purge flow was entrained in the rotor boundary layer and moved radially outward to conserve angular momentum (disk pumping), which caused a radial inward flow on the stator. To satisfy continuity, this flow field induced an axial flow across the cavity. They found that this disk pumping affected the cavity flow field for different rim seal geometries. They provided an improved correlation, over that of Bayley and Owen [3], to predict the minimum sealing flow rate for a variety of simplified rim seal geometries. Their data showed that the minimum flow rate required to seal the cavity increased with rotational speed or rotational Reynolds number [4].

Ingestion has not only been shown to be affected by rotational effects but has also been shown to be affected by the main gas path flow, especially the vane exit pressure field. Phadke and Owen [5] performed experiments for several rim seal geometries at a variety of main gas path conditions. Although the study did not include airfoils, they showed that at low main gas path velocities (related to low airfoil Reynolds numbers), ingestion was driven by rotational effects, such as disk pumping, but at higher main gas path velocities (related to high airfoil Reynolds numbers) ingestion was driven by the nonaxisymmetric pressure difference in the main gas path. Two main regimes were identified: rotationally induced and externally induced ingestion, both with different governing physics. Hamabe and Ishida [6] showed that for externally induced ingestion, the rim seal could be modeled as an orifice and there was good agreement for the sealing effectiveness data using a simplified seal geometry. Chew et al. [7] also showed the presence of two regimes at different purge flow rates. Their results showed that the ingress and egress discharge coefficients varied with the swirl flow angle in the main gas path.

The boundary conditions on the rim seal in the main gas path are not only affected by the upstream vanes but also by the interaction of the vane and blade pressure fields. Unsteady flow field measurements, performed by Bohn et al. [8] in the outer portion of the rim cavity for a simple axial seal, showed the strong influence of the passing rotor on ingestion. The potential field of the passing rotor interacted with the vane wake flow and caused more ingestion than when the passing rotor interacted with the vane core flow. Unsteady computational fluid dynamics (CFD) simulations performed by Wang et al. [9] showed that the interaction of the vane and blade pressure fields caused rotating cells in the main gas path and in the rim cavity. These cells resulted in unsteady boundary conditions on the rim seal that caused alternating pockets of ingestion and egress through the rim seal. Their results also illustrated the importance of simulating a full annulus rather than just a sector.

The geometry of the rim seal has also been shown to have a substantial effect on sealing effectiveness. The effectiveness of multiple rim seal geometries was compared by Graber et al. [10], who showed that reducing the radial clearance of the rim seal was more effective at reducing ingestion than increasing the axial overlap of the rim seal. Sangan et al. [11] also showed that reducing the radial clearance of the rim seal reduced ingestion for a single overlap rim seal. Their data from more complex rim seals, such as double rim seals, exhibited a further reduction in ingestion for a given sealing flow thereby indicating that complex rim seal geometries are needed. Although the literature clearly shows the importance of the rim seal geometry, most studies have used simplified geometries. To generate data and models applicable to engines, it is important to study engine-realistic rim seal geometries.

There have only been a few studies in the literature that have investigated the influence of the delivery method for the purge flow, which is the secondary flow devoted to sealing cavities against hot gas ingestion. Most ingestion experiments have introduced the purge flow at the center of rotation, but the purge flow is generally not introduced to turbine cavities in this manner. Engine designs are complex and often require that the purge flow be delivered in a variety of ways. Coren et al. [12] reported that the incoming angle and the jet momentum of a radially injected purge flow affected the sealing effectiveness and the cavity flow physics in a blade-vane cavity. When the purge flow was directed upstream toward the rotor, the flow was more likely to be entrained into the disk pumping flow. This entrainment phenomenon increased the effectiveness by as much as 50% as a direct result of the purge flow delivery system. Companion numerical studies by Andreini et al. [13] showed that increasing the momentum of the purge flow affected the robustness of the purge flow delivery.

Hot gas ingestion is a complex topic, and fundamental studies have provided useful information that has enhanced our understanding of the cavity flow physics as well as provided the basis for several ingestion models. Many of these studies, however, have been performed at simplified conditions with simplified geometries. Teuber et al. [14] showed that higher external Mach numbers resulted in more ingestion. In regard to the topic of hot gas ingestion, Green and Turner [15] stated that “oversimplified experimental rigs operated far from engine conditions may often only serve to confuse the issue…. Before design rules can be established with confidence, all the influencing parameters should be examined together at conditions as close to modern engine operating levels as possible.”

There is a need to study all the influencing parameters at engine-relevant conditions to allow for the development and validation of ingestion models. This paper is unique in that the effectiveness for two engine-realistic purge flow configurations is presented for an engine-realistic rim seal operated at engine-relevant Mach and Reynolds numbers. The effectiveness data are then compared to an orifice model.

**Description of Facility and Turbine**

The experiments presented in this paper were performed in the Steady Thermal Aero Research Turbine (START) facility, the design of which was previously described by Barringer et al. [16]. The previous papers by Clark et al. [17,18] also described the facility commissioning, the half-stage turbine configuration, and instrumentation. In this section, the facility will be briefly reviewed, and the 1.5 stage turbine used for these experiments will be described in more detail.

**Facility.** The START facility, which houses a 1.5 stage test turbine, is an open, continuous flow loop, as shown in Fig. 1. Ambient air enters a compressor requiring 1.1 MW (1500 hp) of power that provides up to 5.7 kg/s (12.5 lbm/s) of air flow at 480 kPa (70 psia) to the turbine. A portion of that flow is used to provide the turbine secondary air. A heat exchanger and chiller outside the lab cools the compressor and turbine secondary air, respectively. Air from the compressor flows through a supply pipe to an upstream settling chamber, where it is then directed to the turbine test section. The turbine exit air enters a downstream settling chamber after which it exits the facility. Venturi flow meters that are fully calibrated by a NIST certified laboratory [19] provide the turbine inlet and exit flow rates. Operating conditions in the test section are controlled by three flow control valves that independently
control the turbine inlet pressure and the pressure ratio to within ±1.4 kPa (±0.2 psi) and ±0.002, respectively.

The turbine rotor speed, torque, and power are controlled by a water brake dynamometer. The water brake provides braking torque by flowing pressurized water through rotating perforated disks connected to the turbine shaft. A simplified diagram of the water system is shown in the top right of Fig. 1. The dynamometer inlet and exit valves are hydraulically actuated and the set point is maintained by a controller that is provided and tuned by the dynamometer vendor. At a typical operating speed, the water brake dynamometer is shown to hold the turbine rotor speed within a standard deviation of ±0.2% of the mean speed.

The turbine rotor is supported by two radial magnetic bearings through an electromagnetic field. Auxiliary bearings are also available to support the rotor when the magnetic bearings are not active or to catch the rotor in the case of a magnetic bearing failure. Sensors track the radial orbits of the shaft, and a controller maintained the magnetic bearing parameters to ensure safe and stable operation. The shaft centerline is held within ±2.5 μm (±0.0001 in.) of the true centerline by the magnetic bearings during normal operation.

An electromagnetic thrust bearing is used to maintain the nominal axial position of the rotor, but the thrust from the turbine could exceed the 6.7 kN (1500 lbf) capacity of the thrust bearing. A two-stage pneumatic thrust piston system is incorporated into the rotor to provide an additional 8.9 kN (2000 lbf) of counter thrust. The thrust piston operates as pressurized air on the aft side (high pressure side) of the pistons flows to the exhaust cavities (low pressure side) across commercial brush seals. The pressure difference across the two thrust pistons provides sufficient counterthrust in addition to the magnetic thrust bearing for a wide operating range of the turbine.

Test Turbine. The test section included a 1.5 stage turbine with a vane-blade-vane configuration as shown in Fig. 2. Similar to the research of Clark et al. [17,18], partial span airfoils were used for this study to reduce the mass flow requirements while maintaining an engine-relevant axial Reynolds number, rotational Reynolds number, and Mach number. Other researchers have also proven the use of partial span airfoils for studying rim seals and ingestion [20–22].

The airfoils, disk, rim seals and cavities, and secondary flow supplies in this test turbine used modern gas turbine hardware. The first and second vanes were additively manufactured from a nickel alloy in doublets or pairs. The blades were solid single crystal castings attached to the disk through individual fir tree slots. There were also interblade gaps and seals as in an operational gas turbine. Two cover plates with labyrinth seals, shown in Fig. 2, were installed on the front and aft sides of the disk to axially secure the blades and provide an engine representative front wheel-space, front rim cavity, and aft rim cavity.

Two main secondary air supplies were available on the front side of the disk: (1) purge flow from the vane plenum and (2) tangential on-board injection (TOBI) flow as shown in Fig. 2. The purge flow was provided directly to the front rim cavity from the vane plenum in two different configurations: 150 axially oriented holes or 32 axially oriented holes. In an operational gas turbine, a
portion of the TOBI flow, which is a pressurized air flow injected inboard of the front wheel-space, and provide the blade cooling flow. The purpose of the test program was to investigate cause and effect relationships for engine-relevant purge configurations, and this paper separates the effects. As the focus of this paper is on the effects of the purge flow in the rim cavity, the TOBI flow was not used for these experiments. The rationale for not using the TOBI flow in this initial test campaign was the desire to perform a systematic study understanding the effects of increasing complexity. The experiments in the test turbine began with no rotation [18], and the next step was to determine the effects of the number of purge holes without the TOBI flow. Results will be presented in the future for TOBI flow.

The individual blades, gaps, and slots allowed for leakage through the fit tree gaps of the blades as well as from the front rim cavity to the aft rim cavity through the gaps between the blades as in an operational gas turbine. Since the TOBI flow was not used for these experiments, a small portion of the flow (approximately 5% of the minimum flow required to seal the front cavity) in the front rim cavity passed across the front labyrinth seal into the front wheel-space, through holes in the front cover plate, and across the disk through the blade fit tree gaps. A larger portion of the flow, nominally around 15–20% of the minimum flow required to seal the front rim cavity, passed from the front rim cavity to the aft rim cavity through the gaps between the blades. Additional leakage flow was introduced into the aft rim cavity across the lab-rin seal from the thrust piston for the magnetic bearing cooling. The leakage flow from the thrust piston into the aft cavity accounted for approximately 35–40% of the minimum flow required to seal the front rim cavity.

Instrumentation and Uncertainty. The turbine test section was heavily instrumented as shown in Fig. 3 and previously described by Clark et al. [17,18]. Several static pressure taps, pressure probes, and thermocouples composed the instrumentation in the turbine. The first and second vanes were additively manufactured, which allowed for integrated static pressure taps on the airfoil surfaces and near the rim seal of the first vane. These taps are indicated in Fig. 3 as “through AM.”

In addition to pressure measurements, the static pressure taps were used for concentration effectiveness measurements. According to the method described in detail by Clark et al. [17,18], the secondary air supply was seeded with 1% CO2 to be used as a tracer gas in the turbine. Flow was sampled at specific locations in the front rim seal, rim cavity, and wheel space through the static pressure taps. The CO2 concentration of the sampled flow was measured by a gas analyzer. By measuring the background concentration, \(c_\infty\), the supply concentration, \(c_s\), and the concentration at the location of interest, \(c\), an effectiveness based on concentration was determined according to the following definition:

\[
\eta = \frac{c - c_\infty}{c_s - c_\infty} \quad (1)
\]

When making gas sampling measurements, it is important to perform sampling sensitivity studies to ensure that the sampling method does not affect the flow field. The study by Clark et al. [17] showed that concentration gradients were present where the purge flow interacted with the ingested flow in both the rim seal and cavity, and that achieving isokinetic sampling was crucial to obtaining reliable effectiveness measurements. Although data are not shown in this paper for the sake of brevity, sampling sensitivity studies at various purge flow rates for the 1.5 stage turbine were performed according to the method described by Clark et al. [17] to ensure accurate concentration effectiveness measurements were obtained. Concentration effectiveness measurements will be presented in this paper for the locations shown in Fig. 3, namely (A) the front rim seal, (B) the outer radius of the front rim cavity, (C) the purge hole radius of the front rim cavity, and (D) the front wheel-space.

An uncertainty analysis was performed for the facility and turbine measurements reported in this paper according to the method of Figliola [23]. The total uncertainties for these measurements are reported in Table 1, which included both the bias and precision uncertainties for each measurement. The main measurement reported in this paper is concentration effectiveness. The bias uncertainty was minimized by using two concentration ranges and calibration gases on the gas analyzer. The precision uncertainty was minimized by taking a 60 s average of the signal from the gas analyzer (approximately 40k samples). This method resulted in a total uncertainty in concentration effectiveness of \(\pm 0.015\) to \(\pm 0.02\) over the entire range. The purge flow rate was directly measured, yielding the accuracy in Table 1. Several leakage flows in the turbine will be presented later, and their flow rates were inferred and computed from the measurements presented in this paper. The uncertainty of the leakage flow rates was not computed.

Operating Conditions. Effectiveness measurements will be presented in this paper for two purge hole configurations: (1) 150 purge holes and (2) 32 purge holes. The 150 purge hole configuration was tested up to a flow rate of \(\Phi/\Phi_{\text{ref}} \leq 1.5\), where \(\Phi_{\text{ref}}\) was the reference flow rate defined as the flow rate at which the front rim cavity (locations B and C in Fig. 3) was fully sealed \((\eta \geq 0.99)\). The 32 purge hole configuration was tested up to \(\Phi/\Phi_{\text{ref}} \leq 0.6\) with the same reference flow rate as in the case with 150 purge holes so relative comparisons between the two cases can be made given only one scaling number. The experiments presented in this paper were operated at a purge to main gas path density ratio of 1.1–1.3, a blade inlet relative Mach number of 0.2, a blade inlet axial Reynolds number of \(1.4 \times 10^5\), and a rotational Reynolds number of \(3.8 \times 10^5\) as defined in the nomenclature.

<table>
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<tr>
<th>Table 1 Uncertainty in facility and turbine measurements</th>
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<tr>
<td>Parameter</td>
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<tr>
<td>Main gas path flow rate, (\Phi/\Phi_{\text{ref}})</td>
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<td>Shaft rotational speed, (\Omega/\Omega_{\text{ref}})</td>
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<td>Pressures, (p/p_{\text{ref}})</td>
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<td>Temperatures, (T)</td>
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<td>1.5 stage pressure ratio, (PR/PR_{\text{ref}})</td>
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<td>Purge flow rate, (\Phi/\Phi_{\text{ref}})</td>
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<td>Concentration effectiveness, (\eta)</td>
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Engine rotational Reynolds numbers are typically in the range of 2–3 × 10⁷ [24].

Facility and Turbine Benchmarking

This section briefly describes the benchmarking that was performed for the 1.5 stage turbine, including the turbine operating conditions, the first and second vane aerodynamic loadings, and the front cavity boundary conditions. Clark et al. [17,18] previously described benchmarking for the facility for the half stage turbine.

Facility Operation. The facility exhibited stable and repeatable steady-state operation. During a typical test day, the facility was operated for 8–12 h at various conditions. Figure 4 shows the facility flow conditions for a 2.5 h window of a typical test. The turbine inlet mass flow rate, shaft rotational speed, turbine inlet total pressure, turbine inlet total temperature, and the 1.5 stage pressure ratio shown in Fig. 4 were all scaled by reference values. All of these parameters were held constant for this test point. Figure 4 shows that the facility, including the compressor, flow control valves, and dynamometer, exhibited long-term steady-state operation. For the experiments presented in this paper, the main gas path conditions, including 1.5 stage pressure ratio and corrected speed, were all held constant.

Although effectiveness measurements are not presented in this paper for the aft cavity, it was important to maintain the aft cavity flow conditions constant for all of the experiments to ensure consistent operating conditions. The front cavity purge flow rate was set to various conditions throughout testing, and the turbine inlet flow rate was then set to match the desired 1.5 stage pressure ratio. The aft cavity flow rate was determined from the difference between the turbine exit mass flow rate and the turbine inlet plus the front cavity purge flow. Several leakage flows, which will be discussed in more detail later, entered the aft cavity as in an operational engine. As discussed previously, the magnetic bearing cooling air also entered the aft cavity. These flows were accounted for, and the aft cavity egress flow rate was held constant at 0.55 < ϕ / ϕ<sub>ref</sub> < 0.65 for all test conditions presented in this paper.

Turbine Operation. Turbine measurements indicated both repeatability and uniformity in the test section. The first vane aerodynamic loading at 50% span is shown in Fig. 5(a), and the second vane aerodynamic loading at 50% span is shown in Fig. 5(b). The static pressures on the first vane airfoil surface in Fig. 5(a) were normalized by the turbine inlet total pressure, and the static pressures on the second vanes in Fig. 5(b) were normalized by the second vane inlet total pressure as measured at the stagnation point on the second vane leading edge. The solid line represents pretest prediction CFD simulations. Different colored symbols represent measurements from different test days and different symbols represent measurements obtained on separate vanes.

The agreement between the measurements obtained on different vanes for a given test day showed circumferential uniformity for both the first and second vanes. The measurements from different test days indicated good repeatability. Although not shown here, measurements at 10% and 90% span on both vanes also indicated uniformity and repeatability. The agreement between the CFD and the measurements on the first vanes indicated the flow through the first vanes operated as intended, and that the additively manufactured first vanes were built to specifications. The agreement of the measurements for different test days, different vanes, and with the CFD shown in Fig. 5(a) for the first vanes was similar to that shown for the half-stage (vane only) turbine as described by Clark et al. [18]. The pretest prediction CFD and data did not agree as well on the second vanes, but the agreement was sufficient to instill confidence that the second vanes were manufactured to specifications, thus allowing the researchers to proceed with the effectiveness experiments.

The boundary conditions measured at the front rim seal also indicated periodicity in the turbine. Figure 6 shows the static pressures measured on the first vane trailing edge at the endwall, on the platform trailing edge, and in the rim seal, all of which are normalized by the turbine inlet total pressure. Different colored symbols correspond to the measurement locations shown in the inset image. The measurements were obtained with no purge flow. The pressure taps used to obtain these measurements were located...
on two vanes at different circumferential locations as shown by different symbol types. The static pressure measurements on different vanes agreed with each other indicating periodicity. Similar to the same measurements found for the vane only configuration [18], the vane exit pressure distortion was observed. The high pressure region near $\theta = 18$ deg indicated where ingested flow is classically thought to be driven into the rim seal, and the low pressure regions indicated where flow is classically thought to egress into the main gas path. The difference between the maximum and the minimum static pressure was $p/p_{in} = 0.07$ for the measurements at the vane trailing edge. At the trailing edge of the platform, the pressure difference was strongly attenuated to $p/p_{in} = 0.025$, and in the rim seal, the difference was completely attenuated.

The measurements shown in Fig. 6 show that the same time-averaged boundary conditions that existed on the rim seal for the vane only configuration described by Clark et al. [18] were also present for the 1.5 stage configuration presented in this paper. To study ingestion for this 1.5 stage configuration, several additional effects were included in addition to the vane exit pressure field included in the vane only study: rotational effects such as disk pumping, the potential field upstream of the blade, and the unsteady interactions between the vane exit pressure and blade potential field. The main gas path pressure ratio and the corrected rotational speed were held constant for all measurements presented in this paper, and the addition of purge flow had a negligible effect on the pressure measurements shown in Fig. 6.

Sealing Effectiveness for Purge Flow

Concentration effectiveness measurements in the front wheel-space, front rim cavity, and front rim seal are presented in this section for two purge flow configurations: (1) 150 purge holes and (2) 32 purge holes. The effectiveness data are presented versus the scaled purge flow rate $\Phi/\Phi_{ref}$, where $\Phi$ is the purge flow rate and $\Phi_{ref}$ is the reference sealing flow rate defined as the purge flow rate at which the rim cavity was fully sealed for the 150 purge hole configuration (see locations B and C in Fig. 3). Thus by definition, the rim cavity effectiveness of unity corresponds to a purge flow rate of $\Phi/\Phi_{ref} = 1$. The variation in concentration effectiveness with purge flow rate and location is shown for both 150 purge holes and 32 purge holes in Fig. 7. The data shown for both configurations in Fig. 7 were measured at several circumferential positions and averaged circumferentially for each radial location in Fig. 7. Although the data for 150 purge holes exhibited negligible circumferential variation, the 32 purge holes showed a periodic variation in effectiveness, which will be discussed at the end of this section. Figures 8 and 9 are also presented for both configurations to provide a physical understanding of the data being presented.

**Configuration 1: 150 Purge Holes.** In this section, we will discuss the measurements in Fig. 7 for 150 purge holes. The measurements for 32 purge holes will be discussed in the Configuration 2: 32 Purge Holes section. The effectiveness measurements for 150 purge holes in Fig. 7 showed that effectiveness increased with decreasing radius and with increasing purge flow. The highest effectiveness was observed in the front wheel-space (location D), which was fully sealed for $\Phi/\Phi_{ref} = 0.9$, or 10% less than the reference flow rate. Farther outboard in the rim cavity (locations B and C), the effectiveness was slightly lower as more ingestion occurred for a given flow rate than for the front wheel-space. Both
locations in the rim cavity were fully sealed for a purge flow rate of \( \Phi / \Phi_{\text{ref}} = 1 \) according to the definition of \( \Phi / \Phi_{\text{ref}} \). The lowest effectiveness was observed near the main gas path in the rim seal (location A), which was fully sealed for \( \Phi / \Phi_{\text{ref}} = 1.5 \). Due to the high ingestion rates from the main gas path, 50% more flow was required to fully seal the rim seal than the rim cavity. Each of these locations was discussed in more detail in section 3.3.2, starting with location D and moving radially outward to location A.

A major result was that there was appreciable ingestion in the front wheel-space (location D in Fig. 7) for the 150 hole configuration at low purge flows. The front wheel-space was isolated from the rim cavity by a two-stage labyrinth seal. Despite the small clearances on the labyrinth seal, there was still appreciable ingestion that occurred over a wide range of purge flow rates. For example, at \( \Phi / \Phi_{\text{ref}} = 0.1 \), the effectiveness was \( \varepsilon_c = 0.7 \), and at \( \Phi / \Phi_{\text{ref}} = 0.5 \), the effectiveness was \( \varepsilon_c = 0.9 \). Although these effectiveness values were relatively high values compared to the other data in Fig. 7, it was expected that the front wheel-space should be fully purged. For location D, it was not until a purge flow of \( \Phi / \Phi_{\text{ref}} = 0.9 \) that the front wheel-space was fully sealed. If ingestion occurred deep within the turbine wheel-spaces in an engine, then the disk would heat up with potentially catastrophic effects; hence, it is of the utmost importance to fully seal the front wheel-space.

To explain why there was appreciable ingestion in the front wheel-space (location D in Fig. 7), we shall examine the flow schematic in Figs. 8 and 9. The test turbine was an engine-wheel-space (location D in Fig. 7), we shall examine the flow in the rim cavity (location B in Fig. 7) for the 150 hole configuration at low purge flows. The front wheel-space was isolated from the rim cavity by a two-stage labyrinth seal. Despite the small clearances on the labyrinth seal, there was still appreciable ingestion that occurred over a wide range of purge flow rates. For example, at \( \Phi / \Phi_{\text{ref}} = 0.1 \), the effectiveness was \( \varepsilon_c = 0.7 \), and at \( \Phi / \Phi_{\text{ref}} = 0.5 \), the effectiveness was \( \varepsilon_c = 0.9 \). Although these effectiveness values were relatively high values compared to the other data in Fig. 7, it was expected that the front wheel-space should be fully purged. For location D, it was not until a purge flow of \( \Phi / \Phi_{\text{ref}} = 0.9 \) that the front wheel-space was fully sealed. If ingestion occurred deep within the turbine wheel-spaces in an engine, then the disk would heat up with potentially catastrophic effects; hence, it is of the utmost importance to fully seal the front wheel-space.

The deep penetration of the hot gas into the cavity (location D) has major implications for the engine. These effectiveness measurements in Fig. 7 for the front wheel-space highlight the need to provide flow directly to the front wheel-space. The TOBI flow, as shown in Fig. 2, is designed to pressurize the front wheel-space and cause the leakage across the labyrinth seal to flow into the rim cavity (in the opposite direction indicated in Fig. 8), thereby minimizing the ingestion inboard of the seal. Results will be presented in the future that include the TOBI flow. If the seal clearances in an engine were too large due to excessive wear or a poor design, then the pressure in the wheel-space may not build sufficiently, even with the TOBI flow, and ingestion past the labyrinth seal could occur. These measurements show the importance of (1) accurately setting the labyrinth seal clearances, (2) maintaining the engines, and (3) providing sufficient TOBI flow, otherwise ingestion could occur deep within the turbine wheel-spaces with potentially catastrophic effects.

The effectiveness in the front wheel-space (location D in Fig. 7) at the purge hole radius in the rim cavity (location C) exhibited behavior that deviated from the previously measured effectiveness curves [11,21,25], like that shown for the outer radius in the rim cavity (location B). The data shown for location B were similar to the previous effectiveness curves for which the orifice model was derived, and it will be shown later that the model fits the data well. The effectiveness data for locations C and D were quite different from location B, and it will be shown that the orifice model did not fit the data for locations C and D. Compared to location B, the effectiveness for locations C and D increased sharply at low flow rates for \( \Phi / \Phi_{\text{ref}} < 0.2 \). At \( \Phi / \Phi_{\text{ref}} = 0.2 \), the effectiveness leveled off, then began to gradually increase again to fully sealed conditions. Although both locations C and D exhibited similar qualitative behavior with a sharp increase in effectiveness for low flow rates, the effectiveness for location C was slightly lower than location D. For example, at a low flow rate of \( \Phi / \Phi_{\text{ref}} = 0.2 \), effectiveness was \( \varepsilon_c = 0.76 \) at location C and \( \varepsilon_c = 0.83 \) at location D. As the purge flow rate increased, effectiveness for location C gradually increased to unity.

The sharp bend in the effectiveness curve for locations C and D at \( \Phi / \Phi_{\text{ref}} = 0.2 \) was similar to that observed by Clark et al. [18] for the case with no rotation and was attributed to the behavior of a jet-in-crossflow. Recall that the purge flow entered the rim cavity through axially oriented holes normal to the cavity swirling flow. At low purge flow rates, the jet momentum was low, which caused the purge flow to be entrained in the axial flow across the cavity to the rotor side, as shown in Fig. 9(a) at low flow rates. At higher flow rates, the jet momentum was higher, which caused the purge flow to be entrained in the axial flow across the cavity to the rotor side, as shown in Fig. 9(b) at high flow rates. Note that as the purge flow rate increased, the effectiveness at the purge hole radius (location C) approached that at the outer radius (location B). As the purge flow rate increased beyond \( \Phi / \Phi_{\text{ref}} > 0.55 \), the purge jets were likely separated from the stator side of the rim cavity. The purge flow was entrained axially across the rim cavity, as shown in Fig. 9(b). The purge flow was then pumped radially outward on the rotor and then fed the recirculating flow back to the stator side resulting in similar effectiveness at both locations B and C.

The effectiveness at the outer radius in the rim cavity (location B in Fig. 7) was lower than both the purge hole radius (location C) and the wheel-space (location D). The effectiveness measurements at location B displayed a characteristic curve that was similar to those measured by the previous researchers [11,21,25], with a monotonic increase in effectiveness as flow rate increased and an exponential asymptote to an effectiveness of unity. As will be shown in the Empirical Modeling section, the effectiveness data at the outer radius in the rim cavity were well suited to empirical modeling.

In the rim seal (location A in Fig. 7), the effectiveness was zero for \( \Phi / \Phi_{\text{ref}} \leq 0.2 \). Despite providing 20% of the flow required to seal the rim cavity, there was no appreciable effect in the rim seal. The flow schematic of the secondary flows, given in Fig. 8, will be used to discuss this phenomenon. Figure 8 shows a leak across the disk from the front rim cavity to the aft rim cavity. This particular flow passed through the gaps between the blades. The reason for the zero effectiveness in the rim seal was because a significant amount of purge flow, approximately 20% of \( \Phi_{\text{ref}} \), was lost from the rim cavity to feed the blade gap leakage and did not reach the rim seal. Once the blade gap leakage was fully satisfied by the purge flow in the front rim cavity for \( \Phi / \Phi_{\text{ref}} \approx 0.2 \), the effectiveness in the rim seal increased with purge flow rate. As the purge flow rate increased, the effectiveness in the rim seal increased in a nearly linear fashion up to the fully sealed condition. Even though the rim cavity (locations B and C) was fully sealed at \( \Phi / \Phi_{\text{ref}} = 1.0 \), the effectiveness in the rim seal (location A) at the same flow rate was only \( \varepsilon_c = 0.8 \), and for fully sealed conditions, the rim seal required 50% more flow than the rim cavity.

Configuration 2: 32 Purge Holes. In this section, we will discuss the effectiveness measurements for 32 purge holes that are also presented in Fig. 7. The same \( \Phi_{\text{ref}} \) was used to scale the data for 32 purge holes as 150 purge holes to provide a direct comparison between both configurations. As can be seen from the data, the 32 purge holes did not provide enough flow to fully purge the wheel-space, rim cavity, or rim seal for the flow rate range-over which the tests were conducted. Effectiveness measurements for 32 purge holes were obtained for a purge flow rate of \( \Phi / \Phi_{\text{ref}} < 0.6 \), and beyond that flow rate, the pressure ratio across the purge holes was higher than could be expected in an engine.

The effectiveness measurements for 32 purge holes in Fig. 7 again showed that effectiveness increased with decreasing radius.
and with increasing purge flow. In the front wheel-space (location D), the effectiveness was higher than the other locations for all flow rates. The effectiveness also showed a sharp increase in effectiveness at the low flow rates, followed by a more gradual increase in effectiveness as the purge flow rate increased. At the purge hole radius (location C), the effectiveness also exhibited an increase in effectiveness at lower flow rates due to the jet-in-crossflow behavior. The effectiveness at the purge flow rate (location C) approached the effectiveness at the outer radius (location B) as purge flow rate increased. At the outer radius in the rim cavity (location B), the effectiveness characteristic was again similar to those measured by the previous researchers [11,21,25]. In the rim seal (location A), the effectiveness showed zero effectiveness for \( \Phi/\Phi_{ref} \leq 0.16 \), but for \( \Phi/\Phi_{ref} > 0.16 \) the effectiveness increased with flow rate.

Circumferential Variation in Effectiveness. The effectiveness in the rim cavity was found to be circumferentially uniform for 150 purge holes, so effectiveness with circumferential position for 150 purge holes is not shown here for the sake of brevity. The circumferential uniformity was observed both at the purge hole radius (location C) and at the outer radius (location B) of the rim cavity. For 150 purge holes, the distance between the holes was circumferentially spaced less than 4D, so the purge flow entered the rim cavity in a very uniform manner. The circumferentially uniform purge flow in the rim cavity was similar to that previously observed [18].

The circumferential spacing between 32 purge holes was higher than for 150 purge holes, at approximately 16D, which resulted in a circumferential variation in effectiveness in the rim cavity at the purge hole radius. The effectiveness in the front rim cavity as a function of circumferential position is shown in Fig. 10 for four purge flow rates for the 32 purge hole configuration. As indicated in the image, the purge holes were located at \( \theta = 4 \) deg and 15 deg, and the swirl flow produced by the vane was from left to right. At the lowest flow rate of \( \Phi/\Phi_{ref} = 0.05 \), the effectiveness at the purge hole radius (location C) varied greatly with circumferential location. At \( \theta = 13 \) deg, the effectiveness was at \( e_r = 0.25 \), but just downstream of the purge hole at \( \theta = 18 \) deg the effectiveness increased by 200% to \( e_r = 0.75 \). This increase was followed by a decay in effectiveness to \( e_r = 0.3 \) at \( \theta = 23 \) deg. For the same flow rate of \( \Phi/\Phi_{ref} = 0.05 \), the effectiveness at the outer radius (location B) was mostly constant, with a slight increase in effectiveness observed from 13 deg to 23 deg. For a slightly higher purge flow rate of \( \Phi/\Phi_{ref} = 0.1 \), the effectiveness at location C exhibited a similar trend as at the lowest flow rate, but at a reduced level. Effectiveness at \( \theta = 13 \) deg was \( e_r = 0.4 \), followed by a 50% increase to \( e_r = 0.6 \) at \( \theta = 18 \) deg just downstream of the purge holes. Farther downstream at \( \theta = 23 \) deg, the effectiveness again decayed to \( e_r = 0.45 \), which was again slightly higher than what was measured at \( \theta = 13 \) deg. At location B, the effectiveness remained constant near \( e_r = 0.3 \) for \( \Phi/\Phi_{ref} = 0.1 \). For a higher purge flow rate of \( \Phi/\Phi_{ref} = 0.3 \), the effectiveness at locations B and C was a constant \( e_r = 0.6 \) for all circumferential locations. Similarly at \( \Phi/\Phi_{ref} = 0.5 \), a constant effectiveness of approximately \( e_r = 0.75 \) was observed for all circumferential locations.

The effectiveness trends versus circumferential position observed for 32 purge holes at the purge hole radius (location C) were consistent with the behavior of a jet-in-crossflow. At low flow rates or low jet momentum, a jet-in-crossflow would exhibit higher effectiveness just downstream of the hole with a decay in effectiveness with increasing distance from the hole. This same behavior was observed by Clark et al. [18] at the purge hole radius for 16 purge holes in a stationary rim cavity study. At low purge flow momentum, the effectiveness data were shown to increase dramatically downstream of the purge hole, but at high momentum the effectiveness was mostly uniform.

The jet-in-crossflow behavior in the data shown in Fig. 10 did not affect the outer radius in the front rim cavity (location B). The effectiveness was mostly circumferentially uniform, which indicated that the flow was pumping radially inward on the stator side as indicated in Figs. 9(c) and 9(d). If the flow on the stator side were pumping radially outward, then the same trends observed at the purge hole radius (location C) could be expected at the outer radius (location B), but this was not the case.

Comparison of Configurations 1 and 2. At the purge hole radius (location C) and in the front wheel-space (location D), the effectiveness was considerably affected by the number of purge holes. The effectiveness was higher for 150 purge holes than 32 purge holes for locations C and D over the entire flow range but exhibited the largest difference for \( 0.1 \leq \Phi/\Phi_{ref} \leq 0.4 \). At location C, the effectiveness was between \( e_r = 0.1 \) and 0.25 greater for 150 purge holes than for 32 purge holes, and at location D, the effectiveness increase was slightly less at \( e_r = 0.1 \) and 0.18 greater. Clearly, the number of purge holes had an effect on the sealing effectiveness at and inboard of the purge holes, with 150 holes producing higher effectiveness than 32 holes for a given purge flow rate.

As previously discussed, Figs. 9(a) and 9(b) show a flow schematic in the front rim seal and cavity for 150 purge holes at low and high flow rates, respectively. Figs. 9(c) and 9(d) show the schematic for 32 purge holes at the same flow rates. These schematics help to explain why 150 purge holes displayed higher effectiveness at and inboard of the purge hole radius than 32 purge holes. The arrows indicate the bulk flow directions, and the colors of the arrows represent the relative effectiveness of that flow with red indicating ingested flow and blue indicating purge flow. The radial inward pumping on the stator side pulled ingested flow into the rim cavity from the rim seal at low flow rates for both cases, as shown in Figs. 9(a) and 9(c). For 32 purge holes, there was significant spacing between the purge holes (~16D) for the pumped flow to penetrate between the holes farther into the rim cavity, thus lowering the concentration effectiveness, as shown in Figs. 9(c). This lower concentration air also fed the leak across the labyrinth seal to the front wheel space, as shown in Fig. 8, leading to lower effectiveness there. The holes were much more closely spaced for the 150 purge holes (less than 4D), which entrained the radially inward pumped flow from the stator into the axial flow across the cavity to the rotor, as shown in Figs. 9(a) and 9(b). The closely spaced purge holes prevented the ingested flow from being pumped inward past the purge holes, which allowed more of the purge flow to be pumped radially inward, as shown in Figs. 9(a) and 9(b). The purge flow fed the region inboard of the 150 purge holes, as shown in Fig. 8, which also fed the labyrinth seal leakage, thus increasing effectiveness in the wheel-space (location D). For 32 purge holes, the region inboard of the purge

Fig. 10 Circumferential variation in concentration effectiveness for 32 purge holes

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holes was fed by less purge flow and more ingested flow at low flow rates, as shown in Fig. 9(c), which resulted in lower effectiveness at location D. At high flow rates, the stator side flow penetrated inward past the purge holes, but there was higher effectiveness than at low flow rates because the purge flow that moved radially outward on the rotor was recirculated back onto the stator side, as shown in Fig. 9(d).

At the outer radius in the rim cavity (location B in Fig. 7), the effectiveness measurements for both configurations were very similar, indicating that the number of purge holes did not affect the concentration effectiveness measurements at this location. Likewise, in the rim seal (location A), the number of purge holes had a negligible effect on the concentration effectiveness measurements. It can thus be concluded that the number of purge holes had a minimal effect on the sealing effectiveness on the stator side of the cavity outboard of the purge holes.

The flow field at the outer radius of the rim cavity is also shown schematically in Fig. 9 for both low and high flow rates for both configurations. The purge flow entered the rim cavity through axially oriented holes and was entrained axially across the cavity to the rotor, which fed the rotor boundary layer. Part of this rotor boundary layer flow was lost through the blade gap leakage before entering the rim seal as shown in Fig. 9. At low flow rates, most of the flow on the rotor was lost through the blade gap leakage, as shown in Figs. 9(a) and 9(c), and no purge flow ended up in the rim seal. At high flow rates, a portion of the flow on the rotor passed through the blade gap, but most of the flow recirculated back into the cavity or egressed into the rim seal, as shown in Figs. 9(b) and 9(d). The flow rate at which the effectiveness in the rim seal crossed zero was slightly different between both configurations, with 150 purge holes crossing at $\Phi/\Phi_{ref} = 0.2$ and 32 purge holes crossing at $\Phi/\Phi_{ref} = 0.16$.

**Empirical Modeling.** Theoretical models for hot gas ingestion have traditionally been based on an orifice assumption, where the rim seal was assumed to be an orifice with a discharge coefficient for ingress, $C_{d_i}$, and a discharge coefficient for egress, $C_{d_e}$. Several orifice models have been developed with varied success [2,6,7,26,27]. This section compares the experimental data to such an orifice model presented by Owen et al. [2] for externally induced ingress. The model given in Eq. (1) is a simple, yet powerful method for modeling sealing effectiveness as it requires no inputs beyond the minimum sealing flow rate and the ratio of discharge coefficients, which can be empirically determined. Specifically, the relationship between sealing flow rate and effectiveness is given by

$$\frac{\Phi^*}{\Phi_{min}} = \frac{c}{1 + \Gamma_c^{-2/3} (1 - c)^{2/3}}^{3/2}$$

(2)

where $\Gamma_c$ is the ratio of the ingress and egress discharge coefficients, $C_{d_i}/C_{d_e}$, $c$ is the sealing effectiveness, and $\Phi_{min}$ is the minimum flow required to fully seal that location. There is a slight modification here to the original equation presented by Owen et al. [2], and that is the definition of $\Phi^*$, which is the net sealing flow rate given by

$$\Phi^* = \Phi - \Phi_0$$

(3)

where $\Phi_0$ is defined here as the value of $\Phi$ at which effectiveness crosses zero. Note that for both of the rim cavity locations $\Phi_0/\Phi_{ref} = 0.2$ for the data presented in Fig. 7. For the rim seal location $\Phi_0/\Phi_{ref} = 0.16$ for 150 purge holes and $\Phi_0/\Phi_{ref} = 0.16$ for 32 purge holes as shown in Fig. 7. Accounting for the zero-crossing is an important part of modeling the secondary air system for a gas turbine, especially for an engine-realistic geometry where several sources, sinks, and leakages may be present. Converting the gross flow rate, $\Phi$, into a net flow rate, $\Phi^*$, allowed the model to account for the blade gap leakage and the rim seal source flow in this 1.5 stage turbine. Since the front and aft rim cavity pressures remained constant for these experiments even as purge flow rate varied, the blade gap leakage was assumed to be constant.

Figure 11 shows the effectiveness data at the purge hole radius (location C), at the outer radius (location B), and in the rim seal (location A) plotted against $\Phi^*/\Phi_{min}$ for 150 purge holes, and Fig. 12 shows the data for 32 purge holes. The lines represent the models for each data set as indicated in Figs. 11 and 12. The effectiveness data were used to determine the best fit for the ratio of discharge coefficients, $\Gamma_c$, using Eq. (2). Although Zhou et al. [28] recommended a minimum of 16 data points to determine $\Gamma_c$ with high accuracy, the discussion that follows examines the major trends in $\Gamma_c$. Figures 11 and 12 also show the empirically determined values of $\Gamma_c$ for each data set.

In the rim seal (location A shown in Figs. 11 and 12), the model fits the data well over most of the flow rate range. At higher flow rates, from $0.7 < \Phi^*/\Phi_{min} < 0.9$, the effectiveness data for 150 purge holes were $\sim 0.1$ higher than the model, as shown in Fig. 11. The data and empirical model were nearly linear for both configurations, which resulted in a large value for $\Gamma_c$. For 150 purge holes $\Gamma_c = 5.2$, as noted in Fig. 11, and for 32 purge holes $\Gamma_c = 8.0$, as noted in Fig. 12. The effectiveness for 32 purge holes never fully reached unity so $\Phi_{min}$ for the 150 holes configuration was used in the model for both configurations. The high values of $\Gamma_c$ indicated that the ingress discharge coefficient was much lower than the egress coefficient.
higher than the egress discharge coefficient, thus promoting significantly more ingress than egress flow.

As Owen et al. [2] explained, their model uncoupled the pressure difference in the main gas path from the effectiveness, which had the effect of changing the characterization of $\Gamma_c$ compared to previous models. The parameter $\Gamma_c$ presented by Owen et al. [2] empirically included the effects of the pressure difference in the main gas path, while the previous models required the pressure difference as an input. Very high values of $\Gamma_c$ were shown in Fig. 12. As the flow rate increased, the data approached the effectiveness characteristic at the outer radius in the rim cavity (location B), and for $\Phi'/\Phi_{\min} > 0.55$, the data fit the characteristic of $\Gamma_c = 0.98$ shown at location B very well. An order of magnitude variation in the characteristic $\Gamma_c$ over such a small range has not been observed in the literature previously, and the manner through which the purge flow was introduced and the momentum of the purge flow were the causes. The importance of the purge flow momentum was shown previously by Clark et al. [18] for a half-stage turbine with a stationary rim cavity. The momentum of the purge jet, $p_{\text{jet}}V_{\text{jet}}^2$, at the higher value of $\Gamma_c$ at $\Phi'/\Phi_{\min} = 0.55$ was 4.4 to 6.2 times greater than the purge jet momentum at the lower value of $\Gamma_c$ at $\Phi'/\Phi_{\min} = 0.2$.

The data for 32 purge holes at location C in Fig. 12 showed similar behavior as 150 purge holes, although to a reduced degree. At low flow rates, for $\Phi'/\Phi_{\min} < 0.15$, the model seemed to fit the data again with a lower characteristic of $\Gamma_c = 0.25$ as shown in Fig. 12. As the flow rate increased, the data approached the model at location B, and for $\Phi'/\Phi_{\min} > 0.3$, the characteristic changed to $\Gamma_c = 1.25$. The characteristic value of $\Gamma_c$ changed by a factor of five due to the momentum of the purge jets. Again, the momentum of the purge jets was approximately 3.6–5.3 times greater at the higher value of $\Gamma_c$ at $\Phi'/\Phi_{\min} = 0.3$ than at the lower value of $\Gamma_c$ at $\Phi'/\Phi_{\min} = 0.15$.

Conclusions

This paper has presented sealing effectiveness measurements in the front cavity of a 1.5 stage turbine, with engine-realistic airfoils and cavity geometries, operated at engine-relevant Reynolds and Mach numbers. Sealing effectiveness measurements were acquired through the use of CO2 as a tracer gas. Benchmarking of the facility indicated steady-state operation as well as periodic and repeatable conditions in the 1.5 stage turbine.

Sealing effectiveness increased with purge flow rate and with radial distance from the main gas path. Less purge flow was required to produce fully sealed conditions in the rim cavity than in the rim seal. Despite showing higher effectiveness in the wheel-space than in the rim cavity, appreciable ingestion was shown to occur in the wheel-space for low flow rates. These data indicate that ingestion deep within turbine cavities in an operating gas turbine would result in reduced component lifetimes, or possibly catastrophic failures, highlighting the need to provide sufficient TOBI flow to purge the front wheel-space in an operating engine. Of the two purge flow configurations tested, the configuration including more purge holes resulted in higher sealing effectiveness than fewer purge holes inboard of the purge flow injection location. For the larger number of uniformly distributed purge holes, a fully sealed cavity was possible, while for the smaller number of purge holes, a fully sealed cavity was not possible. The results also showed the importance of including the
complexity of the geometry and understanding all the various leakage paths in the turbine.

An orifice model was compared to the sealing effectiveness data with mixed results. It is important to note that the orifice model was developed for simplified geometries, while engine hardware has been developed to be much more complex to avoid hot gas ingestion. The model matched the effectiveness data at the outer radius in the rim cavity and in the rim seal, but did not match the data at the purge flow injection location. The model’s inability to predict the effectiveness is most likely due to the complexity of the geometry and purge flow delivery method. The results suggest that orifice models may work well for matching sealing effectiveness data for some cases, but the models break down for engine-realistic geometries and purge flow delivery as shown in this paper.

Acknowledgment

The authors would like to thank Pratt & Whitney and the Department of Energy National Energy Technologies Laboratory for sponsoring the research presented in this paper, and the Penn State Applied Research Laboratory for their assistance with facility work, including PLC programming. This document has been publicly released.

Nomenclature

- $b$ = hub radius
- $c$ = gas concentration
- $C_{d_1}, C_{d_3}$ = discharge coefficient for ingress, egress
- $C_{s_B}$ = blade axial chord length
- $D$ = diameter
- $m$ = mass flow rate
- $p$, $p_t$ = static, total pressure
- $PR$ = pressure ratio, $p_{t_{in}}/p_{t_{ex}}$
- $Re_{d_m}$ = blade inlet axial Reynolds number, $V_{d_m}C_{s_B}/v$
- $Re_{B_m}$ = rotational Reynolds number, $\Omega B^2/v$
- $S$ = seal clearance
- $S/\text{max}$ = percent wetted surface area on airfoils
- $T$ = temperature
- $V$ = velocity
- $\Gamma$ = ratio of discharge coefficients, $C_{d_1}/C_{d_3}$
- $\epsilon_c$ = concentration effectiveness $(c - c_{m_0})/(c - c_{m_0})$
- $\theta$ = circumferential direction
- $\nu$ = kinematic viscosity
- $\phi$ = sealing flow rate, $m/(2\pi r_p B^2)$
- $\phi_{min}$ = flow rate at which effectiveness crosses zero
- $\phi_{max}$ = minimum flow rate for particular location
- $\phi_{ref}$ = reference flow rate, $\phi_{max}$ for front cavity
- $\phi'$ = net sealing flow rate, $\phi - \phi_{0}$
- $\Omega$ = shaft rotational speed

Subscripts and Abbreviations

1V, 2V = First, second vane
AM = additive manufacturing
B = blade
in, ex = turbine inlet, exit
jet = purge flow jet
min = minimum
ref = reference
rel = relative
sec = secondary air supply
TOBI = tangential on-board injection (prespurled sealing air)
$\infty$ = main gas path, freestream

References