

DESIGN AND COOLING PERFORMANCE OF ADDITIVELY MANUFACTURED CERAMIC TURBINE VANES

Andrew Fox¹, Stephen Lynch¹, Jason C. Young², Carl P. Frick², Jennifer Hankins³, Kyle Kuhn³,
Stephan A. Brinckmann³, Ray S. Fertig III³

¹Mechanical Engineering, Pennsylvania State University, University Park, PA

²Mechanical Engineering, Colorado School of Mines, Golden, CO

³Mechanical Engineering, University of Wyoming, Laramie, WY

ABSTRACT

Ceramic materials are of significant interest in aviation and power generation gas turbine engines due to their low density and ability to withstand high temperatures. Increased cycle thermal efficiency and higher specific power output is possible by incorporating ceramic components that enable high turbine inlet temperatures and lower required cooling airflow levels. However, ceramics can be difficult and costly to form into the complex shapes used in gas turbine components, often requiring specialized multi-step processes. Furthermore, ceramic components in the hottest areas of a gas turbine, such as vanes or blade shroud seals, will still likely require cooling which is challenging to implement in conventional ceramic manufacturing approaches. Therefore, this study presents a multidisciplinary approach that investigates the design, fabrication and overall cooling effectiveness evaluation of additively manufactured (AM), polymer derived ceramic (PDC) turbine vanes. A thermo-mechanically optimized vane design was generated, ceramic additive manufacturing of the complex cooling configuration was developed, and quantification of the increase in overall cooling effectiveness was performed in a 1X scale, high-speed facility using infrared thermography. This study produced a PDC AM process, capable of printing complex internal cooling schemes in 1X scale turbine vanes. It was found that the optimized vane more than doubled the overall cooling effectiveness observed in the baseline design, which reasonably agreed with thermomechanical optimization model predictions. Additionally, the optimized ceramic vane outperformed an identical metal vane, in terms of area averaged cooling effectiveness, suggesting that the ceramic vane could operate at reduced coolant flow rates to achieve comparable levels of cooling performance.

NOMENCLATURE

A_c Cross-sectional area, mm²
 C_{ax} Axial chord, mm

C_p Constant pressure specific heat, J/kgK
 D_h Hydraulic diameter, mm
 h Heat transfer coefficient, W/m²K
 L Length, mm
 $\dot{m}$ Mass flow rate, g/s
 Ma Mach number
 Nu_{Dh} Hydraulic Nusselt number in cooling channel
 P Perimeter, mm
 P_0 Total cascade pressure, kPa
 P_s Blade static pressure, kPa
 ΔP Cooling channel pressure drop, kPa
 Re Airfoil Reynolds number
 Re_{Dh} Hydraulic Reynolds number in cooling channel
 S Airfoil span, mm
 s Airfoil surface length, mm
 s_{max} Maximum airfoil surface length, mm
 T_c Coolant temperature, °C
 T_{in} Measured inlet temperature of the blade, °C
 T_{out} Theoretical heat-up outlet temperature, °C
 T_w Airfoil wall temperature, °C
 T_∞ Freestream cascade temperature, °C
 x Airfoil axial coordinate direction, mm
 z Airfoil spanwise direction, mm

Greek Letters/Overbars

$\bar{T}_s$ Average constant airfoil surface temperature, °C
 $\bar{\phi}$ Laterally averaged overall cooling effectiveness
 $\bar{\bar{\phi}}$ Area averaged cooling effectiveness
 ϕ Overall cooling effectiveness
 μ Dynamic viscosity, kg/ms

Abbreviations

AM Additive manufacturing
CMC Ceramic matrix composite
LE Leading edge
PDC Polymer derived ceramic
Ptap Static pressure tap
TE Trailing edge

1. INTRODUCTION

Ceramics are becoming more prevalent in the design of gas turbines due to their ability to withstand extremely high temperatures. With current aviation gas turbines exceeding inlet temperatures of 3100°F, well beyond the melting point of most metals, a switch to advanced turbine materials, like ceramics, is imperative. Ceramics introduce the opportunity for significantly improved turbine efficiencies when compared to conventional metal alloy turbine vanes in current-generation gas turbines. These improved levels of performance can be attributed to the higher engine firing temperatures that ceramics can withstand, along with the subsequent reduction in cooling airflow needed to meet durability life requirements. Additionally, the relatively low density of ceramic parts lead to lighter turbine stages that can improve aircraft thrust-to-weight ratios.

Manufacturing ceramic turbine vanes can be challenging, however, often requiring specialized molding processes and multiple complex steps to achieve a finalized part. Furthermore, traditional ceramic manufacturing methods are limited in the complexity of the cooling features that can be created. Therefore, additive manufacturing (AM) has emerged as a possible alternative for manufacturing complex ceramics. AM can allow for the printing of complex cooling design features while still leveraging the favorable thermal and weight properties of ceramics.

The primary goal of this study is to develop new insight into how AM can enable transformative levels of cooling performance in ceramic airfoils. This study is enabled by the development of an amorphous silicon oxycarbide ceramic additive manufacturing process that is capable of printing 1X scale turbine vanes with complex, yet relevant internal cooling features. To leverage these printing capabilities, a thermomechanical optimization model is developed, generating an optimized, internal cooling configuration that reduces external surface temperature without compromising structural integrity. This design, along with an internally cooled baseline ceramic vane are then tested in a transonic linear cascade to evaluate the overall cooling effectiveness of each and compare to a metal alloy.

2. REVIEW OF RELEVANT LITERATURE

The benefits of ceramic materials are fairly well understood; Grondahl and Tsuchiya [1] indicated a possible increase of 2 percentage points in cycle efficiency from substituting ceramics in the first two stages of a power generation turbine due to reduced cooling. Because of this, a significant amount of development of ceramic materials for turbines has been conducted, including a multi-year program by Solar Turbines under the Department of Energy's Ceramic Stationary Gas Turbine program [2–6], with field demonstrations of ceramic combustor liners, nozzle guide vanes, and turbine blades. Ceramic matrix composite (CMC) components were tested under NASA's Ultra Efficient Engine Technology program [7]. Most recently, CMC shrouds (stationary component above the rotating turbine blade) have been commercialized by General Electric [8] for its CFM LEAP commercial aircraft engine.

However, ceramics have not seen wide adoption in turbines to date due to low fracture toughness relative to metals and difficulty (cost) of fabrication in the complex shapes often required for turbine applications [9]. Reinforced ceramics (CMCs) have improved fracture toughness relative to monolithic ceramics but are even more complicated to manufacture, requiring multiple processing steps [10–12]. Non-oxide ceramics that have the highest temperature tolerance typically must also have specialized coatings (environmental barrier coatings, EBCs) to prevent oxidative material recession due to water vapor present in the combustion gases [13,14].

Relatively recently, additive manufacturing has been explored for fabrication of ceramic materials [15–19] as it may enable complex features. Of interest here are polymer-derived ceramics (PDCs) with a silicon basis such as polycarbosiloxanes, polysilazanes, or polycarbosilanes that convert to silicon oxycarbide (SiOC), silicon nitride (SiN), or silicon carbide (SiC) after pyrolysis, respectively. Although many additive processes exist [17], vat photopolymerization has the highest potential for fine feature resolution [20]. Brinckmann, et al. [21] recently investigated the effect of print direction on SiOC obtained from vat photopolymerization and found anisotropy in mechanical properties differed for green bodies versus pyrolyzed parts. Recent studies [22,23] have investigated the potential to embed inert ceramic reinforcement (SiC whiskers, particles) in PDCs to reduce shrinkage and improve mechanical properties.

One feature of interest that can be difficult to fabricate in ceramic materials is complex cooling features. Internal and external cooling technologies for superalloy materials are well-established [24] and enable use of those materials at temperatures hundreds of degrees higher than their melting point. However, it is difficult to fabricate small features in ceramic components with conventional manufacturing, and most studies to date have been limited to simple pass-through internal cooling [7,9,25]. Furthermore, the composite microstructure of CMCs results in anisotropy of thermal properties that can result in different cooling behavior depending on the fiber orientation in the matrix [26]. Recent studies have found that the wavy surface morphology of CMCs can have a detrimental effect on external convective heat transfer [27] and disrupt film cooling [28], but the roughness can improve internal cooling [29,30].

A significant body of past work has investigated high temperature ceramic materials for turbine applications, but to date it has been difficult to integrate complex internal cooling features that can enable the ceramic to operate at even higher temperatures (or can further reduce cooling air requirement). The aim of this study is to assess the potential for additive manufacturing of complex internal features in PDCs and quantify the improvement in cooling relative to a simple pass-through approach for a 1X scale ceramic turbine vane operated at representative Reynolds and Mach number conditions.

3. DESIGN

3.1 Cascade Hardware Design

The 1X scale test article is comprised of four main components, shown in Fig. 1. First, at the base of the test article, is a 316L grade stainless steel instrumentation block that routes the cooling air through the vane via two separate u-bends. This instrumentation block also houses four pressure taps and four thermocouple cavities, all marked in Fig. 1, such that the static pressure and total temperature can be measured at the bottom of each passage. Then, within the instrumentation block, is a 12.7mm deep socket that supports the NASA C3X [31] airfoil geometry used for testing; see Table 1 for the airfoil geometric and operating condition parameters.

The remaining three components of the test article make up the turbine vane itself and include: the vane base, the middle ceramic test vane, and the vane tip. The vane base and tip are

similarly made from 316L stainless steel and comprise the bottom and top portion of the airfoil span, respectively. The ceramic vane, which is the region of interest for cooling performance evaluation, is compressed between the metal base and tip pieces via two #2-64 threaded screws that run through each of the three pieces, threading into the vane base. Additionally, the interface between each vane piece is finely layered with a high-temperature adhesive to ensure secure and leak-free contact. This assembly design ultimately allows for a ceramic part that is 16.5mm tall, shown in Fig. 2, and only accounts for about a third of the vane's span. The ceramic was limited here to a small segment of the overall span because of the team's current 3D printer and pyrolysis furnace size constraints.

The internal structure of the baseline ceramic vane consists of four internal cooling channels. As shown in the airfoil surface wall schematic, Fig. 3, the first two channels near the leading edge and the back two channels near the trailing edge are each separated by the cavity walls that house the screws used to support the assembly, while the middle two channels are split via a 1 mm thick partition wall. The external wall of the baseline ceramic vane design is also 1 mm thick. The vane tip piece then segments the flow into two distinct cooling lines; one line feeds

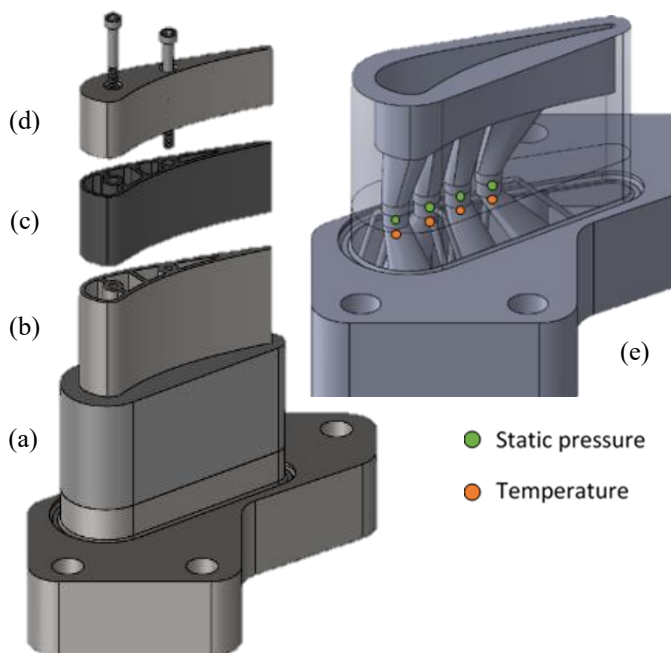


Fig. 1 (a) Measurement airfoil instrumentation block for coolant routing; (b) airfoil base with threaded holes for screws to join components; (c) additively manufactured ceramic vane to be evaluated for cooling performance study; (d) airfoil tip; (e) pressure and temperature measurement locations in instrumentation block.

Table 1. NASA C3X test airfoil parameters.

C_{ax} (mm)	23.4
Pitch/ C_{ax}	1.51
S/C_{ax}	2.25
Design exit Ma	0.90
Design exit Re	390,000



Fig. 2 Additively manufactured ceramic turbine vane.

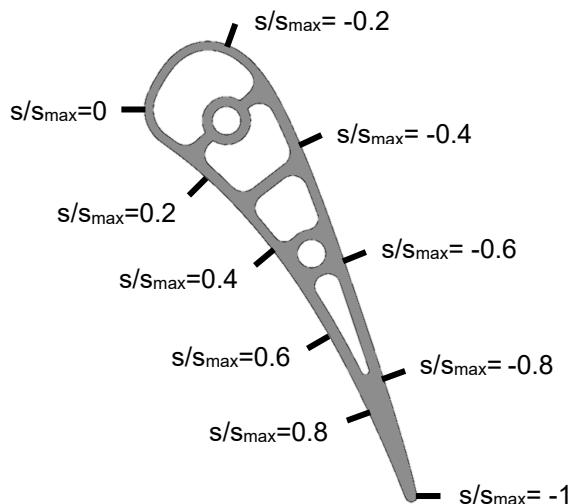


Fig. 3 Airfoil nondimensional surface length schematic.

the two leading edge channels while the other feeds the two trailing edge channels. This flow schematic is represented in Fig. 4, in which the cooling air travels up through channel 1 and channel 3 then loops back down and out of the test section through channel 2 and channel 4, respectively.

3.2 Thermomechanical Optimization

An improved turbine vane design was achieved via a multistep process which analyzed the sensitivity of the geometric parameters of the baseline vane design combined with augmented internal cooling via a pin array. The design process spanned several iterations which also accounted for the evolving constraints imposed by the additive manufacturing process and physical testing requirements. The steady state thermomechanical finite element analysis was conducted in Abaqus. Regarding mechanical loads, the outer surface was given the varying pressure distribution from a CFD analysis at the expected operating conditions. The inner surface was a uniform pressure load of 181 kPa corresponding to the expected internal pressure inside the cooling cavities. Regarding thermal loads, the outer vane surface was supplied with the expected spatially varying convection coefficient from Hylton, et al. [31] and a reference (sink) temperature of 373K (equal to the cascade total temperature), and the inner cavities were supplied with a constant convection coefficient corresponding to the expected channel Reynolds number and sink temperature obtained from a pipe flow analysis (discussed in more detail later). Furthermore, a thermal conductivity of 1.4 W/mK was assumed for the ceramic at the operational temperature of 373K [32].

With a baseline vane model established, an attempt to improve the cooling efficiency of the vane was the next progression so a traditional pin fin cooling array was introduced. Directly implementing a fin array and performing an optimization routine on the full-size vane model incurs large computational expense. Therefore, before directly modeling a full-scale vane model, the design space search process was simplified by introducing a smaller localized pin fin model. The fin array was assumed to run parallel to the struts, and thus a two-plate model approximating the dimensions of the shell of the vane encompassed a twenty-five-pin fin array. Using a steepest descent optimization routine, the maximum nodal temperature was minimized while studying a wide range of pin fin width and spacing for both an aligned and staggered array. The convection

boundary condition imposed on the pin fin array was obtained from curve fits to the cylinder banks in crossflow correlations in Bergman, et al. [33]. From this study it was concluded that a staggered fin array was more efficient for cooling, using less pins overall than the aligned array. Using the average model temperature as an objective (rather than the maximum) predicted a similar trend.

Using the sole objective of minimizing model surface temperature led to the optimization attempting to maximize internal cooling surface area. Given that the exterior of the vane experiences the largest temperature, and the interior cavities are supplied with cold air, a significant thermal stress was observed. Therefore, it was theorized that in order to reduce thermal stress, it would be advantageous to also minimize the maximum thermal gradient in the model. The revised objective function is stated in Equation 1 and 2. Design variables considered were fin diameter, width and length spacing, as well as vane wall thickness. The weighting coefficients allow for the model objective to skew towards the maximum gradient or maximum nodal temperature as desired. After optimizing a pin fin array with this new objective function, it was observed that maximum temperature gradient varies by an order of magnitude depending on the vane wall thickness. Additionally, the thermal stress induced was much smaller than the stress from the pressure loads in the full vane model.

$$L(x_1, x_2, x_3, x_4) = \alpha * \text{Grad}(T) + \beta * \text{Nodal}(T) \quad \text{Eq.(1)}$$

$$\alpha + \beta = 1 \quad \text{Eq.(2)}$$

Returning to a full-scale vane model, it was noted that the maximum stress experienced in the vane occurred in the wall of the vane. The goal was to produce a vane design which improved cooling effectiveness and did not fail under the applied thermomechanical load. Thus, a final objective function was introduced which penalized the objective when the maximum principal stress neared the material's failure stress, reflected in Equation 3. Here σ_n is the maximum principal stress normalized to the material ultimate strength and NT_n is the maximum nodal temperature normalized to the outer sink temperature (373K).

$$L(x_1, x_2, x_3, x_4) = \frac{1}{1 - (\sigma_n)^2} + NT_n \quad \text{Eq.(3)}$$

Finally, an optimal pin fin model was produced from the optimization routine which suggested a vane wall thickness of 0.5 mm with a pin width and uniform spacing of 0.6 mm that would minimize the objective function. For modeling simplicity, a square fin was assumed, and the physical design is shown in Fig. 5. Table 2 summarizes the convection coefficients and sink temperatures for the optimized model based on pin bank correlations.

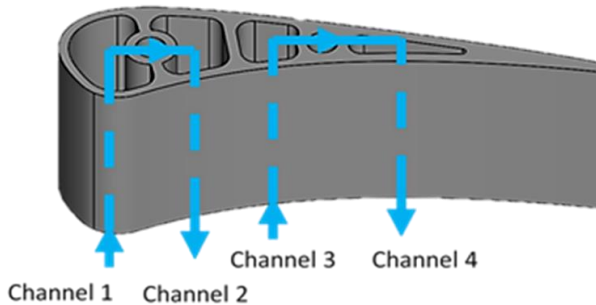


Fig. 4 Double u-bend coolant routing schematic.



Fig. 5 Optimized ceramic vane design with internal pin fin array structure.

Table 2. Sink temperatures and internal convection coefficients for the optimized vane at $Re=8200$.

Channel 1		Channel 2		Channel 3		Channel 4	
$T_{sink}(K)$	$h \left(\frac{W}{m^2 K} \right)$	$T_{sink}(K)$	$h \left(\frac{W}{m^2 K} \right)$	$T_{sink}(K)$	$h \left(\frac{W}{m^2 K} \right)$	$T_{sink}(K)$	$h \left(\frac{W}{m^2 K} \right)$
312	960	323	1121	317	1076	329	1365

4. MANUFACTURING

4.1 Resin Processing

Fabrication of the base resin was made up of four constituents: an -ene containing monomer with an average of six functional groups (poly(vinylmethoxysiloxane), (VMS), and a monomer with 2 acrylate functional groups (Poly(ethyleneglycol)-diacrylate or PEGDA mixed in a 3:2 ratio by mass. The base resin also contains small amounts of photoinitiator (Irgacure® 819), and free radical scavenger (Irganox 1330™) added specifically for photopolymerization during 3D printing.

The two monomers, VMS and PEGDA were combined via a shaker table for 30 min at 650 rpm. The monomers mix well to form a translucent mixture when agitated due to the similarity of both the viscosity and density of both constituents. The free-radical scavenger and photoinitiator were both soluble in the monomer solution. Through a systematic study varying both the free-radical scavenger and photoinitiator, the final resin formulation utilized a ratio of free-radical scavenger to photoinitiator ratio of 3:7 by weight; empirically, this ratio resulted in the most distinct features for 3D printed structures.

This resin mixture, and all turbine vanes tested, were printed on an Asiga Pico2HD digital light DLP printer using a 385 nm wavelength UV light. To ensure proper curing during printing, the resin energy-curing curve was characterized, and an energy-curing curve was generated, Fig. 6. To calibrate the E-C curve, liquid resin was exposed to a range of energy doses of 30 mW/cm^2 to 3000 mW/cm^2 . These energy doses produced curing depths of 0.05 mm at 1 second of total exposure to 1.9 mm at 100 seconds. The printer then extrapolates curing parameters from the E-C curve to properly dose UV. Final parts printed were

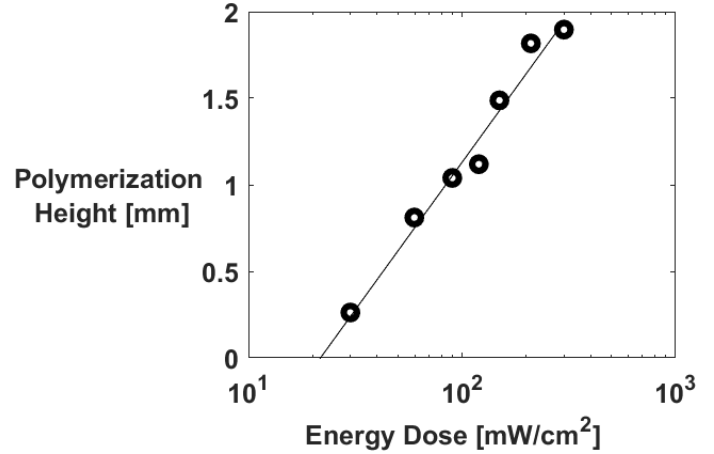


Fig. 6 Energy-curing curve used to characterize the polymerization of the pre-ceramic resin and tune printer.

accurate and compared closely to CAD generated models to within 0.5%.

After printing, parts were rinsed in an isopropanol bath to remove residual uncured resin on the surface of the vane. Subsequently the vane was post-cured under a 20 W UV lamp (Quans High Power UV LED light) with a wavelength of 405 nm for approximately 20 min.

Pyrolysis of the preceramic polymer vanes was carried out in a tube furnace under a controlled flow of argon at a rate of 100 mL/min. The heating profile was determined by analyzing TGA of the resin. The furnace was heated from room temperature to 250°C at a rate of 1°C/min for 5 hours, and then ramped up to 500°C at a rate of 1°C/min. To ensure a complete reaction and controlled radicalization, this temperature remained constant for 1 h before increasing to a maximum of 1200°C at 1°C/min. The maximum temperature remained isothermal for 1 hour before cooling at an average rate of 2°C/min from 1200°C to ambient conditions to avoid cracking due to thermal shock.

5. EXPERIMENTAL METHODS

5.1 Facility Overview

The linear cascade, which can be seen in Fig. 7, is a high-speed wind tunnel that is used to take aerodynamic and heat transfer measurements of true scale, engine airfoil geometries at matched nondimensional gas turbine conditions [34,35]. The facility uses a high flow volume compressor to achieve a range of exit Mach and exit Reynolds numbers that are independently controllable.

The airfoils used in this facility are designed in two separate parts: the center measurement airfoil and six outer periodicity airfoils, shown in Fig. 8. The measurement airfoil is the region of interest when testing that has increased instrumentation based on the experiment, whether that be for flow-field verification, spatial calibration, or heat transfer evaluation. The periodicity airfoils are used to generate a uniform flow field around the measurement airfoil. Additionally, the airfoil pack sits on a

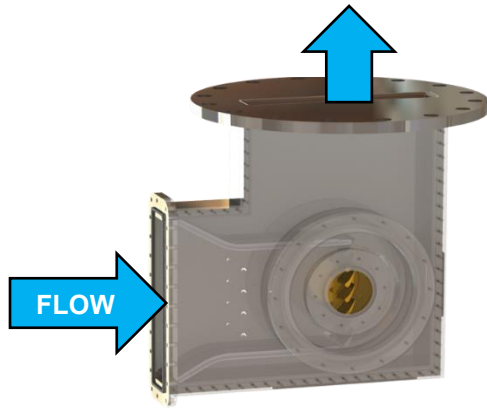


Fig. 7 Model of high-speed linear cascade test section.

rotating turntable that allows for an adjustable incidence angle (not used in this work).

With the mainstream gas path temperature equilibrating around 100°C, desiccant-dried compressed air is supplied using mass flow controllers to cool the airfoils. The mass flow controllers prescribe precise flow rates to cool the measurement airfoil for various heat transfer studies. An infrared camera, mounted on a 360° rotary traverse is used to take both pressure side and suction side infrared measurements via optical access through a zinc selenide window at the blade tip endwall. Fig. 9 displays this imaging setup, which allows for the discernment of blade surface temperatures given the window's infrared transmissivity.

5.2 Flow-field Verification

After installing a pressure-instrumented airfoil as the center measurement airfoil, static pressure measurements were taken at the midspan while the cascade was operating at the desired test conditions for an exit Mach number of 0.9, an exit Reynolds number of 390000, and an incidence angle of 0°. From prior studies, the inlet turbulent intensity is approximately 2.7% [34].

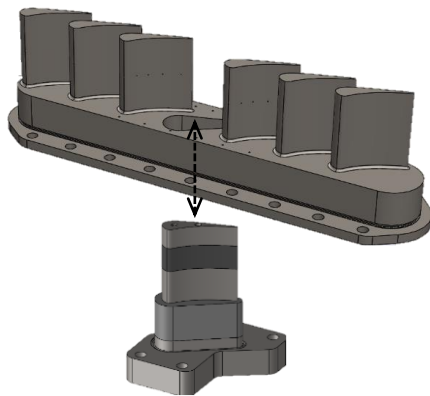


Fig. 8 Cascade airfoil assembly, with measurement airfoil that has increased instrumentation and six outer periodicity airfoils.

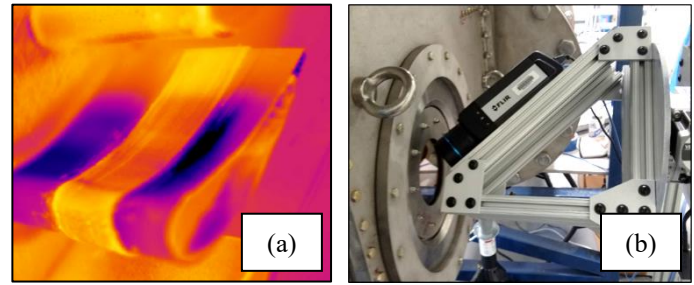


Fig. 9 (a) Raw IR image of ceramic vane assembly in cascade (b) FLIR A655 IR camera mounted to 360° traverse for airfoil viewing through zinc selenide window.

Experimental and CFD results of midspan blade loading at these conditions are shown in Fig. 10. The computational model employs a two-dimensional steady RANS simulation in Siemens Star-CCM+ commercial software; see [35] for more details and validation. The computational domain uses a single NASA C3X vane, with periodic boundary conditions to represent an infinite linear cascade. The CFD was run at the same pressure ratio conditions as the experiment. Fig. 10 shows that the flowfield at the region of interest closely matches the expected behavior.

5.3 Spatial and Temperature Calibration

To accurately evaluate the overall cooling effectiveness, the raw IR images need to be calibrated both spatially and for temperature. To spatially calibrate the airfoil surface image, an identical airfoil with IR registration markers is used as the measurement airfoil, shown in Fig. 11. The IR registration markers are arranged in a known grid pattern such that pixel coordinates corresponding to those marker locations can be used to transform the image from three-dimensional space into a two-dimensional contour plot, shown in Fig. 11.

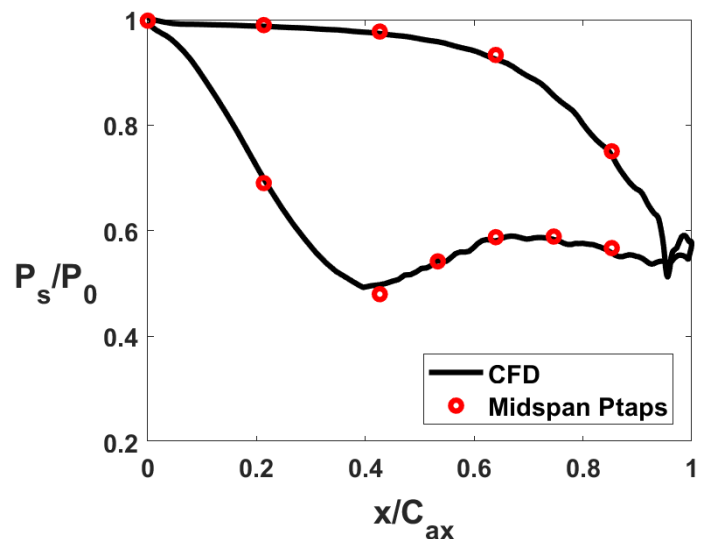


Fig. 10 Comparison of CFD and experimental midspan pressure loading distribution at Ma=0.9 and 0° incidence.

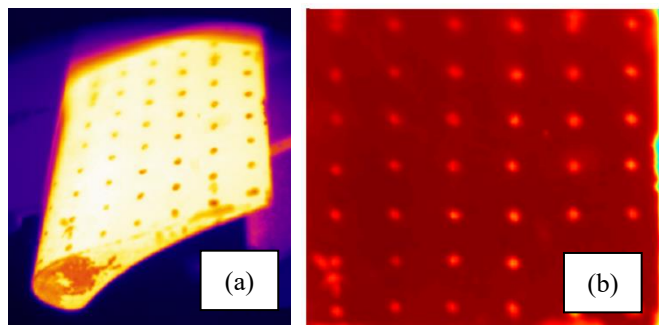


Fig. 11 (a) Spatial calibration airfoil with IR registration markers to transform surface from 3D to 2D space (b) “de-warped” airfoil contour.

Additionally, the IR camera must be calibrated such that it reads the correct airfoil surface temperature. To achieve this, the airfoil is coated in a flat black paint with an emissivity of 0.95 while type E surface thermocouples are attached to the airfoil surface at both the leading edge and trailing edge. The airfoil is then heated, using a heat gun tool, to a temperature exceeding any observed temperature during testing. As the airfoil naturally cools down, the IR camera records the blade surface temperature until it reaches a value below the lowest temperature observed during testing. Then, the actual measured surface temperatures from the thermocouples are compared to the observed IR camera readings at the same location in space and time. A scatter plot, Fig. 12, is then generated to produce a best fit curve that appropriately calibrates the observed camera temperature, in its fixed positions relative to the airfoil, to the actual temperature measured by the thermocouple. The measurement airfoil with the ceramic test piece is also coated in the same high emissivity, flat

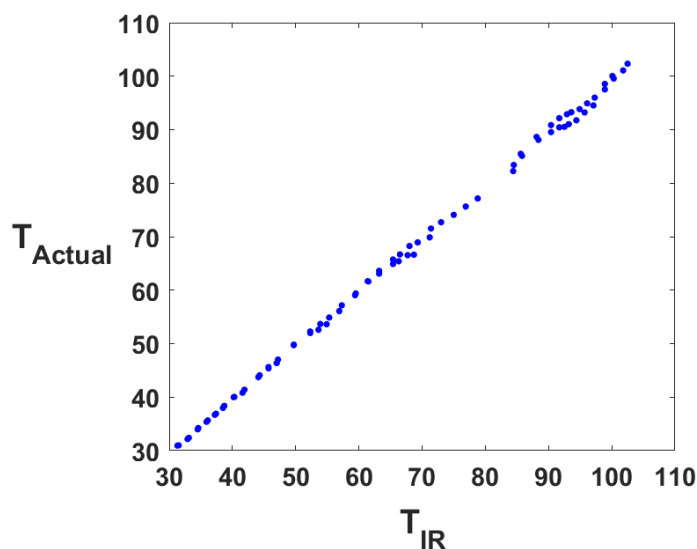


Fig. 12 IR camera temperature calibration curve relating actual, measured surface temperatures to observed IR surface temperatures.

black paint before testing to ensure accurate temperature readings at those specific viewing orientations.

5.4 Test Matrix

Gas path test conditions were limited to an exit Mach number of 0.9 and an incidence angle of 0° with an exit Reynolds number of 390000. To evaluate a range of effectiveness values, different cooling channel flow rates were supplied by targeting specific channel Reynolds numbers. By first measuring the design-intent perimeter and cross-sectional area of each cooling channel, the hydraulic diameter of each was determined. Table 3 documents the properties of each of the four channels. Then, targeting channel Reynolds numbers in the supply cavities (cavities 1 and 3 in Fig. 4) of 1650, 4100, 8200, and 16500, the mass flow rate needed from the mass flow controllers was calculated using Equation 4.

$$\dot{m} = \frac{Re_{Dh} P \mu}{4} \quad \text{Eq. (4)}$$

Note that because of a different hydraulic diameter, the Reynolds numbers in the return cavities (2 and 4 in Fig. 4) are slightly different for the same supplied mass flow. Table 4 documents the calculated flow rates needed to achieve the desired channel Reynolds numbers. The same mass flow rates were maintained for the baseline and optimized vane geometries, even though the optimized geometry had higher blockage due to the internal pin structure.

5.5 Coolant Heat-up Model

To properly evaluate the overall cooling effectiveness, an accurate value for the coolant temperature must be used. However, the temperature instrumentation is at the base of the airfoil, roughly 47.5 mm upstream of the ceramic vane inlet. Therefore, a theoretical coolant heat-up model was applied to track the cooling air heat-up through the metal base and

Table 3. Geometric parameters for airfoil cooling channels.

	A_c (mm ²)	P (mm)	D_h (mm)
Channel 1	41.18	26.85	6.14
Channel 2	30.43	23.74	5.13
Channel 3	20.73	16.90	4.91
Channel 4	13.11	20.55	2.55

Table 4. Calculated channel mass flow rates required to achieve targeted channel Reynolds numbers.

Channel 1		Channel 2		Channel 3		Channel 4	
Re_{Dh}	$\dot{m}$ (g/s)	Re_{Dh}	$\dot{m}$ (g/s)	Re_{Dh}	$\dot{m}$ (g/s)	Re_{Dh}	$\dot{m}$ (g/s)
1650	0.222	1863	0.222	1650	0.139	1347	0.139
4100	0.554	4649	0.554	4100	0.349	3383	0.349
8200	1.108	9297	1.108	8200	0.697	6756	0.697
16500	2.215	18586	2.215	16500	1.394	13513	1.394

determine a more accurate coolant temperature at the inlet of the ceramic. For this model, the two coolant supply channels were treated as pipe flow with diameters equivalent to the previously calculated hydraulic diameters. By knowing the inlet temperature, T_{in} , as the measured temperature at the base of the airfoil, Equation 5 can be used to determine the temperature at the inlet of the ceramic vane, T_{out} , for channel 1 and 3.

$$T_{out} = \bar{T}_s - (\bar{T}_s - T_{in}) \exp\left(-\frac{PL}{\dot{m}c_p} h\right) \quad \text{Eq.(5)}$$

With the known flow conditions in the cooling channel and assuming cooling air properties at 60°C (observed in the experiments), the Nusselt number was calculated for each targeted Reynolds number using the Dittus-Boelter correlation for fully developed, turbulent flow in a smooth tube. In Table 5, those Nusselt number values are documented and converted into corresponding internal convection coefficients which are used in Equation 5. Finally, the observed surface temperature of the metal base from infrared measurements was assumed to be the average constant surface temperature, $\bar{T}_s$, in Equation 5.

Once the heat-up through the two supply channels, 1 and 3, was calculated, the two coolant temperature values were averaged to yield an average coolant temperature, T_c , at the inlet of the ceramic vane. This average coolant temperature was used in the definition of overall cooling effectiveness, Equation 6.

$$\phi = \frac{T_{\infty} - T_w}{T_{\infty} - T_c} \quad \text{Eq.(6)}$$

5.7 Uncertainty Analysis

Table 6 shows the absolute uncertainties for the overall cooling effectiveness measurements at the different channel Reynolds numbers. These results were obtained by using the propagation of measurement uncertainties method [36] with a confidence interval of 95%.

6. COOLING EFFECTIVENESS RESULTS

This section presents the overall effectiveness results for the testing of the additively manufactured ceramic vanes, both baseline and optimized designs. Comparisons were made

Table 5. Calculated Nusselt number and internal heat transfer coefficient for each channel at the targeted Reynolds numbers.

	Channel 1		Channel 2		Channel 3		Channel 4	
Re_{Dh}	Nu_{Dh}	$h \left(\frac{W}{m^2 K} \right)$	Nu_{Dh}	$h \left(\frac{W}{m^2 K} \right)$	Nu_{Dh}	$h \left(\frac{W}{m^2 K} \right)$	Nu_{Dh}	$h \left(\frac{W}{m^2 K} \right)$
1650	7.55	34.6	8.33	45.6	7.52	43.0	6.43	70.8
4100	15.70	71.8	17.32	94.9	15.71	89.9	13.43	147.8
8200	27.33	125.1	30.16	165.2	27.32	156.3	23.36	257.1
16500	47.57	217.7	52.49	287.5	47.56	272.2	40.67	447.6

Table 6. Absolute uncertainty in overall cooling effectiveness.

Re	Absolute Uncertainty in ϕ
1650	± 0.047
4100	± 0.037
8200	± 0.031
16500	± 0.027

between the different targeted channel Reynolds numbers: 1650, 4100, 8200, and 16500, to quantify the effect of coolant flow rate. Additionally, comparisons were made between the baseline and optimized ceramic vanes to quantify the percentage increase in overall cooling effectiveness and match to the thermomechanical optimization model. Finally, the overall cooling effectiveness was evaluated for the metal portion of the airfoil span to draw comparisons to the ceramic vane results.

6.1 Reynolds Number Comparison for Baseline Vane

The overall effectiveness contours of the baseline ceramic vane can be seen in Fig. 13 and Fig. 14. Fig. 13 shows both pressure side and suction side views of the airfoil at a low internal channel Reynolds number while Fig. 14 shows pressure side and suction side views at a high internal channel Reynolds number. The pressure side is represented by a positive s/s_{max} and the suction side is negative s/s_{max} . It is clear from comparing Fig. 13 and Fig. 14 that increasing the channel Reynolds number results in improved cooling near the forward pressure side, where the external gas path velocity and the associated external convection coefficient is relatively low. The internal cooling is less effective at the stagnation point ($s/s_{max}=0$) and at the trailing edge (s/s_{max} close to 1 or -1). In contrast, the overall effectiveness does not change significantly for the suction side of the airfoil, where gas path velocities and external convection coefficients are relatively high, and the overall effectiveness is low since the airfoil surface is close to the gas path temperature.

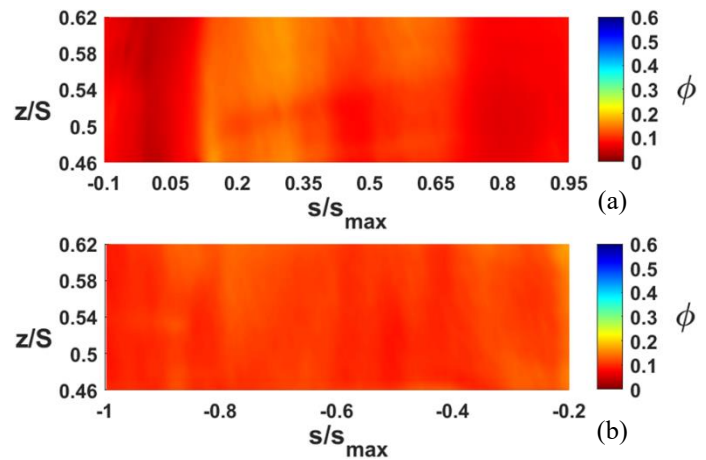


Fig. 13 Overall effectiveness contour plot for (a) pressure side and (b) suction side at $Re=4100$.

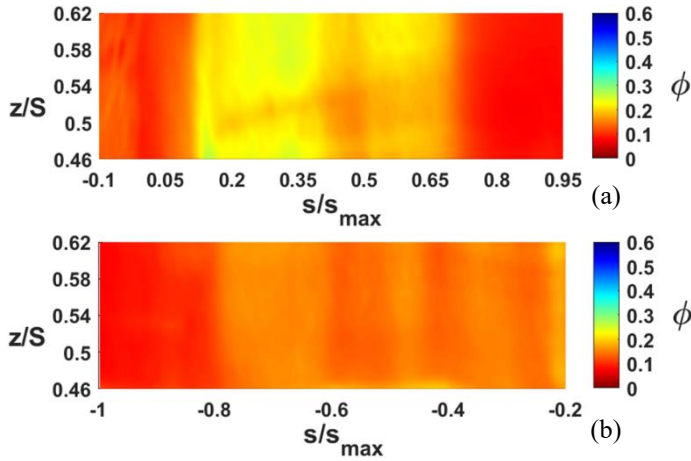


Fig. 14 Overall effectiveness contour plot for (a) pressure side and (b) suction side at $Re=16500$.

Fig. 15 shows the laterally averaged cooling effectiveness values as a function of s/s_{max} for the baseline ceramic vane at each of the different Reynolds numbers, where both the leading edge (LE, channel 1) and trailing edge (TE, channel 3) coolant feeds operate at the same internal channel Reynolds number.

In Fig. 15, the suction side has relatively unchanging overall effectiveness even for a 10X change in internal channel Reynolds number. This suggests that the external convection dominates the conjugate heat transfer through the vane surface for the suction side, bringing the surface close to the gas path temperature. In contrast, the pressure side is highly sensitive to internal channel Reynolds number, suggesting that the internal convection dominates for the pressure side cooling.

In Fig. 15, regions of increased or decreased cooling effectiveness can be attributed to locations on the surface, beneath which there is either free-flowing coolant or a partition wall that segments the cooling passages. Again, refer to Fig. 3

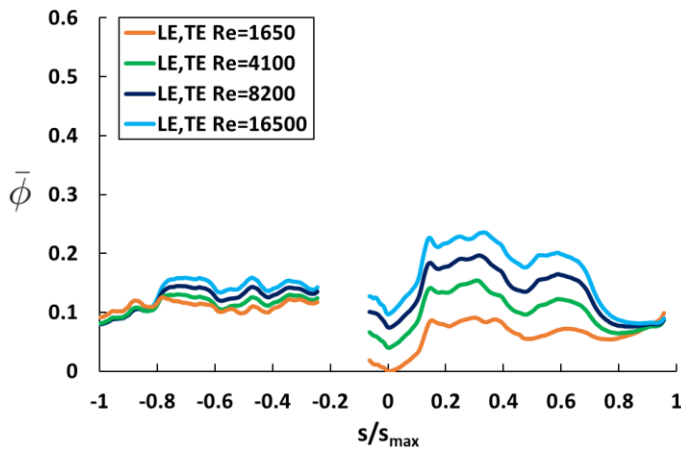


Fig. 15 Laterally averaged, overall effectiveness values for the baseline ceramic vane at varied internal channel Reynolds numbers.

for the airfoil surface wall schematic that marks these transitional s/s_{max} locations. Additionally, a local minimum in laterally averaged, overall cooling effectiveness can be observed at the location where s/s_{max} is approximately zero. This location is very nearly the aerodynamic stagnation point, so the impingement of the hot mainstream gas path is likely the cause for this drop in cooling effectiveness.

6.2 Baseline and Optimized Vane Comparison

For the same internal cooling mass flow conditions, a noticeable increase in cooling effectiveness is observed for the optimized vane. The laterally averaged, overall cooling effectiveness values for the optimized vane can be seen in Fig. 16. The optimized vane utilizes an array of pin fins within the internal cooling channels that augment heat transfer out of the part. For a given channel Reynolds number, the overall cooling effectiveness more than doubles for the optimized ceramic vane in Fig. 16 relative to the baseline in Fig. 15. Interestingly, the suction side is more sensitive to internal channel Reynolds number for the optimized vane suggesting that the high pin density has increased the internal convection to a level more comparable to the external convection. Certainly, for the pressure side, the increase in overall effectiveness with channel Reynolds number is increased further relative to the baseline. Additionally, for the optimized vane, the variation of high and low effectiveness over the surface are more pronounced in comparison to the baseline vane. This difference is due to the 0.5mm thinner external wall of the optimized ceramic vane in conjunction with the internal channel pin fins. With an augmented heat transfer rate out of the part, the vane is cooled more effectively at surface locations that align with cooling passages while still having relatively lower effectiveness values at regions where a partition wall is located.

The results for the baseline and optimized cases are also compared to the predictions from the thermomechanical

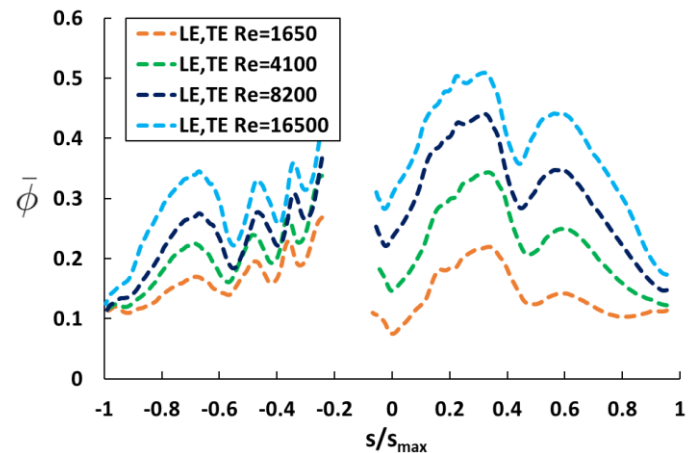


Fig. 16 Laterally averaged, overall effectiveness for the optimized ceramic vane at varied internal channel Reynolds numbers.

optimization model. Fig. 17 shows the simulations for the baseline and optimized vane design at a channel Reynolds number of 8200 as well as the experimental results at the same Reynolds number. There is reasonable agreement in the baseline case, due to the relatively well understood convection coefficients that could be applied as boundary conditions in the FEA (turbulent pipe flow on the internal channels; measurements by Hylton, et al. [31] on the gas path side). In contrast, the optimization model overpredicts the experimental results for the optimized case. It is likely that the textbook correlations for convection coefficients for tube banks [33] used in the initial development of the optimization model do not closely model the actual internal convection coefficient.

Nondimensional pressure drop across both the leading edge cooling channels and trailing edge cooling channels can be compared for the baseline and optimized vanes, as seen in Fig. 18. The pressure drops are normalized by the total pressure in the cascade which is approximately 156 kPa throughout the duration of testing. Naturally, a larger pressure drop is observed across both coolant u-loops in the optimized case for a given channel Reynolds number. This is due to the pin fin array that is employed to augment heat transfer to the surface. Increased channel Reynolds numbers also yields increased pressure drop, with the trailing edge channel experiencing larger pressure drops. While the mass flow rate in the trailing edge coolant loop is comparatively lower to match the channel Reynolds number to the leading edge, the smaller cross-sectional area of those channels is smaller, which yields larger velocities. These larger channel velocities are likely the source of increased pressure drop in the trailing edge.

6.3 Metal and Ceramic Comparisons

In addition to measuring the overall cooling effectiveness of the ceramic vane, the overall cooling effectiveness of the metal base (with the same exact internal cooling structure) was evaluated and compared at the same Reynolds numbers, shown

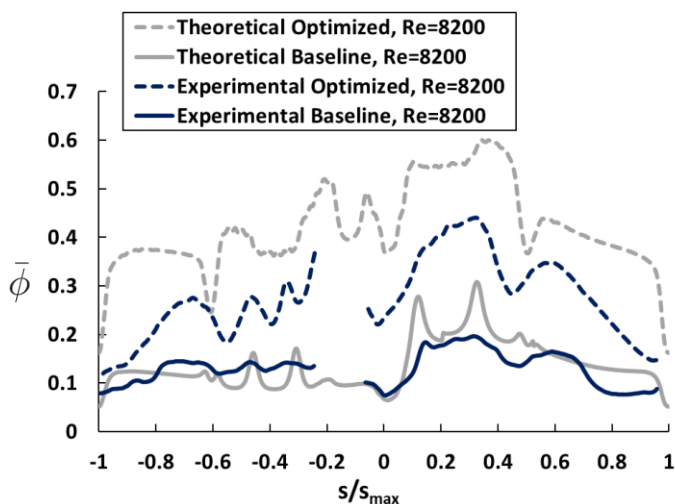


Fig. 17 Laterally averaged overall, effectiveness for the experiment and the optimization model at Re=8200.

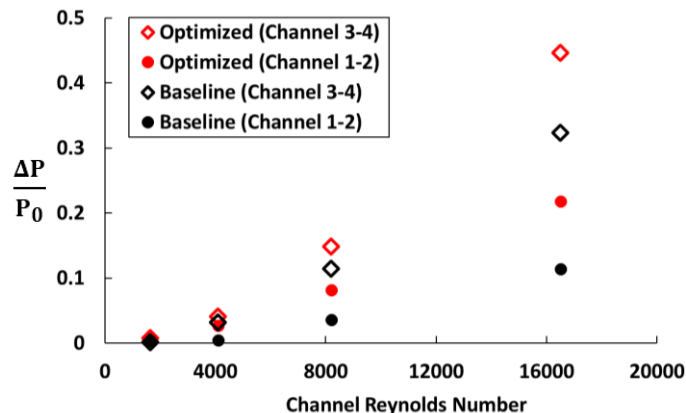


Fig. 18 Nondimensional pressure drop across the cooling channels for the baseline and optimized ceramic vane.

in Fig. 19. The metal vane experiences less effective cooling when compared to the optimized ceramic vane, specifically along the pressure side, while the suction side experiences similar effectiveness levels for the metal and optimized vanes. Additionally, the ceramic vane yields more distinct regions of higher and lower cooling effectiveness than the metal vane. This is likely due to the larger temperature gradients that are present in the lower thermal conductivity ceramic than the metal.

While the optimized internal vane configuration is associated with a larger pressure drop, as seen in Fig. 18, it does provide elevated levels of cooling performance, shown in Fig. 20. The area averaged overall effectiveness values for the optimized ceramic vane, across both pressure side and suction side, exceeds that of the metal vane at a Reynolds number of 8200 and 16500. Based on these area averaged effectiveness values at two different Reynolds numbers and assuming an approximately linear behavior at Reynolds numbers in between,

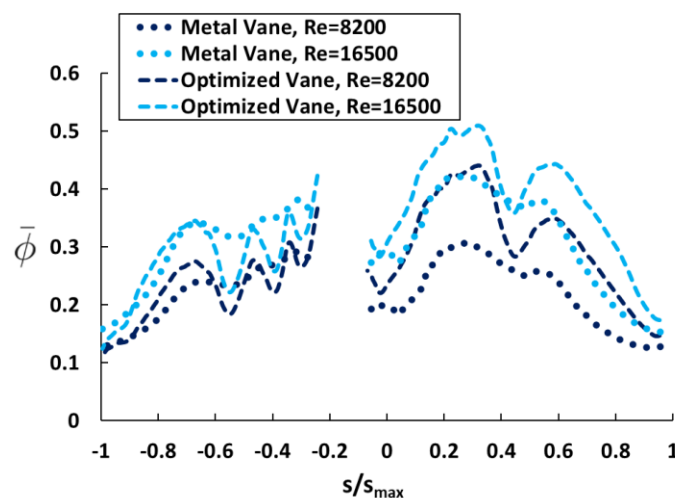


Fig. 19 Laterally averaged, overall effectiveness for the optimized ceramic vane compared to the same metallic vane portion in this study.

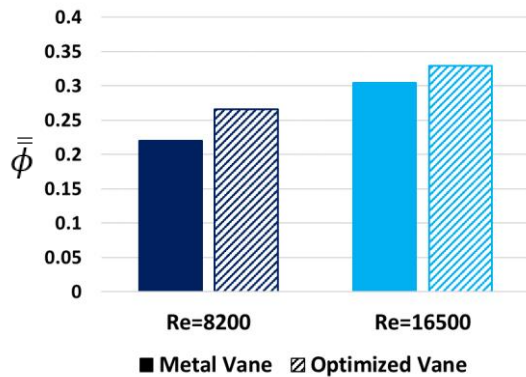


Fig. 20 Average cooling effectiveness values seen along the baseline and optimized vane surface.

it was estimated that the optimized vane could achieve the same area averaged cooling effectiveness as the metal vane with a 20% reduction in coolant flow rate from a Reynolds number of 16500. This reduction in coolant flow rate also results in a 25% reduction in pressure drop for the optimized vane. This reduction in pressure drop corresponds to a leading edge (channels 1-2) pressure drop that is now only 45% larger in the optimized vane than in the baseline vane, as opposed to the original 100% larger pressure drop originally observed in Fig. 18 at a Reynolds number of 16500. And, in the trailing edge (channels 3-4), the pressure drop is only 2% larger for the optimized vane compared to the baseline vane.

7. CONCLUSIONS

A process for additively manufacturing ceramic turbine vanes, thermo-mechanically optimizing the vane's internal cooling scheme, and comparatively evaluating overall cooling effectiveness in a high-speed flow facility is discussed. This study shows that a polymer derived ceramic (PDC) additive manufacturing (AM) process can successfully print a 1X scale, aviation turbine vane that can be internally cooled and perform under relevant nondimensional engine conditions. Furthermore, this AM process can print complex internal cooling schemes, as resolved by a thermomechanical optimization model, that clearly exhibit improved levels of overall cooling effectiveness in comparison to simple internally cooled airfoils.

This study also shows that a ceramic vane with simple internal cooling has a significantly lower overall cooling effectiveness than a metallic vane due to a significant difference in material conductivity. Although ceramics have a higher temperature tolerance than metals, the modest level of cooling from a simple cooling channel may not be sufficient to maintain the ceramic at an acceptable temperature for state-of-the-art turbine inlet temperatures.

To counteract the lower thermal conductivity of ceramic and allow further increase in turbine inlet temperature, internal cooling features in a ceramic can provide cooling performance levels that are comparable to those of a simple internally cooled metal vane for less cooling than would be needed in an engine application to achieve an acceptable material temperature.

ACKNOWLEDGEMENTS

The authors would like to acknowledge the assistance of Alexander Rusted in developing the cascade and airfoil design, as well as the U.S. Department of Energy's National Energy Technology Laboratory for sponsoring research presented in this paper. This paper is based upon work supported by the Department of Energy under Award Number DE-FE0031758.

REFERENCES

- [1] Grondahl, C. M., and Tsuchiya, T., 2001, "Performance Benefit Assessment of Ceramic Components in an MS9001FA Gas Turbine," *J. Eng. Gas Turbines Power*, **123**(3), pp. 513–519.
- [2] Faulder, L., McClain, J., Edwards, B., and Parthasarathy, V., 1998, "Ceramic Stationary Gas Turbine Development Program - Design and Test of a First Stage Ceramic Nozzle," *Proceedings of the ASME Turbo Expo*, American Society of Mechanical Engineers Digital Collection.
- [3] Jimenez, O., McClain, J., Edwards, B., Parthasarathy, V., Bagheri, H., and Bolander, G., 1998, "Ceramic Stationary Gas Turbine Development Program - Design and Test of a Ceramic Turbine Blade," *Proceedings of the ASME Turbo Expo*, American Society of Mechanical Engineers Digital Collection.
- [4] Miriyala, N., Fahme, A., and Van Roode, M., 2001, "Ceramic Stationary Gas Turbine Program - Combustor Liner Development Summary," *Proceedings of the ASME Turbo Expo*, American Society of Mechanical Engineers Digital Collection.
- [5] Price, J., Jimenez, O., Miriyala, N., Kimmel, J. B., Leroux, D. R., and Fahme, T., 2001, "Ceramic Stationary Gas Turbine Development Program Eighth Annual Summary," *Proceedings of the ASME Turbo Expo*, American Society of Mechanical Engineers (ASME).
- [6] Van Roode, M., Jimenez, O., McClain, J., Price, J., Parthasarathy, V., Poormon, K. L., Ferber, M. K., and Lin, H. T., 2002, "Ceramic Gas Turbine Materials Impact Evaluation," *American Society of Mechanical Engineers, International Gas Turbine Institute, Turbo Expo (Publication) IGTI*, American Society of Mechanical Engineers Digital Collection, pp. 77–84.
- [7] Vedula, V., Shi, J., Jarmon, D., Ochs, S., Oni, L., Lawton, T., Green, K., Prill, L., Schaff, J., Linsey, G., and Zadrozny, G., 2005, "Ceramic Matrix Composite Turbine Vanes for Gas Turbine Engines," *Proceedings of the ASME Turbo Expo*, American Society of Mechanical Engineers Digital Collection, pp. 247–251.
- [8] "GE Recognized by the American Ceramics Society for World First Commercializing Ceramic Matrix Composites (CMCs) in Aircraft Engines | GE News" [Online]. Available: <https://www.ge.com/news/press-releases/ge-recognized-american-ceramics-society-world-first-commercializing-ceramic-matrix>. [Accessed: 21-Jun-2021].

- [9] van Roode, M., 2010, "Ceramic Gas Turbine Development: Need for a 10 Year Plan," *J. Eng. Gas Turbines Power*, **132**(1).
- [10] Kopeliovich, D., 2018, "Advances in Manufacture of Ceramic Matrix Composites by Infiltration Techniques," *Advances in Ceramic Matrix Composites: Second Edition*, pp. 93–119.
- [11] Bansal, N. P., and Lamon, J., 2014, *Ceramic Matrix Composites: Materials, Modeling and Technology*, Wiley Blackwell.
- [12] Bansal, N. P., 2005, *Handbook of Ceramic Composites*, Springer US.
- [13] Bhatia, T., Eaton, H., Sun, E., Lawton, T., and Vedula, V., 2005, "Advanced Environmental Barrier Coatings for SiC/SiC Composites," *Proceedings of the ASME Turbo Expo*, American Society of Mechanical Engineers Digital Collection, pp. 253–258.
- [14] Bhatia, T., Vedula, V., Eaton, H., Sun, E., Holowczak, J., and Linsey, G., 2004, "Development and Evaluation of Environmental Barrier Coatings for Si-Based Ceramics," *Proceedings of the ASME Turbo Expo 2004*, American Society of Mechanical Engineers Digital Collection, pp. 431–438.
- [15] Deckers, J., Vleugels, J., and Kruth, J. P., 2014, "Additive Manufacturing of Ceramics: A Review," *J. Ceram. Sci. Technol.*, **5**(4), pp. 245–260.
- [16] Lakhdar, Y., Tuck, C., Binner, J., Terry, A., and Goodridge, R., 2021, "Additive Manufacturing of Advanced Ceramic Materials," *Prog. Mater. Sci.*, **116**, p. 100736.
- [17] Zocca, A., Colombo, P., Gomes, C. M., and Günster, J., 2015, "Additive Manufacturing of Ceramics: Issues, Potentialities, and Opportunities," *J. Am. Ceram. Soc.*, **98**(7), pp. 1983–2001.
- [18] He, R., Zhou, N., Zhang, K., Zhang, X., Zhang, L., Wang, W., and Fang, D., 2021, "Progress and Challenges towards Additive Manufacturing of SiC Ceramic," *J. Adv. Ceram.*, **10**(4), pp. 637–674.
- [19] Eckel, Z. C., Zhou, C., Martin, J. H., Jacobsen, A. J., Carter, W. B., and Schaedler, T. A., 2016, "Additive Manufacturing of Polymer-Derived Ceramics," *Science* (80-.), **351**(6268), pp. 58–62.
- [20] Wang, M., Xie, C., He, R., Ding, G., Zhang, K., Wang, G., and Fang, D., 2019, "Polymer-Derived Silicon Nitride Ceramics by Digital Light Processing Based Additive Manufacturing," *J. Am. Ceram. Soc.*, **102**(9), pp. 5117–5126.
- [21] Brinckmann, S. A., Young, J. C., Fertig, R. S., and Frick, C. P., 2023, "Effect of Print Direction on Mechanical Properties of 3D Printed Polymer-Derived Ceramics and Their Precursors," *Mater. Lett. X*, **17**, p. 100179.
- [22] Brinckmann, S. A., Patra, N., Yao, J., Ware, T. H., Frick, C. P., and Fertig III, R. S., 2018, "Stereolithography of SiOC Polymer-Derived Ceramics Filled with SiC Micronwhiskers," *Adv. Eng. Mater.*, **20**(11), p. 1800593.
- [23] O'Masta, M. R., Stonkevitch, E., Porter, K. A., Bui, P. P., Eckel, Z. C., and Schaedler, T. A., 2020, "Additive Manufacturing of Polymer-Derived Ceramic Matrix Composites," *J. Am. Ceram. Soc.*, **103**(12), pp. 6712–6723.
- [24] Han, J. C., Dutta, S., and Ekkad, S., 2012, "Gas Turbine Heat Transfer and Cooling Technology," *Gas Turbine Heat Transfer and Cooling Technology*, Taylor & Francis, New York, NY, USA, pp. 1–844.
- [25] Zhang, S., Shen, X., Li, Z., Liu, X., and Dong, S., 2022, "Strength Analysis of Two-Dimensional Ceramic-Matrix-Composite Guide Vane Based on Bimodulus Constitutive Model," *AIAA J.*, **60**(11), pp. 6420–6435.
- [26] Bryant, C. E., and Rutledge, J. L., 2021, "Conjugate Heat Transfer Simulations to Evaluate the Effect of Anisotropic Thermal Conductivity on Overall Cooling Effectiveness," *J. Therm. Sci. Eng. Appl.*, **13**(6).
- [27] Wilkins, P. H., Lynch, S. P., Thole, K. A., Quach, S., and Vincent, T., 2021, "Experimental Heat Transfer and Boundary Layer Measurements on a Ceramic Matrix Composite Surface," *J. Turbomach.*, **143**(6).
- [28] Wilkins, P. H., Lynch, S. P., Thole, K. A., Vincent, T., Quach, S., and Kaufman, E., 2022, "Experimental Investigation Into the Effect of a Ceramic Matrix Composite Surface on Film Cooling," *J. Turbomach.*, **144**(12).
- [29] Cory, T. M., Edelson, R. D., Thole, K. A., Vincent, T., Whitney, P. &, and Mongillo, D., 2022, "Impact of Ceramic Matrix Composite Topology on Friction Factor and Heat Transfer," *J. Turbomach.*, **144**(3).
- [30] Edelson, R. D., and Thole, K. A., 2023, "Impact of Ceramic Matrix Composite Topology on Overall Effectiveness," *J. Turbomach.*, **145**(3).
- [31] Hylton, L. D., Mihelc, M. S., Turner, E. R., Nealy, D. A., and York, R. E., 1983, *Analytical and Experimental Evaluation of the Heat Transfer Distribution Over the Surfaces of Turbine Vanes*.
- [32] Stabler, C., Reitz, A., Stein, P., Albert, B., Riedel, R., & Ionescu, E. (2018). Thermal Properties of SiOC Glasses and Glass Ceramics at Elevated Temperatures. *Materials*, *11*(2), 279. <https://doi.org/10.3390/ma11020279>
- [33] Bergman, T. L., Incropera, F. P., DeWitt, D. P., and Lavine, A. S., 2011, *Fundamentals of Heat and Mass Transfer*, John Wiley & Sons.
- [34] Zuccarello, J. T., Saltzman, D., Lynch, S. P., Haydt, S., and Whitfield, C. A., 2020, "A Steady Transonic Linear Cascade for True Scale Cooling Measurements," *Proceedings of ASME Turbo Expo*, ASME, London, UK.
- [35] Rusted, A., and Lynch, S., 2021, "Determining Total Pressure Fields from Velocimetry Measurements in a Transonic Turbine Flowfield," *Proceedings of the ASME Turbo Expo*, pp. 1–12.
- [36] Moffat, R. J., 1988, "Describing the Uncertainties in Experimental Results," *Exp. Therm. Fluid Sci.*, **1**(1), pp. 3–17.