Investigation of Sand Blocking Within Impingement and Film-Cooling Holes

Gas turbines are not generally designed for operation with a particle laden inlet flow but, in fact, are commonly operated in unclean environments resulting in dirt, sand, and other debris ingestion. In addition to the negative effects within the main gas path, for aeroengines these particles are pulled into the coolant system where they can clog cooling passages and erode internal surfaces. Unlike previous research that focused on deposition and erosion within the main gas path, this study evaluated blocking in a double wall liner whereby both impingement and film-cooling holes were simulated. Double wall liners are commonly used in the combustor and turbine for combined internal and external cooling of metal components. Specifically, sand blockages were evaluated through comparisons of measured flow rates for a particular pressure ratio across the liner. Four liner geometries were tested whereby the coolant hole size and orientation were varied in test coupons. At ambient temperature, blocking was shown to be a function of the impingement flow area. A significant rise in blocking was observed as sand and metal temperatures were increased. The overlap between the impingement and film-cooling holes was also found to have a significant effect. [DOI: 10.1115/1.3106702]

1 Introduction

Figure 1 shows two instances of gas turbines operating in dust-laden environments. Complete filtration of the intake air is often deemed infeasible because of the associated pressure drop, filter replacement and cleaning requirements, and overall increase in engine weight. Particle ingestion into a gas turbine can have serious effects on both performance and engine service intervals. While passing through the engine, ingested debris collides with and subsequently erodes the metal surfaces. These particles, which are also pulled into the coolant air bypass, clog the internal channels thereby reducing coolant mass flow and increasing part temperature. Elevated temperatures within the engine can melt the sand, further increasing the likelihood of deposition and blocking within the cooling channels. Two commonly used cooling techniques within a gas turbine are impingement and film-cooling (FC). Impingement cooling is used to cool parts from the inside, as this type of flow would be aerodynamically disruptive within the main gas path. Film-cooling is often employed on the external surface by providing a coolant film along the exterior.

For this study, the combination of impingement and film-cooling holes, often referred to as a double wall liner, was subjected to a sand-laden coolant stream. Measurements of the coolant flow at a given pressure ratio (PR) were made to deduce the reduction in flow that would occur in an engine under sand-laden coolant conditions. The following parameters were investigated: liner geometry, pressure ratio, entering sand temperature, sand amount, and metal temperature. This paper also contains a literature review and description of experimental methods in addition to the experimental results performed at both ambient and engine temperatures.

2 Relevant Past Studies

As stated previously, ingestion of foreign particles can adversely affect a turbine’s performance and lifecycle. The past studies review begins with previous research focusing on actual incidents of in-service turbine hardware experiencing debris ingestion. It is followed by a review of simulated experimental and computational studies on the effects of particle ingestion relevant to gas turbine applications.

2.1 Research Motivation. The most common instances of foreign particle ingestion for a commercial aircraft engine arise from entering a volcanic ash cloud. An early instance of this potentially disastrous situation was on the June 24th, 1982 encounter of British Airways 77 where a Boeing 747, powered by Rolls-Royce RB-211 engines, flew into the eruption cloud of Mt. Galunggung near Indonesia [2]. Upon entering the cloud, the aircraft was forced into an emergency landing after all four engines experienced a temporary flameout. Upon investigation after landing, significant erosion and deposition was observed, as shown in Fig. 2. This encounter, along with several others, prompted the aircraft community to address this new danger [4] and instruct pilots on tactics to avoid ash clouds and emergency procedures during an encounter [5]. The most well known instance of volcanic ash ingestion occurred on December 15th, 1989 when a Boeing 747-400, powered by GE CF6-80C2 engines, entered an ash cloud from Mt. Redoubt volcano near Anchorage, AL [6]. All engines experienced a flame-out after being throttled up in an attempt to climb above the debris. After numerous restart attempts and losing more than 10,000 ft in altitude, the crew was finally able to restart the engines and make an emergency landing. Repairs to the aircraft totaled more than 80 million dollars, which included the total replacement of all four engines. These and other instances of aircraft traversing through volcanic dust clouds prompted studies on the effects of ash ingestion for gas turbine engines and also the development of satellite and radar systems to warn pilots of possible encounters. Despite these precautions, the ingestion of volcanic debris is still considered to be a significant danger to commercial airliners.

With regards to military aircraft, helicopter pilots in Desert Shield and Desert Storm were forced to come up with unorthodox operational procedures to help mitigate the negative effects of sand ingestion for virtually all helicopter platforms [7]. One such procedure was lifting the helicopter off before the recommended engine warm-up period and turning off the engines immediately after a landing. This reduced the total amount of sand being in-
gested into the engine while under load, which was primarily being stirred up by the rotors. Pilots also incorporated strict turbine washing regimes after each flight, which removed some of the deposits in the engine by flushing it with high pressure water.

2.2 Studies on Particle Ingestion. Several investigators have performed research as to the effects of particle ingestion with emphasis on gas turbines. The majority of these studies can be grouped into several categories: full-scale engine tests, accelerated erosion and deposition testing, and internal cooling blockage studies.

The earliest full-scale engine studies on dust ingestion were conducted by Batcho et al. [8] and Dunn et al. [9,10] in 1987. For their tests, several engines were operated with a dust-laden inlet stream while monitoring engine performance. After completion of testing, each engine was disassembled and the effects of sand ingestion were documented. They identified the following damage mechanisms that resulted in deterioration of engine performance: compressor erosion, deposition in the high pressure turbine, blockage of cooling holes in the high pressure turbine, and partial combustor fuel nozzle blockage. For their specific tests, however, the turbine inlet temperature was too low to result in sand glassification downstream of the combustor. While the concentration of sand ingested into the engine’s environmental control system (ECS) was monitored, no attention was given as to the effects of particles traveling through the coolant bypass and into the combustor and turbine internal cooling geometries.

As a result of the high cost associated with these types of engine tests, a hot section test system (HSTS) was developed by Kim et al. [11] to simulate the exit temperature and flow of a T56 can-type combustor and the more modern F100 annular combustor. Actual engine hardware was installed downstream of the HSTS and different dust blends were introduced upstream of the HSTS for each test. It was concluded that deposition was dependent on sand composition, sand concentration, turbine inlet temperature, and metal surface temperatures. Molten deposition was reported in the turbine when inlet temperature and metal temperature were above 1177°C and 816°C, respectively. An independent cooling system was utilized thereby supplying the turbine components with a dust-free coolant stream, as the study was focused only on the effects of a particle laden main flow through the engine. Another study using the HSTS was performed by Weaver et al. [12]. This study evaluated the effects of cooling hole diameter, hole roughness, turbine inlet temperature, and vane geometry with a dust-laden main flow. Test vanes included Inconel® 617 and Lamilloy film-cooled cylinders as well as F100-PW-220 first stage turbine vanes. Unlike previous studies, the coolant stream was modified to accommodate both clean and dust-laden flows. Inspection after testing showed that the cooling holes or passages had negligible clogging from dust while there was significant deposition on the external surfaces at elevated turbine inlet and metal temperatures.

Another subject of study pertaining to dust ingestion is the erosion caused by particle impacts with metal surfaces. Accelerated erosion studies at metal temperatures of up to 815°C have been presented in the literature [13] showing the effects of impingement angle, temperature, particle and eroded surface character, and impact velocity [14,15]. Recently the focus of this group has been investigating the effect of different turbine blade coatings on erosion rates and the resulting surface roughness characteristics. A study by Tabakoff and Simpson [16] compared uncoated turbine and compressor blades to a number of different coatings at elevated temperatures. After testing, the blade weight was taken and compared with the original value. A similar study of coated and uncoated turbine vanes was performed by Hamed et al. [17] with the primary focus being on the resulting vane surface roughness. All of these studies support the argument that particle ingestion causes material erosion and deposition on external compressor and turbine surfaces.

Other researchers have developed a high temperature accelerated deposition facility with emphasis on studying the characteristics and chemical composition of material deposits on turbine surfaces [18]. Jensen et al. [18] recognized the need for accelerated deposition studies as deposition formation could take as much as 25,000 operation hours in an industrial gas turbine. Bons et al. [19] also quantified the surface characteristics and roughness levels for in-service turbine blades and vanes to serve as a basis of comparison for their accelerated tests. The results of the investigation by Bons et al. identified the leading edge region as having the highest levels of external deposition for the evaluated turbine hardware. Another interesting finding of this study was the measurement of substantial deposition on in-service turbine vanes operated at a turbine inlet temperature below 900°C, which was substantially less than the 1177°C threshold for deposition previously stated by Kim et al. [11].

A review of the current research on erosion and deposition in turbomachinery was presented by Hamed et al. [20]. In their paper, the authors presented a review of erosion studies including numerical studies of particle trajectories and collisions for the main gas path. A summary of computational and experimental studies on the mechanisms of particle delivery and deposition was also presented. The authors also addressed the performance and lifecycle reductions, which are associated with erosion and deposition. The paper does not, however, address what effects erosion and deposition have on internal cooling geometries. This is because of the lack of experimental research on internal cooling blockage associated with dirty inlet air.

A numerical study of particles within a square channel having periodically spaced ribs was performed by Tafti and Shah [21]. For their study, 10 μm particles were shown to be more sensitive to large flow structures within the channel. Particle impacts on the rib surfaces were evenly distributed and impacts on the side walls were preferentially concentrated in the region of the top and bottom ribs. This was not true for the 100 μm particles, which showed significantly higher tendency to impact the upstream surface of the rib as well as the reattachment region behind the rib.

Walsh et al. [22] have currently performed the only experimental investigation within the literature directly focusing on the effects of a dust-laden coolant stream. Their study was focused on the effects of metal temperature, coolant temperature, coolant pressure ratio, number of cooling holes, sand amount, and sand diameter on cooling hole blockage for a test coupon with laser drilled film-cooling holes. Walsh et al. concluded that increases in metal temperature had the most significant effect on cooling hole blockage. Walsh et al. reported 1–6% reduction in test coupon
flow parameter (FP) at engine representative metal and coolant temperatures for a pressure ratio range of 1.1–1.6 across the cooling holes.

3 Geometries of the Double Wall Liners

Combinations representing four different designs of a double wall liner utilizing impingement and film-cooling were chosen as representative of either a combustor liner or blade outer air seal. All liners consisted of an impingement plate, spacer plate, and film-cooling plate, as shown in Fig. 3. A periodic representation of the alignment between impingement and film holes is shown in Fig. 4 for each of the four liners tested. As can be seen, for each double wall liner, there was more or less overlap of the impingement jet locations with respect to the film hole entrances.

A 9 × 9 cm² sheet of Inconel® 625, which is a high nickel superalloy with similar physical properties to metals used within the engine, was used as the material base for each plate. The sheets had a machined finish and the thicknesses of the impingement, film-cooling, and spacer plates were 1.27 mm, 0.89 mm, and 1.65 mm, respectively. The impingement and film-cooling holes were machined 90 deg and 30 deg inclined to the surface, respectively. All cooling holes were created using electron discharge machining (EDM), chosen for its high dimensional tolerance at this scale in comparison to traditional and laser machinings. Each spacer plate was machined with a 3.8 cm² hole for the purpose of forming a sealed cavity between the impingement and film plates. All 12 plates were also machined with a six-hole mounting pattern to assure correct alignment for each test. Detailed information on each design is shown in Table 1.

For the geometries shown in Fig. 4, L1 was chosen as representative of an actual engine combustor liner. Each design was varied off of the L1 design to assess the effect of cooling hole diameter, total cooling flow area, and relative alignment between the impingement and film. The number of impingement holes was increased between L1 and L2 while keeping the cooling hole diameter and the number of film-cooling holes constant. This allowed for the investigation of total impingement flow area on sand blocking levels. Cooling hole size was reduced by 20% for L3 while keeping the total film-cooling flow area and impingement hole pattern equal to the values for L1. L4 had an increased cooling hole diameter of 20% above L1 and L2 in addition to a different pattern for the film-cooling and impingement holes.

4 Experimental Facility and Methodology

The level of hole blockage was quantified through the use of the flow parameter, as was done by Walsh et al. [22]. The FP, shown in the Nomenclature, is a convenient definition of a non-dimensional mass flow parameter [23]. In the definition, Pexit refers to the discharge pressure of the cooling hole. During testing each liner was exhausted to the ambient laboratory environment, which allowed Pexit to be replaced by Pam in the definition of flow parameter. There is one unique flow parameter that results for a given geometry, temperature, and pressure ratio.

The pressure ratio is defined as the ratio of supply pressure upstream for the impingement plate to the exit static pressure of the film-cooling plate. The coolant supply pressure Pfc was measured upstream of the impingement plate. If either the film-cooling or impingement holes were partially blocked, one would expect to measure a lower flowrate than for an unblocked hole at the same pressure ratio. This drop in flowrate also corresponded to a drop in flow parameter. Ultimately, matching the flow parameter with engine conditions resulted in a matched air residence time within the liner. For a realistic coolant temperature, matching the air residence time resulted in a particle residence time, which was similar to engine values. Matching the particle and air residence times assured that the thermal response of each particle, set by

### Table 1 Double wall liner details

| Film cooling | | | | |
|---|---|---|---|---|---|---|
| Liner No. | Nfc | Dfc (µm) | Afc (mm²) | Lfc/D | Pfc/D | P2fc/D |
| L1 | 64 | 635 | 20.3 | 2.8 | 7.0 | 7.1 |
| L2 | 64 | 635 | 20.3 | 2.8 | 7.0 | 7.1 |
| L3 | 100 | 508 | 20.3 | 3.5 | 8.8 | 7.1 |
| L4 | 52 | 762 | 23.7 | 2.3 | 5.9 | 6.5 |

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radiation and convection within the part, was equivalent to that occurring within the engine. It was because of these relationships that matching flow parameter as well as coolant and part temperatures were important for studying the blockage of an internal cooling geometry.

Before sand testing was performed, a baseline flow parameter curve, shown in Fig. 5, was established for each liner at ambient and elevated temperatures. At a given coolant temperature, each liner had a unique relationship between pressure ratio and flow parameter. Varying the coolant temperature changed the fluid viscosity within the liner, resulting in a new relationship between the flow parameter and the pressure ratio, as shown in Fig. 5. Each respective baseline was used to evaluate the total reduction in flow parameter FP for a blocked liner, as shown in the Nomenclature.

Prior to each blockage test where sand was injected into the air stream, the measured baseline flow parameter was repeated to ensure that the hole passages were clear of any sand from the previous test. Note that the blockage was based on the flow parameter that occurred at the final pressure ratio of the clogged liner. This procedure is illustrated in Fig. 6 as a zoomed graph of ambient and heated baseline curves with representative data points. Under experimental conditions, the result of sand blocking with the liner was a sudden increase in pressure ratio and decrease in flow parameter. After clogging with sand, liner flow parameter and pressure ratio remained at their blocked values until the liner was cleaned manually. This method of using the reduction in flow parameter to quantify sand blockage within cooling holes was published in a similar study by Walsh et al. [22].

The reduction in flow parameter can be reported several ways, whether an ambient temperature or a heated test was conducted. The previously defined variable RFP was used to compare the reduction in flow parameter under ambient conditions. For all cold cases, the RFP was based on the difference between the blocked flow parameter FP and the equivalent unblocked flow parameter at the blocked pressure ratio FP. The calculation of RFP is illustrated in Fig. 6. At heated conditions, the RFP was used whereby FP was the equivalent unblocked baseline flow parameter at the blocked pressure ratio and FP was the blocked value at heated conditions, as shown in Fig. 6.

4.1 Test Facility. Originally constructed by Walsh et al. [22], a test facility that generated a constant pressure upstream of the test coupons was modified to accept the double wall liner coupons, as seen in Fig. 7. High pressure room temperature air was supplied as coolant to the test coupons. This air was supplied to the laboratory, prior to which it was filtered and dried by an auxiliary compressor facility at approximately 550 kPa. A self-adjusting precision pressure regulator was used to control and regulate a constant coolant pressure. Downstream of the pressure regulator was a laminar flow element, having a maximum capacity of 1400 cm³/s, which measured the total coolant flow rate through each liner. The required flowrate range was 170–830 cm³/s.

Coolant temperatures were recorded within the center of the plenum chamber using a type K sheathed thermocouple probe as shown in Fig. 7. Coolant pressure was also measured at this location using a 1.6 mm diameter Inconel® tube as a pressure probe. The ratio of plenum diameter to impingement hole diameter varied from 144:1 to 96:1, thus assuring that pressure measured within the chamber was equivalent to the total coolant pressure.

Control of the metal temperature was accomplished by placing the liner and sand feed pipe within an electric kiln. The coolant temperature was controlled with a multipass heat exchanger supplied with ambient temperature compressed air, also shown in Fig. 7. Two fiberglass shielded type K thermocouples were affixed to the film-cooling plate using a high temperature ceramic adhesive to measure the metal temperature of the film-cooling plate. Each thermocouple was located in close proximity to a film-cooling hole, but not within the coolant jet. For each test, the difference in metal temperature measured between the two thermocouples was less than 5 °C. Variation of the metal and coolant temperature was accomplished by changing the kiln power and auxiliary heat exchanger flow for the coolant stream.

An electric oven, set at 150°C, was used to dry the sand for 4 h before each test. This preliminary drying was performed to
avoid humidity clumping of the sand. Sand delivery to the liner was accomplished using a sealed gravity feed system, shown in Fig. 7. For each test, the prescribed sand amount was loaded above a valve that controlled access to the sand feed pipe. Opening this valve allowed the sand to pass vertically down the feed pipe, which was approximately 10 cm in length. This pipe terminated just upstream of the impingement plate and above the first row of cooling holes. The sand feed pipe was terminated above the cooling holes to insure that it did not disturb the inlet flow into the impingement plate. It is important to note that this resulted in a slightly higher mass loading of sand through the upper holes than through the lower holes. In addition, visual inspection after testing showed only trace amounts of sand within the coolant plenum, which meant that all the sand had become entrained within the coolant flow.

Since the tube supplying the sand was located within the hot portion of the kiln, as shown in Fig. 7, it is important to recognize that there was a significant heat up of the sand prior to entering the test coupon. Heat transfer to the falling sand was driven primarily through radiation from the section of the feed tube, which was exposed to the heated kiln. It is also important to note that the kiln temperature needed to be adjusted to maintain a constant coupon metal temperature for different coolant flowrates. For a constant coolant and kiln temperature, increasing the coolant flowrate to the coupon resulted in a decrease in the part’s metal temperature. Because of this, the kiln temperature was varied to maintain a constant metal temperature for the different coolant flowrates. As a result, the supply tube temperature was changing with the kiln temperature. With the tube temperature changing, so did the entering sand temperatures. At elevated temperatures the probability of deposition increased when using heated sand. Therefore a transient calculation was performed for each test to estimate the sand temperature as it flowed into the impingement plate. Reported values of sand temperature were based on the mean particle size and assumed that the supply tube temperature was equal to the kiln temperature.

Each liner plate was affixed in the test setup using a ceramic adhesive and sealant that formed an air-tight seal once cured up to 1260°C. Most importantly, the adhesive’s coefficient of thermal expansion was similar to that of Inconel® 625, making it an ideal high temperature sealant. Steps were made to ensure each joint was hermetically sealed before and after each test, which was verified through repeating the baseline flow parameter-pressure ratio curve.

4.2 Sand Characterization. The chosen test sand, Arizona road dust, is comprised primarily of crushed quartz [22]. The analysis of each of the sand samples agree with the manufacturer’s specification stating that it contains different phases of quartz (SiO₂) up to approximately 68–76%. The other major constituent is aluminum oxide (Al₂O₃) between 10–15%, with traces of iron oxide (Fe₂O₃), sodium silicate (Na₂O), lime (CaO), magnesium oxide (MgO), titanium dioxide (TiO₂), and potassium oxide (K₂O) in descending concentration. Arizona road dust has been used extensively for particle ingestion testing by the aviation, automotive, and filtration industries. Its chemical composition closely matches the types of sand found in arid desertlike climates. In addition to having a wide variation in particle size, this particular sand agglomerates readily forming large particles.

In characterizing the sand, a number of measurement methods were originally used to verify the particle size distribution. To reduce the particle breakup during the sizing measurement, a dry analysis was performed by using a series of mesh sieves ranging from 53 μm to 850 μm. The sieves were stacked thus filtering the sand in different bandwidths. These results are given in Fig. 8. The initial weight of each sieve and the sand were recorded. The entire stack was then lightly agitated for 4 h. After the agitation, each sieve was weighed and the amount of sand within each size band calculated. Several large conglomerates of particles were observed in the 850 μm sieve. The maximum linear dimension of the conglomerates was measured by calipers as ~3000 μm and is listed in Fig. 8 as the 100% data point. A minimum particle diameter of 0.6 μm was measured using a Horiba Partica LA-950 laser diffraction analyzer. The LA-950 utilizes both a wet and dry measurement method, both of which caused significant breakup of particle agglomerations and were therefore deemed unsuitable for measurement of the overall size spectrum. The results of the laser diffraction methods are also included in Fig. 8.

Unless indicated, the amount of sand used was 0.35 g for each of the tests. The 0.35 g corresponded to a particle mass loading of 0.8, which was used by Walsh et al. [22] as representative of actual levels within the engine. Particle mass loading is defined as the mass flux ratio of the dispersed phase to the continuous phase [24]. For these tests the sand and cooling air served as the dispersed and continuous phases, respectively. Walsh et al. determined the appropriate mass loading by comparing the flow parameters of clean turbine components with field-operated components in which sand had entered the coolant stream and blocked cooling holes.

Deposition within the engine is most likely to occur over a longer period of time than the method used in this study in which a slug of sand was introduced to the part. To assure realistic results, several tests were performed to determine if a slug of a given sand amount has the same blocking characteristics as when the given sand amount was divided into portions. It was verified that the cumulative sand amount equaled that used for the single slug tests. Representative results of these comparison tests are shown in Fig. 9. The tests indicate an insensitivity to how the sand

Fig. 8 Size distributions for the test sand obtained by several methods

Fig. 9 Cumulative blocking effects versus slug flow for L1 at \( T_{in}=982°C \) and \( T_{in}=649°C \)
was introduced into the liner as both the cumulative and slug tests resulted in the same overall blockage.

4.3 Testing Procedure. Before each test, deposited sand from the previous test was removed from the impingement and film-cooling plates. The liner was considered free of previous sand remnants once its original baseline flow parameters could be reproduced.

After confirming the appropriate baseline flow parameter curve, the desired pressure ratio was set between 1.02 and 1.1. For an unheated test, the system was ready for sand injection. When performing a heated test, the test apparatus was placed in the kiln prior to setting the appropriate pressure ratio. The auxiliary cooling air flowrate and kiln power were varied until the desired coolant and metal temperature were set and steady. Required time to reach steady state for a heated test varied from 3–6 h, dependent on the desired test conditions.

With the pressure ratio set and steady-state temperatures achieved, the sand feed valve was quickly opened allowing the prescribed amount of sand to flow through the part. Once the pressure ratio and flow parameter were steady again, which typically took less than a few seconds, the new pressure ratio and flow parameter were recorded. The appropriate reduction in the flow parameter was then calculated relative to the clean liner flow parameter at the blocked pressure ratio. A 25% repeatability study conducted by Walsh et al. [22] showed that three tests conducted for each case resulted in repeatable results to within 7%. In addition, multiple investigators have performed these studies spanning over 1 year. Their results also indicate that an average of three tests conducted by Walsh et al. [22] showed that three tests provided an adequate repeatability for this type of cooling blockage tests. Average variation of RFP between each group of three tests was 2.5% for both ambient and heated conditions. The propagation of uncertainty associated with the measurement methods was calculated for all test conditions, as outlined by Figliola and Beasley [25], to validate the observed results and trends. At both ambient and heated conditions, the uncertainty in flow parameter was approximately ±2% of the measured value for all pressure ratios. At ambient conditions, the overall uncertainty in RFP was 0.11 ± 0.011 at PR = 1.02, and 0.16 ± 0.01 at PR = 1.10. At heated conditions, the uncertainty in RFP was 0.18 ± 0.016 for PR = 1.03, T_m = 982°C, and T_c = 649°C. Negligible variation in uncertainty was calculated between the different liner geometries. In addition, the uncertainty associated with measuring the prescribed sand amount was 0.35 g ± 0.005 g.

4.5 Derivation of the Test Matrix. Significant consideration was given to the generation of the test matrix, shown in Table 2. The nominal coolant temperature chosen for the high temperature testing was 649°C. With respect to external metal temperatures, the combustor liner typically operates in excess of 1000°C. However, a maximum metal temperature of 982°C was chosen because of the material limitations of Inconel® 625 as it melts at a lower temperature than proprietary engine alloys.

With regards to entering sand temperature, no information was available as to the levels seen within the engine. Whether entering sand temperature was above or below the coolant temperature value was a function of particle residence time, slip velocity, and wall collision rate. Since realistic engine values for these variables are unknown, the maximum limitations of the testing capabilities were chosen to evaluate this parameter resulting in an entering sand temperature range of 316–760°C. This range was set by the minimum and maximum heated lengths of the sand feed tube within the limitations of the kiln’s interior dimensions.

5 Discussion of Results

A number of tests were conducted to analyze the effect of sand being ingested into several double wall liner designs. The experimental results are broken into two sections: ambient temperature and elevated temperature. For the ambient temperature tests, the effects of pressure ratio, liner geometry, and sand amount were investigated. At elevated temperatures, the effects of liner geometry, film-plate metal temperature, and entering sand temperature were evaluated. The ambient results will be discussed prior to the high temperature results. In Secs. 5.1–5.4 and 6, the reduction in flow parameter will be referred to as an increase or decrease in “blockage.”

5.1 Ambient Temperature Results. Figure 10 compares the blockage levels for all four liner geometries at a range of pressure ratios given a nominal sand amount of 0.35 g. An increase in pressure ratio from 1.02 to 1.10 resulted in elevated blockage for all liners with a maximum increase in RFP from 12% to 20% for L3. As seen in Fig. 10, at all pressure ratios the liners showed the same performance trend: lowest values of blockage for L2 and highest blockage levels observed with L3. The results show that the cooling hole diameter, both impingement and film, did not solely dictate the level of blocking for a given design. L3, despite having a 33% smaller cooling hole diameter than L4, varied in RFP less than 1% relative to L4. L1 and L2, despite having the same cooling hole size, showed a difference in blockage of approximately 6% at a pressure ratio of 1.06. However, when considering the total flow area through the impingement plate, consistent trends were identified. Figure 11 shows the same data as Fig. 10 but replotted versus the total flow area of each liner’s impingement plate. Representing the data in this manner indicates that blocking for a given liner design decreases monotonically with increasing impingement flow area.

It was concluded that blocking levels within the liner, at ambient temperatures, were influenced by several different factors that

| L1 | 871 | 982 | 538, 593, 649 | 0.35 | 1.03 |
| L2 | 982 | 649 | 0.35 | 1.03 |
| L3 | 982 | 649 | 0.35 | 1.03 |
| L4 | 982 | 649 | 0.35 | 1.03 |

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Table 2 Parameters used for combustor liner study

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are unique to a double wall liner design. The two dominant factors were filtration of large particles by the impingement plate before entering the impingement cavity (refer to Figs. 3 and 7) and the breakup of particles, which travel through the impingement holes and impact against the upstream side of the film-cooling plate.

It was observed that the impingement plate acted as a filter for the film-cooling plate with respect to sand particles and agglomerations whose maximum linear dimensions were equal to or greater than the impingement hole diameter. Approximately 7% of the particles tested were above 600 \( \mu \text{m} \) in diameter, which was nominally the size of each liner’s impingement holes. Particles in this size range blocked within the impingement holes and were therefore unable to convect into the cavity upstream of the film-cooling holes. It is also important to recall that the film-cooling and impingement hole sizes were matched for each liner configuration. For this particular design, the impingement plate was filtering out all of the larger particles, which could have blocked cooling holes within the film-cooling plate.

Particles, which did not stick to the upstream side of the impingement plate or block within the impingement holes, were carried into the impingement cavity by the coolant flow. Upon exiting the impingement holes, the particles impacted on the upstream surface of the film-cooling plate. This impingement broke up the particles, increasing their likelihood to pass through the film-cooling holes. Post-test inspection of the liner confirmed that the particles were impinging on the upstream side of the film-cooling plate, as shown in Fig. 12. The particle impingement velocities were high enough to locally abrade the metal surface, as shown by the dark circles on the upstream side of the film-cooling plate in Fig. 12.

A study by Land et al. [26], performed congruently with this research, confirmed that particle breakup does occur within the double wall liner. The study compared the relative blocking characteristics of a combined impingement and film-cooling liner with a film-cooling only liner. Land et al. [26] reported a ~300% increase in blockage with the film-cooling only configuration when compared with the double wall liner thus verifying that the impingement resulted in a beneficial breakup of the sand particles within the cavity.

For the presented research, further visual inspection of the impingement and film-cooling holes showed that the bulk of the blockage occurred at the entrance to and within the impingement holes. Conversely, very little blockage was observed within or around the film-cooling holes. The breakup of impinging particles and impingement plate filtering resulted in significantly higher amounts of blockage in and around the impingement holes when compared with the film-cooling plate.

Since all tests were performed with a nominal amount of sand, it is logical that a liner with less impingement flow area would block at a higher rate than a liner with a large impingement flow area. Impingement flow area and particle breakup do not, however, offer insight into the cause of elevated blocking for all liners as pressure ratio was increased. Blocking sensitivity to pressure ratio results from how well particles follow the coolant flow. As the pressure ratio was increased, so did the mean velocity of air through the part therefore resulting in a decreased time characteristic of the flow \( \tau_p \). The particle response time \( \tau_p \) is commonly used to describe the time required for a particle to respond to a change in velocity. In an average sense, \( \tau_p \) remained constant for our testing since it was primarily a function of particle density and diameter. Therefore an increased pressure ratio resulted in an increased average particle Stokes number (St).

Stokes number is commonly used to define how well a given particle will follow the fluid surrounding it. Particles with a St < 1 are assumed to follow the flow perfectly while particles having a St > 1 are considered ballistic and are mostly unaffected by the fluid [24]. Studies by Tu et al. [27] showed that, for a rebounding flow over a curved wall, increased Stokes numbers were associated with particle trajectories deviating from the fluid streamlines thereby increasing the likelihood of wall collisions.

Calculations were performed to determine the Stokes number of 1 \( \mu \text{m} \) and 50 \( \mu \text{m} \) sand particles traveling through the liner. The impingement jet velocity and impingement hole diameter were chosen as the characteristic length and velocity of the coolant flow within the cavity. At 50 \( \mu \text{m} \), particle Stokes numbers were above 500 for all pressure ratios. It was therefore assumed that the trajectories of sand particles at this size or larger deviated greatly from the cooling air path. For a 1 \( \mu \text{m} \) particle traveling through an impingement hole, Stokes numbers were slightly less than unity for all liners. Therefore the Stokes number of the 1 \( \mu \text{m} \) particles remained constant for all four liners. We may infer that increases in pressure ratio corresponding to an increased likelihood that sand particles would collide with the surfaces in and around the cooling holes. The increased number of wall collisions ultimately related to a higher probability of deposition for a given sand amount. Therefore elevated collision and deposition rates were the cause of the increased blocking at higher pressure ratios.

5.2 Ambient Temperature Results With a Varied Sand Amount.

As explained previously, L3 exhibited the highest blockage of the four liner designs tested because of its relatively small impingement flow area. Therefore L3 was chosen to evaluate the sensitivity of the blocking to sand amount. As previously described for the nominal case, sand amounts were determined from the studies by Walsh et al. [22] based on comparing clean and field-operated turbine components. The maximum and minimum particle mass loadings from Ref. [22] were matched for the double wall liners resulting in high and low sand amounts of 0.52 g and 0.21 g, respectively. The effect of sand amount on L3 was then evaluated for all pressure ratios, as shown in Fig. 13.

As shown in Fig. 13, a 51% increase in sand amount resulted in an average increase in blocking of 18% across all pressure ratios.
Lowering the sand amount by 41% resulted in an average 35% decrease in blocking. At pressure ratios above 1.03, these results agree with the results of Walsh et al. [22], which showed a near linear dependence of blocking on the injected sand amount at a given pressure ratio. It is important to note that the studies by Walsh et al. were conducted at pressure ratios above 1.1. The effect of varying sand amount was also evaluated for another liner at a pressure ratio of 1.06 to assure that the results were not dependent on the film-cooling and impingement geometry. As shown in Fig. 14, L1 exhibits the same trend as L3 with linearly increasing blockage as sand amount was increased.

5.3 Elevated Temperature Results. Each liner was evaluated with a constant metal, coolant, and entering sand temperature at $T_M=982\,^\circ C$, $T_c=649\,^\circ C$, and $T_s=386\,^\circ C$. In addition to these tests, as was described previously, the length of the sand delivery pipe was changed such that less of it was inside the kiln to lower the sand temperature for L1 and L2 to $T_s\sim 760\,^\circ C$. As stated previously, a transient heat transfer calculation was performed to estimate the sand temperature flowing into the impingement plate. It is important to note that all elevated temperature tests were performed with a nominal sand amount of 0.35 g.

As discussed previously, the reduction in flow parameter or blockage was calculated differently at heated condition than what was performed for the ambient temperature testing. At heated conditions, the $RFP_H$ was used. In this equation for the reduction in flow parameter at heated conditions and $FP_H$ was the blocked value at heated conditions. Figure 6 illustrates the testing procedure and calculation of this parameter at heated conditions.

The high temperature results for each liner are shown in Fig. 15 with the previously reported values of blockage for the ambient temperature results. Each liner showed an increase in blocking at elevated temperatures. L2 performed the best with the lowest values of blockage for both heated and ambient temperatures. L3 had the highest operational flexibility, exhibiting very little change in sand blockage between heated and ambient temperature testings. The largest increase in blockage, by means of an increase in temperature, occurred with L1. Explanation of the elevated temperature results was accomplished by disassembling each liner after testing and noting differences in the sand blocking patterns.

At elevated temperatures, the liner blocking results no longer scaled with impingement area as it did under ambient conditions. Visual inspection after heated testing confirmed that the sand blockage had moved into the film-cooling holes and continued to occur within the impingement plate. Recall that negligible amounts of blocking were observed within the film-cooling holes at ambient conditions. It was apparent that the impingement plate was still acting as a filter for larger particles, but observations of completely blocked film-cooling holes meant there was a new blocking mechanism occurring within the liner for heated conditions.

Under heated conditions, blockages within the film-cooling plate resulted from the sand becoming sticky at elevated temperatures. This stickiness increased the likelihood of sand adhering to the impingement and film-cooling plates as well as within the cooling holes. It was found, as shown in Fig. 16, that the deposited sand takes the shape of small mounds on the upstream side of the film-cooling plate at elevated temperatures. Figure 16 illustrates these sand mounds in comparison to the case under ambient temperatures, as shown in Fig. 12. As shown, each sand mound was centered on the location of the impingement hole exit. These
mounds formed around and within the entrance to the film-cooling hole when there was overlap between the impingement hole exit and film-cooling hole entrance, thereby contributing to an increase in blocking within the nearby film-cooling holes.

Recalling Fig. 4, L2 and L3 showed little overlap between the impingement and film-cooling holes. The opposite was true for L1 and L4, which had much higher levels of cooling hole overlap. Therefore L2 and L3 were less sensitive than L1 and L4 to operating at elevated temperatures because of the difference in overlap between impingement and film-cooling holes. It was also of note that L4 showed the highest blockage at elevated temperature despite having the largest cooling hole diameter. Again, this was attributed to L4 having the highest level of impingement/film-cooling hole overlap when compared with the other three liners. This result further reinforced the hypothesis that the combination of sand stickiness and the amount of cooling hole overlap were responsible for the increase in blocking observed at elevated temperatures.

5.4 Results With Varied Metal and Sand Temperatures. Varied metal and entering sand temperature tests were conducted with L1 as it showed the greatest sensitivity to being operated at elevated temperatures. The range of metal temperatures tested (871–982°C) correspond to realistic values of metal temperature for the combustor liner. As discussed previously, the range of entering sand temperature was varied to the maximum allowable limitations for the testing setup. This value was set by the heated length of sand feed tube, as discussed previously. All tests were performed with a fixed coolant temperature of \( T_{c} = 649°C \).

The results of this study are shown in Fig. 17 along with the sand temperature entering the liner for each test. The blockage level at ambient conditions is also shown in Fig. 17 to serve as a basis of comparison for the elevated temperature tests. For a relatively low entering sand temperature, the increase in metal temperature from ambient to 871°C resulted in a 13% increase in blocking. An increase of 17% in blocking was observed between impingement and film-cooling holes at ambient conditions. At ambient conditions, sand color darkened appreciably and could not be dislodged by high pressure air alone. The dark mounds of sand were difficult to break up and required manual removal.

6 Conclusions

The effects of sand flowing through a combined impingement and film-cooling double wall liner have been presented. Pressure ratio across the liner, sand amount, metal temperature, and sand temperature were used to evaluate four realistic liner geometries. The four liners were designed to investigate what effect the cooling hole diameter, number of cooling holes, total coolant flow area, and relative alignment between the impingement and film-cooling holes had on blockage levels. The study was divided into two sections: ambient and engine representative temperature results.

For all liner geometries tested at ambient conditions, an increasing pressure ratio resulted in an increase in blockage. Since raising the pressure ratio resulted in an overall increase in fluid velocity, it was hypothesized that the increasing deviation of particles from fluid streamlines, described by the Stokes number, resulted in more wall collisions and therefore higher deposition rates. It was found that blocking levels scaled directly with impingement plate area at ambient conditions. At ambient conditions, the liner with the largest impingement flow area exhibited the lowest blocking for all pressure ratios. Each double wall liner design also had a matched diameter of impingement and film-cooling holes, causing the impingement plate to act as a particle filter for the film-cooling plate. It was also observed that particles small enough to travel through the impingement holes subse-
quently impinged on the upstream side of the film-cooling plate. This impingement broke the particles up, allowing them to more easily pass through the film-cooling holes. Because of the particle breakup and filtering by the impingement plate, a negligible buildup of sand was observed within the film-cooling holes or on the upstream surface of the film-cooling plate at ambient conditions.

At elevated temperatures, blocking levels did not scale with impingement area. The amount of blocking increased for all liners as metal temperature was increased. This was found to be a result of the sand becoming sticky at higher temperatures thus increasing the probability of particle deposition. Visual inspection of the liner after testing confirmed that the sand was becoming sticky, which resulted in elevated deposition levels on the upstream side of the film-cooling plate and within the film-cooling holes. This effect was quite different than at ambient conditions, where no sand buildup was observed on the upstream surface of the film-cooling plate. Since sand was depositing in small mounds on the upstream surface of the film-cooling plate downstream of the impingement jets, it was determined that liners with a high amount of overlap between the impingement and film-cooling holes blocked more when compared with staggered impingement and film-cooling holes.

Overall, the liner with the largest impingement flow area and least overlap between the impingement and film-cooling holes exhibited the lowest blocking overall at both ambient and heated conditions. At all operating conditions, the impingement holes were acting as a particle size filter for the film-cooling plate. This study has shown that impingement could be used to breakup larger particles thereby reducing their possibility of blocking within downstream cooling geometries. In addition, the cooling liners with staggered film and impingement holes were found to be less sensitive to the sand depositing on the upstream side of the film-cooling plate, which occurred at higher temperatures.

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Nomenclature

\[ A = \text{cross-sectional area} \]
\[ d = \text{particle diameter} \]
\[ D = \text{cooling hole diameter} \]
\[ FP = \frac{(m_{\text{hole}}/T_R)(P_{\text{exit}})}{P_{\text{hole}}} \]
\[ L = \text{length} \]
\[ L_c = \text{characteristic flow length} \]
\[ m = \text{mass} \]
\[ \dot{m} = \text{mass flowrate} \]
\[ n = \text{number of holes} \]
\[ P = \text{pressure} \]
\[ P_{\text{pit}} = \text{pitchwise cooling hole spacing} \]
\[ P_{\text{st}} = \text{streamwise cooling hole spacing} \]
\[ PR = \frac{P_{\text{in}}}{P_{\text{exit}}} \]
\[ Q = \text{volumetric flowrate} \]
\[ R = \text{air gas constant} \]
\[ \text{RF} = \frac{P_{\text{in}}}{P_{\text{out}}} \]
\[ \text{RFP} = \frac{P_{\text{in}}}{P_{\text{out}}} \]
\[ \text{Stokes number} = \frac{\tau_1}{\tau_i} \]
\[ T = \text{temperature} \]
\[ U = \text{velocity} \]
\[ \alpha = \text{uncertainty} \]

Greek

\[ \rho = \text{density} \]
\[ \mu = \text{dynamic viscosity} \]
\[ \tau_1 = \text{characteristic flow time scale}, \tau_i = \frac{L_c}{U} \]
\[ \tau_s = \text{particle response time}, \tau_s = \frac{(\rho d_s^2)}{(18\mu)} \]

Subscripts

\[ 0 = \text{equivalent unblocked parameter for ambient conditions at the blocked pressure ratio} \]
\[ 0H = \text{equivalent unblocked parameter for heated conditions at the blocked pressure ratio} \]
\[ OC = \text{coolant total property} \]
\[ \text{amb} = \text{ambient laboratory conditions} \]
\[ B = \text{blocked with sand at ambient conditions} \]
\[ BH = \text{blocked with sand at heated conditions} \]
\[ c = \text{coolant} \]
\[ \text{exit} = \text{film-cooling hole exit} \]
\[ \text{hole} = \text{relating to a single cooling hole} \]
\[ H = \text{evaluated at heated conditions} \]
\[ l = \text{impingement} \]
\[ M = \text{metal} \]
\[ s = \text{sand} \]

References