Numerical Investigation of Thermal Performance for Rotating High Aspect Ratio Serpentine Passages

DISSERTATION

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Abstract

A key performance metric of industrial gas turbines and aircraft engines is efficiency. To increase efficiency, the turbine inlet temperature is increased. As time has progressed, the continual increase in turbine inlet temperature has required the implementation of turbine blades with internal cooling passages supplied with compressor bleed air. The use of compressor bleed air results in an engine power and efficiency detriment, so optimizing the use of the cooling air is required. To do so without causing severe problems to the engine hardware requires knowledge of the thermal performance of the high-pressure turbine blade internal passages.

Current engine blade geometry utilizes serpentine passages with aspect ratios (AR), width to height, in excess of AR 1:6 [1]. The current published experimental work though, varies from AR 5:1 to AR 1:4, with only six measurement programs utilizing an aspect ratio of 1:4. Of those six, one was a three-pass serpentine, three were two-pass serpentines of which two included rotation. For AR 1:6, a single stationary data set exists.

For this research, a three-pass serpentine with a large width-to-height ratio, up to AR 1:6, was numerically studied using computational fluid dynamics and heat transfer (CFD/HT). The thermal performances of high aspect ratios, commonly found in the mid-circuits of current turbine blade cooling schemes, were studied.
A CFD modeling methodology was systematically developed that balanced accurately resolving the flow physics with minimizing the computational cost, which has not been previously documented within the open literature. A mesh with eight nodes along a turbulator edge for a spacing of 0.00625 inches, a first cell height of 3e-5 inches, a growth ratio of 1.25, and a maximum cell size of 0.025 inches was determined to be mesh independent. The Shear Stress Transport model with a Kato-Launder production limiter and curvature correction was selected since it provided accurate predictions with 4-6 times less computation requirement than more complex models.

The methodology was benchmarked against existing experimental data. OSU and HOST aspect ratio 1:1 stationary data were compared to CFD predictions and showed a 10-15% difference beyond experimental uncertainty, with two points within the first passage with a maximum of 37%. HOST aspect ratio 1:1 rotating data comparison to CFD predictions showed similar accuracy with a decrease in the two points within the first passage to a maximum difference beyond experimental uncertainty of 30%. It was concluded that the addition of rotation reduced the difference between the data and predictions due to presence of large scale secondary motion in the bulk flow. Therefore, the agreement of high aspect ratio rotating predictions would be improved over the high aspect ratio stationary predictions.

OSU AR 1:2 and OSU AR 1:6 stationary data was compared to prediction values. The prediction of AR 1:6 stationary or rotating has not been previously reported in the open literature. The AR 1:2 difference between data and prediction was similar to the accuracy of the 1:1 aspect ratio. The difference increased for the AR 1:6, specifically in the middle
of the first passage and at the first point downstream of the tip turn. The differences were accounted for by the inlet geometry, dependence on the measured driving temperature, and the turbulence within the bulk flow. The trends in the measurements for the different aspect ratios were captured by the predictions suggesting that one can use this prediction technique for preliminary design selection.

The high aspect ratio geometries, AR 1:2 and AR 1:6, were investigated with rotational numbers of 0.1, 0.2, and 0.3. The Nusselt number trends throughout the serpentine passages showed differences from the trends seen for the AR 1:1 rotating cases. The AR 1:2 and AR 1:6 had increasing Nusselt numbers on the trailing wall for the entire first passage, while the AR 1:1 case reaches a peak Nusselt number on the second panel and then decreases to a level value. The peak for the AR 1:1 was due to the turbulator vortex structure encompassing the full passage more rapidly and having the lowest mass flow resulting in an increased rate of heating the bulk flow.

The leading wall of the first passage had different trends depending on aspect ratio and rotation number. The AR 1:1 had consistent decreasing Nusselt numbers and a greater decline with increasing rotation number. This was due to the Coriolis forces moving the cooler bulk flow towards the trailing wall. The AR 1:2 had lower starting Nusselt number with a greater decline due to increasing rotation number, but by the third panel the Nusselt number values leveled out. The cause of this behavior was the Coriolis forces moving the separated turbulated vortex flow along the smooth wall and up to the leading wall,reactivating the flow. The flow near the leading wall was warm so the activation was only able to keep the value steady and prevent further decline. The AR 1:6 had the
lowest starting Nusselt numbers for increased rotation number as well, but the 0.1 rotation case had a monotonic decrease, the 0.2 rotation case had an initial decrease followed by an increase, and the 0.3 rotation case had a continual increase. The change in the trend was due to flow recirculation occurring along the leading wall for 0.2 and 0.3 rotation numbers. The point of change from decreasing to increasing Nusselt number corresponded to the start of the recirculation zone.

Conclusions regarding the influence of Reynolds number, rotation number, and aspect ratio are also described in detail. The Reynolds number relationship of Dittus-Boelter/McAdams was found to hold for stationary and rotating serpentine passages of low aspect ratio as well as high aspect ratio rotating passage for Reynolds numbers between 25,000 and 75,000. A comparison of rotation number and aspect ratio for enhancement factors showed that for all passages, turns, and walls, the aspect ratio had a greater effect on enhancement factor than rotation number except for the first passage leading and trailing walls. The leading wall had a greater influence due to rotation and the trailing wall experienced similar influence of rotation and aspect ratio.

This work has improved the state of the art by systematically determining a modeling methodology that accurately predicts flow physics with minimal computational cost for preliminary design purposes. The methodology was used to study the high aspect ratio stationary and rotating configurations through CFD, which had not been done previously. The influence of temperature rise due to pumping was documented and accounted for which also had not been done within the open literature, but is an important factor for engine conditions. The effects of Reynolds number, aspect ratio, and rotation number
were compared to determine influential effects of these parameters. The findings of this work are new to the open literature and are of great importance for industrial preliminary design.
Dedication

To Aaron … For everything you do, I can never thank you enough.
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I would like to thank my advisor Professor Michael Dunn for allowing me the opportunity to work with him and the OSU Gas Turbine Laboratory for this exciting research and the future planned work. I have learned a great deal from you and from working on this project. This experience will serve me well in the next phase of my career.

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Last, but certainly not least, I would also like to thank my husband, Aaron, who has been supportive throughout all of this. I could not have done it without his constant support, encouragement, guidance, and patience.
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<tr>
<td>AR</td>
<td>Aspect Ratio, Width-to-Height (W:H)</td>
<td>[-]</td>
</tr>
<tr>
<td>BSL</td>
<td>Baseline</td>
<td></td>
</tr>
<tr>
<td>BSL RSM</td>
<td>Baseline Reynolds Stress Model</td>
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<tr>
<td>CC</td>
<td>Curvature Correction</td>
<td></td>
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<tr>
<td>CFD</td>
<td>Computational Fluid Dynamics</td>
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<td>DES</td>
<td>Detached Eddy Simulations</td>
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<tr>
<td>EARSM</td>
<td>Explicit Algebraic Reynolds Stress Model</td>
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<td>Hot Section Technology</td>
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<td>HTC</td>
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<td>NASA</td>
<td>National Aeronautics and Space Administration</td>
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<td>O-RSM</td>
<td>Omega Reynolds Stress Model</td>
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<td>OSU</td>
<td>Ohio State University</td>
<td></td>
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<tr>
<td>PIV</td>
<td>Particle Image Velocimetry</td>
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<tr>
<td>RANS</td>
<td>Reynolds-Averaged Navier-Stokes</td>
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<tr>
<td>RPM</td>
<td>Revolutions per Minute</td>
<td>[rev/min]</td>
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<tr>
<td>RSM</td>
<td>Reynolds Stress Model</td>
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<tr>
<td>RTD</td>
<td>Resistance Temperature Detector</td>
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</table>
SST

Shear Stress Transport Model

TKE

Turbulent Kinetic Energy \([\text{in}^2/\text{sec}^2]\)

URANS

Unsteady Reynolds-Averaged Navier-Stokes

**Symbols**

\[
A \quad \text{Area} \quad [\text{in}^2]
\]

\[
BP = BT \times Ro
\quad \text{Buoyancy Parameter} \quad [-]
\]

\[
BT = DR \left(\frac{\Omega R}{V}\right)
\quad \text{Buoyancy Term} \quad [-]
\]

\[
C
\quad \text{Experimental Coefficient} \quad [-]
\]

\[
c_p
\quad \text{Specific Heat} \quad [\text{BTU/lbm-F}]
\]

\[
De = Re \sqrt{\frac{D_h}{r}}
\quad \text{Dean Number} \quad [-]
\]

\[
D_h = \frac{4A}{P_{er}}
\quad \text{Hydraulic Diameter} \quad [\text{in}]
\]

\[
DR = \frac{\rho_b - \rho_w}{\rho_b}
\quad \text{Density Ratio} \quad [-]
\]

\[
e
\quad \text{Turbulator Height} \quad [\text{in}]
\]

\[
e^+ = \frac{e}{D_h} Re \sqrt{\frac{f}{2}}
\quad \text{Roughness Reynolds Number} \quad [-]
\]

\[
f = \frac{\Delta P}{4 \left(\frac{L}{D_h}\right) \rho V^2}
\quad \text{Friction Factor} \quad [-]
\]

\[
F_C = -2m\Omega x \bar{u}
\quad \text{Coriolis Force} \quad [\text{lbf}]
\]

\[
f_{FD} = 0.046 Re^{-0.2}
\quad \text{Kármán-Nikuradse Equation, Fully Developed Turbulent Flow in a Circular Tube} \quad [-]
\]

\[
G = C(e^+)^n
\quad \text{Heat Transfer Roughness Function} \quad [-]
\]

\[
g_c
\quad \text{Gravitational Constant} \quad [\text{lbm-ft/lbf-sec}^2]
\]

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Grashof Number

Gr = BP * Re^2

Heat Transfer Coefficient

h

[BTU/hr-ft^2-F]

Height

H

[ft]

Conversion Factor BTU to ft-lbf

J

[ft-lbf/BTU]

Thermal Conductivity

k

[BTU/hr-ft-F]

Length

L

[in]

Mass

m

[lbm]

Mass Flow

\dot{m} = \rho AV

[lbm/sec]

Experimental Coefficient

n

[-]

Nusselt Number

\frac{hD_h}{k}

[-]

Dittus-Boelter/McAdams Equation, Fully Developed Flow in a Smooth Tube

Nu_{DB} = 0.023Re^{0.8}Pr^{0.4}

Pressure Drop

\Delta P

[lbf/in^2]

Turbulator Pitch

P

[in]

Prandtl Number

Pr = \frac{c_p \mu}{k}

[-]

Perimeter

Per

[in]

Radius of Rotation

R

[in]

Reynolds Number

Re = \frac{\dot{m}D_h}{A\mu}

[-]

Richardson Number

Ri = BP

[-]

Rotation Number

Ro = \frac{\Omega D_h}{V}

[-]

Rossby Number

Rossby = \frac{1}{Ro}

[-]

Stanton Number

St_r = \frac{Nu}{Re Pr}

[-]

Temperature

T

[\text{R}]

Scaled Temperature

T^* = \frac{(T - T_{inlet})}{(T_{max wall} - T_{inlet})}

[-]
\[
dT_{pump} = \frac{\Omega^2 (R_{max}^2 - R_{inlet}^2)}{2 g_c/c_p}
\]

Temperature Rise due to Pumping \([R]\)

\[\bar{u}\] Streamwise Velocity \([\text{in/sec}]\)

\[V\] Velocity \([\text{in/sec}]\)

\[w\] Turbulator Width \([\text{in}]\)

\[W\] Width \([\text{in}]\)

**Greek Letters**

\[\alpha\] Turbulator Angle \([\text{deg}]\)

\[\eta = \frac{Nu/Nu_{DB}}{f/f_{FD}}\] Overall Thermal Performance \([-\]

\[\rho\] Density \([\text{lbm/in}^3]\)

\[\Omega\] Rotation Rate or Angular Velocity \([\text{rad/sec}]\)

\[\mu\] Dynamic Viscosity \([\text{lbm/ft-hr}]\)

**Subscripts**

\[b\] Local Bulk Flow

floor Parameter of Floor (Space Between Turbulators)

\[\text{inlet}\] Value at Test Section Inlet

\[\text{max}\] Maximum Value of Test Section

\[\text{max wall}\] Maximum Wall Value

\[\text{overall}\] Parameter of the Overall Wall

\[\text{overall}\] (Turbulators and Space Between Turbulators)

\[\text{rib}\] Parameter of the Rib or Turbulator

\[w\] Local Wall Flow
Chapter 1

Introduction

1.1 Significance of the Problem

Efficiency, measured through specific fuel consumption, is one of the key performance metrics of industrial gas turbines and aircraft engines. Engine efficiency can be improved by increasing the turbine inlet temperature relative to the compressor inlet temperature based on the fundamental thermodynamics of the Brayton cycle. The continual search for improved efficiency increased the turbine inlet temperature beyond forged blade melting temperatures in the 1960s, necessitating improvements in manufacturing techniques and the genesis of cooling schemes that utilized compressor air to provide internal cooling of turbine blades [2].

Forged blades were replaced by cast blades and later single crystal cast blades [2]. The casting allowed for serpentine type passages to be integrated into the blades to channel cooling air through the blade, as shown in Figure 1. While this cooling air enables higher turbine inlet temperatures, removing compressor air for blade cooling decreases engine power and efficiency because the work the compressor performed on the extracted air is used primarily to drive coolant through the blade supply circuits and internal passages and the turbine can only extract a small amount of work from this portion of the
compressed air. Optimal thermal design must balance increasing turbine inlet temperature and using cooling air to maintain the blade temperatures within safe operating limits.

![Figure 1: Gas Turbine Blade Internal Cross Section View [3]](image)

The blade section of interest for this investigation is the mid-passage where turbulators, or ribs, are shown in Figure 1. The mid-passage is generally three or more straight sections connected by tip turns and root turns. The passages are defined by their aspect ratio (AR) or the width to height of the passage. The width dimension is parallel to the pressure or suction side of the blade. The turbulators are generally only on the internal walls of the pressure and suction side since those are the walls that are exposed to the gas path temperatures and therefore required the highest heat transfer coefficients.
The turbulators for this investigation have either rounded and square cross-sections and are on the internal pressure and suction side walls in a staggered configuration, 45° to the bulk flow direction. The turbulators have a pitch to turbulator height (P/e) spacing of 10 and a rib height to channel hydraulic diameter (e/Dh) of 0.5, 0.67, and 0.86 for aspect ratios of 1:1, 1:2, and 1:6, respectively. The staggered configuration, P/e, and e/Dh specifications are within the configurations determined to improve thermal performance over other geometric combinations.

Beyond optimization of the use of the cooling air with geometric features, understanding the heat transfer mechanisms within the serpentine blade passages can be used to reduce the thermal fatigue caused by strong temperature gradients in the blade material, which is one of the primary life limiting factors [2]. Predictions indicate a 50 - 60°F increase in metal temperature reduces creep life by half [4]. Increases in maximum heat transfer coefficients are not desirable if it also causes increases in the thermal gradients. Knowledge of thermal gradients is especially important for the continuing research into advanced blade materials such as high-temperature ceramics and ceramic-matrix composites where tolerance of thermal gradients is reduced from the current metal blades [5, 6].

1.2 Objectives

The first objective of this research is to determine an optimal modeling methodology that balances accurately resolving the flow physics with minimizing the computational cost for the design of multiple serpentine passages configurations commonly used for internal cooling of modern gas turbine blades. Different modeling fidelities are required
throughout the industrial design cycle; concept selection and preliminary designs are often based on preliminary estimates of the flow pattern and heat transfer characteristics, while very detailed quantification of local heat transfer rates are required for final design iterations and diagnosing field issues. New designs are commonly assessed based on relative differences from a baseline configuration and absolute values are not required until later in the design process. The present investigation is based on developing sufficiently accurate simulation methodologies to resolve the trends of aspect ratio and rotation number and to understand the mechanisms driving new concept performance for preliminary design requirements.

Various modeling methodologies will be compared to evaluate the predictive accuracy and elapsed time to complete the analysis for each approach. The assessment is performed by systematically comparing the simulation results to each other and then to various datasets for stationary and rotating serpentinaes over a range of aspect ratios. The accuracy and the computational cost of these various methodologies will be compared based on CPU hours for a consistent basis to aid in selecting the appropriate modeling approach for the preliminary design of these complex turbulated serpentinaes.

A second objective is to systematically verify the methodology with more than one dataset to improve the confidence in the methodology, which is generally not done in the open literature. The datasets chosen allow the independent assessment of the effects of aspect ratio and rotational speed on heat transfer rate. Stationary modeling of unity aspect ratio serpentinaes will be benchmarked to the NASA HQt Section Technology (HOST) [7] and the OSU [8] low aspect ratio datasets to baseline the low aspect ratio stationary
predictive capability. Rotational modeling predictions will be compared with the HOST rotating datasets and high aspect ratio stationary models will be validated against the OSU datasets to baseline the low aspect ratio rotating and high aspect ratio stationary predictive capability, respectively. Specific details of the geometry and flow conditions of the different databases can be seen in section 2.2.

Finally, the last objective is to utilize the selected modeling approach to obtain pre-test predictions for the upcoming high aspect ratio rotating measurements of the OSU rig. The rigorously determined modeling approach will provide insight on the physical phenomena occurring in an increased aspect ratio serpentine with varying rotation numbers beyond current experimental results and will provide guidance for this combined CFD and experimental effort with regard to boundary conditions that should be measured and other physical parameters of particular interest.
Chapter 2

Literature Review

The current research will focus on rotating, large aspect ratio (AR) serpentine cooling passages as shown in Figure 2. The passages are commonly in the mid-circuit of the blade and generally consist of three or more passages.

A large body of experimental work exists for two passage turbulated ducts where the coolant flows radially outward from blade root to tip, turns through the tip and travels radially inwards towards the root. However, current turbine airfoil cooling schemes implement multiple passes per circuit and serpentines consisting of three or more passes are the preferred data source because they provide root turn data. Many independent measurement programs have established turbulators angled 45° with respect to the bulk flow direction, staggered on opposing walls, having a pitch to turbulator height (P/e) spacing of 7-15, and having a rib height to channel hydraulic diameter (e/Dh) of 0.05-0.10 results in improved thermal performance over other geometric combinations [9].
2.1 Experimental Work

Table 1, color coded for decade, summarizes the published experimental work for 45° turbulated passages, including all aspect ratios, numbers of passages, turbulator cross-section geometries, turbulator placements, and flow conditions quantified by Reynolds numbers and rotation numbers. Aspect ratios varied from AR 5:1 to AR 1:6, with only six measurement programs investigating an aspect ratio of 1:4. Of those programs, one was a three-pass serpentine and three were two-pass serpentines of which two included rotation with rotation numbers of 0.118 and 0.31. A single stationary data set exists for an aspect ratio 1:6.
<table>
<thead>
<tr>
<th>Citation</th>
<th>Authors</th>
<th>Year</th>
<th>Alt.</th>
<th>No. Pass.</th>
<th>Turbulator Angle</th>
<th>Edge Shape</th>
<th>eROI</th>
<th>Prs.</th>
<th>Turbulator Placement</th>
<th>File</th>
<th>ROI Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>8</td>
<td>J.C. Kim, S.A. Park, and G.C. Lee</td>
<td>1993</td>
<td>11</td>
<td>2</td>
<td>45, 60, 66, 72° S</td>
<td>Squares</td>
<td>0.060</td>
<td>16</td>
<td>48/80</td>
<td>Phenomenon</td>
<td>3</td>
</tr>
<tr>
<td>9</td>
<td>J.C. Kim and S.A. Park</td>
<td>1993</td>
<td>11</td>
<td>2</td>
<td>45, 60, 66, 72° S</td>
<td>Squares</td>
<td>0.060</td>
<td>16</td>
<td>48/80</td>
<td>Phenomenon</td>
<td>3</td>
</tr>
<tr>
<td>10</td>
<td>Y. Zhou and J.C. Kim</td>
<td>1993</td>
<td>11</td>
<td>2</td>
<td>45, 60, 66, 72° S</td>
<td>Squares</td>
<td>0.060</td>
<td>16</td>
<td>48/80</td>
<td>Phenomenon</td>
<td>3</td>
</tr>
<tr>
<td>11</td>
<td>J.C. Kim and W. Cho</td>
<td>1993</td>
<td>11</td>
<td>2</td>
<td>45, 60, 66, 72° S</td>
<td>Squares</td>
<td>0.060</td>
<td>16</td>
<td>48/80</td>
<td>Phenomenon</td>
<td>3</td>
</tr>
<tr>
<td>12</td>
<td>M.E. Teufel, L.A. Beach, and J.M. Derber</td>
<td>1931</td>
<td>11</td>
<td>2</td>
<td>45° S</td>
<td>Squares</td>
<td>0.083</td>
<td>16</td>
<td>48/80</td>
<td>Phenomenon</td>
<td>3</td>
</tr>
<tr>
<td>13</td>
<td>B.V. Johnson, J.R. Vogler, and G.O. Schaller</td>
<td>1931</td>
<td>11</td>
<td>2</td>
<td>45° S</td>
<td>Squares</td>
<td>0.083</td>
<td>16</td>
<td>48/80</td>
<td>Phenomenon</td>
<td>3</td>
</tr>
<tr>
<td>14</td>
<td>M.E. Teufel, L. Neil, and J.M. Derber</td>
<td>1931</td>
<td>11</td>
<td>2</td>
<td>45° S</td>
<td>Squares</td>
<td>0.083</td>
<td>16</td>
<td>48/80</td>
<td>Phenomenon</td>
<td>3</td>
</tr>
<tr>
<td>15</td>
<td>J.R. Flaherty</td>
<td>1931</td>
<td>11</td>
<td>2</td>
<td>45° S</td>
<td>Squares</td>
<td>0.083</td>
<td>16</td>
<td>48/80</td>
<td>Phenomenon</td>
<td>3</td>
</tr>
</tbody>
</table>

**Table 1: Experimental Data Sets for 45° Turbulator Passages**
Heat transfer rates are generally reported in terms of Nusselt number or heat transfer enhancement; defined as the local Nusselt number of the test section scaled by the Nusselt number of fully developed flow in a smooth tube, Equation 1. The reference Nusselt number is commonly calculated from the Dittus-Boelter/McAdams equation, Equation 2, using the serpentine inlet conditions to remove the Reynolds number variation as the hydraulic area contracts and expands around turbulators and to demonstrate the improvement over a baseline case of a fully-developed flow in a smooth tube.

\[
\frac{Nu}{Nu_{DB}} = \frac{hD_h/k}{Nu_{DB}}
\]

\[
Nu_{DB} = 0.023Re^{0.8}Pr^{-0.4}
\]

Heat transfer augmentations of 2.2 to 3.0 were reported for stationary square, i.e. AR 1:1 Ro = 0, passages with 45° turbulators on the two opposing walls [11, 43]. The average heat transfer enhancement for the same turbulator geometry with aspect ratio 1:2 and 1:4 passages are comparable to the square passage with enhancements of 2.5 to 3.0 over a smooth circular pipe [37]. For the 1:6 aspect ratio channels, the turbulator secondary flow strength increased with increasing Reynolds number over 50,000, while below 50,000, the turbulated surface was comparable to smooth wall conduit of the same aspect ratio [8].
While the overall average enhancement factors for stationary AR 1:1, 1:2, and 1:4 were comparable, the pressure losses incurred within the channels were not. The 1:4 aspect ratio showed the lowest pressure penalty of the aspect ratios mentioned, when all other geometric factors were held constant [39]. As a result, for a given pressure drop value, the AR 1:4 passage provides the highest overall thermal performance, Equation 3. The pressure drop is expressed as a friction factor in Equation 4, and normalized by the Kármán-Nikuradse equation for fully developed turbulent flow in a circular tube, in Equation 5 [14].

\[
\eta = \frac{Nu/Nu_{DB}}{f/f_{FD}}
\]

**Equation 3: Overall Thermal Performance [14]**

\[
f = \frac{\Delta P}{4\left(\frac{L}{D_R}\right)\rho V^2}
\]

**Equation 4: Friction Factor [14]**

\[
f_{FD} = 0.046Re^{-0.2}
\]

**Equation 5: Kármán-Nikuradse Equation [14]**

A relationship for the relative heat transfer contribution from the floor between turbulators compared to the turbulator surface was proposed by Taslim and Wadsworth, Equation 6 through Equation 8 [59]. The rib averaged heat transfer coefficients were seen to be much higher than those for the area between the ribs. As a result, the area weighted average rib heat transfer coefficients, \(h_{rib}A_{rib}\), would be 33 to 53 percent of \(h_{overall}A_{overall}\) [59]. While the proposed equation provides information regarding the heat transfer coefficient distribution, it does not allow for use of the correlation without experimental data.
Equation 6: Area-Weighted Average of Heat Transfer Coefficient of Ribs to Overall Heat Transfer Coefficient [59]

\[ \frac{h_{rib}A_{rib}}{h_{overall}A_{overall}} = \left( 1 + \frac{h_{floor}}{h_{rib}} \left[ \frac{1}{3} \left( \frac{P}{e} - 1 \right) \right] \right)^{-1} \]

where

Equation 7: Overall Heat Transfer Coefficient [59]

\[ h_{overall} = \frac{(h_{rib}A_{rib} + h_{floor}A_{floor})}{A_{overall}} \]

Equation 8: Overall Area [59]

\[ A_{overall} = A_{rib} + A_{floor} \]

Figure 3 illustrates the development of secondary vortices in both radially outward and radially inward passages due to the domain rotation with the rotation direction change through the turns because the Coriolis force, Equation 9, changes sign as the radial velocity component changes sign from outward to inward [45]. Coriolis effects increase the heat transfer up to five times the level seen in a stationary, fully-developed smooth tube on first passage (radially outward) trailing surfaces, Figure 3, but decrease the heat transfer value to only 65% of the stationary, fully-developed flow on the leading surfaces [7]. The leading surfaces also experience an initial reduction in heat transfer with increasing rotation number, but recover and increase with rotation numbers greater than 0.3 [41]. This effect is due to the stabilization of the near-wall flow and secondary vortices, which causes the heated near-wall fluid from the trailing and side-wall surfaces to accumulate near the leading side of the passage [7]. The Coriolis forces are quantified by the non-dimensional parameter that compares the rotational to inertial forces, commonly referred to as the rotation number, Equation 10.
Equation 9: Coriolis Force
\[ F_C = -2m\Omega \times \vec{u} \]

Equation 10: Rotation Number [7]
\[ Ro = \frac{\Omega D_h}{V} \]

Figure 3: Conceptual View of the Velocity Distribution in a Rotating, Two-Pass Channel [60]

Buoyancy can also affect the heat transfer in a rotating square passage; especially for the trailing edge surfaces with radially outward flow [7], but trailing edge surfaces are relatively unaffected for radially inward flow, Figure 4. For high rotation numbers (Ro = 12...
0.36), and high buoyancy, the heat transfer ratios on the trailing surface for outward flow were only 20-25% larger than the heat transfer ratios for the same conditions and location with smooth walls and 10-15% of the increase can be attributed to increased surface area due to the turbulators [7]. Equation 11 defines the buoyancy parameter as a rescaling of rotation number by a densimetric weighting of the near wall and bulk fluid densities. The buoyancy parameter increases as the bulk-to-wall temperature differences, mass flow through the serpentine, and rotational speeds increase.

**Equation 11: Buoyancy Parameter [7]**

\[
BP = \left(\frac{\rho_b - \rho_w}{\rho_b}\right) \left(\frac{\Omega R}{V}\right) \left(\frac{\Omega D_h}{V}\right)
\]

For larger aspect ratios, AR 1:2 through 1:4, the rotation effect increases the heat transfer on the trailing wall and decreases the heat transfer on the leading surface in the first pass similar to the effect seen in the square rotating turbulated passages, with the first pass leading surfaces showing a greater sensitivity to aspect ratio than all other surfaces [3]. The heat transfer enhancements on the inner and outer side walls also increase with rotation number [39].

Differences between the leading and trailing surfaces are pronounced in the second pass [37]. The cause for this effect was postulated to be Dean vortices dominating the Coriolis-induced vortices. References [37, 61, and 62] suggested Dean vortices formed by the 180° bend, dominate the rotational-induced vortices because the large distance between the leading and trailing walls results in a longer distance before the Dean vortices diminish.
For larger aspect ratios, the leading wall has the strongest dependence on the buoyancy parameter in the first passage for AR greater than 1:1 with the larger AR showing larger heat transfer coefficient differences between the leading and the trailing walls [37]. An increase in the buoyancy parameter also results in an increase in the heat transfer coefficients on the inner and outer side walls [39].

2.2 Benchmark Data

The modeling methodology will be benchmarked to two experimental datasets listed in Table 1, the NASA HOST data [7] and the OSU stationary data [8]. Schematics of the two test sections are show in Figure 5 and Figure 6, respectively. The test section geometries are purposefully similar because one of the primary objectives of the OSU program was to demonstrate a new experimental approach, benchmarking to the HOST industry standard, and then expanding upon it with increased aspect ratios [8]. The following differences exist between the test sections:

- HOST had rounded turns and OSU had squared off turns
- HOST did not have turbulatons on the first copper segment in the first passage
- OSU lengthened the entrance length to approximately six inches where HOST had approximately two inches with a screen at the inlet
- OSU had four turbulated segments, 2 inches long, per passage while HOST had three, 1.94 inches long, per passage [63]
- OSU had variable aspect ratio geometry, allowing AR 1:1, AR 1:2, AR 1:4, and AR 1:6, while HOST only had AR 1:1
• HOST maximum Reynolds number was 75,000 [7] while OSU maximum Reynolds number was 130,000 [8]

• HOST minimum Reynolds number was 12,500 [7] while OSU minimum Reynolds number was 4,000 [8]

• HOST had a heated second tip turn while OSU did not

• HOST had round turbulators while OSU turbulators were square

• HOST had turbulators machined into the copper while OSU turbulators were applied with Loctite 426 adhesive [8]

• HOST had a thermocouple probe in the center of the flow path before the first heated smooth wall at the inlet [7], Figure 5, while OSU had a thermocouple probe from wall D to the center of the flow path 1 inch upstream of the first turbulated panel [8], Figure 6
2.3 Numerical Work

The numerical work performed for turbulated passages can be divided into steady Reynolds-Averaged Navier-Stokes (RANS) analysis and time-accurate analysis; including Unsteady RANS (URANS), Detached Eddy Simulation (DES), and Large Eddy Simulation (LES). There are two key contributors to the steady RANS analysis, the collaboration at University of Manchester Institute of Science and Technology and the collaboration at Texas A&M. For time accurate analysis, primary contributors are at Louisiana State University and at Virginia Tech.

The team led by Prof. Hector Iacovides completed four investigations utilizing Reynolds Stress Models (RSM) [64-67]. Two investigations were of a rotating turbulent passage.
section with a maximum number of two ribs [64, 65]. Two additional investigations were completed on stationary passages, one with six ribs and another with six and a half ribs into a turn and one and a half ribs out of the turn [66, 67]. No qualitative comparison between the predicted models and the experimental data was given. Quantitatively, it was stated that the results showed the RSM models were more successful at predicting the flow development in ribbed ducts than they were at matching wall heat flux [65]. The comparison to data for smooth walls (S) and ribbed walls (R) at the centerline of the walls is shown in Figure 7.

![Figure 7: Smooth (S) and Ribbed (R) Wall Comparison Between Test Data and CFD Predictions Using Reynolds Stress Models [66]](image)

Prof. Je-Chin Han led four investigations of rotating ribbed channels spanning aspect ratios from 4:1 to 1:4 utilizing RSM [68-71]. For the 1:1 aspect ratio channel, a single passage was modeled with 45° turbulators on the leading and trailing surfaces [69]. The
analysis stated the overall predicted Nusselt numbers were relatively close to the experimental data except for the leading surface of the rotating channel, although no quantitative comparison was provided. The research concluded that even in the stationary case, the two asymmetric counter-rotating secondary flows induced by the ribs caused anisotropic turbulence, which influenced the development of momentum and thermal boundary layers along the ribbed duct. With the addition of rotation and centrifugal buoyancy forces, it was concluded that the RSM was required to account for the anisotropy.

Su et al. presented a comparison among AR 1:1, 1:2, and 1:4 using the same methodology as Y.-J. Jang et al. [69], so the results were not compared to experimental data [70]. The AR 1:1 geometry had the highest heat transfer enhancement in the passages for both the stationary and rotating conditions due to the larger rib blockage, higher inlet velocity for the same Reynolds number, and strong rib-induced secondary flows in the core stream. The maximum heat transfer rates were found in the AR 1:2 configuration for both stationary and rotating conditions caused by the secondary flow through the bends reaching maximum strength by occupying the entire channel cross sectional area, as shown in Figure 8 and Figure 9.
Chen et al. numerically investigated the impact of tip turn geometries for an AR 1:1 two-passage serpentine configuration with a rounded turn, Figure 10, and a square turn Figure 11 [72, 73]. The Omega Reynolds Stress Model (O-RSM) provided the best data match, with a 15-30% over-prediction for the rounded turn geometry [72] and 20-30% over-prediction for the square geometry [73]. The Shear Stress Transport (SST) model, however, was selected for the numerical study since it converged with 20-30% of the CPU time required for the O-RSM model [73].
The first two passages of the HOST geometry, AR 1:1, were modeled by Sleiti and Kapat using enhanced wall treatment RSM with tuned model coefficients for a rotation number of 0.238 and a density ratio of 0.13 [61]. The results of the calculation matched experimental data within the experimental uncertainty except for the location at the channel inlet, where the error was attributed to a difference between experimental and assumed boundary conditions [61]. The comparison is shown in Figure 12.
Time-accurate Unsteady Reynolds-Averaged Navier-Stokes (URANS) simulations were reported by Saha and Acharya on unit cell geometries with either two or four turbulators [74-76]. A unit cell geometry is a repeating geometric segment of the turbulator passage. A comparison to test data for both URANS and Large Eddy Simulation (LES) was performed on the two-turbulator unit cell [75]. The Nusselt number ratios for both URANS and LES for stationary and rotating were within 20-25% of the experimental data on the leading and trailing walls and within 15% on the side walls [75], see Table 2. It was concluded that the dissipative nature of the URANS model damped the flow structure resulting in lower heat transfer rates than observed in the LES model; however, both URANS and LES gave reasonable agreement and due to computation expense of LES, URANS was used for the rest of the simulations [75].
Table 2: Normalized Nusselt Number Comparison of URANS and LES to Test Data [75]

<table>
<thead>
<tr>
<th></th>
<th>Sidewall</th>
<th>Leading Wall</th>
<th>Trailing Wall</th>
</tr>
</thead>
<tbody>
<tr>
<td>Re=12,500, (Δρ/ρ)=0.13</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>URANS (no rotation), square ribs (Std κ-ε)</td>
<td>1.51</td>
<td>2.82</td>
<td>2.82</td>
</tr>
<tr>
<td>LES (no rotation), square ribs (dynamic model)</td>
<td>1.65</td>
<td>3.10</td>
<td>3.14</td>
</tr>
<tr>
<td>Johnson et al., rounded ribs (no rotation)</td>
<td></td>
<td>2.47</td>
<td>2.50</td>
</tr>
<tr>
<td>URANS, square ribs (Std κ-ε) (Ro=0.12)</td>
<td>2.11</td>
<td>1.83</td>
<td>4.20</td>
</tr>
<tr>
<td>LES, square ribs (dynamic model) (Ro=0.12)</td>
<td>1.99</td>
<td>2.08</td>
<td>4.45</td>
</tr>
<tr>
<td>Johnson et al., rounded ribs (Ro=0.12)</td>
<td>1.36</td>
<td>1.75</td>
<td>3.72</td>
</tr>
</tbody>
</table>

The two additional URANS simulation were performed by Saha and Acharya for aspect ratios of 1:1, 1:4, and 4:1 for 90° turbulators [74, 76] implementing the aforementioned validated methodology [75]. The AR 1:4 case showed the highest relative difference between the trailing wall and the leading wall heat transfer rates [76]. The leading wall showed low heat transfer enhancement for rotation numbers less than 0.25 caused by the flow stagnating at the wall due to the centrifugal buoyancy and resulting in a conduction limited heat transfer condition [76]. For rotation numbers greater than 0.25, reversed flow occurred at the leading wall resulting in an increase in heat transfer enhancement [74].

Detached Eddy Simulations (DES) and LES have been completed by the group at Virginia Tech working under Prof. Danesh K. Tafti. Two papers have been published on DES for a two-turbulator unit cell and a three-turbulator unit cell computational volume.
A comparison among data by Chanteloup [77], URANS, DES, and LES is shown in Table 3. The DES model was noted as being five times more economical than the Dynamic Smagorinsky model LES case and was able to predict the velocity components on top of the ribs accurately while URANS only predicted the velocity normal to the wall [77]. The final conclusion drawn was that the DES predictions were more consistent with experimental data and LES predictions. While the overall predictions by URANS were within acceptable limits, the localized predictions were not as accurate.

Table 3: Overall Friction Factor and Heat Transfer Augmentation Ratios for DES, URANS, LES, and Test Data [77]

<table>
<thead>
<tr>
<th></th>
<th>DES 128x80x80</th>
<th>URANS 128x80x80</th>
<th>LES 160x128x128</th>
<th>Chanteloup</th>
</tr>
</thead>
<tbody>
<tr>
<td>$U_{av}$ Nu/Nu0</td>
<td>2.5</td>
<td>2.74</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rib Wall % Error</td>
<td>2.92</td>
<td>2.60</td>
<td>2.37</td>
<td>2.78</td>
</tr>
<tr>
<td>Outer Wall % Error</td>
<td>2.37</td>
<td>2.41</td>
<td>2.55</td>
<td>2.23</td>
</tr>
<tr>
<td>Inner Wall % Error</td>
<td>1.75</td>
<td>1.60</td>
<td>1.88</td>
<td>1.65</td>
</tr>
<tr>
<td>$f/f_0$ % Error</td>
<td>14.7</td>
<td>12.5</td>
<td>13.8</td>
<td>12.49</td>
</tr>
</tbody>
</table>

The published works of Tafti’s group using LES are numerous and provide accurate and detailed results but are focused primarily on two to three turbulators within a computational volume, which is limited geometric detail compared to industrial flows [79-87]. Three studies were based on a square channel with six to nine ribs per turbulated surface, Re = 20,000 and a rotation number of Ro = 0.3 [88-90]. The heat transfer
enhancement values were compared to experimental values of fully developed flow and were found to be within 10% on the turbulated wall by the second rib [88]. The smooth walls were within 10% by the fifth rib. It was noted that LES was able to accurately predict the heat transfer characteristics for the developing region of a turbulated duct, but the computational expense of fully-resolved LES currently prevents it from being an industry design tool beyond unit cell investigations.

A stationary simulation of an AR 1:1 two passage, square turn, geometry was completed for Re = 20,000 by Sewall and Tafti [91]. The passages consisted of three ribs on each of the leading and trailing walls of the upward passage and three ribs on each of the leading and trailing walls of the downward passage. The comparison to data was done at the 25% (inner line), 50% (center line), and 75% (outer line) positions of the duct, Figure 13. The comparison showed good agreement between the data and the LES predictions, Figure 14.

Figure 13: Graphical Representation of Inner Line, Center Line, and Outer Line for Figure 14
Numerical approaches range from 2-equation models to LES but lack coherence in the modeling investigations. Steady simulations, 2-equation and RSM, include geometry with up to two passages but are often not validated to experimental data or are vague on the quantification of the difference. As a result, it is difficult to draw conclusions on the accuracy and the predictive capability of the trends. Without predictive capabilities, the trends would not be useful in a design application where they would need to be applied to different geometries. Time-accurate simulations, including URANS, DES, and LES, have shown promising comparisons to data, but have generally been applied to unit cell geometries. Areas of interest for a design environment would include turns, longer passages, and more complex geometry.
2.4 Correlations for Large Aspect Ratio Geometries

For aspect ratios other than 1:1, four proposed correlations are available in the literature, three for stationary data and one for rotational data. Han et al. [14], correlated data for AR 1:1 to 1:4, as detailed in Equation 12. The ribbed side wall Stanton number (St) was correlated to a heat transfer roughness function (G), where the coefficients of G were determined experimentally [14]. Figure 15 shows that the correlation represents the data to within ±8% for the 21 data points used to develop the correlation.

**Equation 12: Correlation for Aspect Ratios 1:4 to 1:2 [14]**

\[
C = 2.24 \left( \frac{W}{H} \right)^{-0.76}, \text{ if } \alpha = 90^\circ \\
C = 1.80 \left( \frac{W}{H} \right)^{-0.76}, \text{ if } 30^\circ \leq \alpha < 90^\circ \\
\]

\[
G = C(e^+)^n \\
1 - \frac{W}{H} \leq \frac{2}{4} \\
\]

\[
e^+ = \left( \frac{e}{D_h} \right) Re \left( \frac{f}{2} \right) \\
n = 0.35 \left( \frac{W}{H} \right)^{0.44} \\
\]
The second skewed turbulator passage correlation available was proposed by Smith and follows the power law form of the Dittus-Boelter/McAdams correlation, Equation 2 [8]. The correlation, shown in Equation 13, is compared with additional published data and the resulting correlation match is shown in Figure 16.

**Equation 13: Skewed Turbulator Passage Proposed Correlation by Smith [8]**

\[ Nu = 1.15(0.11 \, Re^{0.68}) \]
Both correlations are for stationary data only and do not include rotational effects. Rotation introduces complex interactions and Johnson et al. [7] noted that for rotating serpentine passages, rotation number, buoyancy parameter or buoyancy terms were needed for correlating test data [7].

Liou et al. proposed local Nusselt number correlations for stationary and rotational conditions of a single AR 2:1 passage. For the stationary correlations, the form shown in Equation 14 was used with the coefficients shown in Table 4. The correlation values were compared to data from Liou et al. and the first passage of J.C. Han et al., and the result was described as satisfactory [43].
Equation 14: Nu₀ Correlation for Single AR 2:1 Stationary Passage [43]

\[ Nu₀ = A \left( \frac{x}{d} \right) Re^{n(x/d)} \]

Table 4: Correlation Coefficients for A and n [43]

<table>
<thead>
<tr>
<th>Axial location</th>
<th>x/d</th>
<th>Leading-edge A(x/d)</th>
<th>n(x/d)</th>
<th>x/d</th>
<th>Trailing-edge A(x/d)</th>
<th>n(x/d)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rib 1</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.5</td>
<td>2.21</td>
<td>0.43</td>
</tr>
<tr>
<td>Rib 2</td>
<td>1</td>
<td>1.82</td>
<td>0.43</td>
<td>1.5</td>
<td>1.21</td>
<td>0.47</td>
</tr>
<tr>
<td>Rib 3</td>
<td>2</td>
<td>0.82</td>
<td>0.5</td>
<td>2.5</td>
<td>1.12</td>
<td>0.48</td>
</tr>
<tr>
<td>Rib 4</td>
<td>3</td>
<td>0.81</td>
<td>0.51</td>
<td>3.5</td>
<td>0.86</td>
<td>0.50</td>
</tr>
<tr>
<td>Rib 5</td>
<td>4</td>
<td>0.76</td>
<td>0.52</td>
<td>4.5</td>
<td>0.74</td>
<td>0.52</td>
</tr>
<tr>
<td>Mid-rib 1-2</td>
<td>0.5</td>
<td>1.99</td>
<td>0.4</td>
<td>1.0</td>
<td>0.95</td>
<td>0.48</td>
</tr>
<tr>
<td>Mid-rib 2-3</td>
<td>1.5</td>
<td>1.02</td>
<td>0.47</td>
<td>2.0</td>
<td>0.94</td>
<td>0.48</td>
</tr>
<tr>
<td>Mid-rib 3-4</td>
<td>2.5</td>
<td>0.65</td>
<td>0.51</td>
<td>3.0</td>
<td>0.92</td>
<td>0.48</td>
</tr>
<tr>
<td>Mid-rib 4-5</td>
<td>3.5</td>
<td>0.6</td>
<td>0.53</td>
<td>4.0</td>
<td>0.89</td>
<td>0.49</td>
</tr>
<tr>
<td>Mid-rib 5-6</td>
<td>4.5</td>
<td>0.55</td>
<td>0.54</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

The rotational correlations were developed through the combination of regression curves from the experimental data [43]. The resulting correlations are shown in Equation 15. The data from Liou et al. were compared to the correlation and it was determined that 90% of the data is within +/- 25% of the correlation for the entire range of test conditions, Figure 17, [43]. To verify the predictive capability of the correlations, additional test data with different experimental configurations would need to be compared.
Equation 15: Nusselt Number Correlations for Single AR 2:1 Rotating Passage [43]

\[
N_u_L = Re^{0.513} \times [1.19 - 0.543 \times e^{-0.411 \times Re} + \{0.12 - 16 \times e^{-12.7 \times Re}\} \times Bu]
\]

at rib location on leading wall

\[
N_u_L = Re^{0.536} \times [0.697 - 0.363 \times e^{-1.58 \times Re} + \{0.003 - 8.36 \times e^{-11.0 \times Re}\} \times Bu]
\]

at mid-rib location on leading wall

\[
N_u_L = Re^{0.51} \times [1.75 - 0.975 \times e^{-6.25 \times Re} + \{-0.037 - 14.8 \times e^{-5.13 \times Re}\} \times Bu]
\]

at rib location on trailing-wall

\[
N_u_L = Re^{0.45} \times [1.78 - 1.03 \times e^{-9.10 \times Re} + \{-0.108 - 37.3 \times e^{-8.32 \times Re}\} \times Bu]
\]

at mid-rib location on trailing-wall

Figure 17: Comparison of Experimental Measurements With Rotational Correlation [43]

The correlations presented are the only ones within the open literature that incorporate the effects of aspect ratio by expanding beyond AR 1:1. The correlation presented in Equation 12 relates to mass flux experiments and therefore relates to a heat transfer roughness function, not a heat transfer coefficient. The stationary and rotation correlations proposed by Liou et al. are for aspect ratios large in width, which results in different flow physics than what is seen in aspect ratios large in height. The correlation
proposed by Smith addresses the aspect ratios large in height but is only for stationary conditions and for an overall test section Nusselt number.

There is currently no correlation in the open literature that addresses changes in large aspect ratio (in height), passages that include rotation, accounts for the differences between the different walls within different passages, and utilizes multiple data sets for comparison. Those four items are required for a correlation to be of use for industrial preliminary design. Until a correlation is created and validated, an accurate and minimally computationally expensive CFD analysis for individual designs for preliminary analysis is required.
Chapter 3

Modeling Methodology

The modeling methodology investigated in this dissertation will systematically investigate mesh sensitivity, data report procedures, geometric representations, turbulence models, and boundary conditions to validate a CFD/HT approach that is acceptable for the purposes of preliminary design. Table 15 in Appendix A summarizes the parameters of all the runs performed for this purpose.

3.1 Geometry

The HOST experiment, shown in Figure 5, provided data for all smooth walls [92-95], two turbulated walls both skewed and normal to the flow [7, 96, 97], and changes in the model orientation to the axis of rotation [96, 98]. General project information is included in references [99-102]. The only experimental configuration of particular interest to this investigation was the zero axis angle, stationary and rotating, skewed turbulator configuration. It was studied for Reynolds numbers of 25,000 and 50,000 and rotation numbers of 0, 0.12, and 0.25. No detailed drawings of the geometry used for the experiments were reported in open literature, so the CFD/HT model geometry was recreated based on the specifications given in the above references.
The second experimental configuration used in this investigation was the OSU serpentine rig. The OSU test section was patterned off the HOST test section with the differences listed in section 2.2. To date, the OSU serpentine was run stationary with all smooth walls for aspect ratios of 1:1 and 1:6 and two skewed turbulated walls for aspect ratios of 1:1, 1:2, and 1:6 [8]. All of the relevant information for the geometry and experimental conditions of this data set was available.

3.1.1 Nomenclature

A common nomenclature for the HOST and OSU test sections was created to facilitate data processing and comparisons. The convention that goes with Figure 18 is the following: passage 1 (1st), first tip turn (TT1), passage 2 (2nd), root turn (RT1), passage 3 (3rd), second tip turn (TT2), and outlet (Exit). Recall from section 2.2, the second tip turn of the HOST test section was heated while the OSU was not.

![Figure 18: Nomenclature for Passages and Turns for OSU and HOST Geometries](image)

The labeling of the walls was also consistent between the two test sections investigated. The wall to the left of the flow direction is wall A, the wall to the right of the flow
direction is wall B, the wall to the top of the flow direction is wall C, and the wall to the bottom of the flow direction is wall D. Walls C and D are turbulated and when rotating, wall C is the leading wall and wall D is the trailing wall, Figure 19.

Figure 19: Nomenclature for Walls for OSU and HOST Geometries

For consistency, the graphs for HOST and OSU will be done with the convention shown in Figure 20 and Figure 21, respectively. The individual passages and turns will not be labeled on each graph.

Figure 20: Graph Nomenclature for HOST Geometry
Figure 21: Graph Nomenclature for OSU Geometry
3.2 Mesh Sensitivity Study AR 1:1

Five meshes of the OSU geometry were created for a mesh study, four created within the ANSYS software program ICEM v14.5 [103], one created in GridPro v5.1 [104], and all cases were run with the ANSYS solver CFX [105]. The first up pass, tip turn, and down pass of the OSU geometry was used for the initial mesh sensitivity study because additional passages do not introduce new flow physics so a subset of the full rig geometry was used to more efficiently test mesh resolution. Nusselt numbers, bulk temperatures, and bulk pressures were compared among three tetrahedral/prism (tet/prism) and two hexahedral (hex) meshes. Hexahedral meshes for the large aspect ratio geometries produce the same spatial resolution with fewer elements and nodes relative to a tet/prism mesh and therefore decrease the computational cost. A comparison of the HOST geometry with a tetrahedral mesh and a hexahedral mesh with the same wall spacing were compared for wall Nusselt numbers. Finally, a study comparing the results of single precision and double precision runs was done to determine if double precision was required for the geometry and the determined mesh.

3.2.1 Mesh Studies

Three tetrahedral meshes were created for the OSU AR 1:1 geometry that included the first passage, the first tip turn, and the second passage. Meshes are labeled based on the number of computational nodes across the top face of the turbulator. All meshes were computed using ICEM’s Octree methodology.

The 8 node mesh resulted in a surface mesh spacing of 0.00625 in and the 10 node mesh had a surface mesh spacing of 0.005 in. Both meshes were allowed to expand
unconstrained off the turbulators into the free stream, but were found to only grow to a size of 0.025 in, an Octree level 3, before being constrained by elements growing off the opposing wall. The 8 node refine mesh had a surface mesh spacing of 0.00625 in but had the free stream mesh constrained to an Octree level of 3, or 0.025 in. The wall-normal prism mesh parameters were identical among all the cases. Total prism thickness was set to 0.005 in and subdivided into 17 layers with a growth ratio of less than 1.25, resulting in a first cell height less than 3.0e-5 in. The resulting $y^+$ value was less than 1 for all OSU and HOST cases run. A summary of the specifications is shown in Table 5.

<table>
<thead>
<tr>
<th>Nodes on Top Face of Turbulator</th>
<th>Surface Mesh Spacing</th>
<th>Freestream Constraint</th>
<th>Total Prism Thickness</th>
<th>Prism Layers</th>
<th>Growth Ratio</th>
<th>First Cell Height</th>
</tr>
</thead>
<tbody>
<tr>
<td>8 Node</td>
<td>8</td>
<td>0.00625 in</td>
<td>n/a</td>
<td>0.005 in</td>
<td>17</td>
<td>&lt; 1.25</td>
</tr>
<tr>
<td>10 Node</td>
<td>10</td>
<td>0.005 in</td>
<td>n/a</td>
<td>0.005 in</td>
<td>17</td>
<td>&lt; 1.25</td>
</tr>
<tr>
<td>8 Node Refine</td>
<td>8</td>
<td>0.00625 in</td>
<td>0.025 in</td>
<td>0.005 in</td>
<td>17</td>
<td>&lt; 1.25</td>
</tr>
</tbody>
</table>

The Nusselt number comparisons among the three meshes are shown in Figure 22. There are minimal differences between the meshes, with the most variation occurring within the turn of wall A and wall D. Since there is not a consistent shift in values between the different meshes, they are determined to be comparable. The comparison to data was done to verify correct boundary conditions and the accuracy of the model results, but for simplicity the details of the comparison will be discussed in Chapter 4.
To further determine if the meshes are similar, temperature and pressure values were extracted from points directly in the center of the passages along the entire length of the CFD volume, Figure 23 and Figure 26. As was seen in the Nusselt number plots, the meshes show minimal variation between the runs, further demonstrating mesh independence.
The 8 node meshes both utilized approximately 35 million nodes and the 10 node mesh was approximately 53 million nodes, Table 6. The 8 node refined mesh was then determined to be the final mesh. It was chosen because it was still mesh independent with a lower node count than the 10 node mesh and because bulk mesh size control would be needed to insure the element sizes remained small enough to resolve the bulk flow accurately with increasing aspect ratios.

<table>
<thead>
<tr>
<th>Table 6: Node Count for Tetrahedral Meshes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Node Count [Millions]</td>
</tr>
<tr>
<td>8 Node</td>
</tr>
<tr>
<td>10 Node</td>
</tr>
<tr>
<td>8 Node Refine</td>
</tr>
</tbody>
</table>

Visuals of the selected 8 node refine mesh, at the middle of the second panel in the first passage shown in Figure 24 for the HOST geometry, can be seen in Figure 25.
Figure 24: 8 Node Refine Mesh Visual Location in the Mid Second Panel in the First Passage of the HOST Geometry

Figure 25: Mesh Visuals of the 8 Node Refine Mesh for the HOST Geometry. Picture Outlines Correspond to Zoomed In Location.
A further refined mesh was created to again check for mesh independence, specifically in the first passage since the largest amount of error for the AR 1:1 cases was found in the first passage, as will be addressed in Chapter 4 and Chapter 5.

For this test, the hexahedral mesh, CFD 8N R, comparable to the 8 node refine mesh, was created for the HOST AR 1:1 case. Another mesh was then created for the first pass only, using 0.0036 in node spacing, a 1.1 growth ratio, and a first cell height of approximately $3e^{-5}$ in. The mesh node count went from 61 million nodes for the full test section to 43 million nodes for the first passage only. The comparison of the Nusselt number, Figure 26, shows that the 8 node refine mesh is indeed mesh independent.

![Figure 26: Nusselt Number Comparison of Further Refined Mesh for HOST AR 1:1](image)

- **Re = 25k Ro = 0.25**
For further verification, the Nusselt number contours for wall C and wall D are compared in Figure 27 and Figure 28. There are no significant differences in contour patterns or Nusselt number values, which supports the mesh independence conclusion.

**Figure 27:** Wall C Nusselt Number Contours for (A) 8 Node Refine and (B) Further Refined Mesh for HOST AR 1:1 Re = 25k Ro = 0.25

**Figure 28:** Wall D Nusselt Number Contours for (A) 8 Node Refine and (B) Further Refined Mesh for HOST AR 1:1 Re = 25k Ro = 0.25
The bulk temperatures for both mesh cases are shown in Figure 29. There are three points with less than three degrees, or 0.5%, difference between the two mesh results. With all of this information, the mesh is determined to be independent.

![Bulk Temperature](image)

**Figure 29: Bulk Temperature Comparison for 8 Node Refine Mesh and Further Refined Mesh for HOST AR 1:1 Re = 25k Ro = 0.25**

### 3.2.2 Tetrahedral Mesh Versus Hexahedral Mesh

Initial work for the 1:1 aspect ratio geometries was done with tetrahedral meshes. With the expansion to AR 1:2 and AR 1:6, management of the node count with comparable results was desired. A hexahedral mesh allows for a reduction in the node count with the ability to handle high-aspect ratio near-wall cells and continually stretch to the free stream better than a tetrahedral/prism mesh. The cost of generating a hexahedral mesh for complex industrial geometry is often too expensive. Therefore, the HOST AR 1:1 geometry was meshed both with a tetrahedral mesh in ICEM [103] and with a hexahedral mesh in GridPro v.5.1 [104] to determine if both methods would provide similar results and allow either method to be used. The results of the two meshes are shown in Figure 30.
Figure 30: Tetrahedral Mesh Versus Hexahedral Mesh for HOST AR 1:1 Re = 25k Ro = 0.25

The hex mesh produced nearly identical results as the tet/prism for all but three comparison points. The differences for the three points were less than 20% and the overall comparison showed a difference of less than 7%. It was determined that the use of tetrahedral and hexahedral meshes produced equivalent results and justifies using either meshing strategy for these types of flows.

3.2.3 Single Versus Double Precision

Solver precision, single or double floating point accuracy, was investigated because the range of scales present in a CFD/HT simulation dictates the required numerical accuracy to avoid round-off error corrupting the calculations. Double precision allows for a difference of fifteen to seventeen decades between the smallest and the largest scales in the model, while single precision allows for six to eight decades. For the given geometry,
the smallest parameter is the first cell height of $3 \times 10^{-5}$ in and the largest is approximately twenty inches for the longest length of the OSU test section, resulting in six orders of magnitude, which is approaching the single precision limit. To be certain that single precision would be sufficient; the same case was run with single precision and with double precision. The results are shown in Figure 31.

![Figure 31: Nusselt Number Comparison of Single Versus Double Precision for OSU AR 1:1 Re = 50k Ro = 0](image)

The single precision results match the double precision results with minor discrepancy occurring in the second turn of walls B and D. The differences are small in magnitude and both follow the same trend so, on a Nusselt number comparison basis, it was determined that single precision was sufficient.
For completeness, values of temperature and pressure of the bulk flow at the direct center of the passages over the length of the test section were compared, Figure 32. As was the case for the Nusselt number comparison, a small discrepancy between the single and double precision occurs at the root turn but as was concluded for the Nusselt number comparison, the magnitudes and trends are similar allowing for single precision to be sufficient.

![Bulk Temperature Comparison of Single vs Double Precision](image1)

![Pressure Comparison of Single vs Double Precision](image2)

**Figure 32: Bulk Temperature and Pressure Comparison of Single Versus Double Precision for OSU AR 1:1 Re = 50k Ro = 0**

3.3 Data Post Processing

Specific methods for the post-processing and data reporting were done to ensure the comparison of the computational and experimental sources were on a consistent basis. Those methods included Nusselt number calculations, an area correction, and adjustment of the uncertainty analysis for use of a fixed inlet temperature.

3.3.1 Nusselt Number

For the OSU measurement program, the local bulk temperature was determined by a linear relationship for aspect ratios 1:1, 1:2, and 1:6. The linear local bulk temperature
relationship was established by running experiments for a smooth wall serpentine test section with thermocouples inserted into the center of the flow path and comparing the results to energy balance calculations.

For the CFD analysis, monitor points were put directly in the center of the passage and the average temperature for 1,000 iterations, after convergence, were compared with the linear temperature assumption. One thousand iterations were used due to the quasi-steady nature of the flow. The results are shown in Figure 33.

![Bulk Temperature AR 1:1](chart1.png)

![Bulk Temperature AR 1:2](chart2.png)

![Bulk Temperature AR 1:6](chart3.png)

**Figure 33: Bulk Temperature Comparison of OSU Re = 50k Ro = 0 and CFD**

The largest bulk temperature increase occurs within the first passage for the 1:1 aspect ratio causing the temperature distribution to have more of a parabolic shape than linear.
The CFD analysis also does not predict the same overall temperature rise in the bulk flow that was measured in the experiment.

The aspect ratio of 1:2 shows neither a linear nor parabolic trend for bulk temperature. In the first passage, oscillations are seen in the bulk temperature. This is due to occurrence and blending of pockets of warmer air in the center of the bulk flow where the temperature measurement was taken. The pockets can be seen in Figure 34. The temperature is scaled, Equation 16, relative to the maximum, maximum wall temperature, temperature of 549 R and the minimum, inlet flow temperature, temperature of 465 R. A change of 0.05 on the scale equates to a 4.2 R change in temperature.

![Figure 34: OSU AR 1:2 Re = 50k Ro = 0 First Passage Scaled Temperature Contour Cut at Mid Location Between Walls A and B](image)

**Equation 16: Scaled Temperature**

\[
T^* = \frac{(T - T_{inlet})}{(T_{maxwall} - T_{inlet})}
\]

If a streamwise bulk temperature were used for the calculation of heat transfer coefficients, it would be advisable to use a mass weighted area averaged value from the
top of the turbulators to the bottom of the opposite turbulators. This process would allow for a more consistently increasing bulk temperature value traveling through the passages.

Aspect ratio 1:6 bulk temperature was closer to linear in the second and third passage, but had a constant temperature for the first passage. Again this was because the bulk temperature was taken at a point in the center of the passage. As can be seen in Figure 35, the passage centerline does not start to see a temperature difference until the turn but a difference would be seen upstream of the turn if a mass flow area averaged bulk temperature were used. For Figure 35, the maximum temperature is 535 R and the minimum temperature is 465 R resulting in a 0.05 scale change equal to 3.5 R.

![Figure 35: OSU AR 1:6 Re = 50k Ro = 0 First Passage Scaled Temperature Contour Cut at Mid Location Between Wall A and B](image)

The HOST experiments preformed an energy balance for each segment of panels, rather than assuming a linear profile. The energy balance results compared better to the CFD bulk temperature values but there were still some discrepancies, especially before and after the turns, Figure 36.
The two experiments used two different methods for the determination of the bulk temperature. Neither method matched the CFD predictions nor did they exactly follow the same trends. The bulk temperature was also reported to be a primary source of error for both the OSU experiment and the HOST experiment as will be addressed in section 3.3.4. As a result, difference between the bulk temperature value of the experiment and the bulk temperature of the CFD would artificially increase the error between the experiments and the CFD. To prevent the compounding of errors, the inlet bulk temperature, which is measured in both configurations for all experimental runs, will be used to determine the Nusselt number for the CFD. The experimental data was re-post processed from the given heat flux data to determine the experimental Nusselt number based on the fixed inlet temperature. In addition, the use of the fixed inlet temperature is consistent with the Dittus-Boelter/McAdams correlation, which is based on inlet conditions.
3.3.2 Instantaneous Data Versus Averaged Data

Due to the unsteady nature of the investigated flow, even when run with a steady turbulence model, perturbations still appear in the monitored bulk temperature and heat flux values during the solver runs. Due to these perturbations, post-processing the results from the last iteration could be dependent on where the run was ended and potentially produce results deviating significantly from the averaged values. To investigate whether these perturbations could result in variations between the instantaneous reported values, heat fluxes were compared to averaged values for the last 1,000 iterations of the simulation. The 1,000 iterations were always at least 500 iterations after convergence.

Convergence was determined by monitoring the residuals of pressure, mass, and momentums, as well as the monitored bulk temperature and heat flux values. The run was considered to be converged when an average of the residuals, temperature, and heat flux values did not vary when averaged with 500 and 1,000 points.

For the OSU AR 1:1 without rotation, Ro = 0, simulation, the instantaneous values did not show any deviation from the averaged values for any of the walls. The comparison is shown in Figure 37.
Figure 37: Instantaneous Data Versus Averaged Data for OSU AR 1:1 Re = 50k Ro = 0

For the OSU AR 1:2 stationary test, the instantaneous values showed slight variation from the averaged values for the root turn on wall B and for the middle of the second passage for wall C. The difference was minimal and also maintained the trend seen in the averaged values so for the AR 1:2 case, the instantaneous and averaged values were determined to be equivalent.
The OSU stationary AR 1:6 simulation did not show variation between the Nusselt numbers for the instantaneous values and the averaged values for the side walls, wall A and wall B, but did show variation for the turbulated walls, wall C and wall D. The variation occurred within the second and third passages for both walls and within the root turn for wall D. The instantaneous values, while they varied from the averaged values, still maintained the appropriate trends. To minimize the difference between the experimental data and the CFD predictions, however, it was determined that an averaged value of the key parameters should be used, especially ones used within the Nusselt number calculation.
The influence of rotation on the oscillatory nature of the solution was also investigated. Figure 40 compares the HOST AR 1:1 Ro = 0.25 instantaneous and iteration averaged values.
The comparison of the instantaneous values and the averaged values for the HOST case are analogous to the results seen for the OSU AR 1:1 stationary case. This result suggests that the use of instantaneous or averaged values is more dependent on the aspect ratio than on the rotation. To verify the conclusion, the OSU AR 1:6, Ro = 0.3, simulation was used to compare instantaneous and iteration-averaged Nusselt numbers, Figure 41. The difference between the instantaneous values and the averaged values was greater than the difference seen for the OSU AR 1:6 stationary case. The side walls exhibit only slight differences between instantaneous and average values but the turbulated walls showed significant difference, especially in the second passage. This final analysis supports the conclusion that the aspect ratio has more of an effect on the difference between instantaneous and averaged values, and also shows that for the high aspect ratio, the
additional effect of rotation can increase the difference between instantaneous and averaged values.

![Graphs of Wall A, Wall B, Wall C, and Wall D with data points]

**Figure 41:** Instantaneous Data Versus Averaged Data for OSU AR 1:6 Re = 50k Ro = 0.3

### 3.3.3 Area Correction

Both the OSU and the HOST experiments normalized heat fluxes, heat transfer coefficients, Nusselt numbers, and enhancement factors, to a baseline smooth wall condition. The HOST documentation states that the total surface area for angled turbulators would be 1.15 times the smooth wall or projected wall surface area [7]. The OSU total surface area for angled turbulators was approximately 1.3 times the smooth wall surface area, where the difference is attributed to square turbulators instead of rounded ones as in the HOST configuration.
To allow for a direct comparison between the CFD and reported heat fluxes, the CFD was post-processed by calculating the area averaged heat flux on a given surface, multiplying it by the area which the boundary condition temperature was applied to, and then dividing by the experimental project area for the given panel.

### 3.3.4 Uncertainty Analysis

The OSU and HOST experiments both used the uncertainty method of Kline and McClintock [106]. The OSU experiment stated the Nusselt number uncertainty was ±4% at the inlet and for most of the first passage, with increasing uncertainty up to ±15% at the exit for the core experiments of Reynolds numbers between 10,000 and 75,000 [52]. The increased uncertainty is due entirely to the uncertainty of the bulk temperature because all other parameters in the evaluation are constant throughout the test section.

For the HOST experiment, the uncertainty was reported as ±6% at the inlet and ±30% at the exit for stationary experiments and up to ±40% at the exit of the trailing wall for rotating experiments [7]. As was the case for OSU, the calculation of the bulk temperature was the major error contributor with HOST stating it was approximately 75% of the uncertainty [7].

As was discussed in section 3.3.1, it was determined that the bulk temperature would be calculated from the experimentally measured inlet temperature, which is also an input for the CFD analysis. With the change in the calculation of the Nusselt number from a moving bulk temperature to a fixed reference temperature, the uncertainty analysis
needed to be revisited to determine the appropriate uncertainty to apply to the experimental data.

All parameters other than the bulk temperature are constant in the uncertainty analysis. The uncertainty of the parameters, including the uncertainty in the bulk inlet temperature measurement, would occur at the inlet where the bulk temperature was measured. Therefore, when using a fixed inlet bulk temperature, the appropriate uncertainty value to be added to the test data should be ±4% for the OSU data and ±6% for the HOST data.

The application of the uncertainty bands for the OSU AR 1:1 data is shown in Figure 42. The method used in plotting the data results in symbols which are approximately the same size as the error bars for both the OSU data and the HOST data. The only visible error bars are seen in the first pass on walls C and D. Therefore, for the remaining graphs error bars will not be added as the symbols effectively cover the range of the error bars.
3.4 Geometrical Sensitivity

Turbulated serpentine CFD/HT models generally represent the exact geometrical details as faithfully as possible; however, the following two details are difficult to resolve in practice:

- Turbulator profile; while turbulators are square in cross-section, manufacturing variations, especially those resulting from die wear, and material deterioration in the field result in a rounding of sharp corners.
- Supply circuit geometry; how do the models responses differ with inlet boundary conditions, inlet velocity and turbulence profiles, and does explicit modeling of geometric details far upstream of the first turbulators produce more accurate results in the first passage compared to simple flow specifications immediately upstream of the turbulators?
3.4.1 Square Turbulators Versus Rounded Turbulators

To determine if the HOST turbulator geometry being rounded or square is a factor, both geometries were meshed in a tetrahedral mesh to determine the difference. The HOST AR 1:1 geometry was run at a Reynolds number of 25,000 and a rotation number of 0.25 with rounded turbulators and again with square turbulators. The comparison is shown in Figure 43.

![Figure 43: Rounded Turbulators Versus Square Turbulators in a Tetrahedral Mesh for HOST AR 1:1 Re = 25k Ro = 0.25](image)

No difference in Nusselt number is seen for smooth wall A and minimal difference on smooth wall B in the second passage. For walls C and D there are minor variations between the two cases at locations before and after the first tip turn. The point in the middle of the second passage of wall C has an 18% variation and the point immediately
after the tip turn of wall D has a 27% variation. The remaining points are within an average 6% variation. The low variation between the two turbulator geometries is supported by experimental work which shows that low e/Dh turbulators with an e/w equal to 1, have similar average Nusselt number values, within 5%, between square and rounded turbulator geometries [20, 50, 54, 107-109].

While in two locations the variation between the rounded turbulator case and the square turbulator case had variations above 6%, the benefit to being able to mesh the geometry with a hexahedral grid and thereby reduced the mesh size, resulting in a decrease in the number of required CPU and overall simulation time, has a greater benefit to preliminary design type analysis than the increased accuracy for two locations. For the purposes of this investigation, manufacturing variations causing the two turbulator geometries are not significant for RANS models and are considered equivalent for a preliminary design analysis.

3.4.2 Inlet Boundary Conditions

For the OSU stationary experiments, a plenum was used to feed a two inch diameter pipe which in turn fed into another plenum, created by bolting flanges and the test section casing, before the air was fed into the unheated entry length. The unheated entry length was slightly longer than 6 hydraulic diameters for the 1:6 aspect ratio configuration. The air volume of the inlet geometry is shown in off-white in Figure 44 while the air volume of the test section is in blue.
The AR 1:6 geometry was chosen for the inlet geometry study since it was the configuration with the shortest hydraulic diameter entry length and would therefore have potentially the greatest impact on the inlet flow. The Nusselt number values for the two cases are shown in Figure 45.
There were marginal differences between the results with the inlet geometry explicitly modeled and when a uniform inlet profile was defined. As was stated before, if the inlet geometry was going to impact the Nu distribution, the greatest difference would occur in the 1:6 aspect ratio, which would lead to the conclusion that the variation would be even less for the smaller aspect ratio geometries with a hydraulic diameter entry length greater than 6.

For verification of minimal differences with the inclusion of the inlet geometry, profiles taken at the midpoint of each panel spanning from wall C to wall D for the first passage, are shown in green in Figure 46. The first passage only is shown since the Dean vortices within the turn mix the flow so the flow is no longer influenced by the inlet conditions.
The height was normalized to the total passage height and the temperature was scaled using Equation 16.

Figure 46: Profile Locations for OSU AR 1:6

The temperature profile for the entire height of the first passage is shown in Figure 47. The profiles show minor differences, less than 0.1T*, in the bulk scaled temperature downstream as can be seen by the reference scale shown on the plot. The differences are not significant enough to impact the Nusselt numbers or enough to require the inclusion of the full inlet geometry.
To verify that the profiles are not different at the near wall condition, the log scales of the profiles starting at wall D are shown in Figure 48. This comparison confirms that the inlet conditions do not affect the near wall conditions resulting in the same Nusselt number predictions.
The variation seen in the Nusselt numbers and the bulk flow is small enough that the added complexity of adding the additional geometry and the subsequent increase in mesh is not warranted. For the stationary OSU test section cases, the inlet geometry will not be included. For the rotating OSU test section cases, the inlet geometry changes, but not the entrance length. Every attempt will be made to create a uniform inlet to the entrance section for the rotating experiments, so for consistency and for a back-to-back comparison of trends, the geometry of the inlet configuration will not be included.
In the entrance length, there is a thermocouple probe that goes from wall D to the center of the flow path to measure the bulk temperature an inch ahead of the start of the first turbulator panel. It is shown in blue for the AR 1:6 geometry given in Figure 46. The probe was present in all experiments and included in all simulations. The effect of the probe is not noticeable in span wise cut planes shown in Chapter 4, Chapter 6, and Chapter 7 and it also does not show a difference in the comparison to HOST, which had an inlet probe upstream of the entry length. First passage scaled velocity ratio, Equation 17, contour plots, Figure 49, show that the bulk velocity in the region near the probe is affected but once the turbulator vortices begin, the influence of the probe is washed out quickly in the near wall region and then the bulk flow. As a result, it is concluded that the probe does not appreciably affect the Nusselt numbers.

Equation 17: Scaled Velocity Ratio

\[
\text{Scaled Velocity Ratio} = \frac{\text{Velocity}}{(\text{Maximum Velocity} - \text{Minimum Velocity})}
\]
Figure 49: OSU Re = 50k Ro = 0 First Passage Scaled Velocities for (A) AR 1:1, (B) AR 1:2, and (C) AR 1:6

3.5 Steady Turbulence Model Performance

Various Nu number predictions and computation expenses were evaluated for the various RANS based turbulence models available in CFX [105] for the OSU experimental
configuration. The models compared were all omega based models coded with wall integration.

The epsilon based models are stiff in the transition and lower sub-layers and are likely to diverge with integration; therefore, the epsilon models in CFX only utilize the wall function. Even with a fine enough mesh in the near wall region to allow for wall integration, the epsilon-based models will ignore the near wall mesh and apply wall functions. The wall functions are based on the universal law of the wall which does not account for the near wall flow structure that is accounted for with the omega, wall integration, models.

The omega based models are less stiff and are able to be integrated through the log, transition, and viscous sub-layers. The wall integration can account for flow reversal, 3D boundary layers, and high anisotropy, which provide the best possible predictions of wall heat flux for a RANS model.

The time evaluation was done in terms of preliminary design. The value of the solution is the accuracy of the model with respect to the time required to run the model. There is more value in a result with less accuracy that maintains the trends with less resources or time than the model with more accuracy but requiring significantly more resources or time.

Two additional investigations were run with the steady state models. The HOST experimental configuration was run with the SST turbulence model both with and without the curvature correction to determine the impact on the Nusselt number for a rotating
case. For the OSU configuration, the inlet turbulence length scale value was varied to investigate the effect of turbulence dissipation.

### 3.5.1 OSU Turbulence Model Study

The first three models compared are k-omega based models. The first is the k-omega model, $k\omega$, developed by Wilcox [110]. The next two, Shear Stress Transport (SST) and Baseline (BSL), are a blending of k-omega model near the walls and a k-epsilon model in the outer region. The difference between SST and BSL is the blending functions, which transition between the k-omega and k-epsilon models.

A production limiter was used for all three of the models. The production term in the turbulent kinetic energy (TKE) transport equation is based on the model of Kato and Launder [111] production limiter in CFX. The two equation turbulence models can result in an overproduction of turbulent energy in stagnation regions and then causes a buildup of turbulent kinetic energy. The increase in TKE is known to cause too large of a stagnation heat transfer coefficient (HTC). The production limiter is used in all the simulations since stagnation points occur repeatedly in turbulated geometry.
The comparison among the three models shows little to no variation between the models, Figure 50. The regions that show the most variation are wall A tip turn 1 and wall B root turn, but the differences are still small.

Nusselt number contours of wall C, Figure 51, and wall D, Figure 52, confirm the similarity of the models. Slight variations are present within the turns but have a minimal effect on the averaged Nusselt number values.

Figure 50: Two Equation Turbulence Models for OSU AR 1:1 $Re = 50k$ $Ro = 0$
Figure 51: OSU AR 1:1 Re = 50k Ro = 0 Wall C Nusselt Number Contours for (A) SST, (B) $K_ω$, and (C) BSL. Flow In Upper Left and Flow Out Lower Right.
From the three models, the SST model was selected. The k-omega model is comparable to the other models but the comparison is with the 1:1 aspect ratio geometry. For the 1:6 aspect ratio, the sensitivity of the k-omega model to the free stream conditions could result in deviations from the experimental results [110]. The BSL model was not chosen because it does not properly predict the onset of flow separation. The reason for this is that it does not account for the transport of the turbulent shear stress which may result in
an over prediction of eddy viscosity [110]. The SST model incorporates a function to limit the formulation of the eddy viscosity [110].

Next, the SST model was compared to the Reynolds Stress Models: the Explicit Algebraic Reynolds Stress Model (EARSM) and the Baseline Reynolds Stress Model (RSM BSL). The Reynolds stress models solve the transport equations for the individual Reynolds stress tensor components in the fluid. By solving for the stress tensor components the anisotropy of the flow, due to the turbulators and the turns, is solved instead of assuming isotropic flow in the RANS models. The RSM BSL is based on the BSL turbulence model. The EARSM model, instead of directly solving the Reynolds stresses, solves algebraic model equations for the individual Reynolds stresses. The comparison between the results of these models is shown in Figure 53.
The comparison shows similar Nusselt numbers for the SST model and the EARSM model for the smooth walls, wall A and wall B. For the turbulated walls, the SST and RSM predictions were also similar. This is confirmed with the Nusselt number contour plots for wall C and wall D, Figure 54 and Figure 55 respectively, where little to no difference is noted between the models. All models were close in Nusselt number predictions, contours, and all followed the same trends.

While the RSM and EARSM model are more theoretically able to model the complex nature of the turbulated flow, the results do not always show better agreement with the experimental results than the two equation models, such as the SST model [110, 112].
Due to the similar nature of the three models, and the added computational time for the EARSM and RSM BSL model, the SST model was selected to use for future analysis.

Figure 54: OSU AR 1:1 Re = 50k Ro = 0 Wall C Nusselt Number Contours for (A) SST, (B) EARSM BSL, and (C) RSM BSL. Flow In Upper Left and Flow Out Lower Right.
For a final steady model comparison, the SST model was compared to the laminar model, which does not include any turbulence model. By comparing the laminar model to the SST model, the effect of the added turbulence model can be seen. The laminar model predicted systematically lower Nusselt numbers, than the SST model, since it was not modeling the turbulator mixing causing the near wall temperatures to be too warm. The results are shown in Figure 56.
For the turbulated walls, the laminar model has similar predictions to the SST model for the second passage, the root turn, and the third passage. For the turbulated walls, the near wall results (specifically the Nusselt number) are not affected by the added turbulence modeling of the SST model in those areas. For the first passage and the first tip turn, there was a consistent increase in Nusselt number for both turbulated walls from the laminar to the SST model.

For wall A, the models compare for the third passage and for wall B the models compare for the second passage. The similarity is that the near wall Nusselt numbers for the
outside wall after the turn the turbulence model does not affect the results but for the
inner wall downstream of the turn, the turbulence increases the Nusselt number. For the
first passage, as was seen for the turbulated walls, there was a consistent increase in
Nusselt number for these two walls.

Wall Nusselt number contours for the turbulated walls for SST and laminar cases are
shown in Figure 57 and Figure 58. The Nusselt number contours show decreased Nusselt
numbers downstream of the turbulators and an overall smaller contour pattern for the
laminar case. As will be noted in Chapter 5, the Nusselt number contour directly relates
to the turbulator vortex structure, strength, and path. A decrease in the Nusselt number
and the pattern equates to a less developed structure. The results show that the laminar
model is unable to capture the required physics of the case and is insufficient for these
types of flows even though there are many low Reynolds number flow regions present.
Figure 57: OSU AR 1:1 Re = 50k Ro = 0 Wall C Nusselt Number Contours for (A) SST and (B) Laminar. Flow In Upper Left and Flow Out Lower Right.
3.5.2 Steady Turbulence Model CPU Hours

The six models; laminar, SST, k-omega, BSL, EARSM BSL, and RSM BSL, were compared in terms of the number of CPU’s used for the simulation times the total number of lapsed solver hours, which will be referred to as CPU hours. For the laminar, SST, k-omega, and BSL cases, they were able to start up stably with just an overall initialization of the flow volume. They were all able to converge in approximately 500 iterations and were run for a total of 2,000 iterations to allow for consistency and a minimum of 1,000 iterations past the point of convergence to be used for averaging.
The EARSM and RSM BSL models would diverge with only an overall initialization of the flow so they had to be started from other converged models. The EARSM model was started from the BSL, so the amount of time to run the BSL model was included in the EARSM total CPU hours. The EARSM also required more than 2,000 total iterations for 1,000 averaging iterations. As a result, the CPU time for the EARSM was over five and a half times the amount for the BSL. The RSM BSL was started from the EARSM, again due to stability issues. The RSM BSL also took more than 2,000 iterations beyond the EARSM to have 1,000 iterations after convergence occurred for averaging.

The SST model gives comparable results to the higher complexity models without the added computational expense, as shown by the CPU hour comparison.
3.5.3 HOST Turbulence Model Study

For the rotational cases of the HOST measurement program, a curvature correction option was included with the SST model for comparison. The curvature correction option makes the two equation eddy viscosity models sensitive to streamline curvature and rotation by modifying the turbulence production term. An increase in the turbulence production term would result in an increase in the HTC's [110]. In theory, increased Nu should be evident on the turbulated walls, within the turn, and increase more with the addition of rotation. Including the rotational effects is important for the rotating HOST case. The Nusselt number results are shown in Figure 60 and the mid passage TKE values for the tip turn region are shown in Figure 61.

![Figure 60: Two Equation Turbulence Model With and Without Curvature Correction for HOST AR 1:1 Re = 25k Ro = 0.25](image-url)
Figure 61: Turbulent Kinetic Energy HOST AR 1:1 Re = 25k Ro = 0.25 Mid Passage Contours for Passage 1 Panel 4, Tip Turn 1, and Passage 2 Panel 1 for (A) SST and (B) SST with CC. Flow In Bottom Right and Flow Out Top Right.

The added curvature correction shows minor differences from the SST model without the curvature correction in the Nusselt number values but shows more of a difference in the TKE contours with the differences focused around the turns. The differences present in the average Nusselt number values are at the outer wall of the first tip turn, wall A, before and after the first tip turn for the leading wall, wall C, and just before the first tip turn for wall D. The majority of the differences occur on wall C before the turn due to the rotational effects biasing the air towards wall D and after the turn due to the curvature and rotational effect moving the air towards the leading wall. The curvature correction does affect the heat transfer results around the turns and therefore it was chosen to be included for the subsequent models.
3.5.4 Inlet Turbulence Length Scale

The inlet turbulence intensity had been previously set to 5% and the turbulent eddy frequency was estimated from TKE, kinematic viscosity, and an assumed eddy viscosity ratio of 10 [111]. The turbulent parameters are unknown since they are generally not measured as part of the experiments and were not for the OSU or HOST experiments.

To determine the effect that the inlet turbulent boundary conditions could have on the predictive model, the inlet turbulent length scale, which feeds into the eddy frequency, was varied between having the length scale equivalent to the hydraulic diameter to having the length scale equal to 0.01 times the hydraulic diameter. The results are shown in Figure 62.
The variation in the inlet turbulence resulted in a prediction of essentially the same Nusselt number distribution. The reason is that the inlet turbulence is washed out by the probe location, which is an inch before the heated panels. This can be seen in the viscosity ratio contours for the two cases shown in Figure 63.
The conclusion can be made that the inlet turbulence is not a primary driver for RANS calculations in CFX for the Nusselt number predictions. Engineering judgment, therefore, was used to approximate the turbulence value for the given inlet geometry.

3.6 Unsteady Turbulence Models

For the unsteady turbulence model investigation, the URANS SST model was compared to the steady state SST model, with curvature correction, for the HOST AR 1:1 case. This was done to determine if the unsteady model would change the Nusselt number values for the rotating case. The comparison of the time requirement was done for those two cases.

In addition to the HOST case, the OSU AR 1:6 stationary case was also run with the URANS model and the results were compared to the steady SST model results. This comparison was done to determine if the results of the unsteady model would vary from the results of the steady predictions for the high aspect ratio geometries.
3.6.1 HOST Turbulence Model Study

For the HOST case of 25,000 Reynolds number and a rotational number of 0.25, the results of the SST model with the curvature correction and the results of the SST URANS model were compared. The comparison presented in Figure 64 shows that the SST model and the URANS model trend together and deviate from each other slightly immediately after the turn for wall C. For the remainder of the comparison the two models match and the average difference between the results of the two models is less than 6%.

![Graphs of Nu vs Distance for Wall A, Wall B, Wall C, and Wall D showing CFD SST CC and CFD URANS results](image)

**Figure 64: Unsteady Turbulence Models for HOST AR 1:1 Re = 25k Ro = 0.25**

While the iteration averaged Nusselt number values do not show significant differences between the simulations, differences do exist. Steady CFX calculations are time inaccurate transients that result in steady state behavior. The SST time step was automatically calculated and was $1.7 \times 10^{-3}$ seconds. The URANS time step, however, was
set to 1.0 e-4 seconds. The different time step and model results in increased magnitude of oscillations of monitored heat flux values for the URANS simulation.

The increased magnitude is evident in the instantaneous Nusselt number contour plots for the SST and URANS simulations as shown in Figure 65 and Figure 66. Specifically, wall C shows increased Nusselt number values but the iteration averaged Nusselt number plots, Figure 64, do not show as much of a difference. The result is that the URANS simulation is unsteady and the instantaneous contour plot shows Nusselt number values from an oscillation peak of Nusselt number and at another instant, the value will decrease below to average out to the value shown in Figure 64.
Figure 65: HOST AR 1:1 Re = 25k Ro = 0.25 Wall C Nusselt Number Contours for (A) SST CC and (B) URANS. Flow In Upper Left and Flow Out Lower Right.
Figure 66: HOST AR 1:1 Re = 25k Ro = 0.25 Wall D Nusselt Number Contours for (A) SST CC and (B) URANS. Flow In Upper Left and Flow Out Lower Right.

The iteration averaged value of the URANS simulation results in values within an average of 6% of the SST model. As a result, the SST model is selected over the URANS model for the reduced computational expense.
3.6.2 Unsteady Turbulence Model CPU Hours

The two models, steady SST and Unsteady SST (URANS), were compared in terms of CPU hours. The URANS model was initialized with the converged SST solution. In addition to requiring a model to restart from, the URANS model also required additional iterations in order to have 1,000 iterations after convergence.

![Turbulence Model CPU Hours Chart](image)

**Figure 67: Unsteady Turbulence Model CPU Hours for HOST AR 1:1 Re = 25k Ro = 0.25**

The SST model had lower CPU hours as expected due to the URANS model requiring initialization from another model and having a smaller time step. The added CPU time and complexity of the URANS model, without providing significantly different results than the SST model, resulted in the SST model with curvature correction to be chosen over the URANS model for future studies.
3.6.3 OSU Turbulence Model Study

For completeness, the OSU AR 1:6 model stationary case was run with both SST and URANS to determine if there would be a difference in the steady versus unsteady results for the high aspect ratio case. The Nusselt number results, Figure 68, and the Nusselt number contours, Figure 69 and Figure 70, showed slight discrepancies between the two models in the turn regions but the models both showed similar trends, which agree with the results from the HOST study. For the purposes of primary design and consideration of time to run the models, it was decided that the SST model would provide sufficient results for comparison to data for the stationary OSU and stationary and rotating HOST data, as well as predictions for the OSU rotating experiments.

Figure 68: Unsteady Turbulence Model for OSU AR 1:6 Re = 50k Ro = 0
Figure 69: OSU AR 1:6 Re = 50k Ro = 0 Wall C Nusselt Number Contours for (A) SST and (B) URANS. Flow In Upper Left and Flow Out Lower Right.
3.7 Turbulator Heat Flux

In the OSU experiments, the turbulators were attached to the copper panels by a thin layer of conductive adhesive, 426 Loctite. While the adhesive thermal conductivity is significantly smaller than the copper panels and turbulators, less than 1% [8], it does not thermally isolate the turbulators completely and the turbulator surface heat flux must be assessed. Boyle [113] concluded that the fin effect of AR 1:1 integral turbulators was insignificant and the increase in heat transfer was only due to the turbulence created by the turbulators.

For comparison, the OSU AR 1:6 stationary case was run with isothermal turbulators and then with adiabatic turbulators to investigate the maximum possible difference between
the two turbulator boundary conditions. The difference was significant for the turbulated walls as seen in Figure 71. To determine the correct boundary conditions, a turbulated wall, Figure 72, was heated and the surface temperatures of the turbulators, the walls between the turbulators, and the Garolite spacers were compared with an infrared camera [114].

Figure 71: Heated Versus Unheated Turbulators for OSU AR 1:6 Re = 50k Ro = 0

Figure 72: Painted Turbulator Wall. Green Arrows Note Garolite Spacers and Black Arrows Denote Areas With Small Scratches in Paint. [114]
The measurements were taken at four different wall temperature settings for the copper panels: room temperature 71.6F (22C), 86F (30C), 95F (35C), and 104F (40C); and the IR camera results are shown in Figure 73.

![Plate at room temperature (22°C)](image)

![Plate at 30°C](image)

![Plate at 35°C](image)

![Plate at 40°C](image)

**Figure 73: IR Camera (FLIR A655sc) Images of Heated Turbulator Wall [114]**

The conclusion is that the turbulators, along with the Garolite spacers, reach an essentially uniform temperature consistent with the temperature of the base copper panels. As a result, the CFD analysis was done with an isothermal boundary condition on the turbulators and Garolite spacers to match the base temperature of the copper panels.

### 3.8. Temperature Rise Due to Pumping

To achieve the desired conditions for the rotating cases, initially all the boundary conditions were kept the same as the stationary case and rotation was applied to the model. The rotation number set the rotation rate as shown in Equation 18.
Equation 18: Rotation Rate

\[ \Omega = \frac{R_o \times V}{D_R} \]

For AR 1:2 with Ro = 0.3, the predicted Nusselt numbers resulted in trends, Figure 74, inconsistent with ones seen in literature. Previous publications indicate that the first passage has a decrease in Nu for the leading wall and an increase for the trailing wall and the second passage has an increase in Nu for the leading wall and a decrease in Nu for the trailing wall [3, 9, 37, 39]. The trend seen, however, is the same on all the walls, smooth and turbulated, with the Nusselt number going negative in and around the first tip turn, increasing to the root turn, and again decreasing to negative values in the third passage. The results are unphysical and would not occur within an actual engine blade.

![Graphs showing Nusselt numbers for different walls](image)

Figure 74: OSU AR 1:2 Re = 50k Ro = 0.3 CFD Predictions
The trends indicated that Nu was strongly dependent on the radial position of the panels with respect to the axis of rotation. For further confirmation, the wall Nusselt number values for wall C, Figure 75, and wall D, Figure 76, were interrogated.

![Image: OSU AR 1:2 Re = 50k Ro = 0.3 Wall C Nusselt Number Contour. Flow In Upper Left and Flow Out Lower Right.]

The leading wall showed lower Nusselt number values than the trailing wall, which was expected, but by two panels upstream of the first turn and two panels downstream, the Nusselt number values were zero or negative numbers. The plots were limited to positive Nusselt number because once the heat flux reached zero, the experiment would be unable to obtain measurements due to the configuration of the experiment.
The dominant factor for the Nusselt number results was the radial location and the cause was determined to be from a temperature rise due to pumping, \( dT_{\text{pump}} \).

\[ \text{Equation 19: Temperature Rise Due to Pumping} \]
\[ dT_{\text{pump}} = \frac{\Omega^2 (R_{\text{Max}}^2 - R_{\text{inlet}}^2)}{2g \rho L c_p} \]

Temperature rise due to pumping, shown in Equation 19, is commonly accounted for in blade cooling design; however, the difference between the wall and the coolant temperatures are much greater in engine applications. Nusselt numbers in small temperature difference experiments could go negative when the temperature rise due to pumping becomes greater than temperature difference between the bulk flow and the wall. To correctly model the effects of rotation, the \( dT_{\text{pump}} \) should be addressed in the experiment and set to a level that is comparable to actual engine conditions.
The $dT_{pump}$ was not previously addressed in the literature because the experimental rigs are generally shorter in length, closer to the axis of rotation, run at higher operating pressures, run at slow rotation speeds, or a combination thereof [3, 7, 10, 16, 25, 26, 29, 34, 36-41, 43-45, 48, 51, 52, 56, 57]. The pumping in these rigs account for less than 10% of the total temperature difference between the inlet fluid and the heated walls and was therefore unnoticeable by the investigators.

To quantify the $dT_{pump}$, two additional cases were run, BCs 2 and BCs 3, to compare to the initial case, BCs 1, Table 7. The three cases were run with identical boundary conditions except the inlet pressure was varied in order to modify the rotation speed and therefore the $dT_{pump}$. The operating pressure was varied because this is how the experimental facility would adjust the pumping effect. However, adjusting the operating pressure modifies the density. As a result, the bulk inlet velocity needs to be adjusted to maintain the same Re, which causes a mismatch in inlet Mach number. The flow, however, always remains within the incompressible flow regime.
Table 7: Comparison of AR 1:2 Re = 50k Ro = 0.3 Cases BCs 1, BCs 2, and BCs 3

Case Parameters

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The three cases were compared in terms on Nusselt number values for each of the walls and can be seen in Figure 77. As expected, BCs 1 case shows the strongest effect of the dT<sub>pump</sub>, BCs 2 shows the least, and BCs 3 shows intermediate results. The results are comparable to the hand calculated dT<sub>pump</sub> values in Table 7. The BCs 2 case maintained the expected results of lower Nusselt numbers on the leading wall in the first pass than the trailing wall and the reverse in the second pass due to the change in Coriolis force and from buoyancy forces opposing rather than compounding the rotational effects.
The Nusselt number contours for the three cases of wall C, Figure 78, are compared visually to determine the effect of the $dT_{pump}$. As was shown in the Nusselt number plots, the BCs 1 case shows strong dependence on the pumping effects with the entire first tip turn region at a value of zero or less. The BCs 3 case shows an indication of the effects of $dT_{pump}$ but with the revitalization of the Nusselt number in the turn, it is not a dominant factor in BCs 3. In BCs 2, the Nusselt number values and patterns are as to be expected, supporting the conclusion that the Nu number results of BCs 1 was due to the dominance of $dT_{pump}$. 

Figure 77: OSU AR 1:2 $Re = 50k$ $Ro = 0.3$ CFD Predictions for Three Cases With Varying Temperature Rise Due to Pumping
Figure 78: OSU AR 1:2 Re = 50k Ro = 0.3 Nusselt Number Contours for Wall C for Three Cases With Varying Temperature Rise Due to Pumping, (A) BCs 1, (B) BCs 2, and (C) BCs 3. Flow In Upper Left and Flow Out Lower Right.

The trailing wall, D, was also investigated with Nusselt number contours in Figure 79. As was the case for wall C, BCs 1 shows the effects of $dT_{\text{pump}}$ in the first tip turn. Cases BCs 2 and BCs 3 do not show the effects, except for BCs 3 having lower Nusselt number values than BCs 2.
Figure 79: OSU AR 1:2 Re = 50k Ro = 0.3 Nusselt Number Contours for Wall D for Three Cases With Varying Temperature Rise Due to Pumping, (A) BCs 1, (B) BCs 2, and (C) BCs 3. Flow In Upper Left and Flow Out Lower Right.

It was determined that for the CFD analysis, effects of $dT_{pump}$ would be constrained to a low value for the following two purposes:

- A low $dT_{pump}$ will provide a complete set of comparable Nusselt number values from the experiment since a high $dT_{pump}$ would cause the measurements to be zero
The characteristic trends seen in the lower $dT_{pump}$ cases are more representative to what would be seen in an engine.

The ratio of the $dT_{pump}$ to the difference between the inlet temperature and the average wall temperature was used to quantify the relative effect of pumping. For the cases shown above, the values are shown in Table 8. For the CFD analysis of the rotating cases, the ratio was set to 0.05 for margin, to prevent heat flux from going to zero, when performing the experiments and to better replicate the engine environment.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Units</th>
<th>BCs 1</th>
<th>BCs 2</th>
<th>BCs 3</th>
</tr>
</thead>
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<td>7.1</td>
<td>35.5</td>
</tr>
<tr>
<td>$dT_{pump}/(T_{wall}-T_{in})$</td>
<td>-</td>
<td>0.99</td>
<td>0.09</td>
<td>0.44</td>
</tr>
</tbody>
</table>

3.9 Conclusions on CFD Configuration

Through mesh sensitivity studies, it was determined that eight nodes on a turbulator side, for a characteristic near-wall mesh spacing of 0.00625 inches, with a first cell height of 3.0e-5 inches, a growth ratio of 1.25, and a maximum cell size of 0.025 inch would result in mesh-independent solutions over the range of parameters investigated. Single precision numeric was sufficient for the given mesh size and test section size and resulted in no significant differences from double precision results. A creation of a comparable hexahedral mesh with the same node spacing and growth ratio resulted in similar Nusselt numbers to the tetrahedral/prism mesh. Tetrahedral meshes are therefore recommended for RANS calculations early in the design cycles based on the mesh generation time.
The turbulator geometry, whether rounded or square, did not prove to be a primary driver of the Nusselt number predictions. The inclusion of the inlet geometry also did not affect the results for the aspect ratio of 1:6, which would likely have the greatest dependence on the inlet flow conditions due to the shortest entrance length of all configurations.

The turbulators for the OSU test section were attached with a thin layer of Loctite 426 adhesive while the HOST turbulators were machined into the copper panels. As will be seen in section 4.4, the comparison of the two sets of test data show analogous results. Measurement results obtained using the infrared camera indicated that the turbulators attached with adhesive for the OSU test section were isothermal with the copper panels. The CFD model turbulators and Garolite spacers’ surface temperatures were defined to be equal to the controlling panel and included in the heat flux calculations.

The CFD analysis had an area correction applied to account for the inclusion of the turbulators and Garolite spacers in the heat flux. The OSU and HOST test section reported heat flux and Nusselt numbers on a project area, or smooth wall, basis. The CFD analysis, for an equal comparison, applied an area correction so the results were also on a project area basis.

Due to the uncertainty in the measured bulk temperature through the serpentine and the CFD predictions invalidating the linear distribution assumption, heat transfer coefficients were calculated using the inlet, rather than the free stream temperature. With the change to the formulation of the Nusselt number, the uncertainty analysis was revisited and
concluded that the OSU test section would have an approximately ±4% uncertainty and HOST would have an approximately ±6% uncertainty on the Nusselt number data.

The heat flux values and resulting Nusselt numbers will be done on an iteration-averaged basis. The instantaneous values did not show a difference from the averaged values until a high aspect ratio, high rotation number case, but for consistency, all cases will use iteration averaging.

The turbulence model comparison encompassed six steady models and one unsteady model. The Reynolds Stress Models did not show improvement over an SST model with a Kato-Launder production limiter and a curvature correction and also showed that they required over five and a half times the amount of CPU hours. The URANS model also did not show a change in Nusselt number predictions over the SST model, and it also required approximately six times the CPU hours. As a result, the SST model with a Kato-Launder production limiter and a curvature correction was the computation model selected for future analysis.

Temperature rise due to pumping had not been addressed previously in the literature but was determined to be a factor in the future rotating experiment. The rotating cases were set to have a set temperature rise due to pumping compared to the difference of inlet to wall temperature of 0.05. This value allows for margin during the experiment and also better replicates the engine environment.
Chapter 4

Low Aspect Ratio Stationary

The low AR stationary cases were analyzed first to establish SST’s ability to predict the heat transfer rate for stationary turbulated serpentines before adding the complexities of rotation and high AR. Computations were performed for the OSU AR 1:1 stationary experiment for Re = 50,000 and for the HOST AR 1:1 stationary experiment for Re = 25,000 and 50,000 to determine the ability of CFD to predict the Nusselt number dependency on Reynolds number. The OSU and HOST experimental data for Re = 50,000 were first compared to illustrate any differences in the test sections and then compared against CFD predictions.

The focus for this section is the data match. Discussion of flow mechanisms for the full test section will be done in section 6.3.

4.1 OSU Low Aspect Ratio Stationary

The OSU low aspect ratio stationary case at 50,000 Reynolds number is compared to CFD in Figure 80. For the smooth walls, walls A and B, the CFD shows good comparison with the data. For the turbulated walls, good agreement, less than 10% beyond experimental uncertainty, is seen from the first tip turn to the end of the third passage.
In the first passage, reasonable agreement is seen between the CFD predictions and the data. The CFD has a difference, beyond the experimental uncertainty, of less than 15% except for the last point in the first passage for wall C, which is 37%.

![Figure 80: OSU AR 1:1 Re = 50k Ro = 0 Compared to CFD Prediction](image)

The CFD predictions for walls C and D for the first passage are very similar, with an average point-to-point difference of 5% between the walls; however, the data shows an average of 12% variation between the walls. The HOST experiment exhibits the same differences between walls C and D and it was postulated to be due to the staggering of the trips on the surfaces and the presence of the half turbulators on wall D, which were replicated on the OSU test section [7].

The first two panels of the first passage have similarly high Nusselt numbers. Immediately on the first panel, the Nusselt number contours show high values that
encompass the full area between the turbulators, Figure 81 for wall C and Figure 82 for wall D.

Figure 81: Nusselt Number Contour for OSU AR 1:1 Re = 50k Ro = 0 Wall C Passage 1. Entrance Removed for Clarity.

Figure 82: Nusselt Number Contour for OSU AR 1:1 Re = 50k Ro = 0 Wall D Passage 1. Entrance Removed for Clarity.

For additional understanding of the flow between the turbulators for the first two panels, visualization of the Q criterion, colored based on temperature, was done, Figure 83. The Q criterion is a qualitative post-processing technique used to visualize coherent structures.
in turbulent flow by computing the invariants of the velocity Jacobian. In this case it is computed as the absolute value of the vorticity minus the absolute value of strain rate such that it identifies vortices while removing the shear layers [111]. For position reference, planes P1P1 and P1P2, Figure 84, are shown in Figure 83.

![Figure 83: Q Criterion for OSU AR 1:1 Re = 50k Ro = 0 for First Two Panels of Passage 1](image1)

![Figure 84: Plane Locations and Labels for OSU AR 1:1 First Passage](image2)

Figure 83 illustrates that the contour shape is due to the turbulator vortex path. The turbulator vortices encompassing the full area between the turbulators enhance the Nusselt number values.

In addition to the vortex flow path, the high Nusselt numbers are also due to low temperatures near the walls as seen in Figure 85 and Figure 86. The turbulator vortex
structure is still forming in the passage, Figure 85, resulting in higher velocity in the near-wall turbulator region as can be seen by the velocity vector size and in Figure 87.

The full formation of the turbulator vortex structure by the middle of the second panel increases the bulk temperature resulting in warmer flow moving towards the walls. The Nusselt number in the turbulator vortex path decreases due to the increase in near-wall temperature and the averaged Nusselt number decreases as a result.

The asymmetric nature of the temperature contour, Figure 86, from wall C to wall D is due to the initial half turbulator and the additional downstream half turbulators. The initial half turbulator delays the full formation of the turbulator vortex structure in the initial passage resulting in a compressed turbulator vortex structure for wall D, Figure 85.
The asymmetric nature of the velocity contour within the turbulated segment is also due to the partial turbulator, but at the inlet, the thermocouple probe causes the nonsymmetrical velocity. The thermocouple probe is inserted from wall D and goes to the middle of the passage. It is located one inch from the start of the turbulated section.

The comparison with data shows that the CFD is not predicting the increase in Nusselt number occurring from the first to the second panel. The increase in the data is due to the hydrodynamically developed entry length. Saha and Acharya documented that with a
fully developed hydrodynamic entrance length the heat transfer values initially increase and then decrease as the entrance effects are washed out by the turbulator vortices [55]. The previous work in section 3.4.2 and section 3.5.4 shows that the CFX SST model is not sensitive to the primary inlet conditions investigated so it is unlikely that the modeling of inlet conditions is solely responsible for the discrepancy. An additional possible cause is that the RANS model is not capturing the step increase in the mixing rate occurring after the first few turbulators. RANS models have historically underpredicted mixing. To verify this conclusion, PIV experimental data or high fidelity simulations would be needed. Irrespective, while the absolute values are not matched, the general trend is still captured by the analysis.

For the data point immediately before the turn on wall C, the CFD predicted Nusselt number is 37% higher than the measurement. From Figure 86, it can be seen that the CFD has a bias of colder air towards wall C for the entire passage, contributing to the overprediction of the Nusselt number compared to measurement.

### 4.2 HOST Low Aspect Ratio Stationary

CFD predictions for the HOST low AR stationary cases with Re = 25,000 and 50,000 were compared to the experimental results. For the 25,000 Reynolds number case, the smooth side walls showed good agreement with the data along with the turbulated walls from the start of the first tip turn to the end of the second tip turn, Figure 88.
The predictions for the first passage of walls C and D showed the most deviation from the measurements and also showed similar values for both walls C and D. The difference for walls C and D, beyond the experimental uncertainty, was approximately 10-15% except for the third point in passage 1, where the difference was approximately 30%. The third data point in passage 1 equates to the second turbulated panel since the first panel in HOST is smooth and heated.

The formulation of the turbulator vortex path can be seen in the formation of the Nusselt number contours on walls C and D for passage 1 in Figure 89 and Figure 90, respectively. On the first turbulated panel for both walls C and D, the Nusselt number values are high starting immediately downstream of the turbulator at the intersection with wall B. The angled of the turbulator causes the secondary vortex to move diagonally to wall A,
decreasing in Nusselt number but increasing in the size of the affected area. The end of the first turbulated panel and the beginning of the second turbulator panel shows the largest Nusselt number values and the vortices generated by the top and bottom turbulator wall have grown large enough to interact with one another, Figure 92. The magnitude of the Nusselt number then decreases as the flow continues down the second turbulated panel on to the third due to the increased temperature of the bulk flow moving to the near-wall region. The affected area of the turbulator vortices however, maintains the same shape since they are fully developed, Figure 92.

![Wall C Nusselt Number Contours for HOST AR 1:1 Re = 25k Ro = 0 Passage 1](image)

**Figure 89: Wall C Nusselt Number Contours for HOST AR 1:1 Re = 25k Ro = 0 Passage 1**
Figure 90: Wall D Nusselt Number Contours for HOST AR 1:1 Re = 25k Ro = 0 Passage 1

Figure 91: Plane Locations and Labels for HOST AR 1:1 Passage 1

Figure 92: Passage 1 Scaled Temperature Contours With Secondary Flow Velocity Vectors for the Passage 1 of HOST AR 1:1 Re = 25k Ro = 0. Left Side is Wall A and Right Side is Wall B.
In the streamwise temperature contour of the first passage, Figure 93, the start of the formation of the recirculation downstream of the turbulators can be seen in the first turbulated panel. The turbulator vortex formation increases the near wall secondary flow and also entrains bulk air, together resulting in high Nusselt numbers for the first two turbulator panels.

The cut turbulator on wall D results in the higher velocity flow along wall D, as was seen for the OSU case, Figure 85. The higher velocity results in more cool bulk flow transported to the near-wall region, Figure 93, resulting in slightly higher Nusselt numbers for wall D in the first passage.

![Figure 93: CFD Prediction for HOST AR 1:1 Re = 25k Ro = 0 Streamwise Temperature Contour in Passage 1](image)

The predicted velocity contour of passage 1 shown in Figure 94 suggests that the strong shear occurring at the second turbulated panel immediately around the turbulator, along with the low temperature air, is the likely cause of the continued increase in the Nusselt number between the first and second turbulated panels.
The data shows that the Nusselt number should be approximately constant for the first and second turbulated panels. This would indicate that the formulation of the turbulator resulting vortex structure should develop faster on the first panel, be relatively steady for the remainder of the first panel, and relatively steady for the second panel, Figure 89 and Figure 90. The Nusselt number contours show the delayed strength of the turbulator vortex structure by the area and magnitude of the Nusselt number still increasing to the end of the first turbulated panel, as shown in Figure 92.

For the HOST stationary experiment with a Reynolds number of 50,000, the model accuracy follows the same trends as the Re = 25,000 case, Figure 95. There is good agreement with walls A and B and also with walls C and D from the start of the first tip turn to the end of the second tip turn. Again, the largest discrepancy between the CFD predictions and the measurements is in the first pass. The difference is between 10-15%
except for the third streamwise data point, which is approximately 30% for both walls C and D.

![Graphs of Wall A and Wall B Nusselt number contours](image)

![Graphs of Wall C and Wall D Nusselt number contours](image)

Figure 95: HOST AR 1:1 Re = 50k Ro = 0 Compared to CFD Prediction

The Nusselt number contours for walls C and D, Figure 96 and Figure 97 respectively, show a different initial structure to the turbulator vortex flow path contour. On the first turbulated panel by the second turbulator, the Nusselt number contour angled from one turbulator to the next, has encompassed the full region between the turbulators which is much more similar to the OSU 50,000 Reynolds number case than the HOST 25,000 Reynolds number case indicating that the initial formulation of the turbulator vortex path is dependent on the Reynolds number.
While the Nusselt number contour shapes show similarity between OSU and HOST, the trend in the predicted Nusselt number values is not the same. Between the first and second turbulated panels the predicted Nusselt number remains constant for the OSU case while it increases with the HOST case. The cause is due to the CFD analysis not capturing the full effect of the inlet geometry and conditions, which is supported by the
investigation of the inlet geometry in section 3.4.2 and the experimental work by Saha and Acharya as discussed in section 4.1, and the RANS under prediction of the mixing. The reason for the predicted increase in Nusselt number for HOST can be seen in the temperature contour of the first passage, Figure 98.

![Figure 98: CFD Prediction for HOST AR 1:1 Re = 50k Ro = 0 Streamwise Temperature Contour in Passage 1](image)

The cooler inlet air is able to progress further into the turbulated region than what was seen for the OSU case. With the increase in temperature difference between the wall and the bulk fluid, the Nusselt number value increased accordingly. In addition to the expanded reach of the cooler inlet air, the velocity shear is also seen to be higher in the second turbulated panel area than in the first or the third, Figure 99. Both of these items results in the increase in Nusselt number from the first turbulated panel to the second.
As was the circumstance for the 25,000 Reynolds number case, the data shows a small Nu increase from the first to the second turbulated panel while CFD predicts an increase. The first smooth wall panel and the first turbulated panel show reasonable agreement to the data indicating that the turbulator vortex increase and the amount of penetration of the cooler bulk inlet air should be less than what is predicted in the analysis.

4.3 Reynolds Number Effects in Low Aspect Ratio Stationary

The two HOST stationary cases investigated were compared to verify that the measurements scale with the Dittus-Boelter/McAdams relationship, Equation 2, as expected. Figure 100 shows the comparison between the measurements for a Reynolds number of 25,000 and 50,000. It can be seen that there is a definitive additive shift in the data from the lower Reynolds number to the higher Reynolds number.
To determine if this shift is consistent with the Dittus-Boelter/McAdams relationship, the data was normalized as defined by Equation 1 and plotted in Figure 101 to confirm the Reynolds scaling. This shows the data independence of Reynolds number.
The respective CFD predictions are included with the comparison of the HOST data for the two Reynolds number cases in Figure 102. The CFD predictions replicate the shift in Nusselt number as appropriate for the change in the Reynolds number. This shift would be present even if the inlet conditions were not used for the reference temperature and a streamwise bulk temperature were used instead. The CFD results are also reported in terms of enhancement factor, to verify the Re independence, with the results shown in Figure 103.
The CFD results conform to the Reynolds scaling relationship of Dittus-Boelter/McAdams, as was seen with the data. The comparison also shows that while the CFD analysis does not match all points in the first passage on the turbulated walls, the analysis is able to predict the correct shift in the data that would be expected when a change in Reynolds number occurs.

In addition to the differences in the first passage that were discussed in section 4.1 and section 4.2, the comparison shows an increased error in the start of the second passage with increased Re. The cause of the difference is attributed to the fact that with increased Reynolds number the Dean vortices increase in strength. The CFD is over predicting the
heat transfer coefficient due to the Dean vortices. The error decreases as the Dean vortices are washed out by the dominant turbulator vortices within the passage.

Overall, for preliminary analysis and comparison, the current methodology would be able to provide design guidance in terms of qualitative comparison for stationary investigations.

4.4 Comparison of OSU and HOST Low Aspect Ratio Stationary

The OSU experiment was constructed to match the HOST experiment in most geometric aspects, expect those noted in section 2.2, so the HOST and OSU data are generally comparable for stationary aspect ratio of 1:1. As a result, the capabilities of CFD analysis to determine the rotational effects on the HOST configuration would be comparable to
the prediction capabilities for the OSU rotational testing. The OSU rotating experiment is still in design and build up so the CFD analysis benchmark for rotational effects was done with the HOST data. The comparison between the data sets is shown in Figure 104.

![Figure 104: Comparison of OSU AR 1:1 Re = 50k Ro = 0 Data and HOST AR 1:1 Re = 50k Ro = 0 Data](image)

The comparison shows good agreement on the side walls, walls A and B, as well as in the first tip turn through the third passage for the turbulated walls C and D. Smith [8] discussed the differences noted in the first passage for the turbulated walls. The differences were determined to be due to the entrance configurations of the two experiments. The HOST entrance consisted of an approximately two inch heated section with a screen at the inlet to provide uniform flow conditions. This configuration resulted in both hydrodynamic and thermally developing flow. The OSU test section had
approximately six inches of unheated length before the heated turbulated test section began, resulting in a different temperature profile, Figure 105.

![Figure 105: Temperature Contours With Velocity Vectors for a Cut Plane Immediately Before the First Turbulator for AR 1:1 Re = 50k Ro = 0 (A) HOST and (B) OSU](image)

The comparison of the data along with the respective CFD predictions is shown in Figure 106. As would be expected, there is a good agreement between the data sets and the CFD analysis for the smooth side walls, walls A and B. There is also good agreement with the turbulated walls, walls C and D, starting at the tip turn and continuing through the test sections. The discrepancies between the predictions and the data are the same as was addressed in section 4.1 and section 4.2 for the OSU and HOST data comparisons to predictions, respectively.
While for the first passage, the agreement is not as strong as is seen for the remainder of the turbulated walls in the test section, the trends and qualitative changes are similar between both the data sets and the CFD analysis. It is concluded that the data sets are similar enough that the predictive capability of the CFD analysis for the HOST rotating configuration would be equitable to the CFD of the OSU rotating predictions and that the CFD provides reasonable predictions for different inlet heating configurations.
Chapter 5

Low Aspect Ratio Rotating

The HOST database was used to determine the predictive capabilities of the CFD methodology for rotating conditions. The selected cases were \( \text{Re} = 25,000 \) with a \( \text{Ro} = 0.25 \) and \( \text{Re} = 50,000 \) with a \( \text{Ro} = 0.12 \). \( \text{Re} = 25,000 \) had the highest rotation numbers for the temperature differential that matched OSU. \( \text{Re} = 50,000 \) matched the OSU temperature differential and Reynolds number but had the highest rotation number of 0.12.

As was the case in Chapter 4, the focus of this section is the data match. Discussion of flow mechanisms for a rotating AR 1:1 geometry will be done in section 7.3.

5.1 HOST Low Aspect Ratio Rotating

The curvature-corrected SST model was able to predict the \( \text{Re} = 25,000 \) case with the similar accuracy as the stationary case, Figure 107. Again, the model matches the Nusselt numbers on the turbulated walls from the first tip turn through the remainder of the passage.
With the additional complexity of rotation, the agreement of the first passage for turbulated wall D has increased in accuracy with the second turbulated panel decreasing in difference from approximately 30% down to 20%. The first and third turbulated panel predictions are within the experimental uncertainty and the first smooth panel has a difference from measurement to CFD prediction of approximately 12%. Overall, the predictions are better with rotation.

The addition of rotation adds the secondary flow influences of Coriolis and buoyancy forces. The Coriolis forces move the flow from the leading wall to the trailing wall in the first passage and from the trailing wall to the leading wall in the second passage. The buoyancy force opposes the bulk flow in the first passage and with the flow in the second
passage. The CFD is showing increased capability of predictions with the increased large-scale turbulence due to rotation.

The agreement for the first passage for the turbulated wall C is approximately 5-15% beyond experimental uncertainty, which is approximately what was seen for the stationary case for wall C. The second turbulated panel has increased in the difference between data and CFD predictions by approximately 10%.

The likely cause of the increase in discrepancy between the predictions and the data is due to small regions of higher heat transfer predicted on small sections of the upstream and on the leading edge of the turbulators for the second turbulated panel, as seen in Figure 108. The higher Nusselt number in these regions are biasing the area averaged values to the high side resulting in the over prediction of the average Nusselt number for the second panel.

**Figure 108: Wall C Nusselt Number Contour for HOST AR 1:1 Re = 25k Ro = 0.25 Passage 1**
The high Nusselt number is due to the weak formation of the turbulator vortices along the leading wall. The high rotation has increased Coriolis forces pulling the cooler bulk flow away from the leading wall even before the turbulator wall starts causing non-uniformity in the Nusselt number. The continued migration of flow away from the leading wall prevents the full formation of the turbulator vortex structure, Figure 109. In addition, the Coriolis force is causing the wall D vortex flow to continue up along wall A to wall C, Figure 109. The combination of the attempting startup of the turbulator vortex flow and the impingement from the vortex flow up wall A causes local regions of high Nusselt numbers.

![Scaled Temperature Contours](image1.png)

**Figure 109:** Scaled Temperature Contours With Velocity Vectors for the HOST AR 1:1 Re = 25k Ro = 0.25 Passage 1. Left Side is Wall A and Right Side is Wall B.
The expected contour for the first passage leading wall is a more uniform contour with lower Nusselt numbers. The reason is the movement of the colder bulk flow away from the leading wall and weaker turbulator vortices, both equating to lower Nusselt number values. The remaining panel shows the expected contour because the secondary flow has developed, the Coriolis force has moved the bulk flow away, and the local air temperature near the wall has increased resulting in the lower Nusselt number, Figure 108.

The Nusselt number contours for the trailing edge side, wall D, in Figure 110 shows a more consistent pattern from the first turbulated passage through to the third turbulated passage. The Nusselt number structure is similar to the stationary structure with the addition of higher Nusselt number values. The cause is that the Coriolis forces from the rotation increases the Nusselt number on the trailing wall, even at the smooth inlet, and do not hinder the turbulator vortices. The forces increase the turbulator effect by forcing cooler bulk flow towards the trailing wall.
The flow immediately downstream of the split turbulators on the trailing wall shows a different turbulator vortex pattern than the rest of the panel. The split turbulator causes two vortices to form for each part of the split turbulator, Figure 110. The vortices of the split turbulators are not as large as the ones for a full turbulator but they still provide an increased Nusselt number along the vortex path.

The addition of the Coriolis force, Equation 10, and buoyancy force, Equation 11, add large-scale turbulence to the flow field. The Coriolis force adds secondary flows perpendicular to the radial direction while the buoyancy force adds secondary flows in the radial direction with hotter fluids moving radially inward. In the first passage, the Coriolis and the buoyancy forces are additive, and are dominant over the turbulator secondary flow. The CFD analysis is able to better predict the Nusselt numbers for the high rotational case than for the stationary case.
The comparison between measurements and CFD predictions for the HOST Re = 50,000 Ro = 0.12 case is shown in Figure 111. There is good agreement with the side walls, except for the third and fourth side panel for wall B. The first tip turn through to the end of the second tip turn also shows good agreement for the turbulated walls C and D.

![Figure 111: HOST AR 1:1 Re = 50k Ro = 0.12 Data Compared to CFD Predictions](image)

For wall C in the first passage, the stationary case had approximately 15% difference in the predictions and the data, while for the rotating case, the difference increases to about 20%. The outlier is the second turbulated panel where the difference increased 10% over the stationary prediction difference. The addition of rotation, at a rotation number of 0.12, increases the prediction difference minimally except for the second turbulated panel.
The Nusselt number contours for wall C are shown on Figure 112. The contours have changed from the stationary case with the bias towards wall B causing the predicted Nu increase on the side wall B.

![Nusselt Number Contour for HOST AR 1:1 Re = 50k Ro = 0.12](image)

**Figure 112: Nusselt Number Wall C Contour for HOST AR 1:1 Re = 50k Ro = 0.12**

The passage midplane temperature contour, Figure 113, shows the bias of the bulk flow towards the trailing wall, wall D. The bias does not result in an increase in the Nusselt number of wall D from the stationary case, which is consistent with the measurements. The flow bias, however, does not show enough of a decrease in the leading wall Nusselt number values.
Figure 113: CFD Prediction for HOST AR 1:1 \( \text{Re} = 50k \) \( \text{Ro} = 0.12 \) Midplane Scaled Temperature Contour in Passage 1

The leading wall turbulator vortices are pulling the bulk cooler air towards wall C, Figure 114. The trailing wall vortices are not able to reach the leading wall as was seen in Figure 109, especially for the third panel. The result is higher Nusselt numbers than found by the measurements.

Figure 114: Scaled Temperature Contours With Velocity Vectors for HOST AR 1:1 \( \text{Re} = 50k \) \( \text{Ro} = 0.12 \) in Passage 1. Left Side is Wall A and Right Side is Wall B.
The thermal migration shown in Figure 113 also corresponds to a large shear layer, Figure 115. The large shear layer is a result of the Coriolis forces moving the flow towards the trailing wall and the buoyancy force moving against the flow direction. The leading wall region has the highest temperature flow and therefore the highest buoyancy force pushing the flow counter to the flow direction.

![Turbulence Kinetic Energy Contour in Passage 1](image)

**Figure 115: CFD Prediction for HOST AR 1:1 Re = 50k Ro = 0.12 Midplane Turbulent Kinetic Energy Contour in Passage 1**

By looking at the velocity contour of the midplane section of the first passage, Figure 116, the bulk velocity magnitude is similar to the stationary case, as expected for the same Re. The recirculation regions downstream of the wall C turbulators have increased minimally. For increases in rotation, the leading edge wall in the first passage should have an increase in the reattachment length downstream of the turbulators resulting in a decrease in the Nusselt number [7]. From the CFD predictions, it appears that the reattachment length has not increased to the level that would correspond with the data.
For wall D in the first passage, the stationary case had approximately 10-15% difference and the difference remains the same for the rotating case. The outlier of the second turbulated panel also remains the same at approximately 30%. The addition of the moderate level of rotation did not change the prediction capability of the CFD for wall D.

For wall D, the Nusselt number contours magnitude and structure are very similar to the Nusselt number contours shown for the stationary case, Figure 97, but do show a slight increase. The average Nusselt number results are also similar between the two simulations, but with a slight increase due to rotation. For low rotation numbers, therefore, the effect of rotation is primarily seen on the leading wall, where the cooler bulk flow is migrated away from the wall, and to a much lower extent on the trailing wall [3].
5.2 Effect of Rotation for HOST Low Aspect Ratio

In addition to the comparison of the data to the CFD predictions on an individual case basis, the comparison of the effect of the change of rotation number is also investigated.

The comparison of the data for stationary and a rotation number of 0.25 for Re = 25,000 is shown in Figure 118. The smooth walls, walls A and B, show a slight increase with rotation number due to the migration of the turbulator vortices along wall A and the Coriolis forces migrating the flow along wall B in the first passage, Figure 92 and Figure 109.

There is also a difference within the turns for the smooth walls. The addition of rotation causes an increase of Nusselt number with rotation for the first tip turn and a decrease of Nusselt number for the root turn. The strong turbulator vortices against wall A continue into the turn causing the flow to follow the outside wall of the tip turn, A, increasing the heat transfer along the wall due to the Coriolis and buoyancy forces. The decrease for the
root turn is due to the buoyancy forces biasing the cooler bulk fluid towards the inner part of the turn, decreasing the heat transfer on the outer walls.

For walls C and D, the difference is primarily seen within the first passage. The trailing wall, wall D, shows a slight increase in the Nusselt number as the Coriolis force brings the cooler bulk flow towards the trailing side. The leading wall, C, shows more of a differentiation with rotation with a slightly decreasing Nusselt number for the rotational condition. This is due to both the Coriolis force and the buoyancy force resulting in the warmer air recirculating radially inward on the leading edge side.

![Graphs showing Nusselt number vs distance for different walls](image)

**Figure 118: Comparison of HOST AR 1:1 Re = 25k Ro = 0 Data and HOST AR 1:1 Re = 25k Ro = 0.25 Data**

The percent total heat transfer rate for each passage and turn was calculated to illustrate the effect of rotation and the results are tabulated in Table 9. The passages are relatively
insensitive to rotation but the turns have the greater effect due to the Dean vortices, change in Coriolis force direction, and the change of the buoyancy force from opposite the flow direction to same as the flow direction.

### Table 9: Percent Total Heat Transfer Rate Per Passage and Per Turn for HOST AR 1:1 Re = 25k Ro = 0 Data and HOST AR 1:1 Re = 25k Ro = 0.25 Data

<table>
<thead>
<tr>
<th></th>
<th>Passage 1</th>
<th>Tip Turn 1</th>
<th>Passage 2</th>
<th>Root Turn 1</th>
<th>Passage 3</th>
<th>Tip Turn 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Re 25k Ro 0</td>
<td>42%</td>
<td>16%</td>
<td>16%</td>
<td>11%</td>
<td>9%</td>
<td>6%</td>
</tr>
<tr>
<td>Re 25k Ro 0.25</td>
<td>41%</td>
<td>20%</td>
<td>16%</td>
<td>8%</td>
<td>9%</td>
<td>6%</td>
</tr>
</tbody>
</table>

In Figure 119 the CFD predictions are added to the comparison of the change in rotation number. The CFD predictions follow the data for walls A and B as well at the start of the first tip turn for both walls C and D. The difference between the CFD predictions and the data is evident in the first passage, but the qualitative change in the Nusselt number values with respect to rotation is appropriate. The result suggests that CFD predictions can be used to qualitatively determine the changes that would occur in Nusselt number for changes in rotation.
Figure 119: Comparison of HOST AR 1:1 Re = 25k Ro = 0 Data, HOST AR 1:1 Re = 25k Ro = 0.25 Data, and Respective CFD Predictions

The same comparison that was done for the Reynolds number of 25,000 is repeated for the 50,000 Reynolds number case, as shown in Figure 120. The smooth walls show little to no difference between the stationary and rotational case. The same is also true for the turbulated wall D. The difference is evident in the first passage of the leading wall, C.
The CFD predictions are again added to the comparison of the change in rotation number, Figure 121. The CFD predictions for smooth wall A follow the data. For wall B, the CFD predictions vary slightly immediately before the turn as was discussed in section 5.1. For the turbulated walls, walls C and D, the agreement is good for the first tip turn and downstream, as was the case for the 25,000 Reynolds number comparison. Significant variations between model predictions and data only occur within the first passage; however, the qualitative change in the Nusselt number values with respect to rotation is still appropriate. CFD predictions are good downstream of the start of the first turn and are trending correctly in the first passage, and can therefore be used for design purposes.
Figure 121: Comparison of HOST AR 1:1 Re = 50k Ro = 0 Data, HOST AR 1:1 Re = 50k Ro = 0.12 Data, and Respective CFD Predictions
Chapter 6

High Aspect Ratio Stationary

High aspect ratio cases, AR 1:2 and AR 1:6, are evaluated for a stationary condition of Reynolds number equal to 50,000. Details of the flow are described and illustrated and the panel-averaged Nusselt numbers are compared to the experimental measurements. The flow structures of the AR 1:1, AR 1:2, and AR 1:6 aspect ratios are compared to show the changes in the secondary flow structure for the different aspect ratios.

6.1 OSU Aspect Ratio 1:2 and Aspect Ratio 1:6 Stationary

The side walls of the AR 1:2 case show good agreement with the measurements as was the case for the AR 1:1 case. The CFD tends to over-predict the Nusselt numbers, consistent with the AR 1:1 comparison.
For the turbulated walls, predictions for the first three points of the first passage show the greatest discrepancy from data. From the first point to the second there is an increase in the data and then a decrease to the third point. The CFD predicts a constant value between the first two points and then decreases as the third point decreases.

The cause of this trend was documented by Saha and Acharya [55], Fu et al. [37], and Huh et al. [57] who have experimentally studied the entrance effects on the Nusselt number trends in the first pass of a turbulated AR 1:2 and/or AR 1:4 passage geometry. It was concluded that a hydrodynamic fully developed flow entering a turbulated passage would show an increase after the start of the turbulators. The trend in the data is consistent with other reports and therefore determined to be due to the long unheated entry length within the experiment.
The CFD analysis does not show this effect, but instead shows an almost steady level from the first point to the second and then decreases to the turn. This is consistent with the CFD of AR 1:1 in section 4.1. As was stated in section 4.1, the CFD was not sensitive to the range of inlet turbulence perturbations or geometry variations for this test section, detailed in sections 3.5.2 and 3.4.2 respectively. The discrepancy is likely due to the RANS model not capturing the step increase in the mixing rate occurring after the first few turbulators. The localized turbulator wall regions stay the same between the different aspect ratios, but the bulk flow region increases resulting in an increased dependence on the bulk flow mixing with increased aspect ratio. The increased difference between data and prediction for the higher aspect ratio supports the theory of the CFD model under predicting the bulk flow mixing. Even with the increased difference, the general trend is still captured from the second turbulator panel and downstream.

In addition to the possible under prediction of the bulk mixing, the first two panels of the first passage show a discrepancy between the first and second RTD positions on the panels, Figure 123 [8]. Only the second RTD position was controlling the heater for each of these panels so the experimental Nusselt number and the boundary condition for the CFD were both from the second RTD position. However, the measured difference between the RTD in the first position and the RTD in the second position indicates that if these first two panels were split, a difference in required heat flux would be seen between each of the panel readings.
The first panel downstream of the first tip turn also had a documented discrepancy between the first and second RTD positions, with the first RTD driving for wall D and the second RTD driving wall C, Figure 124 [8]. The temperature differences between the two RTDs for each first panel in the second passage for walls C and D are similar indicating that if the RTD locations were consistent between the two panels, the Nusselt number values recorded would likely be the same between the two [8].

The gradients between the two RTDs are also similar; therefore, it is likely that the average panel Nusselt number for the two turbulated walls would be an average value between the driving wall D and driving wall C RTDs. The result would be that wall C
prediction difference from data would slightly increase, with the prediction lower than the data. Wall D prediction compared to data would improve but would still have a slight under prediction.

The smooth wall predictions for aspect ratio 1:6 show greater deviation from data than the AR 1:1 and AR 1:2 comparisons. The predictions have also changed from the AR 1:1 trend of periodically over predicting to the high aspect ratio trend of consistently under predicting the absolute data values. The CFD does, however, continue to follow the trends of the data as was the case for the other aspect ratios.

![CFD predictions vs data for different aspect ratios](image)

**Figure 125: OSU AR 1:6 Re = 50k Ro = 0 Data Compared With CFD Predictions**

There is discrepancy between the CFD predictions and the data in the first passage. The first points on each turbulated wall matches but then the CFD predictions do not predict
the increase in the second point. This is consistent with the 1:2 aspect ratio CFD predictions compared to data and would have the same causes.

As was stated for the AR 1:2 case, the emphasis on the RANS model predicting the bulk mixing correctly increases with increased aspect ratio. The increased miss in the absolute data match of the first passage corresponds with a very limited secondary flow structure within the AR 1:6 bulk flow passage, as will be shown in section 6.3 Figure 132. The 1:1 and 1:2 aspect ratio cases show better matches with measurements for the first passage and also show turbulator secondary flow structure encompassing the full passage of the first pass within the first turbulated panel. Downstream of the turn, the AR 1:6 secondary flow structure spanned the full passage and the comparison of CFD predictions with data improved, Figure 135. These results show the increased dependence on the bulk flow mixing and the increased difference between prediction and data supporting the conclusion of the RANS model under predicting the bulk mixing.

The third and fourth points in the first pass decrease at a similar rate as the data decreases. The CFD is therefore matching the trends of the data even though the predictions deviated from the measured values for the first passage by an average of approximately 23%.

The HOST stationary to rotating comparisons of measurement with prediction supports the improvement in prediction agreement with the increase of the secondary flow. In section 5.1, the data match improved with the introduction of rotation. The addition of rotation increases the secondary flow structures within the bulk flow due to Coriolis and
buoyancy forces. Coriolis and buoyancy add forces in directions not aligned with the bulk flow in the first passage, increasing the mixing levels and making up for the fundamental RANS deficiencies. Therefore, the increase in the secondary flow structure in the bulk flow downstream of the first tip turn improved the comparison, even in stationary cases, and the lack of the secondary flow structure in the first pass contributed to the discrepancy.

After the first tip turn, the temperature of two turbulated walls is controlled by two different RTD positions, as was the case for the 1:2 aspect ratio. As was concluded for the 1:2 aspect ratio, the second passage first two points for the turbulated walls would be similar if they were both driven off the same RTD location, so the wall C data should be slightly higher and the wall D data should be lower. For wall C, this would result in the data following the trend seen in the prediction. If wall D were driven off the second RTD, the first point would be lower than the second point in the second passage, which is unexpected. Based on other 1:2 and 1:4 stationary data, the opposing turbulated walls should show similar values [3, 37, 39]. If the second panel for the second passage wall D value were closer to the wall C value, the CFD predictions would follow the data trend as seen in wall C. Since this data point does not follow expected trends based on the 1:1 and 1:2 aspect ratio data for this experimental set up nor does it follow the literature trends, there is uncertainty in the point and it will be studied in the future test configuration [115].
6.2 Trends for Aspect Ratios 1:1, 1:2, and 1:6 Stationary

The change from aspect ratio 1:1 to aspect ratio 1:2 is shown in Figure 126. The CFD predictions follow the data and also the data trends. The data increase from AR 1:1 to AR 1:2 also has a CFD predicted increase. When the data for the 1:1 and the 1:2 aspect ratios are identical, CFD also predicts identical values between the two aspect ratios. The largest differences occur within the first passage and the first point of the second passage and were addressed in section 6.1. The exact magnitude of the Nusselt number is not always achieved with the CFD compared to data but the capability of predicting the trends is beneficial for preliminary design work.

The CFD predicted change in aspect ratio from 1:2 to 1:6 does not always match the trend in data as was seen for the 1:1 to 1:2 aspect ratio cases. The trend, but not
magnitude, is correctly predicted for the first tip turn and downstream for the change in aspect ratio from 1:2 to 1:6. The slope of the data and the CFD for the second through the fourth panel in the first passage match as well.

For the smooth walls, the first passage shows the 1:6 aspect ratio Nusselt numbers to be lower than the 1:2 aspect ratio values. The capability to correctly predict the trend is reestablished within the first tip turn region. As was stated in section 6.1, the minimal secondary flow within the first passage prevents the heat transfer along the side walls. The turbulator secondary flow enhances the heat transfer and since the secondary flow is still developing, the Nusselt number predictions are lower than the measurements.

While the miss is present within the first passage, the trend is expected to improve under rotating conditions based on the data match for the HOST stationary and rotating cases. In addition to the improvement with rotation, the side walls are also of secondary importance to the turbulated walls. This is because the turbulated walls have a larger temperature gradient having the opposing side exposed to hot gas path temperature. The smooth walls, however, represent blade internal walls and have a smaller gradient due to being exposed to the internal cooling flow.
For the turbulated walls, wall D shows the correct trend from the fourth panel and downstream. For the first three panels of the first passage, the CFD is predicting similar values for both the 1:2 and 1:6 aspect ratio geometries, as also shown in the data. Overall, wall D shows reasonable agreement.

Same as wall D, wall C has the correct trend for downstream of the third panel in the first passage. For the first panel, the CFD predicts similar values for the two aspect ratios and that is also seen with the data. For the second panel, the data shows a slight deviation and the CFD is still predicting a similar value between the two aspect ratios. The third panel has again a CFD prediction of similar values for AR 1:2 and 1:6 while the data shows a separation. The correct trend is recaptured for the fourth panel.
The causes for the incorrect predictions in trends for the first passage of all the aspect ratio 1:6 walls would be the same as those outlined in section 6.1, where the discrepancies between the data and the CFD in the first passage was discussed. Again, based on the HOST comparison, the differences between the data and the CFD predictions are expected to decrease with the incorporation of rotation allowing the rotation predictions to still be sufficiently accurate and useful in preliminary design analysis.

6.3 Aspect Ratio 1:1, 1:2, and 1:6 Stationary Comparisons

The aspect ratios 1:1, 1:2, and 1:6 are compared at each passage and turn individually for the stationary Re = 50,000 case.

6.3.1 First Passage AR 1:1, AR 1:2, and AR 1:6 Stationary

Nusselt number contours for walls C and D for aspect ratios 1:1, 1:2, and 1:6 are compared within Figure 128 and Figure 129, respectively. The aspect ratio 1:1 has a larger area of high Nusselt numbers throughout the first passage. This is due to the turbulator vortices encompassing the passage faster resulting in more cool bulk flow moving towards the walls faster than in the high aspect ratio cases, which will be seen later in Figure 132. The higher Nusselt numbers within the first passage for AR 1:1 results in an overall higher percent total heat transfer rate, Table 10, than for the other aspect ratios.
Figure 128: Wall C Nusselt Number Contours for OSU Re = 50k Ro = 0 (A) AR 1:1, (B) AR 1:2, and (C) AR 1:6. Flow In Upper Left and Flow Out Lower Right.
Figure 129: Wall D Nusselt Number Contours for OSU Re = 50k Ro = 0 (A) AR 1:1, (B) AR 1:2, and (C) AR 1:6. Flow In Upper Left and Flow Out Lower Right.

Table 10: Percent Total Heat Transfer Rate Per Passage and Per Turn for OSU Re = 50k Ro = 0 AR 1:1, AR 1:2, and AR 1:6

<table>
<thead>
<tr>
<th></th>
<th>Passage 1</th>
<th>Tip Turn 1</th>
<th>Passage 2</th>
<th>Root Turn 1</th>
<th>Passage 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>AR 1:1</td>
<td>46%</td>
<td>10%</td>
<td>25%</td>
<td>5%</td>
<td>14%</td>
</tr>
<tr>
<td>AR 1:2</td>
<td>38%</td>
<td>10%</td>
<td>27%</td>
<td>7%</td>
<td>18%</td>
</tr>
<tr>
<td>AR 1:6</td>
<td>31%</td>
<td>11%</td>
<td>28%</td>
<td>8%</td>
<td>22%</td>
</tr>
</tbody>
</table>

The Nusselt number contours also show the path and the influence of the turbulator vortices. The path of high Nusselt numbers between the turbulators corresponds to the
path of the turbulator vortex. To illustrate the flow path three dimensionally, the Q criterion for the first two panels is shown in Figure 130. As mentioned in section 4.1, the Q criterion allows visualization of turbulent flow structures and therefore turbulator and bulk flow vortices can be identified. The cut planes P1P1 and P1P2 from Figure 131 are shown for reference.

![Figure 130: Q Criterion, Colored for Scaled Temperature, for First Two Panels of OSU Re = 50k Ro = 0 (A) AR 1:1, (B) AR 1:2, and (C) AR 1:6](image-url)
The percentage of Q criterion selected is the percent between the minimum and the maximum derivative of swirl. The lower the value, the weaker and the more vortex structures included. The Q criterion was set to 0.1% for aspect ratios 1:6 and 1:2 but increased to 0.25% for aspect ratio 1:1. The reason for the increase was that with the comparable percentage for the 1:1 aspect ratio case the absolute value was significantly larger because the maximum Q criterion value for the AR 1:1 was larger than the other aspect ratios. For equal comparison, the AR 1:1 percentage was increased.

The turbulator vortices can be clearly seen on both turbulated walls in Figure 130. As indicated by the Nusselt number contours of Figure 128 and Figure 129, the angle of the turbulator vortices off the turbulators increases away from the turbulator downstream side as the aspect ratio increases. When the turbulator vortices impinge on the side wall, flow both separates off the turbulator and follows along the edge of the turbulated walls and smooth wall A. For the AR 1:1 case, the previous flow that lifted off becomes a barrier to
the next turbulator flow which attempts to lift off resulting in more flow remaining along the edge and increasing the Nusselt numbers in that area. The AR 1:2 case shows less of the effect due to the additional cross-section area that the separated flow is able to expand into, decreasing the Nusselt number value in the corner. Finally, the AR 1:6 case shows that the majority of the turbulator vortex flow lifts off and joins the previous flow. It is able to continuously grow, resulting in no increased Nusselt number between the turbulated walls and wall A.

To further understand the Nusselt number contours and the resulting averaged Nusselt numbers, the secondary flow vortices across the passage are investigated. The cut planes, where the flow is investigated, are shown in Figure 131. Visually Figure 131 is shown as AR 1:6 but the cut planes are in the same streamwise locations for the other aspect ratios and also utilized the same labels. In addition, all cut planes are shown from a Lagrangian view point so wall A is always on the right, wall B the left, wall C the top, and wall D the bottom. The secondary flow vortices are also of the same magnitude ratio to allow for comparison.

The first passage turbulator vortices are clearly seen in the cut planes in Figure 132. The vortices encompass the full passage within the first panel for the AR 1:1 and by the second panel for AR 1:2. The growth within the first two panels can also be seen in Figure 130. The AR 1:6, however, does not show the turbulator vortices filling the passage even at the end of the fourth panel. The flow incorporating the full passage corresponds to an increased total heat transfer rate in the passage as seen in Table 10. It also results in the side walls between the lower aspect ratios being similar because of the
fully developed turbulator structure activates both side walls. For AR 1:6 the turbulator vortices are biased towards wall A increasing the Nusselt number on wall A more than on wall B.
Figure 132: Scaled Temperature Contours with Secondary Flow Vectors for First Passage for OSU Re = 50k Ro = 0 (A) AR 1:1, (B) AR 1:2, and (C) AR 1:6. Left Side is Wall A and Right Side is Wall B.
The AR 1:1 and AR 1:2 cases not only show the turbulator vortices filling the passage but also an interaction between them, which is more evident for the AR 1:1 case. The lower turbulator vortex structure is imposed upon by the upper vortex. This interaction is due to the initial half-turbulator presence on wall D. The half-turbulator structure causes the initial turbulator vortex structure to start at a limited size and once the opposing side vortex is created, the wall D structure has yet to fully form. The partial formation of the initial vortex structure on wall D, specifically for the 1:1 aspect ratio, can be seen and compared in Figure 130.

The turbulator vortices encompassing the full passage results in a faster increase in the bulk temperature for the AR 1:1 and for AR 1:2. This was also seen in Figure 33 where the vortex interaction location corresponds to an uptick in the bulk temperature. The large aspect ratio, AR 1:6, does not show a drop in the midline temperature from the inlet until the turn. The vortices encompassing the full passage result in large Nusselt numbers initially due to the constant refresh of cooler air towards the walls. However, large AR geometries have larger mass flows at the same Re resulting in the AR 1:1 case Nusselt numbers decaying faster and having a smaller recovery after the turns as seen for the AR 1:6 or even the AR 1:2 case.

As mentioned in section 6.1, the vortices incorporating the full channel for AR 1:1 and AR 1:2 allows for the improved prediction capability of the CFD. It can be seen in Figure 132 that the turbulator vortices do not encompass the full passage and it appears that more segments of the test section would need to be added for the turbulator vortices to interact. However, the vortices are weakening in strength as they are growing, Figure
As a result, even with an increased length, it is likely that the vortices would weaken in strength and start to decay before reaching the opposing vortices. The fact that the turbulator vortices do not reach each other not only affects the turbulated walls but also the side walls since they do not experience the sweeping flow and recirculation observed in the AR 1:1 and AR 1:2 cases.

6.3.2 First Tip Turn AR 1:1, AR 1:2, and AR 1:6 Stationary

While the 1:1 aspect ratio case has increased Nusselt numbers in the first passage on the turbulated walls due to the turbulator vortices between the turbulators and at the wall the turbulator is angled towards, the higher aspect ratios show increased heat transfer within the turn, Figure 128 and Figure 129. This is due to two reasons:
• Effect of the turbulators: As detailed by Elfert et al., if the passage upstream and downstream of the turns were smooth, the Dean vortices would increase the heat transfer within the turns due to the counter-rotating vortices [45]. However, the presence of the turbulators disturbs the vortices. The turbulators upstream of the turn decreases the mass accumulation at the top of the turn and results in a decrease in the separation bubble size in the second passage causing more uniform flow [45]. The angles of the turbulators upstream and downstream can cause turbulator secondary flow opposing the Dean vortices [45], Figure 134. The larger the influence of the turbulator secondary flow, or otherwise the percentage of blockage due to the turbulators, the more destructive the turbulator secondary flow is to the Dean vortices, as can be seen within the tip turn and at the exit. The destruction of the Dean vortices results in a larger pressure drop but a higher increase in the heat transfer [18]. The blockage ratio for the AR 1:1 is the largest, and therefore is the most destructive to the Dean vortices. The blockage ratio for the AR 1:2 is between both aspect ratios and still shows significant changes to the Dean vortices. The AR 1:6 has the lowest blockage ratio and shows the continuation of the Dean vortices at the turn exit. The blockage ratios for the three cases are shown in Table 11.
Figure 134: Scale Temperature Contours With Secondary Flow Vectors for First Tip Turn for OSU $Re = 50k$ $Ro = 0$ (A) AR 1:1, (B) AR 1:2, and (C) AR 1:6. Left Side is Wall A and Right Side is Wall B.
High aspect ratios: The AR 1:1 produces one dominant vortex with two secondary vortices that are remnants of the turbulator vortices, Figure 134. The AR 1:2 case shows a pair of counter-rotating vortices with two smaller vortices within the corners. The AR 1:6 case, with the additional cross-sectional area, shows six larger vortices within the passage. As was detailed for smaller aspect ratios by Su et al., the larger aspect ratios allow for more complete formation of the Dean vortices within the turn increasing the heat transfer rate [70].

### 6.3.3 Second Passage AR 1:1, AR 1:2, and AR 1:6 Stationary

In the second passage, AR 1:6 has the highest total heat transfer rate with AR 1:2 second, Table 10. The lower mass flow and the initial high heat transfer rate of the first passage AR 1:1 case resulted in the higher bulk temperature, Figure 135. The near-wall flow is refreshed with warmer bulk flow resulting in lower Nusselt numbers, Figure 128 and Figure 129.
Figure 135: Scaled Temperature Contours With Secondary Flow Vectors for Second Passage for OSU Re = 50k Ro = 0 (A) AR 1:1, (B) AR 1:2, and (C) AR 1:6. Left Side is Wall A and Right Side is Wall B.
The swirl imparted on the bulk flow in the first tip turn into the second passage for AR 1:6 results in improved agreement between the CFD predictions and the data. The inlet boundary conditions effects, for all cases, are completely washed out by the turn and the CFD is able to predict the Nusselt number values of the data as discussed in section 6.1.

The turbulator vortex structure, separated vortex structure, and Dean vortices remnants within the AR 1:6 second passage increase the bulk temperature as seen in the contours, Figure 135. The AR 1:6 starts to show a decrease in the bulk temperature with the increased vortices but with the most mass flow, the bulk temperature remains at a relatively cool value compared to the other aspect ratios.

6.3.4 Root Turn AR 1:1, AR 1:2, and AR 1:6 Stationary

The root turn is shown for the three aspect ratios in Figure 136. The AR 1:1 and AR 1:2 cases show a different vortex pattern than the first tip turn. On further investigation, it is evident that the structures are similar but with the main vortices not fully formed and a mirror image to the first tip turn vortex structure. The mirror image is expected due to the change in turn from right to left from the Lagrangian view-point.
Figure 136: Scaled Temperature Contours With Secondary Flow Vectors for Root Turn for OSU Re = 50k Ro = 0 (A) AR 1:1, (B) AR 1:2, and (C) AR 1:6. Left Side is Wall A and Right Side is Wall B.
The similarity in the structures to the first tip turn, but not identical, is due to the Dean vortices being unstable structures. The visuals shown are instantaneous images so at another instance in time, the root turn vortex could develop fully into the mirror image of the first tip turn and at the same time the first tip turn vortex structure could diminish to the mirror image of the structures seen in the root turn.

The Dean instability can be perceived in the temperature monitor points located throughout the simulation volume. The percent variation in temperature for bulk flow locations, for each of the aspect ratios, is shown in Table 12. The increased percentages in the turns indicate the instability. Also, as noted in section 3.3.2, the variation increases with aspect ratio, which was the reason the Nusselt numbers and temperature predictions were iteration averaged instead of instantaneous values.

<table>
<thead>
<tr>
<th></th>
<th>AR 1:1</th>
<th>AR 1:2</th>
<th>AR 1:6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Passage 1</td>
<td>0.39</td>
<td>0.42</td>
<td>0.00</td>
</tr>
<tr>
<td>TT1</td>
<td>1.26</td>
<td>1.90</td>
<td>5.09</td>
</tr>
<tr>
<td>Passage 2</td>
<td>1.13</td>
<td>2.47</td>
<td>2.89</td>
</tr>
<tr>
<td>RT1</td>
<td>1.63</td>
<td>2.68</td>
<td>5.12</td>
</tr>
<tr>
<td>Passage 3</td>
<td>1.19</td>
<td>1.53</td>
<td>3.48</td>
</tr>
<tr>
<td>TT2</td>
<td>0.65</td>
<td>1.71</td>
<td>3.84</td>
</tr>
</tbody>
</table>

While the AR 1:1 and AR 1:2 root turn vortex structures are similar to the first tip turn structure, the AR 1:6 root turn vortex structure shows differences. The flow into the first tip turn from the AR 1:1 and AR 1:2 cases were the same as the flow structures into the first root turn so it is expected that the Dean vortex structure would be similar. The flow
structure into the first tip turn for the AR 1:6 case is not the same as the flow structure into the first root turn. There was no bulk secondary flow structure feeding into the first tip turn for the AR 1:6 case. The bulk secondary flow structure from the first tip turn, however, is present at the first root turn.

The additional secondary flow within the bulk flow creates two counter-rotating vortex pairs and two smaller vortices at the turn midpoint for AR 1:6. Wall C has the furthest turbulator into the turn, which is a half turbulator. The flow from the half turbulator maintains the turbulator flow structure into the turn causing the Dean vortex structure towards wall C to be stronger than the structure closer to wall D. At the exit of the root turn, the influence of the start of the turbulators causes the turbulator vortices to push the Dean vortices to the center of the passage causing three counter-rotating vortex pairs in the passage.

6.3.5 Third Passage AR 1:1, AR 1:2, and AR 1:6 Stationary

The third passage results in trends similar to the second passage. AR 1:6 maintains the highest total heat transfer rates of the three aspect ratios with AR 1:2 second, Table 10. Aspect ratio 1:1 maintains the highest bulk temperature, Figure 137, and the lower Nusselt number values in the third passage, Figure 128 and Figure 129.
Figure 137: Scaled Temperature Contours With Secondary Flow Vectors for Third Passage for OSU Re = 50k Ro = 0 (A) AR 1:1, (B) AR 1:2, and (C) AR 1:6. Left Side is Wall A and Right Side is Wall B.
Aspect ratio 1:1 shows the fastest change from the Dean vortex structure to the turbulator secondary flow and it occurs within the first half panel. The AR 1:2 still shows influence of the Dean structure within the first half panel, which is to be expected with the decrease in the blockage ratio from AR 1:1 to AR 1:2, Table 11. The AR 1:6 case has the turbulator vortex structure starting up at the inlet to the passage, Figure 136, which displaces the Dean vortices to the center of the passage where they encompass the full bulk flow region.

Down the passage, the turbulator vortex structure continues to dominate the entire flow passage for AR 1:1 and AR 1:2. As was the case for the first and second passages, the AR 1:1 turbulator flow structure is oval shaped and the AR 1:2 turbulator flow structure is more round due to the higher aspect ratio. The AR 1:1 maintains the highest bulk temperature with the AR 1:2 following behind.

Aspect ratio 1:6 in the third passage shows the secondary flow structure within the bulk flow region of the passage, which continues to decrease the bulk temperature but not to the magnitude of AR 1:1 or AR 1:2. In addition, the decrease in the Dean vortex structure in the first root turn has resulted in bulk secondary flow structures decreasing down the passage, which has not increased the bulk flow temperature as much as was seen in the second passage.

6.3.6 Second Tip Turn AR 1:1, AR 1:2, and AR 1:6 Stationary

The decrease in the bulk flow secondary vortex structure for AR 1:6 results in the last panel of the third passage being more similar in flow structure to the first passage than
the second passage. The result is that the second tip turn vortex structure is very similar to the structure seen in the first tip turn, Figure 139 and Figure 134, respectively. It is likely that the fourth passage would be more similar to the second, if a fourth passage were included.
Figure 138: Scaled Temperature Contours With Secondary Flow Vectors for Second Tip Turn for OSU $Re = 50k$ $Ro = 0$ (A) AR 1:1, (B) AR 1:2, and (C) AR 1:6. Left Side is Wall A and Right Side is Wall B.
The AR 1:2 second tip turn vortex structure is very similar to the first tip turn vortex structure. The slight differences are due to the unstable nature of the Dean vortices causing differences from one iteration to another, Table 12.

The AR 1:1 second tip turn vortex is similar to the first root turn structure only with reversed behavior between walls C and D. The change is due to the location of the half turbulators causing the wall C turbulator vortex structure to be dominant in the second passage going into the first root turn and the wall D turbulator vortex structure to be dominant in the third passage going into the second turn. The single dominant vortex in the turn is replaced by two vortices, which is due to the strong wall D turbulator structure feeding into the turn and not being destroyed by the Dean vortices.

The fourth passage does not have turbulators resulting in all the aspect ratio turn exits not having signs of vortex destruction for the AR 1:1 and AR 1:2 cases or the migration of the vortices to the center of the passage for AR 1:6.
Chapter 7

High Aspect Ratio Rotating

The high aspect ratio cases were studied for a fixed Reynolds number of 50,000 and rotation numbers of 0.1, 0.2, and 0.3 based on actual engine characteristic rotation speed. The trends for varying rotation numbers from 0.1 to 0.3 for AR 1:2 and AR 1:6 are investigated. Average Nusselt number values, wall Nusselt number contours, and secondary flow structures are compared to examine the changes due to rotation. A comparison is then completed for high rotation, rotation number of 0.3, for all three aspect ratios. The effects of Reynolds number, rotation number, and aspect ratio are then summarized.

7.1 Variations in Rotation Number for Aspect Ratio 1:2

Variations in rotation number from 0.1 to 0.3 for an aspect ratio of 1:2 are addressed. The passages and turns are discussed individually in the order that the flow encounters them.

7.1.1 First Passage AR 1:2 Varying Rotation Number

The aspect ratio 1:2 Nusselt number values for varying rotation number are shown in Figure 139. Wall C is the leading wall and in the first passage the Coriolis force pulls the flow away from the leading wall towards the trailing wall, D, with strength increasing...
with Ro, Figure 140. The effect can be seen in the wall C Nusselt number values decreasing with increasing rotation number, Figure 139.

**Figure 139: OSU AR 1:2 Re = 50k CFD Predictions for Stationary and Rotational Numbers of 0.1, 0.2 and 0.3.**
The flow migration from wall C to wall D prevents the turbulator vortex from fully forming, Figure 140, as it occurred in the stationary case, Figure 132, causing higher near-wall temperatures and lower Nusselt numbers, Figure 139. Only the rotation number of 0.1 has enough flow along the leading wall to support the leading wall vortex structure, Figure 141, which is apparent by the angled Nusselt number contour patterns.
The rotation number of 0.2 has a less developed vortex along wall C, which is no longer visible in the Nusselt number pattern of Figure 141, and results in a lower Nusselt number initially. The Nusselt number value drops for the second panel with the destruction of the vortex due to Coriolis forces moving the flow away from the wall. The structure starts to reform in the third and fourth panels resulting in a steady Nusselt number instead of a continued decrease, Figure 140. The Ro = 0.3 starts at a lower Nusselt number value than Ro = 0.2 and shows no vortex structure along wall C initially. The Nusselt number drops for the second panel due to the heating of the near-wall flow due to the lack of a vortex structure. The formation of a vortex structure in the third panel keeps the Nusselt number value steady, but the increased movement of flow from wall C to wall D destroys the vortex formation and the Nusselt number drops again for the last panel.
The trailing wall, wall D, benefits from the secondary flow bias towards the wall. The cooler bulk flow towards wall D increases with increasing rotation number and passage length, Figure 140. The result is an increase in Nusselt number, Figure 139. Unlike the leading wall, the trailing wall turbulator vortex structure is maintained throughout the passage, as seen in Figure 140 and Figure 142.
Wall A shows increased Nusselt numbers for increased rotation down the passage, Figure 139. The turbulators in the first passage are pointed towards wall A and with rotation, the turbulator vortices are separating off and following wall A, as can be seen in Figure 140. The flow is pushed towards wall A due to the Coriolis force migrating the flow from the leading wall, C, to the trailing wall, D, through the middle of the passage. The increase in rotation number results in stronger vortices along wall A, and therefore an increase in Nusselt number.
The travel of the turbulator vortices along wall A are illustrated for each of the rotation numbers in the Q criterion visual, Figure 143. The cut planes shown in Figure 131 are included and the settings are the same as in Figure 130.

Figure 143: Q Criterion, Colored for Scaled Temperature, for OSU AR 1:2 Re = 50k
(A) Ro = 0.1, (B) Ro = 0.2, and (C) Ro = 0.3
Wall B does not see an increase in Nusselt number for the first passage with the increase of rotation number, Figure 139, because the increase of rotation causes the flow from wall C to wall D to be more centered within the passage, Figure 140, and therefore there is less flow movement along wall B.

7.1.2 First Tip Turn AR 1:2 Varying Rotation Number

The flow through the first tip turn experiences several different forces. First is the development of the Dean vortices, which increases the turbulence and promotes mixing. Second is the change of direction of the Coriolis force from the trailing wall to towards the leading wall within the turn. Finally, the buoyancy force changes from opposing the flow direction at the turn entrance to with the flow leaving the turn.

The Coriolis force interacts with the formation of the Dean vortices causing the formation of a main vortex within the center of the turn, Figure 144. The large vortex causes thorough mixing between walls C and D making minimal difference in the Nusselt number for the different rotation numbers.
The Nusselt number for wall A, however, increases with increased rotation number. The strong turbulator vortices against wall A in the first passage continue into the turn causing the flow to follow the outside wall of the turn, wall A, which is visualized in Figure 143, and can be seen at the turn inlet in Figure 144. The buoyancy force promotes the
movement of the cooler bulk flow outwards towards wall A as well. Both cause an increase in the Nusselt number for increased rotation.

The counter-rotating vortices of the Dean vortices at the exit of the turn are altered by the Coriolis force. For Ro = 0.1, counter-rotating vortices are present and are influenced by the change in Coriolis force moving the flow from wall C to wall D. The Ro = 0.2 case also shows the counter-rotating vortices, the bulk movement from wall C to wall D, as well as the formation of the wall D turbulator vortices. The highest rotation number has the strongest movement of flow from wall C to wall D with the wall C vortex stretched due to the migration.

**7.1.3 Second Passage AR 1:2 Varying Rotation Number**

Nusselt numbers for the leading wall are increased relative to the turn value but then proceed to decrease, Figure 139. The addition of rotation increases the second passage wall C values from stationary but not to the magnitude that was lost in the first passage or to amount of the increase in the first passage for wall D. The turbulator vortex structure is maintained throughout the second passage and the increase in rotation number shows a minimal increase in the Nusselt number contours throughout the passage, Figure 141.

While the increasing Ro increases the Coriolis force migration of the cooler bulk flow from wall D towards wall C, the buoyancy force is in the same direction as the flow causing the flow distribution to be more uniform in the passage and the secondary flow to be not as strong, Figure 145. The separated flow off the initial turbulator vortex structure of wall C moves towards wall D with increasing rotation number but the strength of the
return flow does not elongate the vortex structure along the wall as seen in the first passage. The first indication of the elongation and strengthening of the flow towards wall C from wall D is with the highest rotation number of 0.3. The result is that while the cooler bulk flow is moved towards wall C, the bias is not as strong as the first passage movement due to the buoyancy forces and the result is that the Nusselt number has minimal change with rotation number in the second passage.
Figure 145: Scaled Temperature Contours With Secondary Flow Vectors for Second Passage for OSU AR 1:2 \( \text{Re} = 50k \) (A) \( \text{Ro} = 0.1 \), (B) \( \text{Ro} = 0.2 \), and (C) \( \text{Ro} = 0.3 \). Left Side is Wall A and Right Side is Wall B.

With the lack of strong secondary flow from wall D to wall C, wall D does not see the same decline that was seen for wall C in the first passage, but does experience values less than the stationary Nusselt numbers. In Figure 142, it can be seen that while the Nusselt number value is decreasing, the turbulator vortex structure is still present keeping the
wall D Nusselt numbers at a level higher than the values seen for wall C in the first passage, but at lower Nusselt number values than wall C in the second passage.

For the smooth walls, there is not a strong influence due to rotation. The turbulator vortices do not separate and grow off wall C in the second passage, Figure 145, to the magnitude seen in the first passage, Figure 140, so wall B does not see the Nusselt number enhancement in the second passage even though the turbulators are pointed to wall B, Figure 139. Wall A is not expected to have an influence since it is not the wall the turbulator flow is directed towards.

7.1.4 Root Turn AR 1:2 Varying Rotation Number

The aspect ratio 1:2 root turn has the Coriolis forces and the buoyancy forces again switching with the Coriolis forces towards wall D and the buoyancy forces opposing the flow direction. The Dean vortices within the middle of the turn are again disrupted by the Coriolis forces and the result is a single large vortex for $\text{Ro} = 0.1$ and $\text{Ro} = 0.2$, Figure 146. For $\text{Ro} = 0.3$, there is one dominant vortex with another smaller vortex, likely due to the instability of the Dean vortices.
As was the case for the first tip turn, the mixing within the turn results in the leading and the trailing walls having similar Nusselt number values independent of rotation, Figure 139. The Nusselt numbers are lower than the first tip turn due to the warmer bulk flow refreshing the near-wall flow.
Unlike the first tip turn, the outer wall does not see an increase in Nusselt number with rotation. The lack of vortices along wall B does not cause the flow to follow along the outer wall like the first tip turn. In addition, the buoyancy force biases to the cooler flow towards the inner part of the turn, resulting in warmer near-wall flow for wall B.

At the exit of the turn, the Coriolis forces start the movement of flow from wall C to D. The wall D turbulator vortex structure forming moves up along wall A, pushing the Dean vortex to wall C. The buoyancy force is counter to the flow direction enhancing the Coriolis force effect. The higher the rotation number, the more the Dean vortex moves to wall C. For Ro = 0.3, the Dean vortex is already starting to weaken due to the strong movement of flow from the Coriolis and buoyancy forces.

7.1.5 Third Passage AR 1:2 Varying Rotation Number

In the third passage, the wall C Nusselt number magnitude and turbulator vortex pattern size decreases with increasing rotation number, Figure 141, to values similar or below the stationary values, Figure 139. This is consistent with the trend seen for wall D in the second passage. The cause is the increasing strength in the Coriolis force and the buoyancy force opposing the flow direction increasing the flow movement from the leading wall to the trailing wall, Figure 147.
For $\text{Ro} = 0.1$ leading wall, the bias of flow from wall C to wall D is not as strong as in the first passage due to the diminished buoyancy force. The wall C vortex structure is able to grow in size and develop to the flow structure seen in the stationary case. The Nusselt number values for wall C $\text{Ro} = 0.1$ are similar to the stationary case.

For the mid-rotation number, the separated vortex structure from the turbulator vortices of wall D is able to progress further up wall A but is not strong enough to fully reach wall
C. The vortex structure of wall C is compressed but still present, Figure 147 and Figure 141. The combination of the compressed turbulator vortices and the warmer near-wall flow results in Nusselt numbers lower than the values for the stationary case.

For Ro = 0.3, the flow structure is similar to the flow structure of the first passage. The separated vortex flow from wall D moves along wall A but is pushed to an elongated state due to the Coriolis forces moving flow from wall C to wall D. The force of the flow prevents the formation of the wall C vortex structure, Figure 141. The result is the warmest near-wall flow and the lowest Nusselt numbers.

The trailing wall receives the cooler bulk flow to refresh the near-wall region due to the Coriolis and the buoyancy forces, Figure 147. The constant movement of flow along wall D maintains the turbulator vortex structure for each of the rotation numbers, Figure 142. The combination results in Nusselt numbers that are equal to or slightly higher than the stationary values for the higher rotation numbers even with the warm bulk flow refreshing the near-wall flow.

The side walls show no change in Nusselt number values with increased rotation, Figure 139. Even with the turbulators pointed towards wall A, causing the separated vortices off the wall D turbulator vortices to travel up the side wall, it does not result in an increase in Nusselt number with increased rotation. This is because the movement is diminished with the decreased buoyancy force and the bulk flow refreshing the near-wall is warmer.
7.1.6 Second Tip Turn AR 1:2 Varying Rotation Number

In the final tip turn, the change in rotation number changes the vortex structure, Figure 148. The decreased buoyancy force affects both the flow structure into the turn and within the turn. The vortex structure within the turn is similar to the structure seen in the first passage for the next lower rotation number. The Ro = 0.1 case has a vortex structure similar to the stationary case, the Ro = 0.2 is similar to the first tip turn Ro = 0.1 case, and the Ro = 0.3 is similar to the Ro = 0.2 first tip turn structure.
The flow exiting the turn is different than the structure within the first tip turn. This is due to the continued growth of the Dean vortices and the influence of the Coriolis and buoyancy forces without the start of the turbulator vortices.
7.2 Variations in Rotation Number for Aspect Ratio 1:6

Variations in rotation number from 0.1 to 0.3 for an aspect ratio of 1:6 are discussed. As was the case for section 7.1, the passages and turns are discussed individually in the order that the flow encounters them.

7.2.1 First Passage AR 1:6 Varying Rotation Number

The first passage leading wall, C, results in three different Nusselt number trends, Figure 149. For the lower rotation number, the Nusselt number decreases continually down the first passage. This is due to the Coriolis forces pushing the bulk flow from the leading wall to the trailing, Figure 150, as was seen for the 1:2 aspect ratio cases, Figure 140.

![Graphs showing Nusselt number variations](image)

Figure 149: OSU AR 1:6 Re = 50k CFD Predictions for Stationary and Rotational Numbers of 0.1, 0.2 and 0.3

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Figure 150: Scaled Temperature Contours With Secondary Flow Vectors for First Passage for OSU AR 1:6 Re = 50k (A) Ro = 0.1, (B) Ro = 0.2, and (C) Ro = 0.3. Left Side is Wall A and Right Side is Wall B.
The continual movement of the cooler bulk flow away from wall C results in higher local flow temperatures. The vortex structure along the leading wall in the first passage decays until it is nonexistent on the fourth panel, as can be seen in both Figure 151 and Figure 152.

Figure 151: OSU AR 1:6 Re = 50k Wall C Nusselt Number Contours for (A) Ro = 0.1, (B) Ro = 0.2, and (C) Ro = 0.3. Flow In Upper Left and Flow Out Lower Right.
Figure 152: Q Criterion, Colored for Scaled Temperature, for OSU AR 1:6 Re = 50k (A) Ro = 0.1, (B) Ro = 0.2, and (C) Ro = 0.3
For the midlevel rotation number, the Coriolis force again moves the bulk flow away from the leading wall, Figure 150. The turbulator vortex structure is not established until the end of the passage but the structure is upstream of the turbulator instead of downstream. The structure can be seen in Figure 152 and in Figure 151 with the higher Nusselt number values starting near wall B instead of wall A in the middle of the third and the entire fourth panel. The formulation of the vortex structure causes the Nusselt number to increase from the second panel and continue to increase down the passage, Figure 149.

The formulation of the turbulator vortex structure ahead of the turbulator on the leading wall is due to flow reversal, which can be seen in Figure 153, where only the reversed flow is visible. The strong Coriolis force moves the flow away from the leading to the trailing wall creating a low-pressure area. The buoyancy forces moves the warmer flow near wall C radially inward resulting in an increase in the Nusselt numbers, Figure 149. The recirculation of flow for Ro = 0.2 starts at the second panel of the first passage. That is the specific region where the Nusselt numbers for the wall C start to increase, Figure 149.
Figure 153: OSU AR 1:6 Re = 50k Ro = 0.2 Flow Reversal for (A) Passage 1, (B) Passage 2, and (C) Passage 3

For the highest rotation number, the increased rotation moves the start of the recirculation upstream to the first panel, Figure 154. The increased recirculation zone increases the
starting point of the turbulator vortex flow on the upstream side of the turbulator, which can be seen on part of panel two and all of panels three and four in Figure 152 and Figure 151. The flow recirculation and the reformation of the turbulator vortices also decrease the near-wall temperature lower than the other rotation numbers, resulting in the leading wall Nusselt numbers increasing down the first passage.
The presence of flow recirculation for high aspect ratio passages has been proposed previously in the open literature. The recirculation of flow for AR 1:4 case was
hypothesized based on a four-turbulator unit cell URANS simulation performed by Saha and Acharya [74, 76]. The conclusion of the work was that critical conditions consisting of low Reynolds numbers, high rotation numbers, high-density ratios, or a combination thereof could result in stagnation or bulk flow recirculation occurring along the leading wall.

The trailing wall for the first passage has an increase in Nusselt number with increasing rotation number, Figure 149. The increase in rotation number results in a stronger Coriolis force biasing colder bulk flow towards wall D. The flow becomes more biased toward the trailing wall as the flow continues down the passage, Figure 150. The cooler bulk flow with the turbulator vortices along wall D refreshes the near-wall flow increasing the Nusselt number contour values shown in Figure 155. The end result is a continual increase in the Nusselt number with increased rotation number down the first passage, Figure 149.
As was seen for the AR 1:2 case, the turbulators pointed towards wall A in the first passage and with the increase in rotation, the separated vortices from the trailing wall vortices progress further up wall A, Figure 152 and Figure 150. They are also pushed more towards the wall with increased flow migration from the leading to the trailing wall, Figure 150. The increased motion causes an increase in Nusselt number for wall A, both with rotation and with the progression down the first passage, Figure 149.
Wall B does not show variation in Nusselt number with rotation number. This is consistent with the AR 1:2 trends for wall B.

7.2.2 First Tip Turn AR 1:6 Varying Rotation Number

In the first tip turn, the bulk flow mixes the colder fluid near wall D with the warmer fluid of wall C due to the Dean vortices, Figure 156. There is still a bias of colder air toward wall D resulting in similar Nusselt numbers for wall C and wall D but with wall D maintaining the higher Nusselt numbers. The increased rotation number increases the elongation of the Dean vortices in the center of the turn, but there is still a bias of flow towards wall D. The result is that the wall D Nusselt numbers are more affected by rotation through the turn than wall C Nusselt numbers.
Figure 156: Scaled Temperature Contours With Secondary Flow Vectors for First Tip Turn for OSU AR 1:6 Re = 50k (A) Ro = 0.1, (B) Ro = 0.2, and (C) Ro = 0.3. Left Side is Wall A and Right Side is Wall B.
The strong vortex structure along wall A caused by the Coriolis force continues through the turn, Figure 152. The buoyancy force biases the cooler bulk flow towards the end wall. Both result in an increase in the Nusselt number for wall A with the increase of rotation.

At the turn exit, the Dean vortices have elongated from wall C to wall D due to the change in the Coriolis forces. The elongation increases towards wall D with increased rotation. The start of the turbulator vortices for wall C is more prominent with the higher rotation numbers and forces the Dean vortices towards the center of the passage. The wall D vortices ability to move the Dean vortices decreases with increased rotation number.

7.2.3 Second Passage AR 1:6 Varying Rotation Number

Immediately downstream of the first tip turn, both turbulated walls see an increase from the Nusselt number values within the turn, Figure 149. The Coriolis force switches direction from the trailing wall to the leading wall and the buoyancy force changes from counter to the flow direction to with the flow. It is evident from Figure 152 that it takes time for the bulk flow to migrate, and the distance required increases with rotation.

Neither wall sees a significant recirculation due to the buoyancy force since it is with the flow direction and results in a more uniform flow distribution, Figure 153 and Figure 154. The only recirculation evident is either flow recirculating near the turbulators, which is expected, or recirculation near the turns due to the turbulator interaction with either the start or the end of the Dean vortices.
Wall C Nusselt number values have greater decrease in Nusselt number down the passage with increased rotation number, until the last panel. Down the passage, the bulk secondary round vortices start to elongate with increased rotation number indicating the increased migration of flow from wall D, Figure 157. The migration is not as strong nor does it develop as fast as in the first passage due to the buoyancy forces creating more uniform flow in the passage. The result is that the cooler bulk flow does not refresh the near-wall region of wall C until the end of the passage, Figure 157. The Nusselt number decreases faster with increased rotation until the last panel where the Nusselt numbers for the higher rotation numbers recover to stationary values due to the structured flow migration developing.
Figure 157: Scaled Temperature Contours With Secondary Flow Vectors for Second Passage for OSU AR 1:6 Re = 50k (A) Ro = 0.1, (B) Ro = 0.2, and (C) Ro = 0.3. Left Side is Wall A and Right Side is Wall B.
Wall D Nusselt numbers start the passage with higher Nusselt numbers for higher rotation numbers and then the Nusselt number decreases faster down the passage with increased rotation, Figure 149. Immediately after the turn, there are still larger secondary flow vortices towards wall D and also near-wall vortices, Figure 155 and Figure 158, resulting in the initial high Nusselt numbers. As the flow continues down the passage, the increased rotation migrates more cool flow away from wall D to wall C faster and there is a strong decrease in the turbulator vortices, resulting in faster decline in Nusselt numbers.

It is consistent that downstream of all the turns, the opposing wall to the Coriolis force direction after the turn results in high Nusselt numbers immediately downstream of the turn due to the large vortices created by the Dean vortices and the start of the migration of the flow due to the change in Coriolis and buoyancy forces, Figure 157. The speed at which the Coriolis and buoyancy forces affect the flow migration increases with rotation so the rate of decrease of Nusselt number increases with rotation, Figure 152. For the tip turn, this occurred for wall D, as described above. It will be seen that the same effect occurs for wall C after the root turn.

The secondary vortices immediately downstream of the first tip turn not only enhance the turbulated walls, but also the smooth walls, Figure 157. The smooth walls result in increased Nusselt numbers with increased rotation, Figure 149. The Nusselt number values for each rotation number are similar between the walls.
7.2.4 Root Turn AR 1:6 Varying Rotation Number

In the root turn, the flow is again mixed from the leading and trailing wall resulting in similar values between the rotation numbers and walls C and D, Figure 149. The vortex structure within the turn, however, is different for the different rotation numbers, Figure 158. With increased rotation, the Dean vortex structure changes from a circular structure to an elongated structure. The flow is still strongly mixed and the bulk flow temperature is relatively uniform, resulting in similar Nusselt numbers between the leading and trailing walls as well as between rotation numbers, more so than what was seen for the first tip turn.
Figure 158: Scaled Temperature Contours With Secondary Flow Vectors for Root Turn for OSU AR 1:6 Re = 50k (A) Ro = 0.1, (B) Ro = 0.2, and (C) Ro = 0.3. Left Side is Wall A and Right Side is Wall B.
The flow along side wall B for the root turn, Figure 158, is visible to a lesser extent than what is seen for the first turn, Figure 156. This is due to the weaker vortex structure into the turn from the second passage, Figure 157, and the buoyancy force moving the cooler bulk flow towards the center of the turn. The result is a slight increase in Nusselt number with increased rotation, Figure 149.

At the exit of the turn, two counter-rotating vortices are present, Figure 158, increased from the dominant counter-rotating vortices seen in the first tip turn, Figure 150. The vortices still show the elongation due to the Coriolis and buoyancy forces, but the effect on the Dean vortices is less, causing the flow structure to be more similar to the middle of the root turn for the stationary case, Figure 136.

7.2.5 Third Passage AR 1:6 Varying Rotation Number

For wall C in the third passage, there is an increase in the Nusselt numbers with increased rotation due to the turn. This is caused by the slow change in the bulk flow due to the Coriolis force and the weakened buoyancy force. As was the case for wall D in the first tip turn, wall C sees a Nusselt number increase immediately downstream of the turn with higher Nusselt number values with higher rotation numbers, Figure 149. The increased flow by wall C, and dependence on rotation number, can be seen in Figure 152 and Figure 159.
Figure 159: Scaled Temperature Contours With Secondary Flow Vectors for Third Passage for OSU AR 1:6 Re = 50k (A) Ro = 0.1, (B) Ro = 0.2, and (C) Ro = 0.3. Left Side is Wall A and Right Side is Wall B.
The Nusselt numbers for wall C after the turn have a distribution that is similar to that seen for wall D in the second passage. The Nusselt numbers decrease due to the change in the flow bias and also decrease at a faster rate for the higher rotation numbers, Figure 159. This is expected since the strength of the Coriolis and buoyancy forces increase with rotation.

A recirculation region is visible for Ro = 0.3 for wall C in the third passage, Figure 154. The recirculation region is not as large as was observed for the first passage for three reasons. First, the conditions at the inlet to the third passage are different than at the inlet to the first passage. The Dean vortices within the turn produce secondary vortices present in the bulk flow changing the inlet conditions as well as the bulk flow secondary vortices within the passage. Second, the buoyancy force is not as strong in the third passage as it was in the first due to the decrease in the bulk temperature. Third, the Coriolis force has a change in direction at the start of the third passage while it does not in the first passage.

The recirculation region does not affect the Nusselt number in the third passage, Figure 149. There is no observed impact on the Nusselt number because of the smaller size of the recirculation region and the increase in the bulk temperature. With a further increase in rotation, the region would continue to grow and would eventually create an increase in the Nusselt number.

For wall D, the Nusselt number increases with increased distance down the passage at a greater rate with increased rotation numbers, Figure 149. The flow is still biased towards walls C from the second passage due to the Coriolis force and the change in flow
migration is slowed with the decreased buoyancy force. The bulk flow vortices only start to show elongation towards the end of the passage indicating the movement of flow from wall C to wall D, Figure 159. The result is lower Nusselt numbers that increase slightly faster with increased rotation. Not until the final panel does the Nusselt number distribution recover to stationary values, Figure 149.

The Nusselt numbers for the smooth walls are increased with increased rotation number initially downstream of the root turn and then continue to decrease down the passage, Figure 149. The Dean vortices leaving the turn increase the flow along the smooth walls increasing the Nusselt number, Figure 159. The decreased velocity of the bulk flow along with the slow increase of flow from wall C to wall D results in a continual Nusselt number decline that increases with increased rotation.

**7.2.6 Second Tip Turn AR 1:6 Varying Rotation Number**

The flow within the second tip turn, Figure 160, has vortices similar in structure to the first tip turn, Figure 156. Unlike the AR 1:2 second tip turn, the AR 1:6 second tip turn has vortices similar to those seen for the same rotation number within the first turn. The differences seen between the AR trends are likely the result of two factors. First, the inlet conditions to the turns are different. Second, the velocity magnitude of AR 1:2 is decreased further than AR 1:6 due to the decreased strength of the buoyancy force for AR 1:2.
Figure 160: Scaled Temperature Contours With Secondary Flow Vectors for Second Tip Turn for OSU AR 1:6 Re = 50k (A) Ro = 0.1, (B) Ro = 0.2, and (C) Ro = 0.3. Left Side is Wall A and Right Side is Wall B.
7.3 Variations in Aspect Ratio for High Rotation

Variations in aspect ratio from 1:1, 1:2, and 1:6 are discussed for a rotation number of 0.3. The passages and turns are discussed individually in the order that the flow encounters them.

7.3.1 First Passage AR 1:1, AR 1:2, and AR 1:6 for Rotation Number 0.3

The leading turbulated wall, wall C, in the first passage shows the AR 1:1 and AR 1:2 cases with decreasing Nusselt numbers down the passage, Figure 161. The 1:2 aspect ratio has consistently lower Nusselt numbers than the AR 1:1 case because the Coriolis forces are moving the cooler bulk flow from wall C to wall D, Figure 162. With the smaller cross-sectional area of the 1:1 aspect ratio, the bias of the flow is not as large, resulting in higher Nusselt number values.

**Figure 161**: OSU Re = 50k Ro = 0.3 CFD Predictions for AR 1:1, 1:2, and 1:6
Figure 162: Scaled Temperature Contours With Secondary Flow Vectors for OSU Re = 50k Ro = 0.3 First Passage (A) AR 1:1, (B) AR 1:2, and (C) AR 1:6. Left Side is Wall A and Right Side is Wall B.
The first passage leading wall 1:6 aspect ratio case has increasing Nusselt numbers down the passage, which is a different trend than the other aspect ratios. The AR 1:6 Nusselt numbers surpass the 1:2 aspect ratio numbers on the third panel and are equal to the 1:1 aspect ratio Nusselt numbers on the fourth panel, Figure 161. The reason for the change in trend is due to the flow recirculation occurring along wall C as was discussed in section 7.2.1 and shown in Figure 154. The flow recirculation can be seen by the presence of the upstream turbulator vortex Nusselt number pattern in Figure 163.

Figure 163: OSU Re = 50k Ro = 0.3 Wall C Nusselt Number Contours for (A) AR 1:1, (B) AR 1:2, and (C) AR 1:6. Flow In Upper Left and Flow Out Lower Right.
The trailing wall, wall D, shows two trends with the change occurring from the AR 1:1 case to the higher aspect ratios, Figure 161. The AR 1:2 and AR 1:6 cases show similar Nusselt number values but with a slightly faster increase in Nusselt number for the higher aspect ratio. The AR 1:1 case has similar Nusselt numbers to the higher aspect ratios for the first two panels and then decreases in Nusselt number for the last two panels.

The decrease in Nusselt number is due to the small cross-sectional area of the 1:1 aspect ratio and the lower mass flow rate. The increase in Nusselt number for the other two aspect ratios is due to the higher mass flow rates and the bias of the cooler flow towards wall D. As can be seen in Figure 162, the coldest bulk fluid is still present in the 1:6 aspect ratio case at the last panel, minimally present in the 1:2 aspect ratio at the last panel, and nonexistent in the 1:1 aspect ratio by the third panel, which is where the decrease in Nusselt number starts. The trend can also be seen in the Nusselt number contours of wall D, Figure 164. This result indicates that the growth of the Nusselt number in the first passage is finite and dependent on the mass flow rate of the passage.
In the first passage, smooth wall A shows lower values for the higher aspect ratio case, Figure 161. The turbulators are pointed towards wall A in the first passage and the flow separates off the turbulator vortex of wall D and proceeds up along wall A increasing the Nusselt number along that wall. For lower aspect ratios, the vortices encompass all of wall A within the first panel, Figure 162. The AR 1:6 shows strong flow along wall A by the third panel but does not show as much vortex flow as seen for the smaller aspect
ratios, Figure 162. All aspect ratios have the elongation of the vortices along the wall due to the Coriolis and buoyancy forces.

The other smooth wall in the passage does not show much difference among the three aspect ratios, Figure 161. Wall B does not have the vortex flow like wall A and the secondary flow along wall B is from the Coriolis forces pushing flow towards wall D, and not from impinging or creating additional turbulent structures along the wall as was seen for wall A, Figure 162.

7.3.2 First Tip Turn AR 1:1, AR 1:2, and AR 1:6 for Rotation Number 0.3

Within the first tip turn, the Dean vortices mix the bulk flow reducing the Nusselt number difference seen within the first passage between the two turbulated walls, Figure 161. The wall D Nusselt numbers for the lower aspect ratios are more similar to wall C, due to the mixing with a single dominant Dean vortex, but still have a higher value due to the bias of the cooler flow from the Coriolis and buoyancy forces, Figure 165. The AR 1:6 Nusselt number variation between wall C and wall D is more pronounced due to the increased distance of the walls, the still present inlet temperature flow at wall D, and the presence of three dominant Dean vortices. In addition, the AR 1:6 case allows for the largest growth of the Dean vortices increasing the Nusselt number. The result is increased Nusselt numbers with increased aspect ratio, Figure 161.
Figure 165: Scaled Temperature Contours With Secondary Flow Vectors for OSU Re = 50k Ro = 0.3 First Tip Turn (A) AR 1:1, (B) AR 1:2, and (C) AR 1:6. Left Side is Wall A and Right Side is Wall B.
The outer wall of the turn has an increase in Nusselt number for increased aspect ratio, Figure 161. The turbulator separated vortices along wall A continue into the turn increasing the flow along the outer wall. In addition, the Dean vortices form a structure within the turn creating strong secondary flow along the outer wall of the turn and almost impinging on the wall at mid-turn.

At the exit of the first tip turn, effects due to the start of the turbulator vortices vary among the aspect ratios, Figure 165. The strong secondary flow along both smooth walls towards walls C and D dominate the flow for AR 1:6 with only minimal indication of the start of the turbulator vortices. The lower aspect ratio cases have a stronger start to the turbulator vortices.

7.3.3 Second Passage AR 1:1, AR 1:2, and AR 1:6 for Rotation Number 0.3

In the second passage, the influence of the Dean vortices changes with aspect ratio, Figure 161. For the high aspect ratio case, the Dean vortices continue into the passage bulk flow, Figure 166. The high velocity flow towards wall C increases the Nusselt number for the high aspect ratio case to a level that is higher than the other aspect ratio Nusselt numbers.
Figure 166: Scaled Temperature Contours With Secondary Flow Vectors for OSU \( \text{Re} = 50k \text{ Ro} = 0.3 \) Second Passage (A) AR 1:1 (B) AR 1:2, and (C) AR 1:6. Left Side is Wall A and Right Side is Wall B.
Down the second passage, wall C Nusselt numbers decrease, Figure 161. The rate of decrease is the fastest for the high aspect ratio case due to the slower development of flow migration from the trailing to the leading wall, Figure 166, due to the Coriolis force changing direction and the buoyancy force creating a more even flow distribution. The Nusselt number does recover at the last panel with the development of the flow migration.

The lower aspect ratio cases have a lower rate of Nusselt number decline due to the flow migration from wall D to wall C that is present within the first panel, Figure 166. The AR 1:2 case has the lower rate of Nusselt number decline because of the larger amount of cooler bulk flow that is still present within the passage to refresh the near-wall flow temperature.

For wall D, the highest aspect ratio case changes the Nusselt number trend from the trend seen for the lower aspect ratios, Figure 161. Downstream of the turn, the increased turbulence of the Dean vortices, the change in the Coriolis force direction, and the change of buoyancy force from counter to with the flow, increases the Nusselt number for the AR 1:6 case, Figure 161. The proximity of the walls for the lower aspect ratios prevents the continuation of the Dean vortices and allows the forces to impact the opposing wall faster than what is seen for the higher aspect ratio case, Figure 166. As the flow continues down the passage, the high aspect ratio case Nusselt numbers decreases at a faster rate due to the further migration for the cooler bulk flow than the lower aspect ratios, Figure 166, until all Nusselt numbers are similar at the end of the passage, Figure 161.
For the second passage the lower aspect ratios do not show much difference in Nusselt numbers. Both aspect ratios show developed secondary flow structures from wall D turbulators, Figure 164, which keeps the Nusselt numbers relatively steady except for the slight decrease due to the increase in the bulk temperatures, Figure 166.

The smooth walls do not see an affect due to the aspect ratio until AR 1:6, Figure 161. The warmer bulk flow moving along the side walls and the decrease in the vortex velocity causes AR 1:1 and AR 1:2 to have similar Nusselt numbers for both smooth walls and a slight decline due to the continual increase in bulk temperature. In the case of the AR 1:6 configuration the Dean vortices continue into the second passage resulting in higher Nusselt numbers. As the side wall velocities decrease and the flow migration reduces the turbulent vortices in the bulk flow, the Nusselt numbers decrease until at the end of the passage when all aspect ratios have similar Nusselt numbers, Figure 161.

**7.3.4 Root Turn AR 1:1, AR 1:2, and AR 1:6 for Rotation Number 0.3**

In the root turn, the bulk flow is again mixed by the Dean vortices resulting in similar Nusselt numbers for both turbulated walls, Figure 161. For wall C, there are more differences in the results among the various aspect ratios. This is due to the increased amount of cooler bulk flow in the higher aspect ratios and the Coriolis force keeping the cooler flow further separated between the leading and trailing walls within the passage, Figure 167. Wall D Nusselt numbers are all similar and close in value to the low aspect ratio Nusselt numbers for wall C due to the warmer bulk flow near the wall.
Figure 167: Scaled Temperature Contours With Secondary Flow Vectors for OSU Re = 50k Ro = 0.3 Root Turn (A) AR 1:1, (B) AR 1:2, and (C) AR 1:6. Left Side is Wall A and Right Side is Wall B.
For the smooth walls, the increased aspect ratio allows for more developed Dean vortices. The vortices increase the flow velocity along the smooth walls resulting in increased Nusselt numbers. The buoyancy force is towards the center of the turn but is weaker due to the warmer bulk flow, Figure 167. Wall B still sees an increase in Nusselt number with increased aspect ratio, but not to the extent seen in the first passage for wall A.

At the end of the turn, the influence of the turbulator vortices is seen for the lower aspect ratio cases, Figure 167. AR 1:1 shows almost full formation of the turbulator vortices at the exit of the turn. The AR 1:2 case shows the formation of the turbulator vortices moving the Dean vortices to the middle of the passage. The AR 1:6 case shows only small turbulator vortices starting to form in the corners of the passage.

7.3.5 Third Passage AR 1:1, AR 1:2, and AR 1:6 for Rotation Number 0.3
Downstream of the root turn, the change in Coriolis force causes changes in the Nusselt number trends for wall C, Figure 161. The 1:1 aspect ratio shows the migration of bulk flow from wall C to wall D immediately for the first panel, Figure 168. The AR 1:2 case shows cooler flow and turbulator vortices immediately downstream of the turn resulting in a higher Nusselt number than AR 1:1. As flow continues down the passage, the elongation of the vortices along wall A, the decreased turbulator vortices along wall C, and the warmer near-wall flow causes the Nusselt number to decay down to the values seen for the AR 1:1 case, Figure 161. For the 1:6 aspect ratio, immediately downstream of the turn, the Dean vortices are still strong, the Coriolis force is slowly changing direction, and the buoyancy force is decreased due to higher bulk temperature causing high velocity turbulator vortices along wall C resulting in the highest Nusselt numbers of
all three aspect ratios. Down the passage, the Coriolis and buoyancy forces do strengthen the flow migration, resulting in a more rapid decrease of Nusselt number down the passage to the point where all three aspect ratios end at the same Nusselt number. This is consistent with the trend seen for wall D in the second passage.
Figure 168: Scaled Temperature Contours With Secondary Flow Vectors for OSU Re = 50k Ro = 0.3 Third Passage (A) AR 1:1, (B) AR 1:2, and (C) AR 1:6. Left Side is Wall A and Right Side is Wall B.
Wall D in the third passage has similar Nusselt number values for higher aspect ratio cases and lower Nusselt numbers for the AR 1:1 case, Figure 161. For the two higher aspect ratios, the overall bulk temperature is lower than the AR 1:1 case resulting in those cases having a higher Nusselt number, Figure 168. For the highest aspect ratio, the Nusselt number increases at the final panel due to the strengthening migration of the flow from wall C to wall D and the cooler bulk temperature that is still present in the passage which cools the near-wall flow.

Both smooth walls show higher Nusselt number values for the higher aspect ratio cases. This is due to the cooler bulk flow in the higher aspect ratios and the migration of the flow from wall C to wall D to mix the bulk flow with the near-wall flow. The Nusselt number decreases down the passage due to the change in bulk temperature.

7.3.6 Second Tip Turn AR 1:1, AR 1:2, and AR 1:6 for Rotation Number 0.3

As was discussed in section 7.1.6 and section 7.2.6, the second tip turn vortices for AR 1:2 are similar to the next lower rotation number in the first tip turn and the AR 1:6 second tip turn vortices are similar to the same rotation number within the first tip turn with a decrease in the velocity magnitude, Figure 169. The AR 1:1 vortices are most similar to the root turn vortices seen for the stationary case. The result is that the third tip turn vortex is a degradation of the earlier turn flow structures and the degradation decreased with increased aspect ratio.
Figure 169: Scaled Temperature Contours With Secondary Flow Vectors for OSU Re = 50k Ro = 0.3 Second Tip Turn (A) AR 1:1, (B) AR 1:2, and (C) AR 1:6. Left Side is Wall A and Right Side is Wall B.
7.4 Reynolds Number, Aspect Ratio, and Rotation Number Scaling

The CFD predictions obtained as part of this investigation have been interrogated in detail in terms of general effects of Reynolds number, aspect ratio, and rotation number changes on the resulting serpentine passage Nusselt number distributions. The Dittus-Boelter/McAdams correlation is used to determine if it is still applicable for the range of conditions, $Re = 25,000$ to $Re = 75,000$, investigated. The aspect ratio and rotation number effects are then compared to determine the stronger effect.

7.4.1 Reynolds Number Scaling

Figure 101 showed that the Nusselt number dependence on Reynolds number of Dittus-Boelter/McAdams, Equation 2, could accurately reduce the dependence on Reynolds number for low aspect ratio stationary data. To determine if this correlation is valid for the high aspect ratio rotation condition, the 1:6 aspect ratio rotation number of 0.3 CFD solution was obtained for Reynolds numbers of 25,000 and 75,000 to compare to the results of the 50,000 Reynolds number case. The results were then normalized by the Dittus-Boelter/McAdams relationship to determine if the Reynolds number scaling still held.

The result of the comparison is shown in Figure 170. The Reynolds number relationship shows good results. There are only slight differences in the enhancement factor for the turbulated walls. It is thus concluded that the Dittus-Boelter/McAdams relationship is valid for aspect ratios up to 1:6, rotation numbers less than or equal to 0.3, and Reynolds numbers from 25,000 to 75,000.
The Reynolds number correlation is valid for the cases investigated so the comparison for aspect ratios and rotation numbers are done with enhancement values, Equation 1.

### 7.4.2 Aspect Ratio and Rotation Number Scaling

For an overall view of the test section, the overall test section enhancement values, Equation 20, were determined for each aspect ratio and rotation number. The variation for a given aspect ratio for different rotation numbers showed a 9% maximum variation. The variation for a given rotation number and a different aspect ratio showed a 43% variation. The conclusion is that the aspect ratio has more of an effect on the heat transfer enhancement than the rotation number for a full test section comparison. The results are shown in Table 13.
Equation 20: Overall Test Section Enhancement

\[ \text{Overall Test Section Enhancement} = \text{average} \left( \frac{Nv}{Nu_{DB}} \right) \]

Table 13: OSU Overall Test Section Enhancement Values for Varying Rotation Number and Aspect Ratio

<table>
<thead>
<tr>
<th>Overall</th>
<th>0</th>
<th>0.1</th>
<th>0.2</th>
<th>0.3</th>
</tr>
</thead>
<tbody>
<tr>
<td>AR 1:1</td>
<td>1.11</td>
<td>1.07</td>
<td>1.06</td>
<td>1.06</td>
</tr>
<tr>
<td>AR 1:2</td>
<td>1.34</td>
<td>1.34</td>
<td>1.29</td>
<td>1.27</td>
</tr>
<tr>
<td>AR 1:6</td>
<td>1.54</td>
<td>1.49</td>
<td>1.54</td>
<td>1.63</td>
</tr>
</tbody>
</table>

The aspect ratio does not change throughout the test section but the Coriolis and buoyancy forces do affect the walls differently in different locations. The enhancement values for each passage and also for each wall of each passage were calculated for all aspect ratios and rotation numbers. The change in aspect ratio resulted in a greater change in enhancement than the change in rotation number for all passages and turns except for the first passage. The enhancement values for all passages and turns are included in Appendix B.
Table 14: OSU First Passage, TT1, Second Passage, RT1, and Third Passage Enhancement Values for Varying Rotation Number and Aspect Ratio

<table>
<thead>
<tr>
<th>First Passage</th>
<th>0</th>
<th>0.1</th>
<th>0.2</th>
<th>0.3</th>
</tr>
</thead>
<tbody>
<tr>
<td>AR 1:1 Leading</td>
<td>2.37</td>
<td>1.94</td>
<td>1.68</td>
<td>1.39</td>
</tr>
<tr>
<td>AR 1:2 Leading</td>
<td>2.43</td>
<td>1.79</td>
<td>1.36</td>
<td>0.77</td>
</tr>
<tr>
<td>AR 1:6 Leading</td>
<td>2.52</td>
<td>1.25</td>
<td>0.73</td>
<td>0.95</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>First Passage</th>
<th>0</th>
<th>0.1</th>
<th>0.2</th>
<th>0.3</th>
</tr>
</thead>
<tbody>
<tr>
<td>AR 1:1 Trailing</td>
<td>2.28</td>
<td>2.28</td>
<td>2.34</td>
<td>2.57</td>
</tr>
<tr>
<td>AR 1:2 Trailing</td>
<td>2.38</td>
<td>2.81</td>
<td>2.88</td>
<td>3.17</td>
</tr>
<tr>
<td>AR 1:6 Trailing</td>
<td>2.38</td>
<td>2.62</td>
<td>2.95</td>
<td>3.45</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>First Passage</th>
<th>0</th>
<th>0.1</th>
<th>0.2</th>
<th>0.3</th>
</tr>
</thead>
<tbody>
<tr>
<td>AR 1:1 Sidewalls</td>
<td>1.49</td>
<td>1.41</td>
<td>1.39</td>
<td>1.40</td>
</tr>
<tr>
<td>AR 1:2 Sidewalls</td>
<td>1.50</td>
<td>1.42</td>
<td>1.50</td>
<td>1.56</td>
</tr>
<tr>
<td>AR 1:6 Sidewalls</td>
<td>1.22</td>
<td>1.19</td>
<td>1.25</td>
<td>1.33</td>
</tr>
</tbody>
</table>

The first passage does not show that the aspect ratio consistently causes the largest change in enhancement. The first passage is also the reason for the change in the monotonic trend of decreasing enhancement with increasing rotation number for the overall test section. Since the first passage has the deviation from the overall enhancement results, the calculated first passage wall enhancement results are shown in Table 14.

For the first passage leading walls, the change in aspect ratio has a maximum 76% variation in the enhancement value. The change in rotation number, however, has a maximum 130% variation. In the first passage, the Coriolis and buoyancy forces are moving the flow away from the leading wall with increased strength with increased aspect ratio and rotation number. The buoyancy effects are also the strongest in the first passage.
The movement of the bulk flow away from the leading wall with increased Ro is more of a factor for the larger aspect ratios because the cooler bulk flow is farther away from the leading wall in the higher aspect ratio passages resulting in a greater decrease in enhancement due to rotation than for aspect ratio. This movement can be seen in Figure 162.

There is a monotonic decreasing trend for the leading wall for the aspect ratios in the first passage, except for the 1:6 aspect ratio case. The change from decrease to increase from a rotation number of 0.2 to 0.3 is due to the recirculation zone that is created due to the strong movement of flow from the leading wall to the trailing wall, Figure 153 and Figure 154.

The first passage trailing wall has a monotonic increasing trend due to the continuous bias of cooler bulk flow towards the trailing wall in the first passage, Figure 162. The continual cooler flow movement towards the trailing wall results in the continual increase of enhancement values that are higher with increased aspect ratio.

The trailing wall of the first passage has similar variation due to aspect ratio and rotation, but still a bias towards a larger effect due to rotation number. The maximum variation for aspect ratio is 29% and the maximum variation for rotation number is 37%.

The sidewalls of the first passage, unlike the leading and trailing walls, have a greater effect from aspect ratio than from rotation number. The maximum variation for aspect ratio is 20% while the maximum variation of rotation number is 11%.
The higher aspect ratio sidewalls initially show a decrease from stationary to a low rotation number and then proceed to increase. The Coriolis and buoyancy forces initial disrupt the turbulator vortices, which were the main source of wall enhancement for the stationary condition. The increase in rotation number increases the movement of flow from leading to trailing wall and also increases the strength of the turbulator vortices separating off the trailing wall and traveling up wall A so with the increase in rotation number the enhancement recovers and increases with increased rotation, Figure 132 and Figure 162.

The trend is not the same for the 1:1 aspect ratio case. This is due to the small aspect ratio, strong turbulator vortices, and minimal migration of the cooler bulk flow from leading to trailing wall resulting in minor changes in enhancement with increases in Coriolis forces and buoyancy, Figure 132 and Figure 162. As a result, after an initial decrease in enhancement with the start of rotation, the enhancement remains at a similar value for all increases in rotation number.

The enhancement value has greater variation due to aspect ratio than rotation number for all walls except in the first passage. In the first passage, the leading wall has more variation due to rotation number and the trailing wall has similar variation due to rotation number and aspect ratio. The overall enhancement values for the given cases are able to predict the stronger enhancement due to aspect ratio than for rotation number, which is accurate for most of the test section.
Chapter 8

Conclusions

This body of work determined the mesh, pre-processing, post processing, and turbulence modeling requirements for a turbulated serpentine geometry, specifically a three-passage geometry. The turbulence models were systematically evaluated and included steady two equation, Reynolds Stress, and SST URANS. The models were compared on prediction accuracy as well as on a computational basis.

The methodology determined was used to benchmark modeling accuracy to low aspect ratio stationary data, OSU AR 1:1 and HOST AR 1:1; low aspect ratio rotating, HOST AR 1:1; and high aspect ratio stationary, OSU AR 1:2 and AR 1:6. The determined accuracy allowed for the extrapolation of the OSU AR 1:2 and OSU AR 1:6 geometry to obtain predictions for realistic rotation numbers. These predictions provide insight on the physical phenomena occurring for a serpentine passage operating at an increased aspect ratio with varying rotation number beyond what has been available in the open literature either experimentally or computationally.

The mesh studies had the following results. The selected independent mesh had eight nodes on a turbulator side for a mesh spacing of 0.00625 inches, a first cell height of 3e-5 inches, a growth ratio of 1.25, and a maximum cell size of 0.025 inches. Single precision
was sufficient for the minimum mesh size and maximum test section dimension. Comparable tetrahedral and hexahedral meshes resulted in similar Nusselt numbers so either mesh methodology could be used.

The studies on the geometry provided information on turbulator and inlet geometry. The turbulator shape, rounded or square, was determined to not be a primary driver of wall heat transfer, or Nusselt number, for the serpentine passage predictions. The inclusion of the full inlet geometry, upstream of the entrance length, did not affect the predicted heat-transfer results either.

Test sections with turbulators machined into copper panels compared to test sections with turbulators attached by adhesive showed similar Nusselt number results. From infrared camera data, it was determined that the adhesive attached turbulators were isothermal with the copper panels. For both test sections, the turbulators were given isothermal boundary conditions equal to the panel temperature.

Post-processing techniques were determined for the calculation of the Nusselt number. The available experimental results determined Nusselt numbers on a project area, or smooth wall basis, so the CFD analysis had an area correction for equal comparison. A fixed inlet bulk temperature was used to eliminate the uncertainty in the bulk temperature predictions of the experiments and to provide a consistent basis between measurements and CFD predictions. In addition, the averaged heat-flux values were used to determine the Nusselt number instead of instantaneous values from the analysis.
Seven different turbulence models were compared; six steady and one unsteady. The SST model with Kato-Launier production limiter and curvature correction showed equivalent results to the more complex models and had CPU time requirement of 5.5-6 times less.

For the rotating test section predictions, temperature rise due to pumping proved to be an important factor, which had not been previously addressed in the open literature. Experimental configurations currently reported in the open literature showed shorter test-section lengths, smaller axis of rotation distances, higher pressures, maximum 500 rpm rotational speeds (low rotation numbers), or combination of these factors resulting in insignificant temperature rise due to pumping. In actual engine conditions, temperature rise due to pumping is significant and always taken into account. Standard non-dimensional parameters used to define rotating serpentine experiments do not account for the temperature rise effect. As a result, the rotating predictions were done with a fixed ratio of temperature rise due to pumping relative to the difference between the inlet flow temperature and the average wall temperature. If the temperature rise due to pumping became too large, the direction of heating from wall to fluid would reverse and the experiment would not be able to provide useful heat-flux information.

The low aspect ratio stationary baseline cases showed differences between CFD predictions and measurements to be less than 15% beyond experimental uncertainty except for two points within the first passage. The increased uncertainty of these two points did not prevent the correct prediction of the Nusselt number trends. The accuracy and the ability to predict the trends allow the CFD analysis to meet preliminary design requirements. In addition, the Dittus-Boelter/McAdams relationship was verified for both
the data set and the corresponding CFD predictions for the range of $Re = 25,000$ to $Re = 50,000$.

The two low aspect ratio stationary baseline cases, HOST and OSU, were compared and showed good agreement, with the differences mainly due to the differences of the test section entrances. The comparison was also mimicked within the CFD predictions.

The small aspect ratio rotating experiments, HOST, showed similar accuracy to the stationary predictions with improvement seen for the prediction of the two points within the first passage. It is determined that the accuracy improves with rotation so the high aspect ratio stationary data, OSU, comparison to CFD is proposed to improve with rotation. Again, even with the discrepancy in the two points within the first passage, the trends are still captured by the CFD and it is verified to be acceptable for preliminary design analysis.

The AR 1:2 and AR 1:6 predictions for the stationary configuration were compared with experimental results, OSU. The AR 1:2 had similar prediction accuracy as the AR 1:1 and the AR 1:6 had additional discrepancy, but both still captured the trends sufficiently for preliminary design. The discrepancies were determined to be due to fully developed inlet conditions, differences between driving and secondary RTDs, and the lack of predicted bulk flow turbulence specific to the AR 1:6 case.

For the AR 1:2 rotating case predictions, the greatest change in Nusselt number occurred within the first passage for the turbulated walls. The leading wall saw lower initial Nusselt numbers and a greater decrease with the increased rotation number. The cause
was due to the destruction of the leading wall turbulator vortex structure and the increased warm air present due to the Coriolis and buoyancy forces. The trailing wall had an increase in Nusselt number with increased rotation due to the cooler bulk flow biased towards the wall due to Coriolis and buoyancy forces.

The second and third passages of AR 1:2 show less of an effect of rotation. This was due to the Coriolis force change in direction and the reduction in buoyancy force. The buoyancy force is with the flow direction in the second passage and results in a more uniform flow in the passage. In the third passage, the buoyancy force is counter to the flow but decreased due to the warmer bulk flow.

Within the turns, the outer wall sees increases in Nusselt numbers with increased rotation number for AR 1:2. The increased rotation moves the flow towards the outer wall due to the turbulator vortex structure strengthening along the wall and the change in the Coriolis forces within the turn.

For the AR 1:6 rotating case predictions, the Nusselt number trends differ from the AR 1:1 and the AR 1:2 cases. The leading wall sees three different Nusselt number patterns within the first passage. For a 0.1 rotation number, the Nusselt number continuously decreases within the first passage due to the Coriolis and buoyancy forces moving the cooler bulk flow towards the trailing wall. For a 0.2 rotation number, the Nusselt number starts to decrease and then proceeds to increase for the remainder of the passage. For a 0.3 rotation number, the Nusselt number increases for the entire passage. For 0.2 and 0.3 rotation numbers, flow recirculation occurs along the leading wall increasing the Nusselt
number down the passage. The turbulator vortex structure forms upstream of the turbulator as a result. The Nusselt numbers are still low, however, due to the warmer air present on the leading wall side.

For the trailing wall in the first passage for AR 1:6, the predicted Nusselt number distribution is more similar to the trailing wall in AR 1:2. The Nusselt number increases throughout the passage. The absolute values also increase with the increase in rotation number.

Within the second and third passage of the AR 1:6 cases, the Coriolis force changes from trailing to leading wall and then back to trailing wall, respectively. The shift in Coriolis forces results in high Nusselt numbers, higher based on increase rotation number, immediately downstream of the turn for the wall that is opposite the Coriolis force direction. The Nusselt numbers then decrease down the passage with a greater decline for the higher rotation numbers.

The AR 1:6 smooth side walls see an increase in heat transfer for the wall in which the turbulators are pointed towards, wall A for the first and the third passages and wall B for the second passage. The Coriolis forces increase the movement of the separated turbulator vortex flow along the side wall increasing the Nusselt number with increased rotation.

The bias of the flow towards the outer walls within the turn also continues for the AR 1:6 predictions as was seen for the AR 1:2 rotating cases. The Nusselt number increases along the outer wall with the increase in rotation number.
When comparing the aspect ratios for the highest rotation number, the side walls show higher values for higher rotation numbers except for the first passage where the turbulator vortex separations do not encompass the full smooth wall. For the AR 1:6 case, the smooth walls of the second and third passages have the highest Nusselt numbers due to the bulk flow turbulence from the Dean vortices within the turn.

For the first passage turbulated walls, the trailing wall has higher Nusselt number values for higher rotation and higher aspect ratios. The leading wall has a decreasing trend with increasing aspect ratio except for the AR 1:6, which has an increasing trend due to the recirculation.

Within the turns, second passage, and third passage, the turbulated walls have similar trends. Higher aspect ratios consistently result in higher Nusselt numbers.

For overall trends, the Reynolds number dependence of Dittus-Boelter/McAdams, for $Re = 25,000$ to $Re = 75,000$, still holds for the cases investigated. It was verified for low aspect ratio stationary and rotating as well as high aspect ratio rotating.

The overall enhancement values for the test section shows that the aspect ratio has a greater influence on the Nusselt number than the rotation number, and this holds true for all individual walls per passage and turn except for the first passage. In the first passage, the leading wall has a greater dependence on rotation while the trailing wall has a similar level of effect for both rotation and aspect ratio. The side walls follow the overall trend with aspect ratio as a greater factor over rotation number.
A great deal of work has been systematically done to understand the change in the distribution of the local heat transfer within a serpentine passage in transitioning to high aspect ratios and high rotation numbers, beyond what has been done previously. A significant effort was made during the comparison of the various CFD results with the primary data sets to explain everything possible using results available within the open literature that might not have been understood within the comparisons investigated. Future work to improve the state-of-the-art would include completing a series of high aspect ratio rotating experiments that include all rotation numbers within this body of work. The conditions should also include various effect of pumping to better understand the trends and compare to engine conditions.

In addition, infrared visuals and flow visualization would be very beneficial to help with the verification of the CFD results. The infrared visuals of leading and trailing walls and the flow visualization for the high aspect ratio geometry while rotating would be able to verify the aspect ratio and rotational effects on the turbulator vortices as shown in the contour plots and the cut planes within this work. Currently, the visualizations are only possible as separate experiments. The combined use is currently beyond experimental technology, but would allow for the full understanding of the heat transfer and fluid flow interactions occurring within the serpentine passages.

For the CFD analysis, increased computational ability would allow for time-accurate solutions and allow the vortices to be tracked throughout the serpentine passages. Along with the additional resources required, measurements of several additional boundary conditions would be needed. It would be very important for the CFD and experimental
personnel to work closely in planning these experiments well in advance of the start of the measurement program in order to significantly improve the overall successful combination of the CFD and experimental efforts.
References


8. M.A. Smith, “Heat Transfer for High Aspect Ratio Rectangular Channels in a Stationary and Rotating Serpentine Passage with Turbulated and Smooth Surfaces,” Ph.D. Dissertation, Dept. Mechanical Engineering, The Ohio State University, Columbus, Ohio, 2012. All figures used from M. Smith’s dissertation were reprinted with permission from M. Smith.


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70. G. Su, H.-C. Chen, J.C. Han, and J.D. Heidmann, “Computation of Flow and Heat Transfer in Two-Pass Rotating Rectangular Channels (AR = 1:1, AR = 1:2, AR = 1:4)
With 45-Deg Angled Ribs by a Reynolds Stress Turbulence Model,” *Proc. of ASME Turbo Expo 2004: Power for Land, Sea, and Air*, Vienna, Austria, 2004, GT2004-53662. Figure 8 and Figure 9 in this dissertation were reprinted with permission from ASME.


Appendix A

Chapter 3 Model Setup Run Summary
Table 15: Chapter 3 Model Setup Summary

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<th>Name</th>
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Note: The table contains a summary of different model setups for various sections and parameters, including mesh and software details.
Chapter 7 Enhancement Averages
Table 16: OSU Overall Test Section Average Enhancement Values for Overall Test Section, First Passage, TT1, Second Passage, RT1, and Third Passage for Varying Rotation Number and Aspect Ratio

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| RT1            | Overall |     |     |     |
|                | 0       | 0.1 | 0.2 | 0.3 |
| AR 1:1         | 0.49   | 0.59 | 0.63 | 0.59 |
| AR 1:2         | 0.93   | 0.96 | 1.01 | 0.97 |
| AR 1:6         | 1.24   | 1.19 | 1.29 | 1.17 |

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Table 17: OSU Leading Wall Average Enhancement Values for Overall Test Section, First Passage, TT1, Second Passage, RT1, and Third Passage for Varying Rotation Number and Aspect Ratio

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Table 18: OSU Trailing Wall Average Enhancement Values for Overall Test Section, First Passage, TT1, Second Passage, RT1, and Third Passage for Varying Rotation Number and Aspect Ratio

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Table 19: OSU Sidewall Average Enhancement Values for Overall Test Section, First Passage, TT1, Second Passage, RT1, and Third Passage for Varying Rotation Number and Aspect Ratio

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<td>AR 1:2 Sidewalls</td>
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<td>0.74</td>
<td>0.70</td>
<td>0.71</td>
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</tr>
<tr>
<td>AR 1:6 Sidewalls</td>
<td>0.89</td>
<td>0.84</td>
<td>0.99</td>
<td>1.02</td>
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