1 Introduction

The components in the turbine section of a gas turbine engine present several complex problems to engine designers. One of the problems that designers face is the high gas path temperatures that pass through the turbine section. The gas path temperatures continue to be increased in order to increase the gas turbine power output and efficiency. In most engines, the gas path temperature has increased to be well above the melting temperature of most engine components. These high temperatures can drastically decrease the operating life of an engine component if not cooled properly.

To increase the component operating life, designers typically use computational fluid dynamics (CFD) and other analysis tools to predict the regions of a component that will require a cooling scheme to be used. The heat transfer models used in these predictions can be complicated since the heat transfer is affected by several factors, which include Reynolds number, turbulence intensity, turbulence length scale, blade curvature, and pressure gradient just to name a few. The efforts of this work are to examine the effects of high freestream turbulence levels and turbulence length scales on heat transfer from a turbine blade at realistic engine Mach number conditions.

1.1 Summary of Past Literature. The freestream turbulence in the turbine section of an engine is due to the velocity fluctuations of the flow created by the combustor system and vane wakes upstream of the blade passages. While measuring the turbulence level in a gas turbine is extremely difficult, researchers have found that combustion systems typically produce turbulence levels between 7% and 30% [1,2]. The turbulence produced by the combustor decays as the flow passes through the vanes, but additional velocity fluctuations will be created by the vane wakes.

Several experimental studies have been performed in transonic cascades that investigate the effects of exit Reynolds number and freestream turbulence on the surface heat transfer distribution over a turbine blade. Consigny and Richards [3] measured the blade surface heat transfer distribution by varying the freestream turbulence between 0.8% and 5.2%. Consigny and Richards [3] observed that increasing the turbulence level augmented the heat transfer on the pressure and suction sides of the blade as compared with the low freestream turbulence case. At nominal conditions, exit Mach 0.78, average heat transfer augmentations of 23% and 35% were observed on the pressure side and suction side of the blade, respectively.

This paper experimentally investigates the effect of high freestream turbulence intensity, turbulence length scale, and exit Reynolds number on the surface heat transfer distribution of a turbine blade at realistic engine Mach numbers. Passive turbulence grids were used to generate freestream turbulence levels of 2%, 12%, and 14% at the cascade inlet. The turbulence grids produced length scales normalized by the blade pitches of 0.02, 0.26, and 0.41, respectively. Surface heat transfer measurements were made at the mid-span of the blade using thin film gauges. Experiments were performed at the exit Mach numbers of 0.55, 0.78, and 1.03, which represent flow conditions below, near, and above nominal conditions. The exit Mach numbers tested correspond to exit Reynolds numbers of $6 \times 10^5$, $8 \times 10^5$, and $11 \times 10^5$, based on true chord. The experimental results showed that the high freestream turbulence augmented the heat transfer on both the pressure and suction sides of the blade as compared with the low freestream turbulence case.
2 Experimental Setup and Instrumentation

2.1 Wind Tunnel Facility. The two-dimensional transonic cascade wind tunnel, shown in Fig. 1, is a blow-down facility that is capable of sustaining a constant pressure at the cascade inlet for up to 25 s. Prior heat transfer research that has been performed in this facility includes work by Holmberg [6], Nix et al. [7], Smith et al. [8], and Popp et al. [9]. Air is supplied from high-pressure air tanks that are charged up to 1380 kPa (200 psi (gauge)) prior to testing. A control valve regulates the flow from the air tanks to the test section. Cascade inlet pressures range from 20.7 kPa (3 psi (gauge)) to 69.0 kPa (10 psi (gauge)) depending on the objective test conditions. Between the control valve and the test section, the air passes through a passive heat exchanger, which heats the cascade inlet flow up to 120°C. After the air passes through the heat exchanger, the air goes through a contraction and enters the test section before being exhausted to the atmosphere.

The turbine blade tested in these experiments is similar in geometry to a first stage turbine blade for a small industrial gas turbine. The blade was scaled two times so that the nominal exit Reynolds number would be at the desired value. Table 1 summarizes the geometry of the turbine blade.

A diagram of the blade cascade is provided in Fig. 2. From the blade geometry and the test section size, the blade cascade consists of seven full blades and two partial blades, which result in seven full passages and one partial passage. A tailboard placed at the blade exit angle aids in creating periodic flow through the cascade. The blades are numbered starting from the lower left of the cascade with blade 4 being the blade that is fully instrumented to make static pressure and heat transfer measurements. The slot located 0.6C upstream of the cascade is used to measure the turbulence and velocity distributions at the inlet of the cascade.

2.2 Static Pressure Measurements. To calculate the isentropic Mach number distribution on the blade surface, the turbine blade was instrumented with static pressure taps placed at the midspan of the blade. Blade 4 was instrumented with a total of 27 pressure taps with 9 taps on the pressure side, 17 taps on the suction side, and 1 tap near the leading edge. Static pressure taps were also instrumented on suction side of blade 3 and on the pressure side of blade 5 to check the periodicity of the flow. The total pressure of the flow was measured using a pitot static probe located upstream of the cascade inlet. The static pressure measurements were made through independently conducted experiments relative to the heat transfer experiments. In addition to calculating the isentropic Mach number distribution, the acceleration parameter distribution on the blade surface was also calculated.

Static pressure taps on the endwall of the cascade were used to measure the inlet and exit static pressures and characterize the inlet and exit flows. The static pressures measured by these taps are also used to calculate the inlet and exit flow conditions. 12 inlet static taps were located 0.6C upstream of the blade passages and 12 exit static taps were located 0.6C downstream of the blade passages.

2.3 Heat Transfer Measurements. Heat transfer measurements were made with thin film gauges that allow for high spatial resolution measurements on the blade surface with minimal flow disruption. Thin film gauges were originally developed by Schultz and Jones [10] and variations of the original design have been used by Doorly and Oldfield [11] and Dunn [12]. The thin film gauges that were used in these experiments are two-layer thin film gauges similar to the gauges developed by Doorly and Oldfield [11]. The gauges were manufactured according to the procedure described by Joe [13].

Each thin film gauge consists of a platinum sensor that is 3.18 mm (0.125 in.) long that attach copper leads, which are sputtered to a Kapton (k=0.12 W/m K) sheet that is 50 μm thick. The Kapton sheet with the gauges is attached to a blade manufactured from a low thermal conductivity ceramic material called Macor (k=1.46 W/m K). A photograph of the gauges installed on the blade is shown in Fig. 3. The platinum sensor was placed at the midspan of the blade and a total of 36 thin film gauges were instrumented on the blade.

Thin film gauges are used to measure a change in temperature on the surface of the blade. The platinum sensor of the thin film gauge changes resistance with temperature and each gauge is cali-
brated prior to testing. Since the gauge changes resistance with temperature, the gauge is used as one arm of a Wheatstone bridge circuit. The Wheatstone bridge used in these experiments is described by Joe [13]. The change in voltage across the bridge during the experiment is sampled at 1 kHz during the experiment using a 16 bit NI SCXI-1600 data acquisition system. The data from up to 31 gauges can be recorded in this facility during a single test.

To reduce the heat transfer data, several steps must be taken. The voltage output from each Wheatstone bridge is converted into surface temperature using the gauge calibration and basic Wheatstone bridge operating principles. Next, the heat flux for each gauge is calculated by using a finite-difference code developed by Cress [14]. The finite-difference code uses the time history of the surface temperature of a gauge as a boundary condition and solves the one-dimensional, transient heat conduction equation. Over the majority of the blade, the conduction is assumed to be semi-infinite since the Macor conducts heat very slowly. Near the trailing edge, the heat flux is calculated over a finite material thickness with the surface temperatures measured by the gauges on each side of the blade used as the boundary conditions to calculate the heat flux. Once the heat flux is determined, the heat transfer coefficient can then be calculated by using

$$h = \frac{q''}{(T_{aw} - T_{gauge})}$$  \hspace{1cm} (1)

where the adiabatic wall temperature is defined as

$$T_{aw} = T_{aw} \cdot \left( 1 + \frac{\gamma - 1}{2} \cdot \frac{Ma^2}{\gamma - 1} \right)$$  \hspace{1cm} (2)

It was assumed that the boundary layer was turbulent everywhere, so a recovery factor of $r_c = P_{c}^{1/3}$ was applied to all of the gauges. The heat transfer coefficient can then be non-dimensionalized by calculating the Nusselt number by using

$$Nu = \frac{h \cdot C}{k_a}$$  \hspace{1cm} (3)

The heat transfer coefficient was also be non-dimensionalized in terms of the Stanton number given by

$$St = \frac{h}{\rho \cdot C_p \cdot U_L \cdot C_p}$$  \hspace{1cm} (4)

### 2.4 Uncertainty Analysis

The experimental uncertainty of the heat transfer measurements was calculated by using the procedure developed by Moffat [15]. The analysis took into account the bias error and precision error. An uncertainty was calculated for each gauge at every test condition. The total uncertainty of the heat transfer coefficient for the gauges ranged between 8.5% and 11.5% with the bias error contributing the majority to the total uncertainty. For each test condition, measurements were performed at least three times to establish repeatability.

### 3 Turbulence Generation

To generate freestream turbulence levels of 12% and 14%, passive turbulence grids were used. Schematics of the turbulence grids are provided in Fig. 4. The grid designs were based on the correlations reported by Baines and Peterson [16] and research performed by Nix et al. [17] on turbulence grids in the cascade wind tunnel. The first turbulence grid is a square mesh grid. The square mesh grid has bar widths of 1.91 cm (0.75 in.) and spaced to create 3.81 x 3.81 cm$^2$ (1.5 x 1.5 in.$^2$) square openings. The solidity of the square mesh grid is 48%. The second turbulence grid is a parallel bar grid. This grid has bars that are 6.35 cm (2.5 in.) wide and spaced 4.76 cm (1.875 in.) apart to create a solidity of 50%.

The location of the turbulence grids relative to the test section are provided in Fig. 5. Both grids are oriented so that the flow is perpendicular to the bars. Spacers that were 3.18 cm (1.25 in.) thick were added or removed to achieve the desired turbulence level in the test section. The mesh grid was placed downstream of the two-dimensional contraction with two spacers placed between the mesh grid and the test section. The bar grid was placed upstream of the contraction and four spacers were placed between the contraction and the test section.

The velocity fluctuations in the streamwise direction were measured using a single hot-film probe with a 50 μm diameter film that was roughly 1.5 mm long that was connected to a constant-temperature anemometer. Hot-film data were sampled for approximately 1.3 s at 100 kHz and filtered at 40 kHz. The hot-film probe was discretely traversed over one blade pitch along the slot parallel to the blade inlet plane, as shown in Fig. 2. The turbulence intensity and integral turbulence length scales were calculated at each measurement location. The turbulence length scales were calculated using the methods described by Nix et al. [17] that applies Taylor’s hypothesis of frozen turbulence. The turbulence levels and length scales were measured for the square mesh grid, the bar grid, and a baseline case where no turbulence grid was installed in the tunnel. The turbulence intensity and the normalized integral length scale distributions along the blade inlet pitch are provided in Figs. 6 and 7, respectively. The turbulence levels and normalized length scales are provided in Table 2.

The turbulence intensity and length scale measurements were also compared with the literature correlations for grid-generated turbulence in the streamwise direction using the coefficient values reported by Roach [18]. The turbulence decay for both grid geometries is given by

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**Fig. 4 Turbulence grids (a) mesh grid, Tu=12% and (b) bar grid, Tu=14%**

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**Fig. 5 Turbulence grid location relative to the test section for the (a) mesh grid and (b) bar grid**
Tu = 1.13 \cdot \left(\frac{x}{B}\right)^{-5/7} \quad (5)

and the increase in length scale is provided by

\frac{\Lambda_z}{B} = 0.2 \cdot \left(\frac{x}{B}\right)^{1/2} \quad (6)

The turbulence decay for the mesh and bar grids is provided in Fig. 8. The turbulence decay for both the grids show similar trends to the correlation but are a lower turbulence levels as compared with the correlation. The measured turbulence levels being lower than the correlation were also observed by Nix et al. [17] in this facility.

The increase in the diameter of the length scale for the mesh and bar grids is provided in Fig. 9. The measured turbulence length scales show similar trends to the correlation. The mesh grid matches the correlation almost exactly, whereas the bar grid lies below the correlation. The turbulence length scale for the bar grid lies below the correlation since the flow passes through a contraction after the bar grid.

To check that the inlet flow to the blade passages downstream of the turbulence grids was uniform, a Kiel probe was traversed along the measurement slot shown in Fig. 2. A velocity ratio was calculated by dividing the velocity measured from the Kiel probe by the velocity measured from a stationary pitot probe. The velocity ratio for the each configuration tested is provided in Fig. 10. The bar grid shows a slight variation in inlet velocity along the angled slot. The trade-off of generating high turbulence levels with the bar grid was that the inlet velocity was slightly nonuniform.

<table>
<thead>
<tr>
<th>Grid Type</th>
<th>Turbulence Intensity (%)</th>
<th>Length Scale ((\Lambda_z/P))</th>
</tr>
</thead>
<tbody>
<tr>
<td>No Grid</td>
<td>2</td>
<td>0.02</td>
</tr>
<tr>
<td>Mesh Grid</td>
<td>12</td>
<td>0.26</td>
</tr>
<tr>
<td>Bar Grid</td>
<td>14</td>
<td>0.41</td>
</tr>
</tbody>
</table>
4 Blade Static Pressure Distribution

Figure 11 shows the local Mach number distributions on the blade surface for three exit Mach number conditions. The flow accelerates over most of the pressure side except for a short deceleration region just downstream of the stagnation point \((s/C = -0.25)\). The flow on the suction side continuously accelerates up to the geometric throat area \((s/C = 0.84)\). The exit Mach 0.62 and 0.80 cases decelerate immediately after the throat, whereas the exit Mach 1.09 case continues to accelerate and becomes supersonic. A trailing edge shock from the adjacent blade impinging on the suction surface \((s/C = 1.01)\) causes the sudden flow deceleration after the throat at exit Mach 1.09.

The periodicity of the flow at exit Mach 0.85 is provided in Fig. 12. The flow periodicity is shown by comparing the local Mach number distribution over the blade surfaces. The suction side of blades 3 and 4 shows the same local Mach number distributions over the majority of the suction surface. There is only a slight variation in the local Mach numbers near the geometric throat. The pressure side of blades 4 and 5 shows almost identical local Mach number distributions.

The distribution of the acceleration parameter on the blade surface for each exit Mach number is provided in Fig. 13. The acceleration parameter distribution shows the same trends as the local Mach number distribution. A positive acceleration parameter indicates that the flow is accelerating and a negative value indicates that the flow is decelerating. On the pressure side of the blade, the acceleration parameter is above the critical value of \(3 \times 10^{-6}\), which has been observed by Jones and Launder [19] and reported in the transition study by Mayle [20] to relaminarize the boundary layer.

5 Blade Heat Transfer Distribution

5.1 Test Conditions. Heat transfer measurements were performed at exit Mach numbers of 0.55, 0.78, and 1.03. The turbulence levels were varied between 2%, 12%, and 14% at each exit Mach number. The test matrix resulted heat transfer data for nine different flow conditions. For each test condition, heat transfer measurements were performed at least three times to establish repeatability. Table 3 provides the flow and heat transfer conditions for each test. The exit Reynolds number coupled with the exit Mach number is based on true chord and defined by

\[
Re_2 = \frac{p_2 \cdot U_2 \cdot C}{\mu_2}
\]  

(7)

where the average pressure and temperature data during the test run were used to calculate the exit Reynolds number.

Table 3 Test conditions for each case

<table>
<thead>
<tr>
<th>Case</th>
<th>Exit Ma</th>
<th>Tu (%)</th>
<th>Exit Re</th>
<th>(T_{in}) (°C)</th>
<th>(T_{out}) (°C)</th>
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<tr>
<td>1</td>
<td>0.57</td>
<td>2</td>
<td>640,000</td>
<td>106</td>
<td>25.0</td>
</tr>
<tr>
<td>2</td>
<td>0.55</td>
<td>12</td>
<td>610,000</td>
<td>99</td>
<td>24.5</td>
</tr>
<tr>
<td>3</td>
<td>0.55</td>
<td>14</td>
<td>600,000</td>
<td>102</td>
<td>26.5</td>
</tr>
<tr>
<td>4</td>
<td>0.76</td>
<td>2</td>
<td>810,000</td>
<td>103</td>
<td>25.7</td>
</tr>
<tr>
<td>5</td>
<td>0.78</td>
<td>12</td>
<td>790,000</td>
<td>108</td>
<td>26.8</td>
</tr>
<tr>
<td>6</td>
<td>0.78</td>
<td>14</td>
<td>800,000</td>
<td>107</td>
<td>27.4</td>
</tr>
<tr>
<td>7</td>
<td>1.03</td>
<td>2</td>
<td>1,090,000</td>
<td>100</td>
<td>27.0</td>
</tr>
<tr>
<td>8</td>
<td>1.03</td>
<td>12</td>
<td>1,060,000</td>
<td>103</td>
<td>25.9</td>
</tr>
<tr>
<td>9</td>
<td>1.01</td>
<td>14</td>
<td>1,070,000</td>
<td>95</td>
<td>26.5</td>
</tr>
</tbody>
</table>
5.2 Effect of Freestream Turbulence. This section discusses the effect that increasing the freestream turbulence has on the heat transfer distribution over the turbine blade surface. Plots of the heat transfer distribution over the blade surface in terms of Nusselt number and the Nusselt number augmentation relative to the low freestream turbulence cases are shown. The data are also compared with the flat plate correlations for laminar and turbulent boundary layers. Each of these plots is presented for each exit Mach number condition tested.

Figures 14–16 provide the heat transfer distributions for each freestream turbulence level at exit Mach 0.55, 0.78, and 1.03, respectively. The turbulence augments the heat transfer at the leading edge, on the pressure side, and the suction side of the blade for each exit Mach number condition. The general shape of the heat transfer distribution over the blade surface is the same for all cases. The highest heat transfer occurs at the leading edge and decreases as the flow proceeds down the suction and pressure sides. The heat transfer on the suction surface decreases until the boundary layer transitions, where a large increase in heat transfer occurs. The heat transfer on the pressure surface shows a decrease until \( s/C = -0.40 \) and then a slight increase afterwards.

On the suction side, the transition to a turbulent boundary layer occurs near the throat \( (s/C = 0.84) \) for each turbulence level at exit Mach numbers of 0.55 and 0.78. The boundary layer transition length over the blade surface is short for the exit Mach 0.55 and 0.78 cases because the flow is decelerating after the throat. The boundary layer transition for the 2% turbulence at exit Mach 0.55 occurs at \( s/C = 0.58 \). Since the flow is still accelerating at this point, the transition length increases. The effect of acceleration on the transition length was also noted in the study performed by Zhang and Han [21].

The start of transition for the 12% and 14% turbulence tests at exit Mach 1.03 appears to begin at \( s/C = 0.58 \). However, the increase in heat transfer is very gradual for both the 12% and 14% turbulence cases. A sharp rise in heat transfer occurs at \( s/C = 1.01 \), which corresponds to where the trailing edge shock from the adjacent blade is impinging on the suction surface. The slow transition could be attributed to the interaction of the shock. Once the boundary layer goes turbulent on the suction side, the effect of the freestream turbulence level on heat transfer diminishes.

On the pressure side, higher augmentation levels can be seen. The low freestream turbulence data show a local peak in the heat transfer data downstream of the leading edge on the pressure side at \( s/C = -0.26 \). Local peaks in heat transfer have been observed by Consigny and Richards [3], Arts et al. [4], and Giel et al. [5]. This location corresponds to the location of the flow deceleration and has been hypothesized by Arts et al. [4] and Giel et al. [5] to be caused by local flow separation. At the higher turbulence levels, the local peak appears to be diminished. Unfortunately, the gauge located at the peak was damaged after completing the heat transfer measurements at low freestream turbulence. However, the gauges surrounding the damaged gauge do not appear to indicate that a peak is present when the turbulence levels are increased.

The heat transfer data at the high turbulence levels were also normalized with the 2% turbulence data to show the augmentation levels due to turbulence. The augmentation plots for exit Mach 0.55, 0.78, and 1.03 are shown in Figs. 17–19. There is a peak in the augmentation plot on the pressure side where the flow decelerates at \( s/C = -0.26 \). Another peak occurs just after the
throat at $s/C=0.84$ since the transition occurs slightly earlier.

The augmentation levels for the 12% and 14% turbulence cases are almost exactly the same at exit Mach 0.55 and 0.78. At exit Mach 1.03, a difference in the augmentation levels for the 12% and 14% turbulence cases was observed. The 12% turbulence data show a higher augmentation levels on both the pressure and suction sides of the blade than the 14% turbulence data. Since the turbulence levels for each high freestream turbulence test are approximately the same, the length scales are the only difference between the tests. It appears that for the exit Mach 1.03 tests, the smaller length scale augments the heat transfer more than larger length scale.

The heat transfer data were also compared with the flat plate correlations for laminar and turbulent boundary layers. While the flat plate correlations do not take into account the effect of pressure gradient or curvature on heat transfer, the correlations do provide insight on how the boundary layer is behaving. The flat plate correlations reported by Incropera and DeWitt in terms of the local Nusselt number are

\[
\text{Laminar: } \frac{Nu}{Nu_0} = 0.332 \left( \frac{Re}{Re_0} \right)^{1/2} Pr^{1/3} \\
\text{Turbulent: } \frac{Nu}{Nu_0} = 0.0296 \left( \frac{Re}{Re_0} \right)^{4/5} Pr^{1/3}
\]

The local Reynolds number given by

\[
Re = \frac{\rho U L}{\mu}
\]

was used to find the Nusselt number distribution of the correlation. The average pressure and temperature data during the test run were used to in calculating the local Reynolds number. The local Nusselt number was converted into a heat transfer coefficient at each measurement location by using

\[
h = \frac{Nu}{Nu_0} \left( \frac{k}{\rho c_p} \right)
\]

and then the Stanton number is calculated by using Eq. (4).

Figures 20–22 show how the experimental data compare with the turbulent and laminar flat plate heat transfer correlations on the suction side of the blade at exit Mach 0.55, 0.78, and 1.03, respectively. Before transition occurs, the boundary layer lies between the laminar and turbulent boundary layer correlations for each case and follows the trend of the laminar correlation before transitioning. The difference between the laminar correlation and the low freestream turbulence data in the laminar region can be attributed to the effect of favorable pressure gradient. The augmentation due to turbulence on the suction side can also be seen since the high turbulence levels fall closer to the turbulent correlation than the 2% turbulence case. Good agreement between the turbulent flat plate correlation and the experimental data is shown for all of the data once the flow goes turbulent on the suction surface.

The comparison between the experimental data compared with the laminar and turbulent flat plate heat transfer correlations on the pressure side is provided in Figs. 23–25. The measured heat transfer lies near the turbulent correlation indicating that the pressure side boundary layer is fully turbulent. The one exception is the point that lies...
close to the laminar correlation for the 2% turbulence data at exit Mach 1.03. This point is just upstream of the deceleration region. The heat transfer being close to the laminar correlation suggests that the flow is laminar at this location.

The data start above the turbulent correlation and then decay below the turbulent correlation. The data having a steeper slope than the turbulent correlation might suggest that the boundary layer is attempting to relaminarize. From the acceleration parameter in Fig. 14, the acceleration parameter is above the critical value on the majority of the pressure side, which supports the possibility of relaminarization occurring.

5.3 Effect of Length Scale. As shown in Figs. 14–16, the heat transfer is augmented more at the leading edge with the 12% turbulence than the 14% turbulence. Since the turbulence levels are similar, the difference in heat transfer is caused by the size of the turbulence length scale. At similar turbulence levels, the heat transfer is augmented more at the leading edge if the size of the length scale is smaller. Van Fossen et al. [23] also observed that decreasing the length scale will increase the heat transfer augmentation at the leading edge. Both the turbulence intensity and the length scale size have an effect on the heat transfer in the stagnation region.

The effect of the length scale also appears at the exit Mach 1.03 conditions, as shown in Figs. 16 and 19. On both the pressure side and suction side of the blade, the 12% turbulence shows higher augmentation levels than the 14% turbulence.

5.4 Effect of Reynolds Number. The primary objective of this work was to examine the effect of freestream turbulence at on heat transfer at three exit Mach number conditions. Because of this objective, the exit Mach number was not decoupled from the exit Reynolds number. An increase in exit Mach number corresponds to an increase in the exit Reynolds number. To remain consistent with Secs. 5.1–5.3, the data will be compared by stating the Mach number. The actual exit Reynolds numbers measured for each test is provided in Table 3.

The Nusselt number distributions showing the effect of increasing the Reynolds number at turbulence levels of 2%, 12%, and 14% are shown in Figs. 26–28, respectively. As expected, there is an overall increase in heat transfer due to the increase in Reynolds number. For the 12% and 14% turbulence tests, an increase in the heat transfer with Reynolds number is not seen near the throat since the boundary layer transition is gradual at exit Mach 1.03.

At 2% turbulence, the amplitude of the local peak on pressure side caused by local separation increases as the Reynolds number increases. The local peak increasing in amplitude with Reynolds number was also observed by Consigny and Richards [3].

5.5 Leading Edge Correlation. The experimental data at the leading edge were compared with the stagnation region heat transfer correlation for elliptical leading edges developed by Van Fossen et al. [23]. The stagnation region heat transfer correlation takes into account the turbulence intensity, length scale, leading edge diameter, and Reynolds number and is provided by
The experimental leading edge heat transfer data was converted into a Frossling number given by

$$\frac{Fr_{Tu}}{Fr_{Lam}} = 0.00851 \sqrt{\frac{Tu}{Re_{d,in}}} \left( \frac{\Lambda}{d} \right)^{-0.574} + 1.0 \quad (12)$$

The plot of the correlation and the data are provided in Fig. 29. The experimental leading edge heat transfer data also show good agreement with the results reported by Giel et al. [5].

5.6 **TEXSTAN Comparison.** The experimental data was compared with surface heat transfer predictions made using academic version of TEXSTAN developed by Crawford [24]. To model the flow through the blade passages, the two-equation Lam–Bremhorst turbulence model was used with the Schmidt–Patankar transition model. A constant surface temperature boundary condition was applied to the blade and the constant turbulence kinetic energy was set throughout the computational domain by freezing the turbulence dissipation rate.

At low freestream turbulence, the TEXSTAN prediction matches the experimental data well, as shown in Fig. 30. The suction side heat transfer prediction matches levels and transition location of the data very well. On the pressure side, TEXSTAN slightly underpredicts the heat transfer and does not predict the local peak near the leading edge.

At higher turbulence levels, TEXSTAN overpredicted the heat transfer on both the pressure and suction sides, as shown in Fig. 31. The suction side prediction appears to be fully turbulent with the data only coming close to the prediction after the transition.
The pressure side is also well overpredicted. As shown in Fig. 24, the experimental data fall close to the turbulent flat plate correlation and the TEXSTAN prediction is well above the correlation. The leading edge heat transfer is also overpredicted. Similar trends were observed with the predictions at exit Mach 0.55 and 1.03.

6 Conclusions

Aerodynamic and heat transfer measurements were made on a turbine blade at flow conditions representative of engine operating conditions. High levels of freestream turbulence were generated by using passive turbulence grids that produced similar turbulence levels, but different length scales. Increasing the turbulence level was observed to augment the heat transfer over the blade surface.

The turbulence augmentations for the two high freestream turbulence levels with different length scales were almost identical except at the leading edge for the exit Mach 0.55 and 0.78 cases. For the exit Mach 1.03 cases, the turbulence was augmented more for the smaller turbulence length scale. Increasing the freestream turbulence was observed to not significantly influence the location of boundary layer transition.

As expected, increasing the exit Reynolds number was shown to increase the heat transfer levels and cause earlier boundary layer transition. The boundary layer transition was observed to be influenced primarily by the Reynolds number of the flow. An increase in the exit Reynolds (Mach) number caused the boundary layer to transition closer to the leading edge. The local peak on the pressure side increased with increasing Reynolds number for the baseline case.

Several comparisons were made between the data using the analytical flat plate correlations, leading edge correlations developed by Van Fossen et al. [23], and the TEXSTAN predictions. The heat transfer data showed good agreement with leading edge correlation of Van Fossen et al. [23]. The TEXSTAN prediction accurately modeled the heat transfer at low freestream turbulence levels, but was found to significantly overpredict the heat transfer levels at high turbulence levels.

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Nomenclature

\[ d = \text{leading edge diameter of blade} \]
\[ Fr = \text{Frossling number} \]
\[ h = \text{heat transfer coefficient} \]
\[ k_a = \text{thermal conductivity of air} \]
\[ k = \text{acceleration parameter} \]
\[ Ma = \text{Mach number} \]
\[ Nu = \text{Nusselt number} \]
\[ Nu_{12} = \text{Nusselt number at } Tu = 2\% \]
\[ P = \text{pitch of blade} \]
\[ Pr = \text{Prandtl number} \]
\[ q'' = \text{heat flux} \]
\[ r_e = \text{recovery factor} \]
\[ Re = \text{Reynolds number} \]
\[ s = \text{blade surface distance from stagnation point} \]
\[ St = \text{Stanton number} \]
\[ T = \text{temperature} \]
\[ Tu = \text{streamwise freestream turbulence intensity} \]
\[ U = \text{velocity} \]
\[ VR = \text{velocity ratio} \]
\[ x = \text{streamwise distance from turbulence grid} \]
\[ y = \text{pitchwise distance from leading edge of blade} \]

Greek

\[ \gamma = \text{specific heat ratio of air} \]
\[ \Lambda_e = \text{integral turbulence length scale} \]
\[ \rho = \text{density of air} \]
\[ \mu = \text{dynamic viscosity of air} \]

Subscripts

1 = inlet conditions
2 = exit conditions
\[ aw = \text{adiabatic wall} \]
\[ gauge = \text{thin film gauge (surface) measurement} \]
\[ iw = \text{initial wall conditions on blade} \]
\[ L = \text{local conditions} \]
\[ \text{Lam} = \text{laminar} \]
\[ o = \text{stagnation} \]
\[ Tu = \text{turbulent} \]

References

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