Heat Transfer for High Aspect Ratio Rectangular Channels in a Stationary and Rotating Serpentine Passage with Turbulated and Smooth Surfaces

DISSERTATION

Presented in Partial Fulfillment of the Requirements for the Degree Doctor of Philosophy in the Graduate School of The Ohio State University

By

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2012

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ABSTRACT

Heat transfer distributions are presented for a stationary three passage serpentine internal cooling channel for a range of engine representative Reynolds Numbers. The spacing between the sidewalls of the serpentine passage is fixed and the aspect ratio (AR) is adjusted to 1:1, 1:2, and 1:6 by changing the distance between the top and bottom walls. Data are presented for aspect ratios of 1:1 and 1:6 for smooth passage walls and for aspect ratios of 1:1, 1:2, and 1:6 for passages with two surfaces turbulated. For the turbulated cases, turbulators skewed 45° to the flow are installed on the top and bottom walls. The square turbulators are arranged in an offset parallel configuration with a fixed rib pitch-to-height ratio (P/e) of 10 and a rib height-to-hydraulic diameter ratio (e/Dh) range of 0.100 to 0.058 for AR 1:1 to 1:6, respectively. The experiment spans a Reynolds Number range from 4,000 to 130,000 based on the passage hydraulic diameter. Heat transfer measurements were taken using a scaled experimental configuration that was assembled with copper segments, back side bonded foil heaters, and embedded resistance temperature detectors (RTDs) coupled with a computer control system to measure the amount of heat needed to maintain a constant wall temperature. In addition, embedded Kulite pressure transducers and thermocouple probes measured the flow conditions at the inlet and outlet, as well as at several intermediate stations.
While this experiment utilizes a basic layout similar to previous research, it is the first to run an aspect ratio as large as 1:6, and it also pushes the Reynolds Number to higher values than were previously available for the 1:2 aspect ratio. It shows that while the normalized Nusselt Number for the AR 1:2 configuration changes proportionally with Reynolds Number up to 130,000, there is a significant change in flow behavior between Re=25,000 and Re=50,000 for the aspect ratio 1:6 case. This indicates that while it may be possible to interpolate between points for different flow conditions, each geometry configuration must be investigated independently.

For the 1:1 aspect ratio, the high blockage ratio produced by the turbulators created strong secondary flow in the first passage. The counter rotating flow was directed toward the sidewalls increasing the heat transfer on the un-turbulated sidewall. The tip turn generated counter rotating vortices due to the pressure gradient created by the flow impacting the outside wall in the turn, first characterized by Dean, enhanced the heat transfer in the turns, but their strength was mitigated by the secondary turbulator generated flow in the second passage. The tip turn generated vortices enhanced the turn heat transfer for the smooth wall passage more than the skewed turbulator passage configuration since the Dean Vortices were not disturbed by the turbulators in the second passage. However, the high blockage ratio and subsequent enhanced heat transfer due to the turbulators did cause the largest pressure drop of all the aspect ratios.

The larger cross sectional area in the 1:2 aspect ratio configuration allowed the Dean Vortices to develop into a more symmetric shape resulting in higher heat transfer augmentation when compared to the 1:1 aspect ratio. As a result the heat transfer
enhancement immediately after the turns increased disproportionally as the Reynolds Number increased. The larger area however diminished the influence of the turbulators on the sidewalls in the main passage.

The 1:6 aspect ratio configuration had significant flow development heat transfer augmentation in each of the main passages. The enhancement was the largest in the first passage with the simultaneous momentum and thermal boundary layer development combined with the greatest driving temperature. The blockage ratio was the smallest for the 1:6 configuration (3%). As a result, the turbulator enhanced secondary flow development was insignificant until the Reynolds Number was larger than 25,000. At higher Reynolds Numbers the heat transfer augmentation from the main passage turbulator induced secondary flow, the turn vortices, the turn-geometry induced separated flow re-attachment and the smallest pressure drop result in high heat transfer augmentation with minimal frictional losses.

This research is the first internal heat transfer experiment performed in high aspect ratio (AR 1:6) smooth and turbulated rectangular passages. The eight inch long passages are longer than other similar regionally averaged heat transfer experiments. In addition, the experimental assembly utilized three passages. The combined enhanced passage length and the use of three passages significantly adds to the previously published heat transfer data set. The large physical size of the experimental assembly is complimented by the Reynolds Numbers range. For the higher aspect ratios (AR1:2 and AR 1:6), the Reynolds Number range is in excess of previous regionally averaged heat
transfer experiments. The Reynolds Number ranged from 10,000 to 130,000, for AR 1:2.

For AR 1:6, the Reynolds Number range is between 4,000 and 75,000.
DEDICATION

My dissertation is dedicated to my parents Jane C. and Donald E. Smith.
ACKNOWLEDGMENTS

I thoroughly enjoyed my time at The Ohio State University Gas Turbine Laboratory. I feel the practical and theoretical knowledge I obtained at the lab is unparalleled. My time at the lab, even those long hours designing constructing and commissioning the experimental assembly will enable me to excel in my field.

I would like to thank my advisor, Dr. Michael Dunn, for his guidance while I pursued my Ph.D. I am indebted to Prof. Mohammad Samimy, Prof. J. William Rich, and Prof. Sandip Mazumder for serving on my dissertation committee. Also, I would like to give special thanks to Dr. Randall Mathison for the advice and time he spent working with me on this project.

I would also like to thank GE Aviation and GE Energy for funding my research. I especially appreciated the technical oversight provided by Dr. Robert Bergholz of GE Aviation. In addition, I would like to thank Production Control Units, Inc. for providing me time off to pursue my degree and funding my first two years of course work.

I offer my gratitude to the technical staff of The Ohio State University Gas Turbine Laboratory, including Jeff Barton, Ken Fout, Igor Ilyin, Ken Copley, and Cathy Mitchell, for their hard work on this project. In addition, I would like to thank Dr. Charles Haldeman for his work in planning this experiment while he was at the laboratory.
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# NOMENCLATURE AND ABBREVIATIONS

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Units</th>
</tr>
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<tbody>
<tr>
<td>a₁</td>
<td>Scale</td>
<td>(K/ohm)</td>
</tr>
<tr>
<td>A</td>
<td>Projected Area</td>
<td>(m²)</td>
</tr>
<tr>
<td>AR</td>
<td>Aspect Ratio [W:H]</td>
<td></td>
</tr>
<tr>
<td>CFD</td>
<td>Computational Fluid Dynamics</td>
<td></td>
</tr>
<tr>
<td>cₚ</td>
<td>Specific heat</td>
<td>(J/kgK)</td>
</tr>
<tr>
<td>Dₕ</td>
<td>Hydraulic Diameter</td>
<td>(mm)</td>
</tr>
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\[ De = Re \left[ \frac{D_h}{r} \right]^{1/2} \]

DAS Data Acquisition System

\[ e^+ = \left( \frac{e}{D_h} \right) Re \left( \frac{f}{\tau} \right)^{1/2} \]

Roughness Reynolds Number

f Friction Factor

fₜₐ Four Side Turbulated Passage Friction Factor

fₗᵣ Two Opposite Side Turbulated Passage Friction Factor

fₛ Friction Factor for Smooth Wall Tube,

Fully Turbulent Flow

G Gravitation | (m/s²)

GEA General Electric Aviation

H Passage Height | (mm)

h Regionally Averaged Heat transfer coefficient | (W/m²-K)

NASA HOST National Aeronautics and Space Administration

Hot Section Technology

xx
Fluid Thermal Conductivity \( (W/m-K) \)

Length \( (mm) \)

Mass Flow \( (kg/s) \)

National Institute of Standards and Technology

Nusselt Number

Nusselt Number for Smooth Wall Tube, Fully Turbulent Flow

Ohio State University Gas Turbine Laboratory

Turbulator Pitch \( (mm) \)

Pressure \( (kPa) \)

Prandtl Number

Heat Transfer \( (W) \)

Heat Loss \( (W) \)

Radius of Internal Passage Turn \( (mm) \)

Resistance \( (\Omega) \)

Radius of Rotation \( (mm) \)

Reynolds Number

Reynolds Number for Smooth Wall Tube, Fully Turbulent Flow

Rotation Number

Revolutions per Minute

Resistance Temperature Detectors

Room Temperature Vulcanization

Turbine Research Facility

Thermocouple

Average Wall Temperature \( (K) \)

Coolant Bulk Temperature \( (K) \)

Universal Signal Conditioning Instrument Amplifier

Coolant Velocity \( (m/s) \)
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>( W )</td>
<td>Passage Width</td>
<td>(mm)</td>
</tr>
<tr>
<td>( W_d^* )</td>
<td>Dimensionless Divider Thickness</td>
<td></td>
</tr>
<tr>
<td>( \alpha )</td>
<td>Turbulator Angle</td>
<td>(Deg)</td>
</tr>
<tr>
<td>( \eta )</td>
<td>Thermal Performance</td>
<td></td>
</tr>
<tr>
<td>( \mu )</td>
<td>Coolant Dynamic Viscosity</td>
<td>(Pa-s)</td>
</tr>
<tr>
<td>( \rho )</td>
<td>Coolant Density</td>
<td>(kg/m^3)</td>
</tr>
<tr>
<td>( \Omega )</td>
<td>Angular Velocity</td>
<td>(Rad/s)</td>
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CHAPTER 1

INTRODUCTION

For better power output and increased thermal efficiency, the combustor temperature of gas turbine engines has continuously increased since the first designs in the 1940’s [1]. In the early 1960’s, the combustor temperatures exceeded the melting temperature of the simple turbine airfoils, so a combination of material improvements and blade cooling schemes were needed to prevent the airfoils from melting. More exotic materials and casting techniques were used to fabricate the turbine airfoils, culminating in the single crystal nickel based super alloy currently used [2]. Turbine airfoil castings were re-designed to include internal passages for cooling as well as external film holes to create a protective layer of cooler gas over the airfoil surface. The modern gas turbine utilizes coolant air bled from one or more intermediate compressor stages to cool the high-pressure turbine first vane row and first blade row that are located immediately aft of the combustor exit. Bleeding cooling air from the compressor is detrimental to the overall efficiency of the gas turbine, and thus, the heat transfer benefit derived from the use of this coolant must be optimized. Maximizing the heat transferred from the turbine airfoil to the coolant flowing through the internal passages allows the combustion temperature to be increased, which also increases the efficiency of the machine. The
configuration of these internal passages varies widely depending upon the particular engine company.

A typical airfoil is shown in Figure 1.1. The coolant enters the airfoil through the root and is routed through the airfoil maximizing internal heat transfer before the coolant is exhausted out of the airfoil through either film cooling holes, trailing edge cooling slots or tip cap cooling holes. Due to the size and shape of the airfoil, multiple techniques are used to enhance the heat transfer. The trailing edge uses pin fins in the internal passage due to size limitations to enhance heat transfer. This coolant is then exhausted out of the trailing edge for external localized cooling. The leading edge uses the coolant exhausted through cooling holes to form a film around the area of the airfoil with the highest heat load. Additional cooling holes are located on the tip cap and the platform. Each of these locations is the subject of intensive research, however they are not considered as part of this research program.
The research described herein is specific to the center serpentine passages of the turbine airfoil (Figure 1.1). The center section is made up of three pass serpentine passages. The three cooling passages initially were made up of four smooth walls, but research has shown that adding trips or turbulators designed to trip the boundary layer results in higher heat transfer from the metal to the coolant. Further research has shown that the skewed turbulator design enhances the heat transfer when compared with turbulators normal to the flow. Figure 1.1 shows the skewed turbulators that provide the highest heat transfer coefficient combined with the most cost effective casting process.
The current research will compare the Nusselt Number for passages with four smooth-walls and passages with two smooth walls and two walls featuring turbulators skewed to the flow on the leading and trailing (top and bottom) surfaces.

1.1 Significance of Problem

As the sizes of gas turbines have increased for better efficiency, noise reduction, and power generation, the turbine airfoils have also increased in size. The existing data used in the design of these airfoils is largely based on a single data set that dates back to the 1991 timeframe [4, 5] and has significantly lagged behind the development of the larger turbine hardware. The internal passages of these airfoils have grown to aspect ratios in excess of 1:6 [W:H], which is greatly in excess of the 1:1 passages used in the data set just noted. Research has advanced the data set beyond the square aspect ratio experiments from the early ‘90s reaching aspect ratios as large as 1:4. These research programs are typically limited to two short passages with flow conditions that are not representative of modern gas turbines. Therefore, the current research was undertaken to eliminate these deficiencies.

It was important for the assembly used for these measurements to be capable of being configured for multiple aspect ratios with limited time between configuration changes. To be more relevant to modern designs, the assembly was made longer than all other experimental configurations of this type. Included in the extra length was an entrance length equal to six hydraulic diameters for the largest aspect ratio to be run. With the added length and the multiple aspect ratios, the quantity of instrumentation used was far in excess of other regionally averaged internal heat transfer experiments.
This research extends the current design data set to an aspect ratio of 1:6 to be more applicable to the airfoils currently used in the larger high-bypass gas turbines and the passages close to the leading edge of the airfoil. Further, the existing data set spans a limited Reynolds Number range. A major goal of the work described here was to expand on the typical range of (10,000 to 75,000). Figure 1.2 shows the comparison between the new data set for this thesis and existing research that is referenced by industry. For the experiments performed to compare with the HOST (NASA HOr Section Technology) experiments [4, 5], the typical Reynolds Numbers were performed. For higher aspect ratios, experiments were performed for Reynolds Numbers both above and below the typical range. For aspect ratio 1:2, experiments were performed for Reynolds Numbers as high as 130,000. For aspect ratio 1:6, Reynolds Numbers as low as 4,000 and as high as 75,000 were run. This enables data from the stationary runs to be compared with data from the rotating runs since the maximum rig RPM of 3,000 and the target rotation number of 0.30 equate to a Reynolds Number of 75,000 given the maximum internal pressure the rig can withstand.
1.2 Objective

The purpose of this measurement program is to extend the internal heat transfer data envelope for high aspect ratio airfoils. The measurement technique and experimental procedure for a new facility will be validated by initially comparing measurements against the HOST experiments. The sponsors of this research wanted to confirm that the current measurement technique and experimental configuration result in relatively close agreement to HOST heat transfer data and the Nusselt Number for the 1:1 aspect ratio configuration, upon which the industry bases their design system. Furthermore, the uncertainty for the experiment should be minimized with an uncertainty of no more than 10% for the first passage at the HOST baseline Reynolds Number of 25,000. Finally, the
experimental configuration and approach must be capable of creating high aspect ratio data sets for aiding in the design of modern turbine airfoils.

The HOST experiments and all subsequent regionally averaged heat transfer coefficient experiments were performed at steady state conditions having very long run times. All the experimental runs for this program will be performed in a blowdown style utilizing existing facilities at The Ohio State University Gas Turbine Laboratory (OSU GTL). Blowdown facilities create “quasi-steady” conditions where flow properties are nearly constant over a short time period. Because the facility has a limited air supply, there is an ultimate limit to the length of the experiment before the pressure differential across the orifice is no longer sufficient to ensure choked flow. Because of this change in experimental technique, it was deemed necessary to compare data from the short duration experiment to the HOST data obtained using a long duration experiment.

Experiments with the rotating experimental configuration will be conducted upon completion of the analysis of the stationary measurements, but the results of those measurements will not be a part of this dissertation. However, the hardware enabling the experimental configuration housing the serpentine passages and associated instrumentation to rotate were designed and fabricated as part of this dissertation. The hardware had to be incorporated into the existing Fan Spin Pit facility located at The OSU GTL. The requirement was that the entire structure must be capable of rotation numbers up to 0.30 for AR 1:6 at a maximum speed of 3,000 rpm. Included in the design are the rotary manifold, rotor, rotation manifold and the serpentine model (Figure 1.3). Since the internal passages in an actual turbine airfoil are not all normal to the axis of
rotation, the design includes the rotation manifold. The manifold allows the serpentine model to be oriented in 2 discrete orientations: $0^\circ$ and $45^\circ$.

Figure 1.3 Rotating Serpentine Internal Heat Transfer Model

Relevant published internal heat transfer research is reviewed in the following chapter to provide perspective on the current state-of-the-knowledge. Next, the experimental configuration used to advance the current state-of-the-knowledge is
described including the research facility. The method used to collect and reduce the data along with an uncertainty analysis is also described. The following chapter presents the results for this high aspect ratio research project. Finally, the conclusions are summarized.
CHAPTER 2
LITERATURE REVIEW

As noted earlier, the primary objective for this research project is twofold: (1) to demonstrate a new experimental approach for obtaining measurements relevant to current industry interests by initially replicating the NASA HOST data set because it is this data set to which the industry has anchored their predictive techniques and (2) to significantly expand the knowledge envelope by obtaining measurements for much more modern aspect ratios at a significantly wider range of flow conditions for more relevant passage geometries. Therefore, the HOST results authored by Hajek et al. and Johnson et al. [4, 5] in the early 1990’s were an integral part of the design philosophy used for the current experimental configuration.

The progression of research leading up to the high aspect ratio internal heat transfer experiment is shown in Figure 2.1. Prandtl revolutionized fluid dynamics in 1904 when he described the boundary layer. After Prandtl, Nikuradse [6] developed the momentum law-of-the-wall to determine the boundary layer for internal pipe flow roughened with sand grains. The sand was replaced with repeated ribs by Webb and Eckert [7] for use in heat exchangers. In 1978, Han [8] extended the heat exchanger research to the internal passages of an airfoil. By the late ‘80s the NASA HOST had advanced the airfoil internal heat transfer research to include rotation. However, the
research is limited to square passages. As a result, this research project is used to provide data for high aspect ratio internal heat transfer.

![Figure 2.1 Technology Timeline](image)

**Figure 2.1 Technology Timeline**

### 2.1 Hot Section Technology

Similar to The OSU internal heat transfer research program, the HOST experiments were performed in phases. The initial experiments were performed for the smooth wall configuration [9, 10] and these experiments were followed by measurements for the turbulated wall configuration [11, 12]. Unlike the current research program, the HOST experimental assembly had the turbulators machined into the copper panels so the smooth wall experiments were performed stationary and rotating before moving on to the turbulated surface experiments. The process was repeated after the smooth copper panels
were removed and replaced with copper panels containing turbulators normal to the flow. Finally, experiments with copper panels including turbulators skewed to the flow were performed.

Similar to the OSU research program, the HOST experiments started with a baseline condition and compared subsequent experiments to that baseline. For stationary experiments, HOST utilized $\text{Re}=25,000$, $T_w-T_B=44^\circ\text{C}$, with an inlet density ratio of $\frac{\Delta \rho}{\rho} = 0.13$. For rotating experiments, the same conditions as the stationary baseline were used with a rotation number of $\text{Ro}=0.24$ and an orientation of $0^\circ$ to the axis of rotation [13].

The HOST experiments showed that the influence of varying Reynolds Number on heat transfer ratio were reasonably well correlated by a power law dependence of the Nusselt Number on the Reynolds Number ($\text{Nu} \propto \text{Re}^{0.8}$) [12]. A major result of the HOST experiment was to quantify just how much the rotational generated secondary forces influence the heat and mass transfer. For the rotating smooth wall case, the Coriolis and buoyancy forces combine to enhance heat transfer on the trailing surfaces of the first passage with radial outward flow. The difference in heat transfer, over a stationary fully developed flow in a smooth wall tube, was determined to be as much as 3.5 times fully developed flow. Rotating the test section caused a decrease of up to 40% on the heat transfer to the leading surface of the first passage [9]. Heat transfer was influenced differently in the second passage because the flow was moving radially inward and the buoyancy force had little effect on high-pressure surface flow. The heat transfer ratio was
primarily a function of buoyancy parameter on the low-pressure surfaces of the passage, regardless of flow direction [10].

For the turbulated passages, the research showed the heat transfer was greater than the smooth passages at the same flow conditions. The comparison of turbulators normal to the flow with turbulators skewed 45° to the flow showed that the secondary flow created by the skewed turbulators was less sensitive to buoyancy effects, which produced a more consistent heat transfer enhancement. From this research, they confirmed Nu=f(Re, Pr, Ro, Rad/Dh, c/Dh, Δρ/ρ) and it was recommended that skewed turbulators should be used in rotating coolant passages due to their consistently high heat transfer throughout the serpentine passage assembly [12].

2.2 General Serpentine Passage Internal Heat Transfer Review

Aside from the HOST experiments, some of the most significant rotating heat transfer research has been performed by Han and his group at Texas A&M University. They have published numerous papers on both stationary and rotating experiments with Reynolds Numbers ranging from 10,000 to 90,000 and with aspect ratios of 1:4, 1:2, 1:1, 2:1 and 4:1 using regionally averaged heat transfer. Han’s experiments in the late 1980’s and early 1990’s were, for the most part, performed stationary. However, more recent experiments have included rotation, with rotation numbers as high as 0.65 [3, 14-19]. The maximum Reynolds Number for the rotating experiments was 40,000, and the largest aspect ratio was 1:4. For the work of Huh et al. [17], which had the higher rotation number, the aspect ratio was increased to 1:4 to minimize the rotational speed of the
experimental assembly. In other experiments, where the aspect ratio was 1:1, the higher rotation numbers were achieved by limiting the Reynolds Number.

Han also investigated the influence various heating conditions have on the local heat transfer in square rotating passages. For the case of the two turbulated walls being heated with no heat applied to the sidewalls, the result was higher heat transfer on the turbulated walls when compared to the uniform heating on all four walls. However, the heat transfer enhancement was less for the sidewalls under the non-uniform heating condition [20]. For a rotating square serpentine passage, Han et al. found the uneven wall temperature created an uneven buoyancy force between the leading and trailing surfaces. This uneven buoyancy force degrades the effect of rotation resulting in a smaller difference between the heat transfer on the leading and trailing surfaces noted previously for rotation [21].

2.2.1 Computational/Visualization Research

To provide a better understanding of the flowfield, several notable computational/visualization studies have been performed. Bons and Kerrebrock were the first to report the global velocity measurement in a heated, rotating test section [22]. Their research stressed the effect rotational induced Coriolis and buoyancy forces have on the heat transfer in the rotating passages. Like Bons and Kerrebrok, Elfert et al. [23] used particle image velocimetry (PIV) to experimentally investigate the flowfield in a two pass turbulated heat transfer assembly. The flow visualization shows the extent of the separation at the inlet to the second pass along with the turbulator induced vortices. Al-Qahtani et al. used Reynolds-averaged Navier-Stokes (RANS) numerical method with
a near-wall second-moment turbulence closure to develop temperature and velocity contours for rotating and stationary configurations [24]. For the stationary turbulated case, their results indicate the heat transfer increases until the midsection due to the turbulator induced secondary flow becoming stronger in the streamwise direction. For the rotating configuration, they found similar trends mentioned previously: the heat transfer on the trailing surface increased in the first passage while the heat transfer decreased on the leading surface due to the Coriolis forces generated by rotation. Prakash and Zerkle [25] numerically determined similar results along with Liou et al. [26] using Laser-Doppler velocimetry and pressure measurements. Recently, Walker and Zausner [27] used RANS formulations contained within CFX to calculate heat transfer over 90 degree and 45 degree skewed turbulators with mixed results. Their results compared well with experimental data for the 45 degree skewed turbulators but under predicted the normal turbulator heat transfer. Tafti et al. [28-31] have compared results obtained using large eddy simulation (LES) to experimental data. Their results compare well with experimental results for the secondary flow field generated by the turbulators. The computations provide previously unreported major flow structures with fidelity behind the turbulators including recirculation zones on top and after the turbulators along with the corner eddy. Additionally, Viswanathan et al. [32] used detached eddy simulation (DES) and unsteady Reynolds averaged Navier-Stokes (URANS) to calculate the localized turbulator heat transfer. The detached eddy simulation does a good job predicting the average flow characteristics. They both capture the helical vortex behind the turbulators, but local URANS predictions are not accurate. Saha and Acharya [33,
34] used URANS and LES to numerically determine heat transfer around turbulators in a rotating heat transfer model. They achieved predictions within 20% of each other over a wide range of Reynolds Numbers.

2.2.2 Wall Surface Treatments

In order to trip the boundary layer and enhance mixing of cold core flow with warm near wall flow, the surfaces of the internal passages are roughened using various techniques. The first in-depth study of the effects of pipe surfaces roughness was performed by Nikuradse [6]. He used sand to roughen the internal surface of pipes. From this research a modified law-of-the-wall friction correlation was developed based on surface roughness. Since Nikuradse’s historic research, additional research has been performed with the primary goal of improving heat transfer in tube heat exchangers. Nunner [35] was the first to propose a thermal heat transfer correlation. Dipprey and Sabersky applied the law-of-the-wall correlation to the temperature profile of sand roughened pipes [36]. Working with these correlations, Webb [7] generalized it to repeated rib roughness surfaces and applied it to heat exchanger design [37, 38]. The turbulator correlations were next applied to the internal passages of an airfoil by Han [20, 39-41]. He developed equations for the Stanton Number and friction factor for passages with two normal rib roughened surfaces (turbulators) using a weighted average of four-sided smooth passage and four-sided turbulated passage. Shortly thereafter, he added oblique angle ribs into the correlation [42] followed by heat transfer performance comparisons for the developing regions and varying number of turbulated walls [43-45].

Using a similar weighted average technique, Taslim et al. [46-48] measured the heat
transfer of the turbulators and the area between the turbulators. He determined the turbulator heat transfer was significantly more than the area between the turbulators and used an area-weighted average to determine the overall heat transfer for each surface.

The primary method used to enhance turbulent mixing in the center passages of an airfoil is installation of rib turbulators skewed to the flow. The skewed turbulators generate a strong secondary flow directed toward the un-turbulated sidewall. The sidewall forces this flow back towards the center of the passage directing it towards the opposite un-turbulated sidewall. For a stationary serpentine passage, the secondary flow generates two counter-rotating cells of equal size [29, 49, 50]. Figure 2.2 shows the counter-rotating helical vortices with flow into the page. The skewed turbulators are positioned on the top and bottom surfaces along the Z-axis.

Figure 2.2 Skewed Turbulator Secondary Flow[29], Copyright © 2006 ASME, Reprinted by permission of ASME.
This secondary flow enhances the heat transfer on the turbulated surfaces and the adjoining sidewall for low aspect ratio ducts [3]. For regionally averaged heat transfer experiments, the enhancement from the skewed turbulators is not as apparent due to the large smooth sidewall area associated with large aspect ratios.

For ribs on two opposite walls, Han determined the Stanton number on the ribbed sidewall to be between 1.5 and 2.2 times that of the four-sided smooth duct. The Stanton number was also enhanced on the smooth sidewall by up to 25% due to the presence of the ribs in a square duct [39]. The blockage in the passage from the turbulators causes a relatively higher velocity in the passage. This higher velocity enhances the passage turbulence which causes increased heat transfer on the sidewalls. For a square passage with two opposite rib-roughened walls and a P/e of 10, the average Nusselt Number was about two times higher for orthogonal ribbed sidewalls when compared to smooth sided channels. Similar to HOST, Han found the orientation of the ribs to the flow was a factor in heat transfer [8, 42, 43]. In particular, ribs with an oblique angle to the flow in passages with blockage ratios below 0.25 have 10-20% higher heat transfer than the ribs with a 90-degree angle to the flow [48, 51-53].

The size, shape and periodicity of the turbulators have been the subject of many studies. Liu et al. [18] detailed the change in heat transfer when the spacing of the turbulating ribs is varied while keeping the channel aspect ratio fixed. As the rib pitch-to-height ratio (P/e) drops, the thermal performance increases for skewed ribs. However,
this result cannot be extended to orthogonal trips. Whereas the secondary flow pattern
generated by the skewed trips causes the fluid to circulate, the orthogonal trip surface has
a tendency to trap the fluid.

For a given turbulator blockage ratio (e/Dₜₗ), there exists ideal turbulator spacing
since the two extremes tend to smooth surfaces. As the turbulator P/e is reduced from the
high extreme, the heat transfer increases due to the added surface area of the turbulators
and the increased roughness. However, the increase in surface roughness also increases
the pressure drop, so a compromise must be made to achieve high heat transfer with
acceptable frictional penalties [54]. For a configuration with a 1:2 aspect ratio, e/Dₜₗ of
0.094, and square turbulators, Liu et al. found that decreasing P/e increases the friction
factor until P/e=5 [18]. This case has the highest pressure penalty for both rotating and
stationary channels. As P/e is further reduced, the friction factor drops. Taslim and
Spring [55] found similar results for a square turbulator, high blockage ratio e/Dₜₗ (0.22)
rectangular passage. The maximum friction factor occurs at P/e=7.5, and the maximum
heat transfer occurs at approximately P/e=8.0. The friction factor drops off equally as the
turbulator spacing is changed in either direction. The heat transfer enhancement,
however, is less sensitive to an increase in turbulator spacing. This suggests that the
optimum P/e ratio is higher than 7.5. Other research has shown that the ideal P/e occurs
between 7-11 [46, 47, 55]. For the experiment described here, the P/e is fixed at 10.

2.2.3 180-Degree Turns

By definition the serpentine passage contains at least one turn. The turns are
typically 180-degree. These turns generate areas of impingement, separation and
recirculation [19, 23, 54, 56-61]. As the flow passes through the turn, an imbalance of centrifugal forces and pressure gradient develops into counter-rotating vortices. The fluid is forced against the back wall creating a high pressure zone. The back wall redirects the flow toward the bottom and top surfaces, which circulate it back to the center of the turn. These two opposing rotating cells are known as Dean Vortices for which the Dean number is used to quantify the degree of stability [56, 62, 63].

For passages with high blockage ratios due to skewed turbulators, the Dean Vortices and the turbulators may induce conflicting flows. The turn induced secondary flow is forced against the inside wall. However, if the skewed turbulator are oriented so the secondary turbulator induced flow is directed toward the outside wall, the two flow patterns will conflict. The turbulator oriented toward the inside wall in a parallel formation enhance heat transfer more than orthogonal or cross oriented turbulators [20, 64]. For passages with high aspect ratios, Su et al. determined the secondary flow created by the turn developed into a more circular shape that filled the entire passage cross section [65]. The more dense secondary vortices led to higher heat transfer in the second passage when compared to smaller aspect ratios.

The presence of turbulators in the main passage and the shape and size of the turn influence the intensity of the turn induced vortices and the location of the separation zone along the inside wall of the second pass. Figure 2.3 shows the turn-induced vortices along the outside wall and the separation bubble located along the inside wall. The separation bubble and turn-induced vortices dissipate by \(X/D_h=1.0\).
For turns that are square in shape, stronger flow swirl is created resulting in higher heat transfer when compared to rounded corner turns [66]. Liou and Chen concluded that for smooth passages, sharp turns in a multi-pass serpentine passage are responsible for a significant augmentation to the heat transfer in the first part of the second pass [67]. The turn makes the turbulence intensity levels higher and considerably more non-uniform in the first part of the second pass as compared to the first pass. The maximum velocity occurred 1.5 hydraulic diameters downstream of the turn or 0.25 hydraulic diameters upstream of flow reattachment to the inner wall. In addition, Liou et al. [68] determined that the size of the turn dictates the location of the separation zone.
As the dimensionless thickness of the divider between passages increased, the separation bubble moved toward the turn, resulting in the reattachment occurring closer to the exit of the turn as shown in Figure 2.4 [68, 69]. The dimensionless divider thickness ($W_d^*$) for the current research program is 1.0. The dimensionless divider thickness for the three cases shown are as follows: (a) $W_d^* = 0.10$; (b) $W_d^* = 0.25$; and (c) $W_d^* = 0.50$ [70].

Figure 2.4 Turbulent Kinetic Energy Contours Around a Sharp Turn [70], Copyright © 1999 ASME, Reprinted by permission of ASME.
CHAPTER 3
EXPERIMENT

Due to the complex nature and sheer size of the experiment developed for this thesis work, the description is broken up into its key components. The model design encompassed two years of computational fluid dynamics (CFD), mechanical design and fabrication. Beyond that, a heater control system had to be developed that was capable of providing a constant wall temperature for 148 different panels while being flexible enough to compensate for failed RTDs and multiple aspect ratios. Finally, the setup of the research facility required modification to existing hardware, fabrication of nearly 50 multi-conductor cables, and installation of new components to facilitate the coolant supply to the experimental configuration.

3.1 Model Design

A modular design was adopted so that multiple aspect ratios could be investigated using the same experimental test section. The distance between the leading and trailing surfaces can be changed, as illustrated in Figure 3.1. Each passage is encased in insulation approximately 7-mm thick. Garolite sidewalls were used as a second layer of insulation and for support of the main passage assembly. The top and bottom surfaces had only garolite for insulating and support of the copper panels.
Figure 3.1 Passage End View: (a) AR 1:1, (b) AR 1:6
The bottom surface is fixed and the top surface can be adjusted to four discrete positions by inserting spacers on the backside of the copper panels. Figure 3.1(a) shows the configuration with the largest spacer installed to create a 1:1 aspect ratio passage. The spacer allows the leading surfaces to be secured from above along the top of the assembly. No spacers are needed for the highest aspect ratio, as shown in Figure 3.1(b). Intermediate spacers are available for aspect ratios of 1:2 and 1:4, but the 1:4 aspect ratio was not investigated.

The sidewalls are built for the greatest passage height, and the heaters are divided into rows so that only the heaters exposed to the airflow have to be turned on, as shown in Figure 3.2. The 1:1 aspect ratio is designed to match HOST with a ½-inch (12.7-mm) square passage. Thus, the other aspect ratios also have a width of ½-inch, but the height is 1-inch (25.4 mm) for the 1:2 configuration and 3-inches (76.2 mm) for the 1:6 configuration.
Figure 3.2 shows the backside of the copper panel along the first passage with the heaters removed for clarity. In addition, the four AR 1:1 sidewall copper panels have been removed along the first passage to provide visibility inside the passage. The garolite grid separating and supporting the copper panels is also shown.

The copper panels exposed to the coolant flow for AR 1:1 have two RTDs embedded in the vertical center of the plate and separated in the direction of flow by one inch (25.4 mm). The higher aspect ratios have only one RTD centered in the cooper panel. For the panels with one RTD, it is used to measure the copper temperature for the calculation of the heat-transfer coefficient and to control the input power of the heater.
On the copper panels with two RTDs, both temperatures are recorded and only one RTD is used to control the heater. The RTD used to control the heater is also used as an input to calculate the heat transfer coefficient. The temperature measurement from the second RTD in the AR 1:1 passage is used to evaluate the assumption of uniform temperature across the high conductive copper panel. During the design phase, the temperature difference across each copper panel was estimated to be less than ±0.5 Kelvin throughout a majority of the serpentine assembly. The additional RTD on the AR 1:1 panel is also used as a backup for the controlling RTD. The copper panels are 3/16-inch (4.8mm) thick to optimize the temperature distribution and minimize the overall weight of the model. The 110 copper alloy is used throughout the entire assembly to mimic the interior surface of the airfoil exposed to the coolant.

In addition to being able to run multiple aspect ratios, the test section is designed to investigate multiple turbulator configurations including no turbulators. The turbulators are secured to the top and bottom copper panels with a thin layer of adhesive in the desired configuration as shown in Figure 3.3. They are mounted in an offset configuration 0.25-inch (6.4 mm) between the leading and trailing surface in a collinear orientation as shown in Figure 3.3(a).
While there are many possible configurations, the experiments reported here focused on smooth walls (no turbulators installed) and skewed (45-deg) turbulators with smooth sidewalls. The turbulators are made from copper and are 0.050-inch (1.3mm) square, spaced 0.5-inch (12.7mm) apart resulting in a pitch-to-height ratio (P/e) of 10 as shown in Figure 3.3 (b). The turbulator size did not change as the aspect ratio changed, so the blockage ratio dropped from 20% for AR 1:1 to 3% for the AR 1:6 case. There are no turbulators in the root turn or the tip turns, and the turbulators did not span the gap.
between the copper segments. The 1/16-inch (1.5mm) gap allowed room for instrumentation (pressure transducers and thermocouples) and prevented the conduction of heat between adjacent panels.

Shown in Figure 3.4, the model inlet is designed to be at a minimum six hydraulic diameters long for the AR 1:6 case. Garolite G-10/FR4 is used for the inlet and outlet passages as well as the support grid for the copper panels. It provides high strength, reduced weight, and low thermal conductivity (0.27 W/m-K). In addition, the garolite is easily machinable, readily available, and inexpensive. It is secured in place with screws, and the gaps in-between the passages are sealed with a combination of O-rings and high strength RTV. The garolite side panels from the adjacent passages were secured next to each other.
The model was housed in an aluminum case designed to be suitable for the G-forces applied at maximum rotation. The entire heat transfer assembly, including the case, was designed using Autodesk Inventor. The three dimensional drawings were imported into a stress analysis system (ANSYS) to verify the design was suitable for the rotational forces. A detailed description of the stress analysis for the rotating facility is given in section 3.4.2. The top of the case was open to provide room for the instrument and heater wires to exit the model. The sidewalls and top surface spacers were secured in
place with multiple crossbars. The crossbars were also used for cable connector mounting.

For the stationary experiments, the model was connected directly to the cooling facility via an inlet and outlet hose. The initial experiments were conducted at an AR of 1:1, with smooth walls. The Reynolds Numbers were adjusted from 13,000 to 75,000 using discrete interchangeable chokes installed in the outlet manifold assembly downstream of the mass flow meter. There were pressure transducers located on either side of the choke to verify choked flow and record the outlet static pressure. Upon completion of the AR 1:1 experimental runs, the model was reconfigured to AR 1:2 and the process repeated until the AR 1:6 experiments were completed.

The test section used in this experiment was designed with three 8.0-inch (203 mm) long passages in a serpentine formation, as illustrated in Figure 3.5. These passages were connected by two squared-off turns with a centerline radius of 0.75 inches (19 mm). There is an unheated 6.0-inch (152 mm) long entrance passage upstream.
As shown in Figure 3.5, the model’s three main passages were each broken into four 2.0-inch (51 mm) long segments so the heat transfer could be localized to specific areas. These segments were made of copper panels separated by a 1/16” (1.5 mm) piece of garolite. The high conductivity of the copper helped maintain a constant temperature across the heated segment, and the low conductivity of the garolite minimized heat conduction between adjacent panels. Each copper panel was instrumented with at least one RTD and had a thin film heater bonded to the backside to enable control of the temperature of the panel.

3.2 Heater Control

The model used 144 Minco thin-film heaters bonded to the back of the copper plates. A photograph of the backside of a passage sidewall with the heaters installed is
shown in Figure 3.6. Also visible are the wires exiting at the top of the heaters from the embedded RTDs. Four custom heaters were designed and manufactured for the top and bottom panels in the turns by Birk Manufacturing. The nominal power density for the heaters was 45-kW/m², and the overall surface area of the experimental configuration is roughly 0.11-m². The total power applied to the assembly at peak heating was roughly 5-kW, and the maximum power required during the experimental runs was roughly 2-kW.

Figure 3.6 Bonded Minco Heaters on Back of Main Passage Sidewall

The number of heaters used for any one experiment is determined by the aspect ratio desired. For AR 1:6 all the heaters are utilized, but for AR 1:1 when only the first row of panels is exposed to the flow, just 58 heaters are utilized. Each heater is controlled independently in a digital on or off fashion. The five different resistances used in the model are listed in Table 3.1. The second tip turn is neither heated nor instrumented. The inside radius of the turns is, also, unheated and not instrumented.
The embedded RTDs were supplied with a constant 1-mA current. The change in temperature caused a corresponding change in RTD resistance, which was converted to voltage range suitable for digital monitoring. The conditioned signal passed through one of two 80 position patch panels before reaching the National Instruments PCI-6255 analog input cards. The heater power was controlled by an industrial rackmount computer with two 80-channel National Instruments PCIE-6509 digital output cards. The time the heater was energized was controlled by a custom National Instruments LabView program.

The heater control system queried the RTDs and if the copper panel temperature was below the target temperature, the heater was energized. The command signal to energize the heater passed from the National Instruments output card to an optically coupled switching card that was custom built for this experiment. Each optical switching
panel had 12-channels which isolated the computer from the heater supply voltage. With the command signal active, the heater power was supplied to the heaters via four constant voltage direct current power supplies. The voltage output was optimized depending on the heater resistance. After 50 command loops running at 250-Hz, the amount of time the heaters were energized was output to a storage file.

3.3 Instrumentation

The greatest number of measurements in this experiment comes from the power levels recorded for the different heated copper panels, but Kulite pressure transducers and miniature butt-welded thermocouples also provide important measurements of the flow conditions, including the inlet and exit total temperatures and static pressures. The mass flow exiting the test section was measured using a Micro Motion Coriolis mass flow meter. Additional facility pressure transducers and thermocouples provided a check of condition repeatability and provided an alternate method of calculating mass flow. The quantity of each type of instrument contained within the experimental assembly is given in Table 3.2. The instrumentation density used for this measurement program is far greater than used for the HOST measurements, and the quality of the instrumentation is consistent with 2012 technology. The AR 1:1 case has two RTDs embedded in each of the four copper sidewalls. The higher aspect ratio configurations have one RTD embedded in each of the copper side panels. The class A, 1000 ohm RTDs were manufactured by Sensing Devices, Inc. Their physical size was 2.3mmx2.1mmx0.9mm with 450-mm 32AWG leads. They were secured in a pocket machined in the back of
each copper panel with a thin layer of adhesive. They were potted in place with a high thermally conductive adhesive (3.6 W/m-K).

<table>
<thead>
<tr>
<th>Model</th>
<th>RTDs</th>
<th>Thermocouples</th>
<th>Pressure Transducer</th>
<th>Heater Circuits</th>
</tr>
</thead>
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<td>136</td>
<td>11</td>
<td>6</td>
<td>58</td>
</tr>
<tr>
<td>AR 1:2</td>
<td>166</td>
<td>2</td>
<td>6</td>
<td>88</td>
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<td>226</td>
<td>2</td>
<td>6</td>
<td>148</td>
</tr>
</tbody>
</table>

Table 3.2 Instrument Overview

The pressure was measured using six Kulite (0-50 psia) pressure transducers. Four transducers were embedded in the top surfaces in-between the passages and the remaining two were embedded in the inlet and outlet passages. In addition to the model integrated pressure transducers, three external Heise pressure transducers were used. One of the Heise pressure transducers measures the tank blowdown pressure, another the outlet static pressure through a pressure tap in the model case, and the third measures the vent pressure after the choke.

The 1-mil, butt-welded miniature E-type thermocouples (T/C), shown in Figure 3.7, were housed in a 3-inch (76.2 mm) long, 1/16-inch (1.6 mm) diameter stainless steel tube for support and protection during model assembly. The T/C was positioned in the tube so that it aligns with holes allowing the coolant flow to pass through the tube and past the T/C. The stainless steel tubes were inserted in the bottom of the assembly and
can be inserted to different depths to measure the coolant centerline temperature for calculating the coolant to sidewall delta temperature.

![Figure 3.7 Thermocouple with Steel Support Tube](image)

The bulk coolant temperature was measured with an inlet thermocouple and an outlet thermocouple. Nine additional thermocouples were equally spaced in the three serpentine passages. The first three passage thermocouples are shown in Figure 3.8. The passage thermocouples were only used for selected experimental runs to prevent turbulence enhancement they cause from influencing the overall heat transfer.
The passage thermocouples were used to corroborate the overall energy balance calculation performed using the inlet and outlet thermocouples.

Each of the instruments used to measure the temperature and pressure were calibrated before any experimental runs were performed. Some of the instruments, such as the Heise pressure transducers and class A Sensing Devices’ RTDs, were calibrated at the manufactures’ facility with labs that are traced back to the National Institute of Standards and Technology (NIST). The bulk flow thermocouples and the Kulite pressure transducers were calibrated at The Ohio State University Gas Turbine Laboratory based on NIST traceable instruments.

The platinum RTDs operate based off the propensity of metal to exhibit a change in resistance with a change in temperature. The class A RTD change in resistance follows the Callendar-Van Dusen equation. Since the temperature will not be below 0º
C, the modified Callendar-Van Dusen equation was used for this program.

\[ R_t = R_o (1 + A \cdot Temp + B \cdot Temp^2) \]  \hspace{1cm} (1)

\[ R_o = \text{Resistance at } 0^\circ C \]

\[ A = 3.9083 \times 10^{-3} \text{ } ^\circ C^{-1} \]

\[ B = 5.775 \times 10^{-7} \text{ } ^\circ C^{-2} \]

Since the RTDs are supplied with constant current from custom amplifiers, RTD resistance will change according to the modified Callendar-Van Dusen equation as the temperature changes. This change in resistance results in a voltage output that is measured by the data acquisition system after being conditioned.

The accuracy of the class A RTD is:

\[ \Delta T^\circ C = \pm (0.15 + 0.002T) \]  \hspace{1cm} (2)

Since the RTDs were calibrated by the manufacturer no additional calibration was performed. There was an offset applied to voltage output to correct for minor differences in line resistances. The offset was based on a 4-wire class A RTD applied to the model and exposed to the same ambient conditions. The 4-wire RTD was connected to a Keithley multimeter with an NIST traceable calibration. The 4-wire RTD automatically compensates for the line resistance loss.

The pressure transducers take advantage of the property that a conductor will change in resistance when tension is applied. Typically a conductor force-summing device is applied to a diaphragm in a pressure transducer. It is used to convert applied pressure into a stress proportional to the pressure. This stress is applied to an electrical transduction device which generates the electrical signal. Kulite combines the
transduction and force-summing into a stiff silicone diaphragm. With a change in pressure, the resistance of the silicone diaphragm will change. Thus the measured voltage will change proportional to the pressure applied.

For this program, the Kulite excitation voltage was +5 volts. The Kulite transducers were calibrated by pressurizing the entire experimental assembly to 1.25 times the typical run pressure. The pressure was allowed to vent out of the assembly in a controlled manner. The pressure drop was simultaneously recorded using the data acquisition system for all transducers including the NIST traceable Heise transducer used as a master. The Kulite pressure transducers were calibrated based on a linear curve fit. The calibration was deemed successful if the curve fit was within ±0.3 kPa of the master pressure transducer.

The E-type thermocouples were made up of dis-similar metals butt-welded together. The dis-similar metals generated a voltage when the tip was exposed to a temperature different from the reference temperature at the opposite end of the thermocouple. The thermocouples were conditioned with an electronic cold reference located in a custom circuit board mounted on the model support case. Their output voltage was amplified before passing through the interconnecting cables to the data acquisition system. The voltage was converted to temperature using a 5th order polynomial curve fit to the E-type voltage to temperature relationship. The polynomial was centered on the temperature range used for this program. Similar to the RTDs, an offset was applied to the measured temperature to compensate for line losses. This offset was based off of the 4-wire RTD mentioned previously.
3.4 Research Facility

The Ohio State University Gas Turbine Laboratory is made up of multiple permanent facilities both rotating and stationary. The internal heat transfer stationary experiments were performed in the turbine research facility (TRF) due to the support hardware already in place. Since the TRF is above ground, it provided easy access to the experimental assembly for initial debugging.

The rotating experiments will be performed in the existing fan spin pit. Like the TRF, the fan facility has a majority of the necessary support hardware already in place. Since the rotating mechanism is designed to support large fan blades at relatively low RPM, the design of the new internal heat transfer experimental hardware has to accommodate the existing envelope.

3.4.1 Stationary Facility

The coolant supply for all the experimental runs was dry air pressurized to 206.8-kPa stored in a 1.285-m$^3$ (3.336-m$^3$ for high Reynolds Number runs) pressure vessel. Prior to a run, the heaters were energized to the target temperature. They were allowed to stabilize before the coolant flow started. Before the flow starts, the power required to maintain the model at a constant pressure was recorded and used as heat loss. With the storage tank, heat exchanger and model pressurized; the experimental run started when the outlet isolation valve was opened (Figure 3.9).

The coolant passed through a liquid cooled heat exchanger before entering the serpentine model. The chiller was connected to the heat exchanger via insulated recirculation hoses. The chiller was maintained at approximately 223K for most of the
runs. Due to heat loss and heat exchanger efficiency, the coolant temperature at the inlet to the serpentine assembly was approximately 260K. The difference between the coolant temperature and the wall temperature was maximized to reduce the uncertainty without exceeding the allowable heater power.

![Figure 3.9 Stationary Test Facility](#)

As the coolant exited the serpentine assembly, the mass flow was measured by a 1.0-inch (25.4-mm) F-series Micro Motion mass flow meter. Finally, the coolant exhausted into a 360-m$^3$/hr Stokes vacuum pump used to ensure the flow was choked through the interchangeable orifice. Due to the blowdown style of experiment, the mass flow gradually declined during the 110 second run time. As the pressure drops in the
pressure vessel, the passage Reynolds Number transitioned through the target value for each run. The time averaged heat transfer data was reduced using a 10-second time window centered on that target Reynolds Number. A photograph of the stationary internal heat transfer assembly is shown in Figure 3.10.

![Stationary Serpentine Heat Transfer Assembly](image)

**Figure 3.10 Stationary Serpentine Heat Transfer Assembly**

### 3.4.2 Rotating Facility

The serpentine assembly shown in Figure 1.3 will be installed in an existing spin pit facility located at The OSU GTL, Figure 3.11. The new design had to fit within the constraints of the existing outboard bearing and upper drive housing. In addition, the
existing shaft had a flared mounting flange which was used to secure the rotor supporting the serpentine internal heat transfer assembly. The shaft, lower and upper bearing assemblies are designed for 5,000 rpm. The coolant will enter the rotating assembly just above the outboard bearing assembly. To accommodate the mass flow for the high aspect ratio (1:6), high Reynolds Number experiments; multiple coolant lines are used for the inlet and outlet. The inlet and outlet lines are connected to the fixed manifold from an air manifold (not shown). The coolant is isolated from the evacuated spin pit and from cross flow by three Stein seals. The coolant passes up through the rotary manifold to the rotor. One end of the rotor is used to secure the serpentine assembly while the opposite end will be used to counterbalance the serpentine assembly. The counterweight is made up of a battery assembly used both as a counterweight and to power the heaters used for simulating the heat of combustion.
The case housing the serpentine assembly was machined from forged aluminum 6061-T6 due to its high strength properties. The total assembly weighs approximately 27-Kg. At maximum design rotational velocity, the force exerted by the serpentine assembly is in excess of 4,800 G’s (1.28x10^6 Newton’s). Shown in Figure 3.12, the stresses for this design condition were analyzed using ANSYS once the original case design was completed.
After the ANSYS review, the case was strengthened with side ribs and the end wall of the case was made thicker. Finally, the two gussets were added to support the top and bottom of the case mount. The crossbars were also changed from the angle iron to square tubes as shown in the figure. The case attaches to the tilt manifold with sixteen grade eight bolts.

The tilt manifold, shown in Figure 3.13, is sandwiched between the case and the rotor. The sixteen bolt pattern in a circular formation is used to secure the tilt manifold to the rotor and allows for the model to be rotated between 0° and 45°. The tilt manifold is machined from stainless steel 17-4ph for its favorable strength properties. The plate is 1.0-inch (25.4mm) thick allowing maximum engagement for the thirty-two (32) 1”-12 bolts.

Figure 3.12 Case Stress Analysis
Figure 3.13 Tilt Manifold Stress Analysis

The rotor was also analyzed using ANSYS, shown in Figure 3.14. The rotor was designed to withstand the combined centripetal forces created by the serpentine model/case assembly and the tilt manifold. Like the tilt manifold, the rotor was machined from a stainless steel 17-4ph casting due to its favorable strength properties. In addition to supporting the serpentine model assembly, the rotor provides communication for the coolant passing to and from the model. The opposite arm supports the counter balance to the model. Like the case, the rotor design was enhanced after the structure was analyzed by significantly increasing the fillets used to relieve the stress around the counterweight and model mounts.
Surrounding the main shaft is the rotary union. The rotating portion of the rotary union is the rotary manifold. It connects to the bottom of the rotor and has three raised portions with tungsten carbide coating to mitigate seal wear. For better heat dissipation and maximum strength the rotary manifold was machined from 304 stainless steel. The heat generated by the Stein seal (maximum speed of 3,000-rpm) rubbing against the rotary manifold is expected to be in excess of 450-BTU/hr. For this reason, the seal wear area was coated in tungsten carbide. Due to this heat, the coolant will pass through heat exchangers cooled with liquid nitrogen.
The stationary manifold has sixteen 0.75-inch x 14-threads/inch female pipe thread ports for coolant communication (8 for inlet & 8 for outlet). The stationary manifold is also used to secure the Stein seals, and is machined from 304 stainless steel.

3.5 Data Collection

The voltage output from the RTDs controlling the heaters was routed to both the high-speed data acquisition system (DAS) and to the heater control industrial rackmount computer. All other transducers were recorded only on the DAS, shown in Figure 3.15. The thermocouple output was electronically cold referenced and amplified close to the serpentine assembly and further amplified by signal conditioners inside the control room. The pressure transducers and RTDs received power and signal conditioning by the Universal Signal Conditioning Instrument Amplifier (USCIA) modules in the control room. These conditioners were housed in 19 racks with 12 slots each. Each USCIA is able to accommodate two channels. These channels generally operate independently, but for the heater controlling RTDs, they were tied together so that a single input could be output to two locations. They converted the analog signal to a 10.00-volt scale centered on +1.50 volts by offsetting the analog voltage signal from 1000-Ω RTDs supplied with a 1-mA current. The thermocouples were centered on 0 volts. For the Kulite pressure transducers, the signal conditioners provided the 10-volt power and converted the analog pressure measured to narrow band of +2.25-volts and –1.25-volts for higher resolution. The transducer sensitivity was roughly 3-mV/PSIA.
For a majority of the inputs, the DAS was set to +6.5-volts and -3.5-volts. The conditioned signal was routed to the DAS through patch panels. For this experiment, 280 of the 288 available DAS channels were used. The DAS had 16-bit digitizers used to convert the conditioned analog signal output by the USCIA to digital. Although the DAS is capable of simultaneous sampling all channels at 2.5-MHz, the sampling for this experiment was fixed at 12.5-kHz so that any high-frequency response that developed could be resolved.

3.6 **Experimental Procedure**

Before the blowdown process was initiated, the heaters were energized until they reached the target wall temperature. The temperature was allowed to stabilize for several
minutes before starting the coolant flow. After the heaters stabilized, the heater control program was switched to run-time mode so that heater cycling could be maximized while minimal time was spent updating displayed information. The coolant flow was initiated once the vacuum isolation valve was energized. During the blowdown, data was collected for 110-seconds by both the DAS and the heater-controlling industrial rackmount computer. A trigger voltage signal was sent to both the high speed DAS and the heater control industrial rackmount computer when the isolation valve was energized so that the data could be properly aligned.

Once the fixed run time had expired, the isolation valve was closed and the heaters were turned off. While the data was being saved, the blowdown reservoir was refilled and the heaters were allowed to cool down. The entire process took approximately 30-minutes.

3.7 Data Reduction

To convert the raw measurements to the heat transfer and friction factor data of interest, the data time window must be identified so values can be averaged and the heat flux from each panel including losses to the surroundings can be calculated. It also requires understanding the uncertainty in the measurements and the calculation of relevant parameters.

3.7.1 Generation of Time-Averaged Values

This experiment is performed as a blowdown, so data from a single run is able to cover a range of Reynolds Numbers. Time windows are selected to average the data as the Reynolds Number passes through a desired condition. Figure 3.16 illustrates the
change in Reynolds Number (Part A) over time in conjunction with the change of the pressure in the test section (Part B).

![Graph showing blowdown decay](image)

**Figure 3.16 Typical Blowdown Decay: (a) Reynolds Number (b) Pressure**

After the isolation valve is opened, the Reynolds Number increases to its maximum value as coolant begins flowing through the model. The passage pressure decays with the supply tank pressure, and the mass flow through the choke and the
Reynolds Number also decrease. In the case shown here, the target Reynolds Number is 25,000, so a ten second time interval is chosen stretching from near 70-seconds to 80-seconds. All runs started with an initial passage pressure of 207-kPa, and the pressure during the time window was typically 160-kPa, although this varied slightly with the location of the time window. Figure 3.16(b) also shows the pressure in the vent line downstream of the choke, which is used to verify that the flow remains choked. Once the ten-second time window is determined, it is used to average the other flow measurements as well as the wall temperatures and the power applied to the heaters.

3.7.2 Heat Transfer Calculation

As previously noted, the heaters are energized before initiation of the experiment, and the wall temperature is allowed to stabilize at the desired value before the start of the coolant flow. Figure 3.17 presents the wall and coolant temperatures (Part A) and the total power delivered by all heaters (Part B) for an entire heat up and run sequence. In this sample run, power is first applied to heat the walls approximately 3.5-minutes before the flow is started. Since the time scale is defined to have zero seconds align with the start of the flow, this spike in power and increase in temperature can be observed at -220-seconds. The next several minutes are used to allow temperature and power levels to stabilize. Flow is started at zero seconds, and the coolant inlet temperature quickly drops to 264K and stabilizes before the beginning of the averaging time window at 70-seconds. This creates a 46K differential between the inlet coolant temperature and the wall temperature during the time window, as shown in Figure 3.17 (a). The power applied is also stable over this time window at nearly 200-Watts.
The heat transferred to the fluid by each heater panel is calculated using Equation 3. It represents the electrical power delivered by the heater minus the heat lost to the surroundings for that heater.

\[
q_{net} = \left( \frac{v^2}{R} \right) \times \% \text{ Heater On} - q_{loss} \tag{3}
\]
The electrical power is calculated using the voltage drop across the heater corrected for the line resistance from the power supply to the heater. Each heater resistance was measured, and the change of heater resistance with temperature was measured and found to be negligible for the temperature range of these experiments. Because the digital heater control system regulated power by turning the heater on or off in a high frequency loop, it is also necessary to multiply the electrical power by the fraction of time the heater was on over the time window. The heat lost to the surroundings was determined based on the power required to maintain the copper panels at a constant temperature without coolant flow immediately before initiation of an experiment, as indicated by the heat loss window on Figure 3.17. Because the heat transfer to the surroundings is nearly identical for the 10-seconds before the run and the 110-second run, this provides a very good indication of the amount of heat lost by means other than the main flow. To minimize the potential for heat loss, the runs were performed close to room temperature. It should be noted that the power required during this heat loss period is minimal compared to the power during the time window.

With the heat transfer rate determined, the increase in bulk fluid temperature was calculated for each segment by summing the power input to all four walls and dividing it by the mass flow and the specific heat of the fluid. For the first segment, the calculated value was added to the inlet thermocouple to get the bulk temperature at the outlet of the first segment. This process was repeated using the calculated bulk temperatures along the passage length to determine the streamwise bulk temperatures, as stated in Equation 4.
To ensure the energy balance relationship matches the actual bulk coolant temperature, thermocouples were inserted into the center of the passage for several AR 1:1 experimental runs. Since the thermocouples enhanced the turbulence in the passage, they were not used for all experimental runs. Figure 3.18 shows the comparison of the actual bulk coolant temperature measured by the thermocouples inserted into the center of the passage and the calculated bulk temperature using the energy balance. The specific heat is determined based on the local bulk temperature. As shown in Figure 3.18, the energy balance results can be approximated by a linear temperature relationship. For this program, the linear relationship was used to calculate the bulk temperature throughout the serpentine passage. The results are plotted against the distance from the inlet through the test section to each copper panel, expressed as a multiple of the hydraulic diameter of the passage (X/Dₜ). The location of the tip and root turns are indicated on the figure along with the passage numbers. The negative X/Dₜ distance indicates the inlet thermocouple is located upstream of the heated copper passage.
With the wall temperature measured by the embedded RTDs, the heat transfer calculated from Equation 3 and the bulk temperature determined from Equation 4, the heat transfer coefficient can be calculated using Equation 5.

\[
h = \frac{q_{\text{net}}/A}{(T_w-T_B)} \quad (5)
\]

The area is the projected area of the copper panel on the coolant side.

The Nusselt Number can be normalized using the Dittus-Boelter/McAdams [71-73] equation with \( \text{Pr}=0.71 \). The Dittus-Boelter/McAdams correlation is used to compare normalized data with other published results. The Dittus-Boelter/McAdams correlation
characterizes fully developed internal turbulent flow in circular passages with smooth surfaces. The properties for air were calculated using the local bulk temperature.

\[
\frac{Nu}{Nu_\infty} = \frac{hD_h/K_f}{0.023Re^{0.8}Pr^{0.4}}
\]  

(6)

The normalized Nusselt Number is helpful because it represents the level of enhancement over a smooth channel and can collapse the general effects of changes in Reynolds Number to highlight local changes in flow behavior. However, the power law dependence of the Nusselt Number on the Reynolds Number for turbulated rectangular serpentine passages is better characterized by \(Re \propto 0.6\) rather than the Dittus-Boelter/McAdams with \(Re \propto 0.8\) (Refer to Section 4.14 for more details). The difference in exponents indicates the heat transfer is better at lower Reynolds Numbers for the turbulated rectangular serpentine passages. Normalizing the rectangular serpentine passage with Dittus-Boelter/McAdams correlation results in a decreasing enhancement as the Reynolds Number increases.

### 3.7.3 Friction Factor

The dynamic pressure is calculated using the coolant velocity and density. The pressure difference was taken from the first passage inlet to the third pass outlet. Using the dynamic pressure, the friction factor is calculated based on the distance between the two pressure transducers and the passage hydraulic diameter.

\[
f = \frac{(\Delta Press)}{4(\frac{L}{D_h})^{1/2} \rho u^2}
\]  

(7)
The pressure penalty can be determined by normalizing the friction factor by a close fit of the Kármán-Nikuradse equation for fully developed turbulent flow in a circular tube [74]. The normalized friction factor is valid for Reynolds Numbers between $3 \times 10^4 < \text{Re} < 10^6$.

$$\frac{f}{f_\infty} = \frac{f}{0.046 \text{Re}^{-0.2}}$$

The overall thermal performance factor is determined by dividing the overall averaged normalized Nusselt Number by the normalized friction factor [37].

$$\eta = \left(\frac{\text{Nu}}{\text{Nu}_\infty}\right)/\left(\frac{f}{f_\infty}\right)^{1/3}$$

This performance factor helps quantify the balance of heat transfer increase against friction factor penalty and is helpful for determining the optimal design.

### 3.7.4 Theoretical Analysis

The theoretical analysis initially performed by Nikuradse and later by Dipprey and Sabersky on sand roughened pipes, modified by Webb et al. for heat exchanger use and adapted by Han for internal passages of an airfoil is compared with the data set from this research project. Han used the equations developed by Webb et al. [7] for round tubes and added duct dimensioning for square and rectangular passages normally found in an airfoil. Han also formulated a weighted area for friction factor of a passage with two opposing turbulated walls based off of the four sided turbulated passage and included the adjustment for skewed turbulator. The friction factor calculated with equation (7) for both the smooth passage and the turbulated passage AR 1:1 and 1:6 were weighted by area using the following equation:

$$f_{rr} = f_{rs} + (H/W)[f_{rs} + f_{ss}]$$

59
Using the dimensionless velocity profile normal to the wall a momentum transfer roughness function \( R(e^+) \) representing the velocity at the top of the turbulator is added to the law-of-the-wall and integrated over the distance between the turbulated wall and the height where there is no shear stress. Substituting for the dimensionless velocity in terms of the friction factor the following equation results:

\[
R(e^+) = (2/f_{rr})^{1/2} + 2.5\ln(2e/D_n) + 2.5\ln(2W/(W + H)) + 2.5 \tag{11}
\]

Using a similar method, the dimensionless temperature profile normal to the ribbed wall, a heat transfer roughness function was added to the temperature law-of-the-wall similarity. The heat transfer roughness function \( G(e^+, Pr) \) developed by Dipprey and Sabersky [36] was determined to be independent of the flow cross-sectional area and dependent only on the rib geometry for roughness Reynolds Numbers above \( e^+ > 25 \).

\[
G(e^+, Pr) = 3.74(\infty/90\,\text{deg})^{0.3}(e^+)^{0.28}Pr^{0.57} \tag{12}
\]

Combining equation (12) with the momentum transfer similarity law, the Nusselt Number can be calculated.

\[
\frac{Nu}{1+(f_{rr}/2)^{1/2}[G(e^+, Pr)\cdot R(e^+)]} \tag{13}
\]

3.8 Uncertainty Analysis

The uncertainty of the heat transfer coefficient is broken up into three major components: wall temperature, coolant temperature and heat flux. The uncertainty in the heat transfer coefficient increases as the driving temperature drops in the direction of the coolant flow. In addition, the uncertainty increases as the applied heater power drops for
the lower Reynolds Number runs. The uncertainty is calculated using the Kline and McClintock second-power method [75].

\[
\frac{\Delta h}{h} = \left\{ \left( \frac{\Delta q}{q} \right)^2 + \left( \frac{\Delta T_{wall}}{(T_{wall} - T_{fluid})} \right)^2 + \left( \frac{\Delta T_{fluid}}{(T_{wall} - T_{fluid})} \right)^2 \right\}^{1/2}
\]  

(14)

The significant component uncertainties that contribute to the overall uncertainty are listed in Table 3.3. The wall temperature uncertainty is largely dependent on the RTD calibration, constant current amplifiers and the resistance correction for line losses. The coolant temperature uncertainty has two major components. Since one thermocouple positioned in the middle of the passage is used to determine the bulk coolant temperature, it did not fully measure the spatial variation in the temperature. The variation is estimated at 0.80% after reviewing comparable computational fluid dynamics plots. The remaining portion of the uncertainty in coolant temperature is due to the calibration of the thermocouples. The heat flux uncertainty is made up of the uncertainty in heater resistance, copper panel area, the time the heater is energized and the voltage applied to the heater. In the early portion of the serpentine passage, the difference between the wall temperature and the bulk coolant temperature is high enough to have minimal impact on the overall uncertainty. Therefore the uncertainty remains constant until the difference drops sufficiently low to surpass the fixed thermocouple calibration uncertainty.
Table 3.3 Summary of Significant Uncertainties

As shown in Figure 3.19, the uncertainty in Nusselt Number is roughly ±4% through most of the first pass. The outlet of the third passage is higher for the lower Reynolds Number flow with an uncertainty of under ±15% for the core experiments (Re=10k-75k).
The Nusselt Number uncertainty is calculated based off the uncertainty in the heat transfer coefficient and the estimated uncertainties in the hydraulic diameter and thermal conductivity.

\[
\frac{\Delta Nu}{Nu} = \left\{ \left[ \frac{\Delta h}{h} \right]^2 + \left[ \frac{\Delta k_f}{k_f} \right]^2 + \left[ \frac{\Delta D_h}{D_h} \right]^2 \right\}^{1/2}
\] (15)
CHAPTER 4
RESULTS

Experiments were first performed in the AR 1:1 configuration so the results could be compared to the HOST data set. Upon completion of the AR 1:1 measurements, the model was reconfigured to AR 1:2 and the process was repeated until the AR 1:6 experimental runs were completed.

4.1 Run Matrix

Multiple experimental runs were performed for each configuration. The repeatability of the experiment was critical in determining the overall stability of the experimental setup. Based off the runs repeated with the same boundary conditions, the standard deviation remained low for all the different configurations. Combined with the single-sample uncertainty analysis [75], the multiple reviewers felt the number of repeated runs was satisfactory. Three aspect ratios and two different wall conditions were run with the same experimental model. In-between each phase, the model was reconfigured for different aspect ratios or different wall configurations (turbulated or smooth). Table 4.1 lists the specific Reynolds Number run for each aspect ratio and the associated wall configuration. If present, the turbulaters were only on the top and bottom surfaces.
4.2 Wall Identification

For all plots to follow the left-side wall relative to the coolant flow is labeled wall A (Left) and the right side is wall B (Right), as shown in Figure 4.1. The top is labeled wall C (Top) and the bottom surface is wall D (Bot). The turbulators, when present, are applied to the top and bottom walls.
4.3 Smooth Passage, Aspect Ratio 1:1

The initial experiments were performed with an aspect ratio of 1:1 (square passage) with smooth walls (no turbulators installed). The wall temperature was set to 300K, and the coolant inlet temperature was nominally 252K. The coolant inlet temperature changes with Reynolds Number due to changes in the degree to which the flow is warmed passing through the supply plumbing, and the wall temperature was adjusted to maintain a constant temperature difference when possible. The streamwise increasing coolant temperature is shown in Figure 4.2. The average wall temperature during the 10 second time window is also shown in the figure. The difference between the wall temperature and bulk fluid is known as the driving temperature. As the coolant receives heat from the sidewalls the coolant temperature increases reducing the driving temperature from 48K at the serpentine assembly inlet to 20K at the exit. Since the inlet thermocouple was located upstream of the heated section, the location was less than zero.
Figure 4.3 plots the calculated Nusselt Number averaged over the multiple runs for all four smooth walls for the Re=50,000 case. Nusselt Number is represented by unique symbol shapes and colors for each wall, and the value for each heater panel is plotted against the distance from the inlet through the test section to that panel, expressed as a multiple of the hydraulic diameter of the passage (X/D_h). The location of the tip and root turns are indicated on the figure along with the passage numbers.

From Figure 4.3, it is clear that the heat transfer declines through the beginning of all three passages as the coolant flow develops. The unheated entrance region is 12 hydraulic diameters long for the 1:1 aspect ratio configuration, which minimizes aerodynamic entrance effects but does not change the thermal entrance region at the beginning of the heated segments. The second passage has the largest change with flow development due to the combined turn effects and the higher driving temperature than the third passage. The turns experience the highest heat transfer, and the effect of the Dean
Vortices is clearly visible in the latter half of the turn where the maximum Nusselt Number occurs. As the coolant enters the turn, it is forced against the back wall. The imbalance of the centrifugal force and the pressure gradient causes the flow to impinge on the back wall increasing the pressure. The pressure gradient forces coolant impinging on the back wall to turn toward the top and bottom surfaces. This motion sets up counter rotating vortices in the turn.

Since no walls are turbulated for this configuration, the heat transfer is similar for all surfaces in the main passages. The only exception occurs at \(X/D_h=24\) immediately after the tip turn on wall D due to the location of the RTD controlling the heater which will be discussed in more detail in section 4.3.1.

![Figure 4.3 Smooth Passage Combination, AR 1:1](image)

The fractional dependence of the Nusselt Number on the Reynolds Number is shown in Figure 4.4. The general trends are consistent as the Reynolds Number
increases. The higher inertial forces enhance the turbulent mixing which is the primary mechanism driving the increase in heat transfer for the smooth passage. This increased turbulence causes the faster moving cooler core flow to mix with the warmer near wall flow which increases the heat transfer. Without any other mechanism to trip the boundary layer, the heat transfer remains constant after fully developed flow has been established. A high-pressure zone forms on the outside wall as the coolant enters the turn. This high-pressure zone allows the Dean Vortices to grow in strength as the fluid traverses the turn and they are at their maximum by the outlet of the turn. Once the coolant enters the second passage, these vortices rapidly decline.
Figure 4.4 Reynolds Number Combination, Smooth Passage
4.3.1 Driving RTD Location

As shown in Figure 4.5, the RTDs controlling the heaters are set at 305K. The first two segments show there is higher heat transfer due to developing flow as the coolant enters the first pass of the serpentine assembly. Since the RTD controlling the heater is located in the second position, the heat transfer is the lowest in this area for the segment. Thus the heater power input is lower than if the controlling RTD were in the first position. This causes the first position RTD to measure a lower temperature for the first two segments because the requested power is not sufficient to maintain the target temperature. This also leads to a lower Nusselt Number in the developing region of the first passage.

Figure 4.5 First Passage Temperature, AR 1:1, Re=50,000
In the area immediately after the tip turn on wall D, the combination of the Dean Vortices and the reattachment of the separation bubble at the inside of the turn resulted in a high Nusselt Number as shown in Figure 4.6. A similarly high Nusselt Number was not observed for wall C at this location because in this region of high heat transfer a temperature gradient develops within the copper plate, and the control RTD used for wall C is farther from the turn than the one used for wall D. This means that the heater control system will see a lower temperature for wall D than for wall C and will therefore put in more power to compensate. The secondary RTDs for wall C and wall D indicate that the temperature gradients across each plate are similar, indicating that the heat transfer levels for these plates are also similar.

![Figure 4.6 Second Passage Temperature, AR 1:1, Re=50,000](image-url)
4.3.2 Thermocouple Influence

The thermocouples used to verify the bulk temperature calculation influenced the bulk coolant flow enhancing the mixing of the warm near-wall coolant with the core region. For this reason, the thermocouples were only used to verify calculations and removed for all subsequent experiments. Figure 4.7 depicts how the thermocouples were inserted into the bottom of the serpentine assembly. For clarity the four copper sidewall panels that makeup the AR 1:1 configuration have been removed.

Figure 4.7 Thermocouple Layout

There were no thermocouples inserted in the turns. The influence of the thermocouples is localized to wall D main passages as shown in Figure 4.8. In each of the three passages, there is noticeable heat transfer augmentation for wall D caused by the
increased turbulence from the thermocouple support tubes. The largest difference is in the first passage with nearly 20% enhancement due to the thermocouples disrupting the coolant flow.

4.4 Skewed Turbulator Passage, Aspect Ratio 1:1

After completing the smooth wall experiments, the experimental assembly was dis-assembled. Turbulators were added to the top and bottom surfaces (wall C and D) skewed 45° to the flow in a parallel configuration (See Figure 3.3). As shown in previous research programs, the turbulators trip the boundary layer increasing the mixing of the

Figure 4.8 Thermocouple Heat Transfer Influence, Smooth Passage
near wall warm fluid with the cooler central core. The skewed turbulators are preferred over other configurations due to the secondary flow they setup. This secondary flow results in consistent heat transfer enhancement in all passages under rotation.

The wall temperature ranged from 291K for the high Reynolds Number runs to 319K for low Reynolds Numbers runs. The coolant inlet temperature was approximately 260K for these runs.

The Nusselt Number results for the Re=50,000 case are plotted together in Figure 4.9. The heat transfer enhancement for the turbulated wall is as much as two times the sidewall in the first passage. The sidewall heat transfer in the first passage increased when compared to the smooth wall case in Figure 4.3 due to the secondary flow development caused by the turbulators. The heat transfer in the turns was roughly 25% less than the smooth passage configuration from the high blockage ratio turbulator induced secondary flow interaction with the turn vortices. The heat transfer enhancement from the turbulators is the largest in the first passage due to the higher driving temperature. The peak in the first passage at X/D_h=5.8 is due to the combination of the high heat transfer from the developing flow at the entrance and the development of the secondary flow from the turbulated surface as the coolant advances down the passage. The decline in the heat transfer farther down the passage is due to the combined decline in the developing flow and the turbulator secondary flow.
Figure 4.9 Skewed Turbulator Passage Combination, AR 1:1

Similar to the smooth wall comparison, the turbulated wall Nusselt Number enhancement increased with increase in Reynolds Number as shown in Figure 4.10. Due to the large uncertainty from the low driving temperature in the last passage, the heat transfer for the walls A, B and C have similar results.
Figure 4.10 Reynolds Number Comparison, Skewed Turbulator Passage AR 1:1
4.5 Aspect Ratio 1:1, HOST Comparison

While the HOST experiments were used as the basis for the design of the 1:1 aspect ratio, there are some differences between the two assemblies that were dictated by the sponsors of this project, see Figure 4.11. Specifically, the entrance length is 6.0-inches (152.4 mm) for the OSU design. The HOST design utilizes a short settling length which is roughly a third of the OSU design. The turns have a minimal radius in the corner for the OSU design. Each main passage in the OSU assembly is roughly 50 mm longer. As a compromise for the large number of instruments required for the multiple aspect ratio capable OSU design, the second tip turn is not heated or instrumented in the OSU design. For the turbulated passage experiments, all four segments have turbulators on the top and bottom surface in the first passage of the OSU design while the settling length in the first passage of the HOST design is smooth. Both designs incorporate three serpentine passages broken up into multiple two inch segments to enable locally averaged heat transfer. The OSU assembly has four segments whereas the HOST assembly has three segments in passage two and three. If present, the turbulator pitch-to-height ratio (P/e=10) and the turbulator height-to-hydraulic diameter (e/D_h =0.1) are identical between the two assemblies. The turbulator alignment to the flow and the staggered pattern also match. The experiments were run with identical Reynolds Number for optimal comparison.
4.5.1 Smooth Passage Comparison with HOST, AR 1:1

Comparison of the results for all four walls as shown in Figure 4.12 illustrates that the only significant difference is in the early portion of the first passage and in the turns. The only other notable difference is due to the RTD location on wall D immediately after the turn. The HOST data exhibits more flow development in the first passage due to the short entrance region. The HOST assembly has combined hydrodynamic and thermal development in the first part of the first passage. The combined flow development results
in more heat transfer enhancement than flow with hydrodynamic or thermal development alone. The thermal entrance length begins at the same location for both assemblies. The 152.4 mm OSU entrance length is a significant length of the hydrodynamic entrance length. Therefore, the hydrodynamic flow is more developed in the OSU assembly at the point where the heated copper plates start.

The heat transfer is higher in the OSU assembly along the sidewall in the first turn. The difference is due to the different turn profile. The OSU design incorporates a sharper corner and the HOST assembly utilizes a rounded corner shape. As Wang et al. [66] found, the turn shape influences the heat transfer enhancement. The author compared three different turn profiles progressing from a circular turn to turns with square corners. The sharper turn resulted in higher heat transfer.

Overall, the HOST and OSU results are within 10% of each other. HOST quantified the uncertainty in terms of the heat transfer coefficient. The HOST uncertainty typically varied from ±6% at the inlet to ±30% at the exit of the heat transfer model baseline stationary test conditions [5]. The uncertainty for the OSU experiment AR 1:1 is similar to HOST for the inlet, but roughly half of the HOST uncertainty for the exit. Combining the uncertainty from both experiments, the OSU results were close enough for the smooth passage configuration to continue onto the skewed turbulator configuration and subsequent additional comparison with the HOST results.
Figure 4.12 Smooth Passage, HOST and OSU Comparison
4.5.2 Skewed Turbulator Passage Comparison with HOST, AR 1:1

Like the smooth wall comparison, the HOST and OSU results for the skewed turbulator configurations have similar heat transfer enhancement on all four walls. The most notable difference is in the first passage for the two walls (wall C and wall D, Figure 4.13) with turbulated surfaces. The OSU assembly experiences a higher heat transfer enhancement in the first passage than HOST. The difference is due to the settling section in the first passage of the HOST assembly (See Figure 4.11). The first segment of the HOST assembly was heated, but did not have turbulators.

The sidewall heat transfer has stable enhancement throughout the serpentine passage for both HOST and OSU. The second turn sidewall (wall B) for HOST again exhibits heat transfer enhancement that is 50% higher than the first turn and 80% higher than the OSU data at the same location. For those reasons, the HOST data is considered questionable on the sidewall root turn. There is a small difference in the sidewall enhancement in the first passage due to the turbulator influence. The OSU enhancement is larger than for HOST because there are turbulators along the entire heated first passage. The secondary flow, setup by the turbulators, is apparent in the area before X/Dh=10.

As with the smooth wall case, the heat transfer enhancement for the OSU turbulated passage case compares well with HOST, being within 10%. With the OSU data comparing well with the HOST data for both smooth and turbulated passages, the research program continued to higher aspect ratios.
Figure 4.13 Skewed Turbulator Passage, HOST and OSU Comparison
4.6 Skewed Turbulator Passage, Aspect Ratio 1:2

For the aspect ratio 1:2 experiments, the spacers holding the top of the serpentine passage were replaced. The new spacers were 12.7 mm shorter exposing 30 additional copper panels along the sidewall to the coolant flow. The heater control system was modified to include the control of the additional heaters secured on the backside of the copper panels. The heater control system hardware was already present to control the additional heaters, and it was necessary to only modify the software used to energize the heaters controlling them to constant wall temperature. For the sidewall heat transfer data, the heat flux results from the stacked copper panels (original AR 1:1 sidewall panel and the newly exposed AR 1:2 sidewall panel) were averaged for each segment.

The wall temperature ranged from 294K for the high Reynolds Number runs to 322K for low Reynolds Numbers runs with the coolant inlet temperature being approximately 265K for these runs. Again the heat transfer values are plotted against the distance from the inlet through the test section to that panel, expressed as a multiple of the hydraulic diameter of the passage (X/Dh). For the AR 1:2 the hydraulic diameter has increased to 16.7 mm.

Figure 4.14 is the combined Nusselt Number data for the four sidewalls of the aspect ratio 1:2 turbulated configuration with a Reynolds Number of 50,000. In the first passage, the noted difference in the sidewall heat transfer is apparent. Both the top and bottom surfaces achieve a Nusselt Number of 400 in the first passage resulting in an enhancement on the turbulated surfaces in excess of two times the sidewalls. Again, the location of the RTD immediately after the turn is visible by the high heat transfer to wall
D. There is a significant difference in the enhancement on the turbulated surfaces. In the second passage the heat transfer for wall C is 50% higher than it is for wall D. This indicates the coolant flow is biased toward the top surface in the latter portion of the passage. The sidewall heat transfer enhancement is similar for wall A and B. However, the secondary flow setup by the turbulators directs the flow toward sidewall A with recirculation zone directed back toward the opposing sidewall. As a result, there is roughly a 20% larger heat transfer on wall A when compared to wall B in the first passage. Figure 2.2 shows an example of the secondary flow induced by the skewed turbulators for a stationary case. The turn enhancement is 33% more than the sidewalls in the main passage. The larger cross sectional area in the turns allows the Dean Vortices to develop in the turns enhancing the heat transfer.

Figure 4.14 Skewed Turbulator Passage Combination, AR 1:2
In Figure 4.15 all the Reynolds Numbers investigated are compared. The heat transfer enhancement noted for the 50,000 Reynolds Number runs continued across all the other Reynolds Numbers. The smooth sidewalls show an increasing hydrodynamic flow development length as the Reynolds Number increases. The hydrodynamic entrance length increases from roughly $X/D_h=20$ for $Re=10,000$ to roughly $X/D_h=31$ for $Re=130,000$. With the inlet length equating to $X/D_h=9$ for the AR 1:2 case, the coolant hydrodynamic development has significantly diminished by the middle of the first pass for the lower Reynolds Numbers and is not fully develop before the flow enters the first turn on the Reynolds Numbers above 15,000.

In the middle of the second passage, the higher turbulence noticeably enhances the heat transfer difference between the top and bottom surfaces. For the highest Reynolds Number, the top surface has roughly 40% higher heat transfer than the bottom surface. The turn induced Dean Vortices and the separation bubble reattachment is causing an unbalanced velocity response. The turn-induced vortices diminish in strength toward the end of the passage so both the top and bottom surface heat transfer enhancement diminish. The reason for the high heat transfer located at the first measurement on wall D has been noted previously. However, the second panel measurement is also elevated at $X/D_h=20$ so the heat transfer enhancement shown is not only due to instrument location. There is higher turn induced turbulence on the bottom surface immediately after the turn. This secondary flow enhancement moves to the top surface farther down the passage.
Figure 4.15 Reynolds Number Comparison, Skewed Turbulator Passage
AR 1:2
4.7 Smooth Passage, Aspect Ratio 1:6

The final aspect ratio was run in both the smooth passage and turbulated passage configuration. For aspect ratio 1:6, the spacers holding the top surface were removed. In addition, the turbulators on both the top and bottom surfaces were removed; the system was re-assembled and placed back in the stationary test facility. The heater control program was modified to control all 148 heaters to the constant temperature boundary condition. The wall temperature ranged from 300K for the high Reynolds Number runs to 311K for low Reynolds Numbers runs with the coolant inlet temperature being approximately 265K for these runs.

The combination of all four walls is shown in Figure 4.16 for the Re=50,000 smooth wall case. The significant development length is apparent in each passage along with the turn induced secondary flow development in the second half of each turn. As expected for the smooth wall case, all four walls have similar heat transfer. The difference in heat transfer between the sidewalls and the top and bottom surfaces at X/Dh=1.5 is caused by the location of the supply pipe connecting the stationary assembly to the coolant supply. Because the supply pipe is located in the vertical center of the passage, the coolant is in full contact with the sidewalls immediately upon entry into the experimental assembly but some amount of redistribution is necessary to reach the top and bottom surfaces. This is mostly handled in the unheated entrance region, but some effect is still clear for this first panel.
The comparison of the wide range of Reynolds Numbers plotted in Figure 4.17 demonstrates the increasing hydrodynamic development length as the Reynolds Number increases. The Dean Vortices, separation bubble at the outlet of the first turn, and Reynolds Number relationship are also apparent by the significant increase in heat transfer enhancement after the tip turn for the Reynolds Numbers equal to or above 50,000.

The low Reynolds Number flow is just above the typical turbulent threshold of 2,300, but the Nusselt Number is significantly above the fully developed laminar value. While the laminar entrance length is significantly longer than the corresponding turbulent length, the slope of the Nusselt Number declined to zero before the end of the passage so it is unlikely it would decline to 3.66. Therefore, the Re=4,000 is determined to be fully turbulent flow.
Figure 4.17 Reynolds Number Comparison, Smooth Passage
AR 1:6
4.8 Skewed Turbulator Passage, Aspect Ratio 1:6

For the AR 1:6 skewed turbulator passage, the wall temperature ranged from 300K for the high Reynolds Number runs to 311K for low Reynolds Numbers runs with the coolant inlet temperature being approximately 265K for these runs.

Figure 4.18 shows the Nusselt Number for the AR 1:6 configuration with a Re=50,000. The heat transfer enhancement on the turbulated surfaces is roughly 2 times the smooth sidewalls. The largest enhancement occurs in the second passage immediately after the tip turn. Tip turn induced vortices combine with the turbulator induced secondary flow on wall D to enhance the heat transfer. However, on wall C the two secondary flows work counter to each other causing very little enhancement compared to the smooth sidewalls. The turn vortices quickly dissipate in the second passage allowing the turbulator induced secondary flow to enhance the heat transfer on wall C by X/D_h=17. In the third passage, a similar interaction occurs with the turn induced vortices and the turbulator secondary flow. Since the stagger of the turbulators is reversed from the second passage, the counter secondary flow interaction occurs on wall D. The Dean Vortices once again rapidly dissipate allowing the heat-transfer enhancement on wall D to increase at X/D_h=29.5.
Figure 4.18 Skewed Turbulator Passage Combination, AR 1:6

The data imply a changing flow structure generated by the turbulators and turn geometry, as suggested in Figure 4.19. The flow is into the page in passage one, out of the page in passage two and back into the page in passage three. The first passage secondary flow is caused by the turbulators directing the flow toward the outside wall. Due to the staggered arrangement of the skewed turbulators, the panels downstream of the turn begin with either a full-length turbulator or a half-length turbulator, as can be viewed in Figure 3.5. After passing through the tip turn, the vortex caused by the full-length turbulator on the bottom surface re-establishes in location 6 immediately after the turn (as labeled on Figure 3.5) and the turn vortices begin decaying. The combined turn and turbulator secondary flow enhance the heat transfer immediately after the turn on the bottom surface. As noted in Figure 4.18, the impact of the turn vortex dissipates quickly because the two secondary flows work counter to each other. In passage three, the
vortices oppose each other along the bottom surface, preventing the secondary flow caused by the half-length turbulator from developing immediately after the turn.

![Figure 4.19 Turbulated passage flow structure](image)

The combined results for the span of Reynolds Numbers investigated are shown in Figure 4.20. Measurements for five Reynolds numbers are shown with different colors for each flow condition. Again, the Reynolds Number case of 4,000, while close to the typical laminar threshold is still considered fully turbulent. The presence of the turbulators ensures the flow will be turbulent. Furthermore, the Nusselt Number drops to a level sufficiently above the fully developed laminar Nusselt Number. The near wall secondary flow development in the streamwise direction combines with the inlet thermal development region resulting in the maximum enhancement occurring in the middle of each of the three passages.
Figure 4.20 Reynolds Number Comparison, Skewed Turbulator Passage AR 1:6
The combined surfaces in Figure 4.21 for Re=4,000 show the turbulators have minimal influence on the turbulence intensity. The sidewalls and the top and bottom surfaces have minimal difference throughout the first two passages. In the third passage the heat transfer varies due to the low power input and the low temperature difference between the wall and bulk coolant flow. These two factors combine to drive the uncertainty up to nearly 20%.

![Figure 4.21 Skewed Turbulator Passage Combination; Re=4,000, AR 1:6](image)

The small blockage ratio in the 1:6 case caused the turbulators to have minimal effect on the heat transfer plotted in Figure 4.21. However, the secondary flow generated by the turbulators progressively got stronger as the Reynolds Number increased. As shown in Figure 4.22, the effect of the turbulators is distinct in the Re=75,000 skewed turbulator passage. The turbulated surfaces have nearly two times the heat transfer compared to the smooth sidewalls. The heat transfer enhancement for the Re=4,000 case
is 50%, at best. The higher Reynolds Number flows have more inertia aiding in the turbulent mixing of the colder core flow with the warmer near wall coolant. The disproportional relationship of the heat transfer with change in Reynolds Numbers for high aspect ratio profiles suggests there is a minimum inertial requirement to realize maximum heat transfer enhancement.

![Image](Figure 4.22 Skewed Turbulator Passage Combination Re=75,000; AR 1:6)

4.9 Analysis and Comparisons

As part of this research program, multiple comparisons are made to gain insight into the impact of geometry changes, localized heat transfer effects, and comparisons with previous research. As a result of increasing the efficiency of turbo machinery, the size of airfoils has continued to increase. The internal passages have increased from square to rectangular with large aspect ratios. To achieve a better understanding of which design provides the highest heat transfer with the smallest pressure penalty for these
larger internal passages, the size and orientation of turbulators along with the aspect ratio they are installed in are compared.

4.10 Skewed Turbulators vs. Smooth Passage Walls

The skewed turbulators on the leading and trailing surfaces have been found to provide stable heat transfer enhancement for rotating airfoils with minimal pressure penalty. To provide a baseline for the effectiveness of the skewed turbulator in a stationary reference frame, smooth passages were compared with skewed turbulators applied to the top and bottom surfaces for each aspect ratio examined as part of this research. Since the same size turbulators are used for three different aspect ratios three different blockage ratios result.

4.10.1 Smooth Passage and Skewed Turbulator Passage Comparison, AR 1:1

Figure 4.23 shows the heat transferred from the surfaces of the same size passage with and without turbulators applied to the top and bottom surfaces. The heat transfer on the sidewalls differs even though there is no difference in their surface finish. The turbulated top and bottom surfaces are influencing the heat transfer enhancement on the sidewalls especially in the first passage. The smooth passage sidewalls (wall A and B) have higher Nusselt Number in the turns due to development of the stronger Dean Vortices in the first passage. The remaining portion of the serpentine passage shows little difference between the smooth sidewall and the sidewalls in the turbulated passage.

The turbulated top and bottom surfaces (wall C and wall D) enhanced the heat transfer over the smooth top and bottom surfaces by a factor of 2 in the first passage. The
turbulator enhancement is present in the second and third passages, but it is reduced due to the decreasing driving temperature.
Figure 4.23 Smooth Wall and Skewed Turbulator Passage Comparison; AR 1:1, Re=50,000
4.10.2 Smooth Passage and Skewed Turbulator Passage Comparison, AR 1:6

The comparison of the smooth and skewed turbulated surfaces in the smallest blockage ratio configuration is shown in Figure 4.24. The sidewall (A and B) heat transfer plots indicate the heat transfer enhancement is nearly identical for all three passages. The only difference noted is the significant enhancement in the first passage due to the flow development at the entrance of the smooth wall serpentine assembly. The flow development length is shortened by the presence of the turbulators. The enhanced turbulence mixes the cooler core flow with the warmer near wall flow resulting in shorter development lengths. There is minimal difference in the enhancement on sidewall A after the second passage. These differences are small enough that they are within the uncertainty of the Nusselt Number, but the comparison still lends some additional insight into the unbalanced secondary flow development.

The top and bottom surface comparison between the smooth and turbulated passage show distinct turbulator induced enhancement. The first passage turbulated surface has nearly two times the augmentation compared to the smooth surface. Both surfaces have equal enhancement in the first passage unlike the enhancement on the turbulated surfaces in the second passage. As previously noted, the tip turn interaction with the secondary turbulator induced flow resulted in nearly identical heat transfer on wall C immediately after the turn for both the smooth and turbulated surface. The same interaction with the turn vortices and turbulator induced flow is noted in the third passage. However, the effected wall changes from C to D. Again, the interaction on wall
D results in nearly identical heat transfer as the smooth passage immediately after the root turn.

The heat transfer in the turns is similar for both the turbulated and smooth passages. The turns do not have turbulators so the enhancement is due to the turn vortices.
Figure 4.24 Smooth Wall and Skewed Turbulated Passage Comparison; AR 1:6, Re=50,000
4.11 Aspect Ratio Comparison

The increase in blockage ratio ($e/D_h$) of the internal passages in an airfoil enhances the heat transfer due to the increased turbulence induced mixing that results. There is, however, a pressure penalty that results from the increased blockage. The following discussion compares the difference in heat transfer for three different blockage ratios ranging from 0.06 for AR 1:6 to 0.1 for AR 1:1.

Since these plots are comparing measurements from the same X location but with different hydraulic diameters, the plotted X/$D_h$ locations have been adjusted slightly so that the turns and the entrance of each passage are aligned between aspect ratios.

4.11.1 Aspect Ratio 1:1 and 1:2 Skewed Turbulator Passage Comparison

The two aspect ratios with high blockage ratios are compared first. The ratios are compared using the Nusselt Number and dimensionless passage length in Figure 4.25 for Re=50,000. Overall there is minimal difference in heat transfer between the two aspect ratios. The heat flux for aspect ratio 1:2 is greater than 1:1, but inclusion of the hydraulic diameter in the Nusselt Number calculation eliminates the heat flux difference. The sidewall comparison between AR 1:1 and 1:2, indicates the heat transfer enhancement from the presence of the turbulators has more of an influence in the smaller AR 1:1 in the first passage. The smaller passage has a higher blockage ratio, which enhances the turbulence intensity and increases the coolant velocity.

Aspect ratio 1:2 has higher heat transfer in the turns than the smaller aspect ratio. The difference is attributed to the Dean Vortices since they are able to develop more in the larger cross sectional area. The counter rotating vortices are more circular in shape.
resulting in stronger rotation and heat transfer augmentation. The combined stronger turn vortices and reduced blockage ratio results in higher heat transfer in the second and third passages on the turbulated surfaces. There is a small difference on the top and bottom surfaces in the second passage as the strength of the Dean Vortices dissipates through the second passage. The third pass heat transfer enhancements are too close to determine if there is a notable reason. The uncertainties overlap in the third passage for the Re=50,000 case.
Figure 4.25 Aspect Ratio 1:1 to 1:2 Heat Transfer Comparison, Skewed Turbulator Passage
4.11.2 Aspect Ratio 1:1 and 1:6 Smooth Passage Comparison

The two smooth passage aspect ratio configurations (AR 1:1 and AR 1:6) are compared in Figure 4.26 for Re=50,000. Since neither of these passages have turbulators, all four walls behave in a similar manner. The Nusselt Number is significantly higher for the 1:6 aspect ratio for the first two points in the first passage because the six-inch long unheated entrance upstream of the passage is only seven hydraulic diameters in length for the 1:6 configuration compared to twelve for the 1:1 configuration. Therefore, the flow is less developed for the 1:6 aspect ratio and the heat transfer is increased. This difference is more pronounced on the sidewalls since the flow is developing from a round supply pipe centered on the rectangular passage.

The enhancement in the tip turn is approximately the same for both configurations. In the root turn the heat transfer on the top and bottom surfaces is 1.5 times greater for the larger 1:6 AR. As the flow develops for the high aspect ratio top and bottom surfaces, the heat-transfer enhancement decays to the AR 1:1 level. The sidewall heat transfer for AR 1:6 is larger than AR 1:1 throughout all three passages. The larger sidewall heat transfer and larger development length results in greater overall heat-transfer enhancement for the AR 1:6 smooth passage configuration.
Figure 4.26 Aspect Ratio 1:1 and 1:6 Heat Transfer Comparison, Smooth Passage
4.11.3 Aspect Ratio 1:1, 1:2 and 1:6 Turbulated Passage Comparison

The same comparison for Re=50,000, but with turbulated top and bottom surfaces and additional data from the 1:2 aspect ratio configuration is presented in Figure 4.27. The heat-transfer enhancement in the first passage is similar among all four walls and all three aspect ratios. The addition of the turbulators shortens the flow development length significantly compared to the smooth wall case, so the effect of passage size on flow development in the first passage is nearly eliminated. However, in the turns and the second and third passages, where turn vortices develop, the turbulator blockage ratio is more important. The greatest heat transfer is measured for the 1:6 aspect ratio (e/Dh=0.06) and the smallest heat transfer for the 1:1 aspect ratio (e/Dh=0.10). In the confined passage of the AR 1:1, the turn induced Dean Vortices are not able to form due to strong opposition from turbulator-induced secondary flows. The larger AR 1:2 passage allows the Dean Vortices to develop higher strength rotation, but the disruption from the turbulator-induced vortices dissipates them quickly in the second passage. For AR 1:6, the larger area allows more independent development of the turn-induced Dean Vortices and the near-wall turbulator-induced vortices, as illustrated in Figure 4.19. The stronger Dean Vortices cause the increased heat transfer observed for the 1:6 aspect ratio for the sidewalls in the second and third passages. In addition, the interaction of these stronger vortices causes larger differences among the aspect ratios for the top and bottom walls.
Figure 4.27 Aspect Ratio 1:1, 1:2 and 1:6 Heat Transfer Comparison, Skewed Turbulator Passage
Figure 4.28 continues with this aspect ratio comparison for a Reynolds Number of 10,000 and a skewed turbulator passage. At this lower Reynolds Number, the secondary flow created by the turbulators is weaker, and the balance with the Dean Vortices is shifted. In the first passage, the flow behaves almost as if the low blockage-ratio turbulators are not installed. The heat transfer on the turbulated surfaces for the AR 1:6 configuration is 1.5 times less than the AR 1:1 configuration and nearly two times less on the sidewalls. In the second passage, the reduced turbulator-induced flows are offset by the strengthened Dean Vortices resulting in similar Nusselt Numbers for all four walls. However, when compared to the Re=50,000 case (Figure 4.27), where the highest Nusselt Number occurred in the second and third passages and reached up to 450, the heat transfer is clearly reduced for this lower Reynolds Number. In the third passage, the AR 1:2 configuration achieves the highest Nusselt Number and the best compromise between turn-induced and turbulator-induced vortices for this low Reynolds Number condition.
Figure 4.28 Turbulated Passage Comparison, Re=10,000
4.12 Tip Turn Induced Separation Bubble

As noted earlier, the turn induced vortices develop roughly halfway around the turn. As the coolant exits the first turn these vortices are forced toward the outside wall by a separation bubble that develops due the turning geometry. The heat transfer on the sidewalls where the re-attachment of the separation bubble has the largest influence is compared for the aspect ratio 1:1 and 1:6 cases for both smooth and turbulated passages. Since the dimensionless divider thickness is larger than the divider thickness shown in Figure 2.4, the separation bubble is most likely pulled into the turn along the inside turn wall. The reattachment of the separated flow occurs on the inside wall (wall B) immediately after the turn because the bubble is in the tip turn. Figure 4.29 provides a graphical interpretation of the flow exiting the first turn and reattaching onto wall B.
The heat transfer along the outside sidewall (wall A) for passage two is shown in Figure 4.30. The heat transfer results are presented with respect to the individual segments along the passage in the streamwise direction. The larger cross sectional area of the AR 1:6 allows the turn vortices to influence the heat transfer on wall A resulting in a 40% higher heat transfer at Re=75,000 over the other three segments in the second
passage. Even with a much smaller cross section, the AR 1:1 configuration exhibited elevated heat transfer on the first segment due to the turn vortices.

Figure 4.30 Smooth Second Passage Sidewall A Aspect Ratio Comparison

The heat transfer enhancement on the inside wall immediately after the turn is shown in Figure 4.31. The separation bubble re-attachment zone in the first segment of the second passage causes increased heat transfer in this area. The high aspect ratio reattachment zone stretched into the second segment unlike for the AR 1:1 case.

Figure 4.31 Smooth Second Passage Sidewall B Aspect Ratio Comparison
For the skewed turbulator case, wall A heat transfer enhancement is shown in Figure 4.32. Again, the larger cross sectional area of the AR 1:6 passage allows the turn vortices to exit into the second passage increasing the heat transfer on the first segment outside wall, but these vortices quickly dissipate. By the second segment the heat transfer has dropped to match the rest of the passage. The turbulators had minimal influence on the sidewall heat transfer for the high aspect ratio passage. The high blockage in the AR 1:1 passage does not exhibit the distinctive heat transfer difference between segment one immediately after the turn and the rest of the segments.

![Figure 4.32 Skewed Turbulator Second Passage Sidewall A Aspect Ratio Comparison](image)

Unlike wall B in the smooth passage, the turbulators in the AR 1:1 case mix the coolant causing the reattachment heat transfer enhancement to be absent across all four wall segments as shown in Figure 4.33. This disruption of flow in the boundary layer was also noted by Liou et al. [60]. However, the heat transfer enhancement is prominent
in the AR 1:6 low blockage ratio first segment. For the Re=75,000, the enhancement is roughly 30% larger than the rest of the sidewall segments.

![Figure 4.33 Skewed Turbulator Second Passage Sidewall B Aspect Ratio Comparison](image)

4.13 Local Effects

To better understand the differences in specific areas along the serpentine assembly, locations were selected and compared for the three blockage ratios and two surface conditions. Two positions were selected in the first two passages sufficiently far away from the entrance in the first passage and the exit of the turn in the second passage. After entering the serpentine assembly in the first passage and after the tip turn in the second passage, these two positions also allow the turbulator induced secondary flows more distance to develop. The first two passages were chosen due to the relatively higher driving temperature when compared to the third passage. In the first passage, the second
segment top and bottom surface locations are shown in Figure 4.34 along with the second segment location in the second passage.

![Diagram of Regional Heat Transfer](image)

**Figure 4.34 Regional Heat Transfer**

### 4.13.1 Nusselt Number Comparison in 1\textsuperscript{st} Passage and 2\textsuperscript{nd} Passage, 2\textsuperscript{nd} Segment, Smooth Passage

Results for wall C and D are shown in Figure 4.35 and Figure 4.36 because they represent all four walls since all four walls have equal surfaces conditions. Figure 4.35 shows similar results for both aspect ratios until the Reynolds Number exceeds 50,000. This result is due to the added development length for the larger hydraulic diameter.
The second passage results shown in Figure 4.36 are similar to the results for the first passage. The results are consistent with the flow development length out of the turn being longer for the larger Reynolds Numbers and for the bigger cross-sectional area case. This combination of the physical flow environment and Reynolds Number causes the divergence as the Reynolds Number increases.
4.13.2 Nusselt Number Comparison in 1\textsuperscript{st} Passage and 2\textsuperscript{nd} Passage, 2\textsuperscript{nd} Segment, Turbulated Passage

For the turbulated passage comparisons, only the top and bottom surfaces are compared in Figure 4.37 and Figure 4.38 because the sidewalls had similar heat transfer enhancement as the smooth passage shown in the previous two figures (Figure 4.35 and Figure 4.36). The following figures show the heat transfer comparison for skewed turbulator passages for the same locations as the smooth passage case just described. In Figure 4.37, the first passage with the high aspect ratio configuration lags behind the
other two aspect ratios below Re=50,000 for both the top and bottom surfaces. For the lower Reynolds Numbers the turbulators are not as effective for the 1:6 aspect ratio case until the Reynolds number increases.

![Graph showing heat transfer](image1)

**Figure 4.37 Top and Bottom Surface 1st Passage, 2nd Segment Localized Skewed Turbulator Passage Heat Transfer**

The second passage combines the development from the turbulators with the secondary flow developed in the turn. Aspect ratio 1:2 has higher heat transfer than AR 1:1 due to the ability for the tip turn vortices being nearly circular resulting in higher strength and the favorable secondary flow from the turbulator direction. The advantage of the high aspect ratio, higher Reynolds Number in the secondary flow is clear in Figure 120.
4.38. The heat transfer enhancement is 50% larger than AR 1:2 at Re=75,000 for both walls. The turbulent mixing caused by the tip turn vortices and the turbulator secondary flow combine to enhance the heat transfer. This result suggests that higher aspect ratios with blockage ratio similar to the 0.06 ratio used for this experiment are preferred especially after a tip turn for Reynolds Numbers in excess of 25,000.

Figure 4.38 Top and Bottom Surface 2nd Passage, 2nd Segment Localized Skewed Turbulator Passage Heat Transfer
4.13.3 Localized Sidewall Heat Transfer Enhancement

The design of the serpentine model incorporates multiple rows of copper panels to make up the sidewall. This allows heat transfer to be localized not only along the passage, but also vertically. Figure 4.39(a) shows the backside of the tip turn and passage number one (wall A). The bottom row of copper panels is used for the AR 1:1 runs. When the top row of copper panels is exposed to the coolant along with the bottom row they make up AR1:2 sidewall. Figure 4.39(b) shows the plan view of the serpentine model. The areas selected for localized sidewall heat transfer are noted on the figure.
Figure 4.39 (a) Side View First Passage and Tip Turn (b) Plan View
Note: Garolite Insulation Removed from Side View for Better Visibility

Figure 4.40 (a) and (b) show the coolant entering the first passage has equally elevated enhancement on both sidewall rows. However, the heat transfer enhancement separates as the turbulator secondary flow develops. In segment three (Figure 4.41(a)), the heat transfer on the top of wall A is 25% greater than the bottom row for Re=75,000. The trend continues through the fourth segment of the first passage. On wall B, the
opposite occurs as shown in Figure 4.41 (b). The higher heat transfer occurs on the bottom wall. The difference takes longer to develop on wall B. By the third segment it is 20% greater on the bottom wall for $Re=75,000$.

![Figure 4.40 First Passage, First Segment Sidewall Heat Transfer Enhancement](image)

Figure 4.40 First Passage, First Segment Sidewall Heat Transfer Enhancement
The higher heat transfer on row 2, wall A continues into the tip turn. The secondary flow developed in the turn combines with the turbulator-induced flow. As a result, the heat transfer is enhanced on both sidewalls. Like the entrance to the first passage, there is minimal difference between the two rows in the second passage. The interaction between the turbulators and turn secondary flow pattern prevents the same heat transfer enhancement from developing in the second passage.

The coolant exits the second turn and impinges on the outside wall (wall A) with the separation bubble reattachment occurring on wall B. The heat transfer on the
sidewalls in the first segment is higher than the rest of the wall segments (Figure 4.42 (a) and (b)). The difference in the heat transfer enhancement increases with Reynolds Number. At 130,000, the difference is approximately 20%. So the first segment heat transfer enhancement is due to a combination of the Dean Vortices being pushed toward to outside wall and the separation bubble reattachment. Segments two through four, however, are only dependent on change in Reynolds Number. They have roughly equal heat transfer enhancement at specific Reynolds Numbers, which indicates the turn vortices have dissipated in strength by the second segment in the second passage (Figure 2.3). The coolant enters the root turn without any specific bias toward a sidewall. In the root turn, the coolant moves toward the top row as it moves through the turn. The higher heat transfer on the top row does not extend into the third passage. The turn secondary flow and turbulators again interact to provide equal enhancement along the third passage sidewalls.
4.14 Comparison with Published Data

To compare this data with other studies besides HOST, it is helpful to plot the Nusselt Number averaged over the entire passage normalized by the Dittus-Boelter/McAdams correlation, as shown in Figure 4.43. Additional data for two passage assemblies extracted from figures in Liu et al. [18], Huh et al. [17], and Zhou [76] are also included on this figure. The Zhou data was included to compare similar hydraulic diameters with turbulators applied to the wider wall. The plot indicates that while there
are some differences in detail among the experiments, the overall levels and trends are similar.

As the Reynolds number increases beyond an inflection point, the change in normalized Nusselt Number behaves almost linearly for nearly all conditions. The slope of this decay varies by aspect ratio. Data are available for the 1:2 aspect ratio configuration up to a maximum Reynolds Number of 130,000, nearly doubling the range of previously available data. The linear slope of the Normalized Nusselt Number shows that the heat transfer is still governed by the power law relationship over this wide range.

In contrast, data for the 1:6 aspect ratio configuration show a slight increase in normalized Nusselt Number between a Reynolds Number of 25,000 and 50,000. This is consistent with the change in behavior with Reynolds Number which will be discussed in section 4.15.5 (Figure 4.55). It shows that for the passage geometry studied here, there are still changes taking place in the overall flow structure and that each flow condition of interest should be investigated separately.
Figure 4.43 Heat Transfer Enhancement Comparison, Skewed Turbulator Passage

The 1:2 aspect ratio program performed by Liu et al. utilized a two passage experimental assembly [18]. The non-dimensional length of the assembly was $X/D_h=19$. The stationary data, scaled from a figure, with a $P/e=10$ was used for this comparison. The 1:2 aspect ratio under investigation for this program has a non-dimensional length double that of Liu’s assembly. In addition, the results reported in this thesis contain experiments with Reynolds Number in excess of 3 times Liu’s.

The 1:4 aspect ratio research was performed by Huh et al. [17]. It was also made up of a two-passage assembly. This assembly passage length is equivalent to the research program performed by Liu. The stationary data, scaled from a figure, had a $P/e=10$ with turbulator skewed $45^\circ$ to the flow. This aspect ratio was not examined during this research program, but it is capable of being reconfigured for this aspect ratio.
The research performed by Zhou [76] based on a 4:1 aspect ratio assembly was included to allow comparisons of the same hydraulic diameter and blockage ratio (e/Dh=0.1) as those used by the OSU and Texas A&M research programs. The data was scaled from a figure at Ro=0.0. The assembly used for this research was similar in length to the OSU assembly, but the sidewalls were not heated.

For all the data sets shown in Figure 4.43, the heat transfer enhancement rapidly increases as the Reynolds Number decreases below 20,000 due to the Dittus-Boelter/McAdams correlation going to zero faster than the heat transfer data. For this reason, an alternate correlation has been proposed. The increase in the normalized Nusselt Number with a drop in Reynolds Number is also noted by Fu et al. and AL-Hadhrami et al. [3, 77]. The Dittus-Boelter/McAdams equation is for fully developed turbulent internal flow above Re=10,000. Due to the short length of the passages and the turns, the average Nusselt Number for the entire passage for both smooth and turbulated passages is in excess of the traditional Dittus-Boelter/McAdams correlation. The Reynolds numbers examined for this thesis work were in several instances below the acceptable range for the Dittus-Boelter/McAdams correlations. Another correlation which is suitable for Reynolds Numbers below 10,000 is the Gnielinski correlation [78].

\[
NU_D = \frac{(f/8)(Re_D-1000)Pr}{1+12.7(f/8)^{1/2}(Pr^{2/3}-1)}
\]  

\[
f = (0.790 \ln Re_D - 1.64)^{-2}
\]

\[3000 < Re_D < 5 \times 10^6\]
The Gnielinski correlation resulted in similar spread of the data as the Dittus – Boelter/McAdams correlation. Both correlations approached zero faster than the experimental results. This resulted in the lower Reynolds Number runs appearing to have a higher enhancement than the higher Reynolds Number cases. Because of this tendency, the alternate correlation proposed above was also used to normalize the data. This alternate correlation, shown in Figure 4.44 along with the Dittus-Boelter/McAdams correlation, was determined using a least squares curve fit of the smooth passage data. Since the turbulated passages have higher overall averaged heat transfer enhancement, there is a multiplier of 1.15 for the skewed turbulator cases as shown in Figure 4.45. The Prandtl number is considered constant at 0.71. The correlation was determined to be the following:

\[ Nu = 0.11 Re^{0.68} \] (18)

The correlation is of the same form as the Dittus-Boelter/McAdams relationship (\(Nu \propto Re^X\)), but the exponent of 0.68 more closely matches the correlation from Mayle with an exponent of 0.65 [79].

The authors of the HOST experiment noted that the Dittus-Boelter/McAdams correlation did a good job collapsing the data and eliminating the Reynolds Number effect, which makes sense on Figure 4.44 since the HOST data is parallel to the Dittus-Boelter/McAdams relationship. However, Fu et al. [3] noted the drop in the normalized Dittus-Boelter/McAdams data set as the Reynolds number increases similar to the trend noted for this measurement program.
Figure 4.44 Heat Transfer Enhancement Comparison with Alternate Correlation, Smooth Passage

The work done by Liu et al. [15] is for the smooth wall and is similar to the previous Texas A&M work. The AR 1:4 configuration was made up of two passages. Each passage length was shorter than the OSU design and similar to their previously published programs so that it would fit into their spinning rig.

For the skewed turbulator configuration the same correlation can be applied with an enhancement factor of 1.15. Shown in Figure 4.45, the alternate correlation with the enhancement factor covers a wide range of aspect ratios. Again, the HOST data has a slope, which is similar to the Dittus-Boelter/McAdams correlation. This close agreement led to the propagation of the correlation used for airfoil research.
4.15 Normalized Results

In order to better understand the effect (other than Reynolds Number) on flow behavior the following figures present measurements for each Reynolds Number with the Nusselt Number normalized (different colors for each flow condition). The Nusselt Number was normalized using both the Dittus-Boelter/McAdams correlation and the alternate correlation. The comparison of the two normalizing correlations is shown to emphasize the usefulness of the alternate correlation.

4.15.1 Aspect Ratio 1:1, Smooth Passage

The Nusselt Numbers calculated from AR 1:1 smooth passage runs were normalized using the Dittus-Boelter/McAdams correlation to provide enhancement
factors comparable to allow comparisons to other research programs. Each wall is shown individually in Figure 4.46.
Figure 4.46 Aspect Ratio 1:1, Smooth Passage Normalized Nusselt Number
The same data normalized using the alternate correlation is shown in Figure 4.47. The alternate correlation does a better job removing the Reynolds Number effect from the data so that other effects may be isolated. For example, the outside wall of both turns exhibits a heat transfer enhancement that is not strictly a function of the Reynolds Number power law relationship. The developing turn vortices cause different heat transfer response for the five different Reynolds Numbers shown. Also, wall B in the third passage exhibited a heat transfer response which did not collapse due to the increased uncertainty of the data for the low heat transfer associated with low Reynolds Number runs.
Figure 4.47 Aspect Ratio 1:1, Smooth Passage Alternate Correlation Normalized Nusselt
4.15.2 Aspect Ratio 1:1, Skewed Turbulator Passage

The normalized skewed turbulator data for the high blockage ratio passage are shown in Figure 4.48. Again, the Dittus-Boelter/McAdams correlation was used to normalize this data. Like the smooth wall case, it is difficult to discern any differences in the heat transfer enhancement data, which are not just a function of the Reynolds Numbers. However, the Dittus-Boelter/McAdams does a better job of collapsing the higher Reynolds number data for the skewed turbulator passage configuration than it did for the smooth passage configuration (Figure 4.46).
Figure 4.48 Aspect Ratio 1:1, Skewed Turbulator Passage Normalized Nusselt Number
The skewed turbulator data for the AR 1:1 configuration was normalized using the alternate correlation. The data collapsed much better using the alternate correlation as can be seen from Figure 4.49. Like Figure 4.47, the inconsistent enhancement is immediately apparent. Unlike Figure 4.47, the turn vortices do not develop in the AR 1:1 configuration because the high turbulator blockage. The significant differences noted on all four walls in Figure 4.49 are due to the increased uncertainty in the third passage for the low heat transfer runs (Re=13,000). The differences in the heat transfer augmentation for the turbulated surfaces and the smooth sidewall are nearly two times the smooth sidewalls. However, the augmentation due to the turbulator quickly diminishes farther down the serpentine passage. By midway through the second passage, the enhancement is nearly equivalent.
Figure 4.49 Aspect Ratio 1:1, Skewed Turbulator Passage Alternate Correlation
Normalized Nusselt

(a) Wall A
AR 1:1
Nu=0.11Re^{0.68}

(b) Wall B
AR 1:1
Nu=0.11Re^{0.68}

(c) Wall C
AR 1:1
Nu=0.11Re^{0.68}

(d) Wall D
AR 1:1
Nu=0.11Re^{0.68}
4.15.3 Aspect Ratio 1:2, Skewed Turbulator Passage

The AR 1:2 Nusselt Number was normalized, again, using the Dittus-Boelter/McAdams correlation. As noted for the skewed turbulator AR 1:1 configuration, the Dittus-Boelter/McAdams correlation collapsed the data better for the Reynolds Numbers above 25,000. Figure 4.50 shows the Re=10,000 case with significantly higher enhancement than the other five cases. This separation is due to the validity of the Dittus-Boelter/McAdams correlation discussed above. Even with the Gnielinski correlation the trend is similar.
Figure 4.50 Aspect Ratio 1:2, Skewed Turbulator Passage Normalized Nusselt Number
Comparing Figure 4.50 and Figure 4.51, the significant difference in enhancement noted for the Re=10,000 case has been eliminated. Also, the difference in enhancement between the turbulated and smooth walls is evident. The turbulated surfaces achieve an enhancement two times greater than the sidewalls. While most of the data collapses using the alternate correlation, the area immediately after the second turn on wall D exhibits an enhancement that is not consistent with the fractional power law dependence on the Reynolds Number. This difference is due to the strength of the rotation vortices. With the larger cross sectional area, the vortices are able to strengthen more than the higher blockage ratio AR 1:1 configuration. The results of Figure 4.50 shows that the enhancement for the Re=10,000 case is the largest in the third passage for all the walls, but Figure 4.51 indicates the minor differences are due to the increased uncertainty for the lower Reynolds number run where there was a smaller driving temperature. Overall, Figure 4.51 shows the alternate correlation collapses the data for all the Reynolds Numbers including those in excess of 100,000 showing the design correlations used for lower Reynolds Numbers may be extended to Reynolds Numbers as high as 130,000.
Figure 4.51 Aspect Ratio 1:2, Skewed Turbulator Passage Alternate Correlation
Normalized Nusselt Number
4.15.4 Aspect Ratio 1:6, Smooth Passage

For the AR 1:6 smooth passage configuration shown in Figure 4.52, the Nusselt Number is normalized using the Dittus-Boelter/McAdams correlation. The results for Reynolds Numbers above the validity range of the Dittus-Boelter/McAdams correlation collapse fairly well. The Dittus-Boelter/McAdams correlation is used to normalize the data for the low Reynolds Number cases for consistency. As noted previously, the Gnielinski correlation results in similar enhancement.

With the inclusion of the low Reynolds Number cases, one must be careful in reaching conclusions from the data. The differences in heat transfer between the different Reynolds Numbers can be examined only after the heat transfer has been normalized using the alternate correlation. The calculated Nusselt Number for the AR 1:6 smooth passage has been normalized using the alternate correlation and the result is shown in Figure 4.53. Like the smooth AR 1:1 passage, the sidewall in the turns exhibit enhancement not directly correlated to the Reynolds Number. As expected, the smooth passage configuration enhancement is nearly equal for all four walls and equal to unity. All the walls exhibit similar heat transfer enhancement in the first passage due to the long development length given the large hydraulic diameter.
Figure 4.52 Aspect Ratio 1:6, Smooth Passage Normalized Nusselt Number
Normalization using the alternate correlation, shown in Figure 4.53, collapses the sidewall data at most locations for each wall condition, indicating that the heat transfer increases due to a direct relationship to Reynolds Number. As the flow develops in each passage, the enhancement drops to near one, matching fully developed flow in a smooth tube. There are small differences between the Reynolds Numbers on the sidewalls due to the turn vortices.
Figure 4.53 Aspect Ratio 1:6, Smooth Passage Alternate Correlation Normalized Nusselt Number

Nu/Nu = 0.11Re ^ 0.68

(a) Wall A

(b) Wall B

(c) Wall C

(d) Wall D
4.15.5 Aspect Ratio 1:6, Skewed Turbulator Passage

The Nusselt Number for the AR 1:6 skewed turbulator passage normalized by the Dittus-Boelter/McAdams correlation is shown in Figure 4.54. Like the other Dittus-Boelter/McAdams normalized data, only general conclusions are possible.
Figure 4.54 Aspect Ratio 1:6, Skewed Turbulator Passage Normalized Nusselt Number
Using the alternate correlation shown in Figure 4.55, the significance of the higher Reynolds Number enhancing the turbulator’s effectiveness is apparent. The figure shows a clear difference in enhancement for the Reynolds Numbers above 25,000. The low blockage ratio for the low Reynolds Number case gets little enhancement from the presence of the turbulators. The enhancement resembles a smooth passage case more than a turbulated passage. The augmentation is approximately one for all four walls for the low Reynolds Number case matching the smooth passage enhancement shown in Figure 4.53. Like the previous turbulated configurations, the enhancement in the first and second passage due to the turbulated surfaces is close to two times for Reynolds Numbers above 25,000. The sidewall data collapses, indicating the heat transfer increased based on the power law correlation. However, there are small differences between the Reynolds Numbers on the sidewalls due to the turn vortices. The turbulated surfaces enhance the Nusselt Number by two times for the first and second passages on walls C and D. As noted, there is a clear difference between the low Reynolds Number cases (10,000 and 25,000) and the higher Reynolds Number cases (50,000 and 75,000). The higher Reynolds Number cases show significantly greater enhancement, indicating that the flow behavior has changed in a way not fully captured by the Dittus-Boelter/McAdams normalization. This likely indicates a shift in the balance between the turbulator-induced vortices and the turn-induced vortices. This shift is characterized by the Reynolds Roughness Number.
Figure 4.55 Aspect Ratio 1:6, Skewed Turbulator Passage Alternate Correlation
Normalized Nusselt Number
4.16 Frictional Losses

The pressure drop measured through the entire passage was small for all aspect ratios and surface conditions investigated as shown in Figure 4.56. In some cases such as the AR 1:6 smooth wall, the pressure drop was so low across the entire three-pass assembly that it was within the calibration limit of the pressure transducers (±0.345 kPa with an uncertainty of ±0.483 kPa). The pressure drop increased with increasing Reynolds Number. The pressure drop was higher for the smaller aspect ratio passages and for cases with the turbulators installed. The largest pressure drop occurred for the smallest aspect ratio 1:1 with the skewed turbulators installed. Overall the pressure drop for all the configurations falls nicely within an exponential relationship.
The pressure penalty was determined by normalizing the dynamic pressure with the simplified Kármán-Nikuradse equation as shown in Figure 4.57. The AR 1:1 skewed turbulator passage had the highest pressure penalty of all the configurations. The pressure penalty started at 2.5 times that of a smooth wall passage at Re=13,000 to in excess of 6.5 times a smooth wall fully turbulent passage by Re=75,000. The normalized friction approached one for the lower Reynolds Numbers and larger aspect ratios.
The overall thermal performance of all the configurations and surface conditions reviewed is shown in Figure 4.58. The thermal performance is based on the overall averaged Nusselt Number for the entire passage for each configuration. The Nusselt Number was normalized using the Dittus-Boelter/McAdams correlation. Even though the heat transfer is similar for aspect ratios 1:1 and 1:2 as shown in Figure 4.25 and Figure 4.27, the thermal performance of the larger aspect ratio degrades less than the smaller aspect ratio as the Reynolds Numbers increases. For all aspect ratios the thermal performance decreased with increasing Reynolds Number. As noted, some of the drop in thermal performance with increase in Reynolds Number is due to the Dittus-Boelter/McAdams going to zero faster than the experimental data. The largest aspect ratio 1:6 configuration for which pressure data is available resulted in thermal
performance equal to AR 1:2. Thus the turbulated high aspect ratio, low blockage ratio and low Reynolds Number configurations are the best method for transfer of heat from the walls.

Figure 4.58 Thermal Performance

4.17 Theoretical Analysis

The comparison of the theoretical weighted area for the smooth and turbulated surfaces for AR 1:1 and AR 1:6 is shown in Figure 4.59. Since smooth passage data was not taken for AR 1:2, the smooth wall friction factor is not available for use in equation (10). The theoretical analysis comparison with the alternate correlation is within 10% for AR 1:1 and 15% for AR 1:6. The deviation from the theoretical estimate in the AR 1:6 configuration is most likely due to the small pressure difference in the large aspect ratio

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passage. Due to this close agreement, the theoretical Nusselt Number for a serpentine passage can be determined based off the dimensionless geometric parameters, which describe the passage along with the Prandtl Number and Reynolds Number.

Figure 4.59 Theoretical Analysis Comparison, Skewed Turbulator Passage
CHAPTER 5
CONCLUSIONS

This experimental program significantly extends the range of geometry and flow conditions available for a serpentine cooling passage and lays the groundwork for future experiments for which the serpentine passage will be rotated at engine-like Rotation Numbers. The normalized Nusselt Number for the aspect ratio 1:2 case was demonstrated to change linearly with Reynolds Number up to Re=130,000. In contrast, experiments with the novel aspect ratio 1:6 configuration showed that for this low blockage ratio, there is a significant shift in flow behavior between a Reynolds Number of 25,000 and 50,000 due to the increased roughness. This indicates that while basic design tools can be useful for interpolating between operating points, it is necessary to specifically investigate new geometry configurations to avoid surprising deviations in heat transfer performance.

This research is a significant addition to the stationary data set for high aspect ratio internal heat transfer with serpentine configurations consistent with current engine designs for cooled turbine blades. The key findings are as follows:

1. The largest aspect ratio configuration had the highest heat transfer for both the smooth passage and the skewed turbulator passage at Reynolds Numbers above 25,000.
2. The blockage ratio for the AR 1:6 allowed the turn induced vortices to develop with the secondary induced turbulator flow enhancing the heat transfer in the second passage.

3. The heat-transfer enhancement for the AR 1:2 configuration continued to follow the power law dependence ($\text{Nu} \propto \text{Re}^x$) as the Reynolds Number was increased to 130,000.

4. The turbulator induced secondary flow grows in strength as the Reynolds Number increases for AR 1:6. For low Reynolds Numbers the turbulated surfaces mimic smooth wall heat-transfer enhancement.

5. The data compared well with other published data and a new alternate correlation has been proposed for normalizing the heat transfer to eliminate the misleading enhancements at lower Reynolds Numbers.

6. The secondary turbulator induced flow development in AR 1:6 is notably increased for Reynolds Numbers above 50,000.

In summary, for the new 1:6 aspect ratio configuration, the level of heat transfer is strongly dependent on the balance of the interaction between the turbulator-induced vortices and the turn-induced Dean Vortices. The Nusselt Number distribution is substantially more dependent on Reynolds Number than the lower aspect ratio configurations since it almost behaves as a smooth passage until the Reynolds Number reaches 50,000 and the turbulators begin to create their own strong vortices. There is much to be gained from investigating high aspect ratio passages since a modern cooled
turbine airfoil will have a wide range of aspect ratios within a single airfoil, some of which are well outside the current database.
REFERENCES


