Forced Convective Heat Transfer to Supercritical Water

in Micro-Rocket Cooling Passages

by

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Abstract
An investigation of heat transfer to supercritical fluids in micro-channels was completed to assess the cooling characteristics of the MIT micro-rocket engine. Previous results from supercritical ethanol heat transfer tests were compared to water tests to establish a baseline for future fuel testing. Existing literature on supercritical heat transfer was also consulted to corroborate the water test results. It was found that the characteristics of the water tests matched those observed in the literature, as well as those of ethanol tests run at similar conditions.

Thesis Supervisor: Jack L. Kerrebrock
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Chapter 1

Introduction

1.1 Background
Microfabrication techniques used to manufacture silicon microprocessors are now being applied to microelectrical and mechanical systems (MEMS). MEMS technology is capable of manufacturing many wafers of planar geometry simultaneously. These wafers are then combined to form 3-D devices. The Massachusetts Institute of Technology Gas Turbine Lab (MIT GTL) has applied this technology to propulsion systems such as the MIT micro-gas turbine and the MIT micro-rocket. MIT is currently testing a bi-propellant regeneratively cooled μ-rocket engine. The current μ-rocket is a nozzle, combustion chamber, and fuel and cooling passages etched onto a silicon chip approximately 8 mm by 12 mm and weighing 2 g. It is designed to produce 15 N of thrust.

There are several advantages to applying MEMS technology to a rocket engine. They can be manufactured in large quantities in a short period of time at low cost, and the strength of silicon lends itself to large thrust to weight ratios. The μ-rocket will ultimately consist of a nozzle, pumps, and valves packaged on the same chip. This will eliminate the need for integration of engine components. Multiple engine packages can be added to a space vehicle to increase the total thrust.

Since the combustion chamber walls are made from silicon, the surface temperature is limited to approximately 1000 K. The chamber pressure will be approximately 125 atm and the heat flux at the walls is expected to reach values as high as 200 W/mm². Heat transfer is therefore a critical issue in the design of the μ-rocket. Heat transfer experiments were necessary to collect data on the behavior of fluids in micro-channels at conditions above the critical point. Ethanol experiments were completed by Jacob Lopata in Septem-
ber, 1998. Ethanol is one of the possible fuels being considered for the μ-rocket, and was chosen because it is relatively safe to test in the MIT GTL facilities.

1.2 Motivation

Because the wall temperature of the chamber and nozzle must be kept relatively low, heat transfer in the wall cooling passages is a primary concern in the design of the μ-rocket. In addition, the high heat fluxes and high pressures mean that the coolant will be at supercritical conditions. Research on heat transfer to supercritical fluids flowing in circular tubes exists, however, the heat fluxes used in these experiments are small compared to what the coolant will see in the μ-rocket, and the tube dimensions are several orders of magnitude larger than the μ-rocket cooling passages. Furthermore, the ethanol tests were originally run because no prior research on heat transfer to supercritical ethanol was available. The ethanol data were checked against an empirical correlation for supercritical helium. Data and empirical correlations do exist for supercritical water, however. It was decided that water should be tested in the μ-rocket heat transfer test rig because the results could be compared to previous water heat transfer research.

1.3 Objective

The objective of this research was to establish a baseline for the ethanol heat transfer tests by running heat transfer tests on supercritical water and applying a single data reduction scheme to both fluids. Heat transfer data on supercritical water flowing through circular tubes is available, so the results of the water tests can be compared to results from previous research. The water tests also establish a benchmark for future heat transfer tests of other possible coolants: JP-7, hydrogen peroxide (H₂O₂), hydrazine (N₂H₄), and nitrogen tetroxide (N₂O₄).
1.4 Supercritical Fluid Properties

The coolant in the μ-rocket will be at supercritical pressures and temperatures because of the high chamber pressures envisioned for the devices. When a fluid is above critical conditions, there is no phase transition between liquid and vapor. The critical point is the pressure and temperature at which a phase change will no longer take place, as shown in figure 1.1. The critical conditions for water and ethanol are listed in table 1.1.

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Pressure (bar)</th>
<th>Temperature (C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>220.9</td>
<td>374.14</td>
</tr>
<tr>
<td>Ethanol</td>
<td>62.55</td>
<td>242.85</td>
</tr>
</tbody>
</table>

Table 1.1: Critical Conditions

Figure 1.1: P-V plot for a typical fluid

---

13
Supercritical fluids are characterized by rapid changes in the fluid properties, as figure 1.2 shows. The temperature at which the sharp peak in specific heat ($C_p$) occurs is called the pseudocritical temperature. A fluid at the pseudocritical temperature demonstrates remarkable cooling properties due to the $C_p$ maximum and resulting increase in the heat transfer coefficient. In addition, the low viscosity, $\mu$, results in an increase in turbulence, and therefore the cooling abilities, of the fluid. Small changes in temperature translate to large variations in the fluid properties and instabilities in the fluid flow. These discontinuities become less drastic as the pressure of the fluid is increased. The $C_p$ maximum decreases as pressure increases, which leads to a deterioration in the heat transfer coefficient. Figures 1.3 and 1.4 show the fluid properties of ethanol at pressures of 100 bar and 300 bar to compare the magnitude of the property variations.

![H$_2$O Property Data for 300 bar](image)

**Figure 1.2:** Thermodynamic properties of water for 300 bar ($P_f = 1.36$)
Figure 1.3: Thermodynamic properties of ethanol at 100 bar ($P_r = 1.6$)

Figure 1.4: Thermodynamic properties of ethanol at 300 bar ($P_r = 4.8$)
The ethanol heat transfer tests were run at pressures of approximately 100 and 300 bar, or reduced pressures \( (P_r = P/P_c) \) of 1.60 and 4.80. The high pressure ethanol data show a gradual drop in wall temperature at the pseudocritical point, while the low pressure ethanol tests show a sharp drop in wall temperature at the pseudocritical point. The water tests were run at approximately 300 bar, a reduced pressure of 1.36. The conditions were chosen to match those of the low pressure ethanol tests in order to corroborate the ethanol test results. These water tests showed a similar drop in the wall temperature to the low pressure ethanol tests, as well as instabilities in the temperature readings corresponding to the observations in the literature. The high pressure ethanol tests could not be duplicated with water since the test rig was not rated for pressures above 6000 psi (414 bar).
Chapter 2

Experimental Apparatus

2.1 Test Rig

The test rig located in GTL was designed to measure the outside surface temperature of the test section tube while varying heat flux and keeping the pressure and mass flow constant. From the fluid pressure, outside surface temperature, and heat flux into the tube, the inside wall temperature, bulk fluid temperature, and heat transfer coefficient were determined. A schematic of the test rig is pictured below.

Figure 2.1: Schematic of test rig

[Diagram of test rig]
The rig occupies two rooms, the μ-rocket test cell, and the control room. Instruments and measurements were represented graphically in Labview on a computer in the control room so that flow conditions could be altered remotely. The test rig is located in the test cell and no human intervention is required while a test is running. The fuel tank cylinder was filled with fuel via the sight glass downstream of the cylinder. Fuel was injected using a plastic syringe with a 2 μm filter attached to remove particles from the fluid. The fuel tank was then pressurized using a 6000 psi helium tank located inside the control room. This pressure was measured with a pressure transducer located at the top of the fuel tank and displayed as line pressure on the Labview console. Solenoid valve #1, called the line flow solenoid, must be opened to begin the flow of fuel through the rig.

Mass flow was controlled by either an orifice located downstream of the test section, or by a valve located immediately upstream of the orifice. The valve was added to the rig so that fuel could be vented if the orifice clogged. Mass flow was measured using a Micro-Motion, Inc. high pressure, low flow meter. Prior to running the ethanol tests and the water tests, the mass flow meter was calibrated by measuring the amount of fluid that ran through the test rig for a set amount of time and comparing that volume to the volume calculated using the average mass flow reading from the meter.

Test section heat flux was applied using resistive heating. Copper leads from a Hewlett-Packard 1000 W constant voltage DC power supply were attached to the test section on either side of the test section tube. The power supply is located in the control room so that the voltage may be increased remotely. This voltage reading was saved by Labview. The current running through the test section must also be known to find the power going into the tube. This was measured using the voltage drop across a metal shunt of known resistance. Because the test section was heated resistively, an electric insulator was required along the flow path. A block of G10 fiberglass was installed downstream of the
test section. The G10 is pressed between two steel plates to keep it from delaminating at high pressure.

2.2 Test Section

The test sections were manufactured by MicroGroup, Inc. Each test section consisted of a 10 mm long 300 μm outer diameter, 95 μm inner diameter 304 stainless steel tube. This thin tube was silver soldered into a larger 1/16 inch outer diameter tube so that 3 mm on either side of the test section were inside the larger tube and the center 4 mm length was exposed. This center length was heated by current introduced by the copper leads on either side. The 3 mm inlet length allowed the hydrodynamic profile to develop before heating began.

![Diagram of test section](image)

**Figure 2.2:** Test section
2.3 Thermocouples

The outside surface temperature of the test section was measured and reported in Labview. Three 2 mil diameter K-type thermocouples were spot welded under a microscope along the 4 mm test section at upstream, midpoint, and downstream positions. Another thermocouple in the fuel tank cylinder read the fuel temperature inside the tank, which was assumed to be the inlet fluid temperature.

2.4 Thermocouple Calibration

The non-zero size of the thermocouple spot weld to the outer surface of the test section led to the thermocouple reading being corrupted by the electric field due to the resistive heating. This voltage drop across the bead caused the temperature to appear different than the actual temperature by an amount proportional to the heating current. This error could be corrected by applying a calibration to each thermocouple.

Figure 2.4 shows the relationship between thermocouple voltage and temperature for K-type thermocouples.
The first step in the calibration was to determine the constant of proportionality. This constant was a function of the quality, or surface area, of the bead weld. The constant was multiplied by the power supply voltage to get the actual mV drop across the thermocouple. This mV drop was then applied to the plot in figure 2.4 to get the corresponding temperature loss in degrees C. This temperature loss was added to the surface temperature recorded by Labview to get the actual temperature of the test section.

The calibration constant is unique to each thermocouple and must be found by running a calibration test. In previous work, the constant was found by placing a small voltage on the order of about 0.001 V across the test section. It was assumed that such a small voltage did not actually raise the temperature of the tube, and any temperature change recorded
was purely the result of a voltage drop across the bead. The ratio of temperature drop to voltage was used to determine the thermocouple mV reading from the relation in figure 2.4 and therefore, the constant in mV/V.

It was discovered, however, that this method did not correct the thermocouple readings enough. Upon running several water heat transfer tests at sub critical pressures, it was observed that the change in slope of the inside wall temperature curve indicating the change from liquid to vapor did not correspond to the saturation temperature for the given pressure. The calibration constant was then iterated until the film boiling point occurred at the saturation temperature. This was determined to be a more accurate calibration method, so a sub critical pressure calibration test was run before each critical test for a new test section. It was assumed that the physical characteristics of the weld did not change during a test, and the calibration constants therefore remained constant throughout the life of a particular test section. This appeared to be a good assumption until the last water test, which saw much higher heat fluxes than previous runs. It was observed that the midpoint thermocouple on the last test section required a larger correction after the first supercritical test was run. The test section eventually failed at the midpoint because of the high thermal stresses, and it is assumed that the deformation at the midpoint was responsible for the change in the thermocouple weld. Previous heat transfer studies have used this same calibration method to correct for thermocouple voltage drop on a resistively heated test section tube. Figure 2.5 is an example of temperature readings from a calibration test. The film boiling point is readily apparent.

The ethanol test temperatures had to be re-calibrated according to this method without the existence of calibration tests. This was done by noting that the drop in wall temperature in the supercritical tests corresponded to the pseudocritical temperature, and the calibration constants were adjusted accordingly.
Figure 2.5: Calibration test 11 for water. The change in slope indicates the film boiling at the saturation temperature (222°C).
2.5 Experimental Procedure

2.5.1 Test Rig Operation

The following procedure was used to operate the test rig:

The manual shut-off valves should be in the following positions prior to running a test. The valves are labeled to correspond with figure 2.1.

<table>
<thead>
<tr>
<th>Valve</th>
<th>Position</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>CLOSED</td>
</tr>
<tr>
<td>B</td>
<td>CLOSED</td>
</tr>
<tr>
<td>C</td>
<td>CLOSED</td>
</tr>
<tr>
<td>D</td>
<td>CLOSED</td>
</tr>
<tr>
<td>E</td>
<td>SLIGHTLY OPEN</td>
</tr>
<tr>
<td>F</td>
<td>CLOSED</td>
</tr>
<tr>
<td>G</td>
<td>CLOSED</td>
</tr>
<tr>
<td>H</td>
<td>OPEN IF ORIFICE IS BLOCKED</td>
</tr>
</tbody>
</table>

Table 2.1: Manual shut-off valve positions

Activate the instruments in the control room by turning the chassis power and the pressure transducer power supply on. Activate the main power button in the Labview console, then press the button on the chassis to illuminate the Main Power indicator light. Verify that the three test section thermocouples are reading room temperature. To fill the fuel tank cylinder, activate solenoid #2 via the Labview console in the control room. On the test rig, open valve D. Fill the sight glass with fuel using the syringe with the 2μ filter. When full, close valve D, and de-activate solenoid #2. To begin the flow of fuel through the rig, power solenoid #1. To pressurize the system, open the helium tank, and turn the pressure regulator until the desired pressure (in psi) appears on the Labview console. Turn on the power sup-
ply, and begin to slowly increase the voltage, stopping approximately every 1 W/mm$^2$ to allow the temperatures to come to equilibrium. Once the data are recorded, shut off the power supply, and close the pressure regulator on the helium bottle. To vent the line, open valve A. Power solenoids 1, 2, and 3 to vent all remaining fuel and helium.

2.5.2 Labview file

All measurements taken during a test were displayed graphically in the Labview console. These measurements were line pressure, test section pressure drop, tank temperature, power supply voltage, current, heat flux, mass flow, and temperature at upstream, midpoint, and downstream positions along the test section. The console also indicated whether or not main power, and the solenoid valves are on or off. Previously, for the ethanol tests, each of these measurements was recorded by hand after increasing the heat flux. There was only one value for mass flow and line pressure recorded for these tests. The Labview code was modified during the water testing. A save button was added so that measurements could be written to an output file with the date and time when the button was pushed. After temperature oscillations were observed around the critical point, the code was further modified to save every time the measurements were re-calculated in Labview at a frequency of 1 Hz.
Chapter 3

Data Reduction

The data reduction process was a lengthy one since the heat transfer coefficient must be determined through only a few measured quantities. Voltage, current, outside surface temperature, and tank fluid temperature were measured directly. From these, test section power flux, inside wall temperature, bulk temperature, and heat transfer coefficient were determined.

3.1 Power

The power into the test section, \( Q \), is a function of the voltage drop across the tube and the current flowing through the tube:

\[
Q = IV. \tag{3.1}
\]

It was assumed that the copper block leads on either side of the test section were in good electrical contact and caused no voltage loss. Previous research assumed losses and calculated the power using a formula for the resistance of the stainless steel as a function of temperature, however, it was observed that this resistance model broke down at high heat flux.

3.2 Wall Temperature

A formula for the inside wall temperature of the tube as a function of the outside surface temperature was derived. It was assumed that the electric field in the tube and therefore, the current flowing through the tube were constant. A diagram of the heating conditions is shown in figure 3.1.
The change in temperature across the tube is equal to the energy dissipated in the tube. The energy balance equation is

$$\frac{r k_s}{dr} \left. \left( \frac{dT}{dr} \right) \right|_r = \int_r^{r_0} \sigma E^2 r dr. \quad (3.2)$$

where $r$ is the radius from the center line, $k_s$ is the thermal conductivity of 304 stainless steel, $T$ is the temperature of the metal, $\sigma$ is the charge density in the tube, $E$ is the electric field in the tube, and $r_0$ and $r_i$ are the outside and inside surface radii. This energy balance was integrated to get the tube temperature as a function of the radius. The thermal conductivity of steel increases with temperature, so a relation for $k_s$ as a function of $T$ was necessary to complete the integration. Figure 3.2 shows this relation.
Figure 3.2: Thermal conductivity of 304 stainless steel as a function of temperature. A best fit line was fit to the data points to get the following formula for $k_s$:

$$k_s = 0.0152T + 14.2444$$

(3.3)

where $T$ is in degrees C and $k_s$ is in W/mK. Equation 3.2 becomes

$$r \frac{dT}{dr} (0.0152T + 14.2444) = \int_r^0 \sigma E^2 r dr.$$  

(3.4)

$E$ and $\sigma$ were assumed to be constant in this model, so the integral could be reduced and the variables separated to get the following:
\[(0.0152T + 14.2444)dT = \frac{\sigma E^2}{2} \left( \frac{r_0^2}{r} - r \right)dr. \quad (3.5)\]

This can be integrated to get

\[\frac{0.0152}{2} T^2 + 14.2444T = \frac{\sigma E^2}{2} \left( r_0^2 \ln r - \frac{r^2}{2} \right) + C. \quad (3.6)\]

C is a constant of integration which can be found using boundary conditions. By setting \(T = T_0\) at \(r = r_0\), and \(T = T_i\) and \(r = r_i\), where \(r_0\) is the outside surface of the tube and \(r_i\) is the inside wall. The final equation for \(T_i\) becomes

\[T_i = -937.13 + 65.79 \sqrt{202.90 + 0.0304X} \quad (3.7)\]

where

\[X = \frac{-Q}{2\pi l} \left( \log \frac{r_0}{r_i} - \frac{1}{2} \right) + 0.0076 T_0^2 + 14.2444 T_0. \quad (3.8)\]

\(Q\) is the power into the test section in Watts, \(l\) is the length of the heated tube in meters, \(T_0\) is the surface temperature in degrees Celsius, \(r_0\) is the outside surface radius in meters, and \(r_i\) is the inside wall radius in meters. This formula was used to calculate the inside wall temperature in degrees C.

### 3.3 Bulk Temperature

The bulk temperature was calculated by equating the enthalpy difference between the tube inlet and outlet to the power per mass flow:
\[ \frac{Qx}{\dot{m}} = H_o - H_i, \]  
(3.9)

where \( Q \) is the power in W, \( x \) is the fractional distance along the tube length, \( \dot{m} \) is the mass flow in kg/s, and \( H_o \) and \( H_i \) are the outlet and inlet enthalpies in J/kg respectively. This formula was used to find the outlet enthalpy, which was then used to find the temperature at that point, or bulk temperature. The inlet enthalpy was known because the inlet temperature, or fuel tank temperature, and the pressure were known. The enthalpy was read from a fluid property table which lists enthalpy as a function of \( T \) and \( P \). Once the outlet enthalpy was calculated, the bulk temperature was read from the same enthalpy table.

The average of the wall and the bulk temperatures is the film temperature:

\[ T_f = \frac{T_w + T_b}{2}. \]  
(3.10)

### 3.4 Heat Transfer Coefficient

The heat transfer coefficient was simple to calculate once the inside wall temperature and bulk temperature were known. Heat transfer coefficient is given by

\[ h = \frac{Q}{A_i(T_w - T_b)}, \]  
(3.11)

where \( Q \) is the power into the tube in W, \( A_i \) is the inside surface area of the tube in m\(^2\), \( T_w \) is the inside wall temperature (\( T_i \) from equation 3.7), and \( T_b \) is the bulk temperature.

### 3.5 Stanton Number

Previous research displayed end results in non-dimensional quantities of Reynolds number, \( Re \), and Nusselt number, \( Nu \). It was decided that data would be reduced to heat transfer coefficient and Stanton number to eliminate uncertainties property tables of viscosity and thermal conductivity introduce. Many property tables do not include data in the
supercritical regime, and a linear extrapolation may not be accurate. The Stanton number (St) requires enthalpy table values, however, the enthalpy tables were already necessary in the calculation of bulk temperature. The following defines St:

\[
St = \frac{Q}{A_i \rho u C_p (T_w - T_b)}
\]  

(3.12)

where \( Q \) is the power in W, \( A_i \) is the inside surface area of the tube in m², \( \rho \) is the fluid density in kg/m³, \( u \) is the fluid velocity in m/s, \( T_w \) is the temperature of the fluid at the inside wall, and \( T_b \) is the bulk temperature in degrees C. \( \rho u \) can be found by using

\[
\dot{m} = \rho u A_e
\]  

(3.13)

where \( A_e \) is the tube inlet/outlet area in m². \((H_w - H_b)\) can be substituted for \( C_p(T_w - T_b)\).

The equation used for Stanton number is then

\[
St = \frac{Q}{A_i \dot{m} (H_w - H_b)}
\]  

(3.14)

St is a ratio of the amount heat transferred through the walls to the thermal capacity of the fluid. In these experiments, \( \dot{m} / A_e \) was held constant, so variation in St was a function of heat flux and the difference between the wall and bulk temperatures.

### 3.6 Losses

Potential heat losses include heat flux from radiation and free convection. Radiative heat loss was previously determined to be approximately 20 kW/m² assuming a generous outside surface temperature of 1000 K. This equates to a test section tube loss of 0.075 W, which is small compared to the actual heat flux, which reaches 380 W. The forced convective heat loss was previously determined to be negligible as well. Assuming again, an outside surface temperature of 1000 K and a bulk temperature of 288 K, the convective
heat loss would be approximately 0.067 W. Buoyancy effects in the fluid flow are also negligible for these experimental conditions.\textsuperscript{11} Although buoyancy effects increase with heat flux, and the heat flux is high, they are also proportional to the tube dimensions.
Chapter 4

Results and Discussion

The data reduction algorithms discussed in chapter 3 were applied to both the water tests as well as the ethanol data measured previously by Lopata.

4.1 Results of Water Tests

A total of six heat transfer tests of water at supercritical conditions were completed. The pressure for all water tests was kept constant at approximately 300 bar, a reduced pressure of 1.36, to correspond to the low pressure ethanol tests run at a reduced pressure of 1.60. The mass flow was varied between 100 mg/s and 623 mg/s. This pressure was close enough to the critical point that the fluid properties varied severely over small changes in temperature, as illustrated in figure 1.2. The water test results were therefore characterized by regions of instabilities caused by the rapid changes in properties. The tests are labeled 10, 12, 14, 16, 17, and 18. Tests 1 though 8 were preliminary subcritical calibration tests of the rig, and are therefore not presented here. Tests 9, 11, 13, and 15 were calibration tests of the four test sections used. It should be noted that the upstream thermocouple in tests 9 and 10 did not work. This was attributed to a poor quality weld.

4.1.1 Wall and Bulk Temperature

The temperature plots show the characteristics of supercritical heat transfer clearly. The graphs presented in figures 4.1 through 4.6 are plots of the measured outside wall temperature and the calculated inside wall and bulk temperatures. The bulk temperature never reaches the pseudocritical temperature or the critical temperature in any test, water or ethanol. Despite attempts to achieve a critical bulk temperature by increasing the mass flow and allowing higher heat fluxes, the test section failed due to thermal stress before reaching this point. In most cases, there is an obvious change in the slope of the wall tem-
perature curves around the pseudocritical point. Most tests also show some oscillation and drift in the wall temperature as well. This oscillation was noticed during tests 10 and 12, which show some unsteadiness, after the wall temperature reached the critical point. To get a better representation of this phenomenon, several points were sampled at each heat flux during test 14. Some instability is apparent around the critical point, where the data points begin to spread out. The data acquisition program was then altered to record a data point every time the program ran a calculation cycle at a frequency of 1 Hz. Tests 16 through 18 therefore consist of far more data points than the first 3 tests, and instabilities are readily apparent, especially in test 16. The rapid increase in the midpoint wall temperature in test 18 is not fluid property related, however. It is believed that this is an anomalous temperature reading caused by a malfunctioning thermocouple. It was observed that the test section tube shape warped significantly due to the large thermal stresses when this temperature jump occurred. The midpoint thermocouple was attached at this failure point and was most likely affected by the changing geometry of the tube.

It is also important to note that some of the water tests experienced a significant pressure drop across the test section. The pseudocritical temperature changes with pressure, so the pressure drop had to be factored into the data reduction. Tests 12 and 18 had particularly large test section pressure drops accompanied by mass flow rates several times higher than other tests, both of which were caused by leaks in the G10 block downstream of the test section. The pressure drop for test 12 remained fairly constant between 55 bar and 65 bar until the test section began to fail around 130 W/mm², at which point the pressure drop increased rapidly to 140 bar. The epoxy fittings in the G10 block had cracked, and the G10 block was replaced. This G10 block began to leak as well following test 16 due this time to delamination, but the pressure drop peaked at only 17 bar around 100 W/mm² for tests 16 and 17. An attempt was then made to run at a pressure higher than 300 bar for test 18,
and this caused the G10 to delaminate substantially, resulting in a mass flow of 623 mg/s, which increased to more than 800 mg/s after 260 W/mm². The test section pressure drop for test 18 began at 138 bar and dropped down to 80 bar until the test section began to fail at 260 W/mm².

**Figure 4.1:** Temperature plots for water at $P_r=1.32$, mass flow=141 mg/s
Figure 4.2: Temperature plots for water at $Pr=1.2$, mass flow=397 mg/s
Figure 4.3: Temperature plots for water at $P_r=1.3$, mass flow=100 mg/s
Figure 4.4: Temperature plots for water at $P_r=1.45$, mass flow=149 mg/s
Figure 4.5: Temperature plots for water at $P_r=1.4$, mass flow=180 mg/s
Figure 4.6: Temperature plots for water at $P_f=1.3$, mass flow=623 mg/s
4.1.2 Heat Transfer Coefficient

At a reduced pressure of 1.36, the heat transfer coefficient is expected to peak at the pseudocritical temperature, marking the $C_p$ peak maximum shown in figure 1.2. There is a noticeable peak in the heat transfer coefficient plot for test 12, but for the most part, the curves are more well-behaved than expected. This pattern was observed in the literature on supercritical water heat transfer.

Figure 4.7: Heat transfer coefficient for water at $P_r=1.32$, mass flow=141 mg/s
Figure 4.8: Heat transfer coefficient for water at $P_r=1.2$, mass flow=397 mg/s

Figure 4.9: Heat transfer coefficient for water at $P_r=1.3$, mass flow=100 mg/s
Figure 4.10: Heat transfer coefficient for water at $Pr=1.45$, mass flow=149 mg/s

Figure 4.11: Heat transfer coefficient for water at $Pr=1.4$, mass flow=180 mg/s
Figure 4.12: Heat transfer coefficient for water at $P_f=1.32$, mass flow=623mg/s

4.1.3 Stanton number

The data were reduced to the Stanton number because a non-dimensional quantity can be used for direct comparison with any other fluid. Stanton number is proportional to \( \left( \frac{\text{heat transferred}}{\text{thermal capacity of the fluid}} \right) \). In these experiments, the heat transfer is the dependent variable and the mass velocity, \( \rho u \), is held constant, therefore the variations in the plots of St are a function of \( C_p \) and the temperature difference \( (T_w-T_b) \) according to equation 3.12. As the fluid temperature approaches the pseudocritical point, the \( C_p \) rises sharply, which would result in a decrease in St. However, because the heat transfer around this temperature is enhanced, the wall temperature drops sharply, reducing the \( (T_w-T_b) \) term. This results in an overall increase in the St curve. Figures 4.13 through 4.18 are plots of St for the water tests. When compared to the temperature plots in figures 4.1 through 4.6, the St plots all show an increase at the heat flux corresponding to the wall temperature drop. Tests 14 and 17 are particularly clear examples.
**Figure 4.13:** Stanton number for water at $P_r=1.32$, mass flow=141 mg/s

**Figure 4.14:** Stanton number for water at $P_r=1.2$, mass flow=397 mg/s
Figure 4.15: Stanton number for water at $P_r=1.3$, mass flow=100 mg/s

Figure 4.16: Stanton number for water at $P_r=1.45$, mass flow=149 mg/s
Figure 4.17: Stanton number for water at \( P_r = 1.4 \), mass flow=180 mg/s

Figure 4.18: Stanton number for water at \( P_r = 1.32 \), mass flow=623 mg/s
4.1.4 Comparison with Literature

Previously conducted water heat transfer research was reviewed to compare to the water tests. Establishing a baseline was difficult, however, due to several differences between the test rigs described in the literature and the µ-rocket test rig, as well as differences in the test conditions.

Several articles on heat transfer to supercritical water flowing in circular tubes were used for comparison. Each experimental test apparatus described differed from the µ-rocket test rig in several ways. First, several rigs described used vertical tubes instead of horizontal tubes. The test section tube dimensions were substantially larger, on the order of 10 mm diameter and 1 to 2 m in length. More importantly, in each case, the fluid was heated to near critical temperatures prior to entering the test section, making the heat fluxes much lower (~0.1 - 2 W/mm²) than the µ-rocket test conditions (~1 - 350 W/mm²). The inlet temperature was regulated while the heat flux, pressure, and mass flow rate were held constant. In the case of the µ-rocket test rig, the heat flux is the independent variable and the inlet temperature was kept constant.

Several papers discussed a heat transfer deterioration phenomenon, which occurred at high heat fluxes. High heat flux, in their case was approximately 0.5 - 1 W/mm², which is far exceeded in the µ-rocket tests. It is therefore expected that a deterioration in the heat transfer coefficient can be observed in the water tests.

Deterioration of the heat transfer coefficient is a phenomenon governed by the heat flux only. Swenson, et al. noticed that the heat transfer coefficient peaked when the film temperature reached the pseudocritical temperature, however, increasing the pressure of the fluid lowered the peak, and increasing the heat flux from 0.788 W/mm² to 1.74 W/mm² lowered the peak heat transfer coefficient from 45,400 W/m²K by a factor of 2 to 22,700 W/m²K. The tests for which the film temperature reached the pseudocritical tem-
perature (−400°C for water at 300 bar), tests 10 and 14, are shown in figures 4.19 and 4.20. The heat transfer coefficient does have a maximum at approximately 400°C, however, it is not a sharp peak discontinuity, like the $C_p$ curve. Koshizuka, et al. and Tanaka, et al. offer an explanation for the mechanism of heat transfer coefficient deterioration. As the heat flux increases, the difference between the wall temperature and the bulk temperature increases. The fluid in the μ-rocket test sections is heated from room temperature, so the bulk temperature is much lower than the wall temperature, which reaches a gas-like state almost immediately upon entering the heated tube length. The increasing temperature drop across the gas-like wall fluid causes the heat transfer coefficient to decrease. It was observed in the literature that the heat transfer coefficient will increase, resulting in enhanced cooling capabilities as the bulk temperature exceeds the pseudocritical temperature. This never occurred in the μ-rocket experiments.

![Graph showing heat transfer coefficient vs. film temperature](image_url)

**Figure 4.19:** Heat transfer coefficient vs. film temperature for water at $P_r = 1.32$, mass flow = 141 mg/s
Yamagata et al. wrote that the deterioration occurred when a particular heat flux, called the critical heat flux, $q_c$, was exceeded. The authors developed a relationship for the critical heat flux is a function of the mass velocity, $G$:

$$q_c = 0.20G^{1.2}.$$

The critical heat flux is labeled on the plots of temperature in figures 4.1 through 4.6. This relationship was developed using data from vertical tube experiments with bulk temperatures that reached the critical point, so this correlation might not be accurate for this situation. It is worth noting, however, that for both tests 10 and 14, the critical heat flux was surpassed, meaning the deterioration in the heat transfer peak was expected.

Rapid fluctuations in wall temperature were observed during several of the water tests while the heat flux was held constant. The effect is most prominently shown in figure 4.4.
the 1.45 reduced pressure, 149 mg/s mass flow test. At constant heat flux, the temperature plot shows a drift-like pattern around the critical point. This phenomenon was also observed in the literature at high heat flux. Temperature oscillations were recorded in vertical tube supercritical water heat transfer experiments for heat fluxes above the critical heat flux. The oscillations are the result of the mixing of the gas-like wall layer and the cold bulk temperature within the viscous layer at the walls. The high wall temperatures cause the viscosity of the water to drop, the turbulence to decrease, and the heat transfer coefficient to break down. This further raises the wall temperature and increases the thickness of the boundary layer. The larger boundary layer then begins to mix with the cooler bulk temperature, and the heat transfer coefficient in the boundary layer increases, lowering the temperature. This shrinks the thickness of the wall layer, creating a cycle for temperature oscillation at the wall. The effect becomes more pronounced as the bulk temperature reaches the pseudocritical temperature where the heat transfer is enhanced.

The thermal entrance region is also of concern. The literature states that the thermal entrance region length is extended significantly for near-critical or supercritical fluids. This means that the thermal profile may still be developing through most of the measured test section tube length. The 3mm long section was placed in front to avoid entrance region effects due to a developing velocity profile, but the heating begins at the inlet of the measured 4mm long section. One experiment observed that the thermal entrance region effects progressed as far as 1/3 of the length of the test section. This figure may be larger for the $\mu$-rocket experiments, however. One article indicates that the entrance region also increases significantly if the bulk temperature is lower than the wall temperature and the critical temperature. This was the case for all $\mu$-rocket tests. The thermal entrance region is characterized by a drop in the heat transfer coefficient, which rises again once the flow has developed. It is therefore expected that the downstream heat transfer coefficient
curves and Stanton number curves should be lower than the midpoint and upstream curves if the thermal entrance region extends that far into the tube. This does not occur in the water tests. The upstream heat transfer coefficient is greater than that at the midpoint and downstream positions. It can therefore be concluded that the entrance region stays within the first segment of heated length upstream of the first thermocouple.

4.2 Results of Low Pressure Ethanol Tests

The reduced pressure for the low pressure ethanol tests (1.6) corresponds approximately to the reduced pressure for the water tests. It is expected that the ethanol tests should show similar heat transfer coefficient increases at the pseudocritical temperature.

4.2.1 Wall and Bulk Temperature

The temperature plots of the low pressure ethanol tests show a dramatic decrease in the wall temperature at the pseudocritical temperature, about 270°C at this pressure. This is caused by the sharp increase in the $C_p$ curve, shown in figure 1.3. A trend in the curve that resembles film boiling is visible in the downstream temperature plots. As the temperature approaches the $C_p$ maximum, the heat transfer coefficient begins to rise at an increasing rate with the $C_p$ curve, creating a steady drop in the rise of the wall temperature until the peak is reached.

Unlike the water temperature plots, the ethanol data show no sign of temperature drift or oscillation. The previous research on the ethanol tests reported some pressure oscillations, but the temperature readings remained steady. The critical heat flux is not displayed on either set of ethanol plots because it is not useful to know where oscillations may begin.
Figure 4.21: Temperature plots for ethanol at $P_r = 1.64$, mass flow = 63 mg/s
Figure 4.22: Temperature plots for ethanol at $P_r = 1.67$, mass flow = 60 mg/s
Figure 4.23: Temperature plots for ethanol at $P_r = 1.66$, mass flow = 32 mg/s
Figure 4.24: Temperature plots for ethanol at $P_r = 1.65$, mass flow = 76 mg/s
Figure 4.25: Temperature plots for ethanol at $P_r = 1.65$, mass flow = 57 mg/s
4.2.2 Heat Transfer Coefficient

As expected, the heat transfer coefficient shows a sharp increase corresponding to the point where the wall temperature dropped. The heat transfer plots also indicate the presence of the thermal entrance region as the heat flux increases. The literature indicated that heat flux did not have a large effect on the thermal entrance region length, however, the heat fluxes reached here are considerably larger than those in the literature. The heat transfer coefficient downstream drops below the upstream measurement slowly over a range of more than 10 W/mm², so the dependence of thermal entry length on the heat flux may not have been noticed over the test conditions in previous experiments.

![Diagram](image)

**Figure 4.26:** Heat transfer coefficient for ethanol at $Pr = 1.64$, mass flow = 63 mg/s
**Figure 4.27**: Heat transfer coefficient for ethanol at $P_r = 1.67$, mass flow = 60 mg/s

**Figure 4.28**: Heat transfer coefficient for ethanol at $P_r = 1.66$, mass flow = 32 mg/s
Figure 4.29: Heat transfer coefficient for ethanol at $P_r = 1.65$, mass flow = 76 mg/s

Figure 4.30: Heat transfer coefficient for ethanol at $P_r = 1.65$, mass flow = 57 mg/s
4.2.3 Stanton Number

The behavior of the Stanton number is governed by the difference between the bulk and the wall temperatures, and therefore shows an increase around the pseudocritical temperature as the wall temperature drops. In addition, the Stanton number plots show the development of a thermal entrance region at the upstream and midpoint positions indicated by the decrease in the Stanton number below the downstream position.

![Stanton number for ethanol at Pr = 1.64, mass flow = 63 mg/s](image)

**Figure 4.31:** Stanton number for ethanol at $P_r = 1.64$, mass flow = 63 mg/s
Figure 4.32: Stanton number for ethanol at $P_r = 1.67$, mass flow $= 60$ mg/s

Figure 4.33: Stanton number for ethanol at $P_r = 1.66$, mass flow $= 32$ mg/s
Figure 4.34: Stanton number for ethanol at $P_r = 1.65$, mass flow = 76 mg/s

Figure 4.35: Stanton number for ethanol at $P_r = 1.65$, mass flow = 57 mg/s
4.3 Results of High Pressure Ethanol Tests

The graph of fluid properties for ethanol at 300 bar in figure 1.4 indicates that there are no large variations in the properties at a reduced pressure of 4.80. There is a maximum in the $C_p$ curve at approximately 320 C, but the peak $C_p$ value has been severely degraded at these extreme pressures. The $C_p$ curve in figure 1.4 is actually for a pressure of 250 bar, since $C_p$ data for 300 bar was unavailable. Figure 3.3 shows $C_p$ plotted over a range of pressures above the critical point. The maximum value of $C_p$ decreases as pressure increases. From this plot, it can be concluded that the $C_p$ curve at 300 bar would closely resemble the $C_p$ curve at 250 bar, and the pseudocritical temperature would remain close to 330 C.
Figure 4.36: Specific heat of ethanol at supercritical pressures

4.3.1 Wall and Bulk Temperature

As expected, the lack of variation in the ethanol fluid properties at 300 bar results in a smooth temperature profile with no sudden decrease due to specific heat and viscosity changes. There is little variation in the slope of the wall temperature to mark the pseudocritical temperature, as observed in figures 4.22 through 4.25. The temperature plots show that the fluid at this pressure is stable and predictable, although there is no enhanced heat transfer in the pseudocritical region.
Figure 4.37: Temperature plots for ethanol at $P_f = 4.86$, mass flow = 102 mg/s
Figure 4.38: Temperature plots for ethanol at $P_f = 4.86$, mass flow = 121 mg/s
Figure 4.39: Temperature plots for ethanol at $P_r = 4.85$, mass flow = 71 mg/s
Figure 4.40: Temperature plots for ethanol at $P_r = 4.89$, mass flow = 255 mg/s
4.3.2 Heat Transfer Coefficient

Both the pressure and the heat flux affect the heat transfer coefficient by reducing the pseudocritical peak value as pressure and heat flux are increased. These tests were run at pressures significantly above the critical point and at high heat flux, so, as expected, the heat transfer coefficient curves are smooth and show little variation. These plots also show a clear thermal entrance region development, particularly test 2017. The downstream heat transfer curve increases at a smaller rate due to the increasing length of the thermal entrance region. Eventually, the midpoint and downstream curves overtake the downstream curve indicating that the thermal entrance region has expanded into the test section as far as the downstream and eventually midpoint thermocouples. This seems to indicate that the thermal entrance length is dependent on heat flux and the fluid, since it appears in the ethanol tests and not the water tests. Pressure does not seem to be as important because both the thermal entrance region transitions for the low and high pressure ethanol tests occur at nearly the same heat flux.
Figure 4.41: Heat transfer coefficient for ethanol at $Pr = 4.86$, mass flow = 102 mg/s

Figure 4.42: Heat transfer coefficient for ethanol at $Pr = 4.86$, mass flow = 121 mg/s
Figure 4.43: Heat transfer coefficient for ethanol at $P_r = 4.85$, mass flow = 71 mg/s

Figure 4.44: Heat transfer coefficient for ethanol at $P_r = 4.89$, mass flow = 255 mg/s
4.3.3 Stanton Number

The Stanton number curves, like the heat transfer coefficient curves, are well behaved. For the midpoint and downstream points, the Stanton number rises as the film boiling phenomenon starts, since the difference between the bulk and the wall temperatures drops. The upstream curve shows a different trend. It drops, indicating the continued increase in the temperature difference. This, again, is the result of the extended thermal entrance region.

Figure 4.45: Stanton number for ethanol at \( \text{Pr} = 4.86 \), mass flow = 102 mg/s
Figure 4.46: Stanton number for ethanol at $Pr = 4.86$, mass flow $= 121 \text{ mg/s}$

Figure 4.47: Stanton number for ethanol at $Pr = 4.85$, mass flow $= 71 \text{ mg/s}$
Figure 4.48: Stanton number for ethanol at $Pr = 4.89$, mass flow = 255 mg/s
Chapter 5

Conclusions

5.1 Summary

The purpose of the water heat transfer tests was to extend to higher heat fluxes existing data on supercritical water heat transfer and compare the previously acquired ethanol data with these results. The water tests showed some features that were observed in the literature. The phenomenon of deterioration of the heat transfer coefficient at high heat fluxes was present in the water data, as well as oscillations in the temperature within a constant heat flux value. A discontinuity in the wall temperature plot as the wall temperature reached the pseudocritical point was also observed.

The water test reduced pressures were similar to those of the low pressure ethanol tests, and similarities existed between the two data sets. The ethanol wall temperature plots showed the same discontinuity around the pseudocritical temperature. The ethanol data, however clearly showed a thermal entrance region while the water data did not.

The high reduced pressure ethanol tests cannot be compared to water tests because the test rig is unable to duplicate the test conditions. Although, the high pressure ethanol tests showed characteristics observed in the literature. The heat transfer coefficient was degraded due to the high pressure. The thermal entrance region was present and occurred at heat fluxes close to those of the low pressure ethanol.

The water tests seemed to display characteristics seen in previous research, indicating that the difference in test conditions did not make a comparison impossible. The literature, as well as the water tests show that high heat fluxes cause instabilities in the fluid flow, which appear when the bulk temperature is much lower than the wall temperature. To achieve large heat transfer coefficients, the bulk temperature must reach the pseudocritical
temperature. In addition, discontinuities decrease as the reduced pressure increases. The water and low pressure ethanol tests demonstrate that heat transfer at reduced pressures close to 1 is unstable and unpredictable. The high pressure ethanol tests showed stable flow, but the enhanced heat transfer coefficient cannot be achieved since the fluid properties have no large changes leading to a sudden increase in cooling ability.

5.2 Future Work

The next step in the heat transfer experimentation is to begin a series of tests using other propellants to observe the supercritical behavior of the fluids. JP-7 is one of several fuels being considered for the μ-rocket. It has the advantage that it does not coke, which is vital because the cooling passages are so small, and therefore are easily clogged. The heat transfer tests will most likely begin after the current μ-rocket testing has been completed.

The test rig was designed to investigate the cooling passage effectiveness by varying the coolant pressure and the mass flow, but keeping the tube dimensions constant. The actual μ-rocket cooling passages have a set coolant pressure and mass flow, so the passage dimensions must be changed to alter the cooling abilities. It is unclear at this time how accurately the test rig can predict the heat transfer conditions in light of this fundamental difference. It is therefore suggested that running a series of tests using test sections with different internal diameters at a constant pressure would provide a better understanding of how supercritical heat transfer scales as a function of tube dimensions. This may provide a more accurate representation of the rocket engine.
Appendix A

Data Reduction Programs

These programs were written for Matlab.

% HEAT TRANSFER DATA REDUCTION PROGRAM FOR WATER

clear;

%%%%%%% CONSTANTS %%%%%%%

tc_cor = 24576;  % thermocouple correction for K type t.c.'s
OD = 300e-6;  % outer diameter of tube in m
ro = OD/2;  % outer radius of tube in m
ID = 95e-6;  % inner diameter of tube in m
ri = ID/2;  % inner radius if tube in m
l = 4e-3;  % length of the tube in m
Vol = l*pi*(ro)^2-(ri)^2);  % volume of steel tube wall (m^3)
Ai = pi*ID*l;  % inside surface area of the tube (m^2)
Ao = pi*OD*l;  % outside surface area of the tube (m^2)
Ae = pi*ri^2;  % inlet / exit area of the tube

Pc = 220.9;  % Critical pressure of water in bar
Tc = 374.14;  % Critical temperature of water in C

%%%%% Read data from spreadsheet-like file%%%%%

file_name = ['data/water/test09.txt'
'data/water/test10.txt'
'data/water/test11.txt'
'data/water/test12.txt'
'data/water/test13.txt'
'data/water/test14.txt'
'data/water/test15.txt'
'data/water/test16.txt'
'data/water/test17.txt'
'data/water/test18.txt'];

output_file_name = ['data/reduced_data/water/test09.dat'
'data/reduced_data/water/test10.dat'
'data/reduced_data/water/test11.dat'
'data/reduced_data/water/test12.dat']
for i = 1:10,
% CLEAR OLD ARRAYS
clear T3, clear T4, clear T5;
clear T3w, clear T4w, clear T5w;
clear T3b, clear T4b, clear T5b;
clear T3f, clear T4f, clear T5f;
clear yr, clear day, clear hr, clear minute, clear sec;
clear T_in, clear m_dot, clear P, clear ts_P, clear V, clear I, clear power;
clear P3, clear P4, clear P5;
clear Q, clear heat_flux, clear h_in, clear C3, clear C4, clear C5;
clear h3_out, clear h4_out, clear h5_out, clear h3, clear h4, clear h5;
clear cp, clear St3, clear St4, clear St5, clear G;
clear data;

test_num = file_name(i,16:17);
disp(test_num);
fid = fopen(file_name(i,1:21),'r');
test_cond(1:6) = str2num(fgetl(fid));
c3 = test_cond(1); % thermocouple calibration constant
C4 = test_cond(2);
c5 = test_cond(3);
x3 = test_cond(4); % l/d for thermocouples
x4 = test_cond(5);
x5 = test_cond(6);
x3 = x3*ID/l; % fractional distance of t.c. location
x4 = x4*ID/l;
x5 = x5*ID/l;
line = fgetl(fid);
j = 1;
while line ~= -1
    line_str = str2num(line);
    yr(j) = line_str(1); % year
day(j) = line_str(2); % day
hr(j) = line_str(3); % hour
minute(j) = line_str(4); % minute
sec(j) = line_str(5); % second
T_in(j) = line_str(6); % supply tank temp (degrees C)
m_dot(j) = line_str(7)/1e6; % mass flow (kg/s)
P(j) = line_str(8)*6.894757e-2; % line pressure in bar (from psi)
ts_P(j) = line_str(9)*6.894757e-2; % test section pressure drop in bar (from psi)
V(j) = line_str(10); % voltage (V)
I(j) = line_str(11); % current (A)
power(j) = line_str(12); % heat_flux (W/mm^2)
T(j) = line_str(13); % T5 (degrees C)
T4(j) = line_str(14); % T4 (degrees C)
T3(j) = line_str(15); % T3 (degrees C)

%%%%%%%%%%%%%%%%%%%%%%%%% PRESSURE DROP ACCOUNTING %%%%%%%%%%%%%%%%%%%%%%%%%%
P3(j) = P(j)-ts_P(j)*x3;
P4(j) = P(j)-ts_P(j)*x4;
P5(j) = P(j)-ts_P(j)*x5;

%%%%%%%%%%%%%%%%%%%%%%%%% THERMOCOUPLE CALIBRATION %%%%%%%%%%%%%%%%%%%%%%%%%%
milli_V = V(j)*c3;
Tc_factor = tc_cor*milli_V;
T3(j) = T3(j) + Tc_factor;
milli_V = V(j)*c4;
Tc_factor = tc_cor*milli_V;
T4(j) = T4(j) + Tc_factor;
milli_V = V(j)*c5;
Tc_factor = tc_cor*milli_V;
T5(j) = T5(j) + Tc_factor;

Q(j) = I(j).*V(j);
heat_flux(j) = Q(j)/(Ai*1e6); % W/mm^2

%%%%%%%%%%%%%%%%%%%%%%%%% PROPERTY TABLE INDEPENDENT %%%%%%%%%%%%%%%%%%%%%%%%%%
h_in(j) = tp2h_h2o(P(j),T_in(j)); % enthalpy at tube inlet (J/kg)
C3(j) = -1*(Q(j)/(2*pi*I))*)((log(ro/ri)/(1-(ri/ro)^2))-0.5)
+0.0076*T3(j)^2+14.2444*T3(j);
T3w(j)=(-14.2444+sqrt(14.2444^2+4*0.0076*C3(j)))/(2*0.0076); %wall temp(C)
C4(j) = -1*(Q(j)/(2*pi*1)*((log(ro/ri)/(1-(ri/ro)^2))-0.5))
+0.0076*T4(j)^2+14.2444*T4(j);
T4w(j)=(-14.2444+sqrt(14.2444^2+4*0.0076*C4(j)))/(2*0.0076);
C5(j) = -1*(Q(j)/(2*pi*1)*((log(ro/ri)/(1-(ri/ro)^2))-0.5))
+0.0076*T5(j)^2+14.2444*T5(j);
T5w(j)=(-14.2444+sqrt(14.2444^2+4*0.0076*C5(j)))/(2*0.0076);

%wall temp(C)

h3_out(j) = hin(j) + ((Q(j).*x3)/m_dot(j)); % enthalpy at tube exit (J/kg)
h4_out(j) = hin(j) + ((Q(j).*x4)/m_dot(j));
h5_out(j) = hin(j) + ((Q(j).*x5)/m_dot(j));
T3b(j) = hp2t_h2o(h3_out(j),P3(j)); % bulk fluid temperature (degrees C)
T4b(j) = hp2t_h2o(h4_out(j),P4(j));
T5b(j) = hp2t_h2o(h5_out(j),P5(j));
h3(j)= Q(j)/Ai*(T3w(j) - T3b(j)); % heat transfer coefficient (W/m^2.K)
h4(j)= Q(j)/Ai*(T4w(j) - T4b(j));
h5(j)= Q(j)/Ai*(T5w(j) - T5b(j));

G(j) = m_dot(j)/Ae; % mass velocity (kg/s.m^2)

%% PROPERTY TABLE DEPENDENT %%%%

if test_num == '09' | test_num == '10'
    St3(j) = 0;
else
    h3w(j) = tp2h_h2o(P3(j),T3w(j)); % Wall enthalpy (J/kg)
    h3b(j) = tp2h_h2o(P3(j),T3b(j)); % Bulk enthalpy (J/kg)
    St3(j) = (Q(j)/Ai)/((m_dot(j)/Ae)*(h3w(j)-h3b(j))); % Stanton number
end
h4w(j) = tp2h_h2o(P4(j),T4w(j));
h5w(j) = tp2h_h2o(P5(j),T5w(j));

h4b(j) = tp2h_h2o(P4(j),T4b(j));
h5b(j) = tp2h_h2o(P5(j),T5b(j));

St4(j) = (Q(j)/Ai)/((m_dot(j)/Ae)*(h4w(j)-h4b(j)));
St5(j) = (Q(j)/Ai)/((m_dot(j)/Ae)*(h5w(j)-h5b(j)));
j = j + 1;
line = fgetl(fid);
end
L = j-1;

P3_ave = 0;
P4_ave = 0;
P5_ave = 0;
m_dotave = 0;
for index=1:L,
    P3_ave = P3(index)+P3_ave;
P4_ave = P4(index)+P4_ave;
P5_ave = P5(index)+P5_ave;
m_dotave = m_dot(index)+m_dotave;
end
P3_ave = P3_ave/L;
P4_ave = P4_ave/L;
P5_ave = P5_ave/L;
p3_str = num2str(P3_ave./Pc);
p4_str = num2str(P4_ave./Pc);
p5_str = num2str(P5_ave./Pc);
m_dotave = m_dotave/L*1e6;  \% converts to mg/s from kg/s
m_dot_str = num2str(m_dotave);
m_dot_str = m_dot_str(1:3);

st = fclose(fid);

%%%%%%WRITE DATA TO FILE%%%%%%

data = [ID];
fid = fopen(output_file_name(i,1:34),’w’);
fprintf(fid, ’%12.6f
’,data);
st = fclose(fid);
data = [P3; P4; P5; m_dot; heat_flux; T3; T4; T5; T3w; T4w; T5w; T3b; T4b; T5b; St3; St4; St5; G];
fid = fopen(output_file_name(i,1:34),’a’);
fprintf(fid, ’%12.4f %12.4f %12.4f %12.4f %12.4f %12.4f %12.4f %12.4f %12.4f %12.4f %12.4f %12.4f %12.4f %12.4f %12.4f
’,data);
st = fclose(fid);

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fig = 0;

% plot surface temp, inside wall temp, and bulk fluid temp vs. heat flux
plt = input('Do you want to plot temperature vs. heat flux? (y/n def = n) ','s');
if isempty(plt) == 1
    plt = 'n';
end
if plt == 'y'
    fig = fig+1;
    figure(fig)
    hold off, clf
    subplot(3,1,1), plot(heat-flux(1:L), T3(1:L)./Tc, 'bo');
    hold on
    plot(heat-flux(1:L), T3w(1:L)./Tc, 'rs');
    plot(heat-flux(1:L), T3b(1:L)./Tc, 'gd');
    ti(1:2) = testnum;
    ti(3:44) = 'Temperature upstream vs. Heat Flux  P_r=';
    ti(45:49) = p3_str(1:5);
    ti(50:52) = ' ';
    ti(53:55) = m_dotstr;
    ti(56:60) = 'mg/s';
    title(ti);
    clear ti;
    xlabel('Heat Flux (W/mm^2)');
    ylabel('T/T_c');
    legend('Outside Surface Temp', 'Inside Wall Temp', 'Bulk Fluid Temp',2);

    subplot(3,1,2), plot(heat-flux(1:L), T4(1:L)./Tc, 'bo');
    hold on
    plot(heat-flux(1:L), T4w(1:L)./Tc, 'rs');
    plot(heat-flux(1:L), T4b(1:L)./Tc, 'gd');
    ti(1:2) = test_num;
    ti(3:44) = 'Temperature midpoint vs. Heat Flux  P_r=';
    ti(45:49) = p4_str(1:5);
    ti(50:52) = ' ';
    ti(53:55) = m_dotstr;
    ti(56:60) = 'mg/s';

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title(ti);
clear ti;
xlabel('Heat Flux (W/mm^2)');
ylabel('T/T_c');
legend('Outside Surface Temp', 'Inside Wall Temp', 'Bulk Fluid Temp',2);

subplot(3,1,3), plot(heat_flux(1:L), T5(1:L)./Tc, 'bo');
hold on
plot(heat_flux(1:L), T5w(1:L)./Tc, 'rs');
plot(heat_flux(1:L), T5b(1:L)./Tc, 'gd');
ti(1:2) = test_num;
ti(3:46) = ' Temperature downstream vs. Heat Flux P_r=';
ti(47:51) = p5_str(1:5);
ti(52:54) = ' ';
ti(55:57) = m_dot_str;
ti(58:62) = ' mg/s';
title(ti);
clear ti;
xlabel('Heat Flux (W/mm^2)');
ylabel('T/T_c');
legend('Outside Surface Temp', 'Inside Wall Temp', 'Bulk Fluid Temp',2);
end

% plot heat transfer coefficient vs. heat flux
plt = input('Do you want to plot heat transfer coefficient vs. heat flux? (y/n def = n) 
','s');
if isempty(plt) == 1
    plt = 'n';
end
if plt == 'y'
    fig = fig+1;
    figure(fig)
    hold off, clf
    plot(heat_flux(1:L), h3(1:L), 'bo');
    hold on
    plot(heat_flux(1:L), h4(1:L), 'rs');
    plot(heat_flux(1:L), h5(1:L), 'gd');
    ti(1:2) = test_num;
    ti(3:49) = ' Heat Transfer Coefficient vs. Heat Flux P_r=';
    ti(50:54) = p4_str(1:5);
ti(55:57) = ' '; 
ti(58:60) = m_dot_str; 
ti(61:65) = ' mg/s'; 
title(ti); 
clear ti; 
xlabel('Heat Flux (W/mm^2)'); 
ylabel('Heat Transfer Coefficient (W/m^2.K)'); 
legend('upstream', 'midpoint', 'downstream',2); 
end

% plot Tw-Tb vs. heat flux 
plt = input('Do you want to plot Tw-Tb vs. heat flux? (y/n def = n ) ','s'); 
if isempty(plt) == 1 
    plt = 'n'; 
end 
if plt == 'y' 
    fig = fig+1; 
    figure(fig) 
    hold off, clf 
    plot(heat_flux(1:L), T3w(1:L)-T3b(1:L), 'bo'); 
    hold on 
    plot(heat_flux(1:L), T4w(1:L)-T4b(1:L), 'rs'); 
    plot(heat_flux(1:L), T5w(1:L)-T5b(1:L), 'gd'); 
    ti(1:2) = testnum; 
    ti(3:29) = ' Tw-Tb vs. Heat Flux P_r='; 
    ti(30:34) = p4_str(1:5); 
    ti(35:37) = ' '; 
    ti(38:40) = m_dot_str; 
    ti(41:45) = ' mg/s'; 
    title(ti); 
    clear ti; 
    xlabel('Heat Flux (W/mm^2)'); 
    ylabel('Tw-Tb (C)'); 
    legend('upstream', 'midpoint', 'downstream',2); 
    end

% plot Stanton Number vs. heat flux 
plt = input('Do you want to plot St vs. heat flux? (y/n def = n ) ','s'); 
if isempty(plt) == 1 
    plt = 'n'; 
    end
if plt == 'y'
    fig = fig+1;
    figure(fig)
    hold off, clf
    plot(heat_flux(1:L), St3(1:L), 'bo');
    hold on
    plot(heat_flux(1:L), St4(1:L), 'rs');
    plot(heat_flux(1:L), St5(1:L), 'gd');
    ti(1:2) = test_num;
    ti(3:26) = ' St vs. Heat Flux P_r=';
    ti(27:31) = p4_str(1:5);
    ti(32:34) = ' ';  
    ti(35:37) = m_dot_str;
    ti(38:42) = ' mg/s';
    title(ti);
    clear ti;
    xlabel('Heat Flux (W/mm^2)');
    ylabel('St');
    legend('upstream', 'midpoint', 'downstream',2);
end
clear;

% HEAT TRANSFER DATA REDUCTION PROGRAM FOR ETHANOL

%%% CONSTANTs %%%% CHECK%%% 
tc_cor = 24576;  % thermocouple correction for K type tc’s
OD = 300e-6;  % outer diameter of tube in m
ro = OD/2;  % outer radius of tube in m
ID = 95e-6;  % inner diameter of tube in m
ri = ID/2;  % inner radius if tube in m
l = 4e-3;  % length of the tube in m
Vol = l*pi*((ro)^2-(ri)^2);  % volume of steel tube wall (m^3)
Ai = pi*ID*l;  % inside surface area of the tube (m^2)
Ao = pi*OD*l;  % outside surface area of the tube (m^2)
Ae = pi*ri*l;  % inlet / exit area of the tube
Tpc = 243;  % Critical temperature of ethanol in C

%%% Read data from spreadsheet-like file%%% 
filename = ['data/ethanol/1010.txt' 
'data/ethanol/1017.txt' 
'data/ethanol/1021.txt' 
'data/ethanol/2017.txt' 
'data/ethanol/3010.txt' 
'data/ethanol/3017.txt' 
'data/ethanol/4010.txt' 
'data/ethanol/4017.txt' 
'data/ethanol/5010.txt'];

output_file_name = ['data/reduced_data/ethanol/1010.dat' 
'data/reduced_data/ethanol/1017.dat' 
'data/reduced_data/ethanol/1021.dat' 
'data/reduced_data/ethanol/2017.dat' 
'data/reduced_data/ethanol/3010.dat' 
'data/reduced_data/ethanol/3017.dat' 
'data/reduced_data/ethanol/4010.dat' 
'data/reduced_data/ethanol/4017.dat' 
'data/reduced_data/ethanol/5010.dat'];
for i = 1:9,

% CLEAR OLD ARRAYS
clear T3, clear T4, clear T5;
clear T3w, clear T4w, clear T5w;
clear T3w_o, clear T4w_o, clear T5w_o;
clear T3b, clear T4b, clear T5b;
clear T3f, clear T4f, clear T5f;
clear V, clear I, clear power;
clear Q, clear heat_flux, clear h_in, clear C3, clear C4, clear C5;
clear h3_out, clear h4_out, clear h5_out, clear h3, clear h4, clear h5;
clear cp, clear St3, clear St4, clear St5, clear G;
clear data;

% testnum = filename(i,14:17);
disp(test_num);
fid = fopen(filename(i,1:21),'r');
line = fgetl(fid);
test_cond(1:9) = str2num(fgetl(fid));
P = test_cond(1) * 6.894757e-2; % line pressure in bar (from psi)
p_str = num2str(P);
m_dot = test_cond(2) / 1e6;
c3 = test_cond(3);
c4 = test_cond(4);
c5 = test_cond(5);
xd3 = test_cond(6);
xd4 = test_cond(7);
xd5 = test_cond(8);
x3 = xd3*ID/l;
x4 = xd4*ID/l;
x5 = xd5*ID/l;
T_in = test_cond(9); % Temperature at the inlet of the test section in degrees C
h_in = tp2h_c2h6o(P,T_in); % enthalpy at tube inlet (J/kg)
line = fgetl(fid);
j = 1;
while line ~= -1
   line_str = str2num(line);
   V(j) = line_str(1); % voltage (V)
% THERMOCOUPLE CORRECTION

```
milli_V = V(j)*c3;
Tcfactor = tc_cor*milli_V;
T3(j) = T3(j) + Tcfactor;
milli_V = V(j)*c4;
Tcfactor = tc_cor*milli_V;
T4(j) = T4(j) + Tcfactor;
milli_V = V(j)*c5;
Tcfactor = tc_cor*milli_V;
T5(j) = T5(j) + Tcfactor;
```

```
Q(j) = I(j).*V(j);
heat_flux(j) = Q(j)/(Ai*1e6); % W/mm^2
```

```
PROPERTY TABLE INDEPENDENT

C3(j) = -1*(Q(j)/(2*pi*l)*((log(ro/ri)/(1-(ri/ro)^2))-0.5)) + 0.0076*T3(j)*A2 + 14.2444*T3(j);
T3w(j) = (-14.2444 + sqrt(14.2444*A2+4*.0076*C3(j)))/(2*.0076); % wall temp (C)
C4(j) = -1*(Q(j)/(2*pi*l)*((log(ro/ri)/(1-(ri/ro)^2))-0.5)) + 0.0076*T4(j)*A2 + 14.2444*T4(j);
T4w(j) = (-14.2444 + sqrt(14.2444*A2+4*.0076*C4(j)))/(2*.0076);
C5(j) = -1*(Q(j)/(2*pi*l)*((log(ro/ri)/(1-(ri/ro)^2))-0.5)) + 0.0076*T5(j)*A2 + 14.2444*T5(j);
T5w(j) = (-14.2444 + sqrt(14.2444*A2+4*.0076*C5(j)))/(2*.0076);
h3_out(j) = h_in + ((Q(j).*x3)./m_dot); % enthalpy at tube exit (J/kg)
h4_out(j) = h_in + ((Q(j).*x4)./m_dot);
h5_out(j) = h_in + ((Q(j).*x5)./m_dot);
T3b(j) = hp2t_c2h6o(h3_out(j),P); % bulk temperature in degrees C
T4b(j) = hp2t_c2h6o(h4_out(j),P);
T5b(j) = hp2t_c2h6o(h5_out(j),P);
h3(j)= Q(j)./(Ai.*(T3w(j) - T3b(j))); % heat transfer coefficient (W/m^2.K)
h4(j)= Q(j)./(Ai.*(T4w(j) - T4b(j)));
h5(j)= Q(j)./(Ai.*(T5w(j) - T5b(j)));
```
\[ G(j) = \frac{m_{\text{dot}}}{Ae}; \text{ mass velocity (kg/s.m^2)} \]

PROPERTY TABLE DEPENDENT

\[ T_{3f}(j) = \frac{T_{3w}(j)+T_{3b}(j)}{2}; \text{ film temperature of fluid (C)} \]
\[ T_{4f}(j) = \frac{T_{4w}(j)+T_{4b}(j)}{2}; \]
\[ T_{5f}(j) = \frac{T_{5w}(j)+T_{5b}(j)}{2}; \]

\[ h_{3w}(j) = t_{p2h}-h_{2o}(P_{T3w}(j)); \text{ wall enthalpy (J/kg)} \]
\[ h_{3b}(j) = t_{p2h}-h_{2o}(P_{T3b}(j)); \text{ bulk enthalpy (J/kg)} \]
\[ St_{3}(j) = \frac{Q(j)/A_i}{(m_{\text{dot}}/Ae)*(h_{3w}(j)-h_{3b}(j))}; \text{ Stanton number} \]

\[ h_{4w}(j) = t_{p2h}-h_{2o}(P_{T4w}(j)); \]
\[ h_{5w}(j) = t_{p2h}-h_{2o}(P_{T5w}(j)); \]

\[ h_{4b}(j) = t_{p2h}-h_{2o}(P_{T4b}(j)); \]
\[ h_{5b}(j) = t_{p2h}-h_{2o}(P_{T5b}(j)); \]

\[ St_{4}(j) = \frac{Q(j)/A_i}{(m_{\text{dot}}/Ae)*(h_{4w}(j)-h_{4b}(j))}; \]
\[ St_{5}(j) = \frac{Q(j)/A_i}{(m_{\text{dot}}/Ae)*(h_{5w}(j)-h_{5b}(j))}; \]

\[ j = j + 1; \]
\[ \text{line} = \text{fgetl(fid)}; \]
\[ \text{end} \]
\[ L = j - 1; \]
\[ \text{st} = \text{fclose(fid)}; \]

WRITE DATA TO FILE

\[ \text{data} = [P; m_{\text{dot}}; ID]; \]
\[ \text{fid} = \text{fopen(output_file_name(i,1:34),'w')} ; \]
\[ \text{fprintf(fid, '12.4f 12.6f 12.6f
',data); \]
\[ \text{st} = \text{fclose(fid)}; \]
\[ \text{data} = [\text{heat_flux}; T3; T4; T5; T3w; T4w; T5w; T3b; T4b; T5b; St3; St4; St5; G]; \]
\[ \text{fid} = \text{fopen(output_file_name(i,1:34),'a')} ; \]
\[ \text{fprintf(fid, '12.4f 12.4f 12.4f 12.4f 12.4f 12.4f 12.4f 12.4f 12.4f 12.4f 12.4f 12.4f 12.4f',data); \]
\[ \text{st} = \text{fclose(fid)}; \]
fig = 0;

% plot surface temp, inside wall temp, and bulk fluid temp vs. heat flux
plt = input('Do you want to plot temperature vs. heat flux? (y/n def = n) ','s');
if isempty(plt) == 1
    plt = 'n';
end
if plt == 'y'
    fig = fig+1;
    figure(fig)
    hold off, clf
    subplot(3,1,1), plot(heat_flux(1:L), T3(1:L), 'bo');
    hold on
    plot(heat_flux(1:L), T3w(1:L), 'rs');
    plot(heat_flux(1:L), T3b(1:L), 'gd');
    ti(1:4) = testnum;
    ti(5:42) = ' Temperature upstream vs. Heat Flux  ';
    ti(43:45) = p_str(1:3);
    ti(46:49) = 'bar';
    title(ti);
    clear ti;
    xlabel('Heat Flux (W/mm^2)');
    ylabel('Temperature (C)');
    legend('Outside Surface Temp', 'Inside Wall Temp', 'Bulk Fluid Temp',2);

    subplot(3,1,2), plot(heat_flux(1:L), T4(1:L), 'bo);
    hold on
    plot(heat_flux(1:L), T4w(1:L), 'rs');
    plot(heat_flux(1:L), T4b(1:L), 'gd');
    ti(1:4) = testnum;
    ti(5:42) = ' Temperature midpoint vs. Heat Flux  ';
    ti(43:45) = p_str(1:3);
    ti(46:49) = 'bar';
    title(ti);
    clear ti;
    xlabel('Heat Flux (W/mm^2)');
    ylabel('Temperature (C)');
legend('Outside Surface Temp', 'Inside Wall Temp', 'Bulk Fluid Temp.2);

subplot(3,1,3), plot(heat_flux(1:L), T5(1:L), 'bo');
hold on
plot(heat_flux(1:L), T5w(1:L), 'rs');
plot(heat_flux(1:L), T5b(1:L), 'gd');
ti(1:4) = test_num;
ti(5:44) = ' Temperature downstream vs. Heat Flux ';
ti(45:47) = p_str(1:3);
ti(48:51) = ' bar';
title(ti);
clear ti;
xlabel('Heat Flux (W/mm^2)');
ylabel('Temperature (C)');
legend('Outside Surface Temp', 'Inside Wall Temp', 'Bulk Fluid Temp.2);
end

% plot heat transfer coefficient vs. heat flux
plt = input('Do you want to plot heat transfer coefficient vs. heat flux? (y/n def = n) ','s');
if isempty(plt) == 1
    plt = 'n';
end
if plt == 'y'
    fig = fig+1;
    figure(fig)
    hold off, clf
    plot(heat_flux(1:L), h3(1:L), 'bo');
    hold on
    plot(heat_flux(1:L), h4(1:L), 'rs');
    plot(heat_flux(1:L), h5(1:L), 'gd');
ti(1:4) = test_num;
ti(5:47) = ' Heat Transfer Coefficient vs. Heat Flux ';
ti(48:50) = p_str(1:3);
ti(51:54) = ' bar';
title(ti);
clear ti;
xlabel('Heat Flux (W/mm^2)');
ylabel('Heat Transfer Coefficient (W/m^2.K)');
legend('upstream', 'midpoint', 'downstream',2);
% plot Tw-Tb vs. heat flux
plt = input('Do you want to plot Tw-Tb vs. heat flux? (y/n def = n) ','s');
if isempty(plt) == 1
    plt = 'n';
end
if plt == 'y'
    fig = fig+1;
    figure(fig)
    hold off, clf
    plot(heat_flux(1:L), T3w(1:L)-T3b(1:L), 'bo');
    hold on
    plot(heat_flux(1:L), T4w(1:L)-T4b(1:L), 'rs');
    plot(heat_flux(1:L), T5w(1:L)-T5b(1:L), 'gd');
    ti(1:4) = test_num;
    ti(5:27) = 'Tw-Tb vs. Heat Flux';
    ti(28:30) = p_str(1:3);
    ti(31:34) = 'bar';
    title(ti);
    clear ti;
    xlabel('Heat Flux (W/mm^2)');
    ylabel('Tw-Tb (C)');
    legend('upstream', 'midpoint', 'downstream',2);
end

% plot Stanton Number vs. heat flux
plt = input('Do you want to plot St vs. heat flux? (y/n def = n) ','s');
if isempty(plt) == 1
    plt = 'n';
end
if plt == 'y'
    fig = fig+1;
    figure(fig)
    hold off, clf
    plot(heat_flux(1:L), St3(1:L), 'bo');
    hold on
    plot(heat_flux(1:L), St4(1:L), 'rs');
    plot(heat_flux(1:L), St5(1:L), 'gd');
    ti(1:4) = test_num;
ti(5:24) = ' St vs. Heat Flux ';
ti(25:27) = p_str(1:3);
ti(28:31) = ' bar';
title(ti);
clear ti;
xlabel('Heat Flux (W/mm^2)');
ylabel('St');
legend('upstream', 'midpoint', 'downstream',2);
end
end
end
Appendix B

Temperature Plots for Calibration Tests

These plots are the temperature as a function of heat flux plots for the calibration tests that were conducted for the water runs. Test 09 is the calibration test for water test 10, calibration test 11 corresponds to test 12, calibration test 13 corresponds to test 14, and calibration test 15 corresponds to tests 16, 17, and 18. The dotted line on each plot shows the location of the saturation temperature for the given pressure.

Figure B.1: Calibration test at 20 bar, saturation T = 212 °C
Figure B.2: Calibration test at 23 bar, saturation T = 220°C
Figure B.3: Calibration test at 24 bar, saturation T = 222°C
Figure B.4: Calibration test at 24 bar, saturation $T = 222^\circ C$
References


18: Popov, Distinctive Features of Heat Exchange During Turbulent Forced and Mixed Convection of a Fluid in the Supercritical Region


