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HEAT TRANSFER COEFFICIENT IN SERPENTINE COOLANT PASSAGE FOR CCDTL*

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Abstract

A series of heat transfer experiments were conducted to refine the cooling passage design in the drift tubes of a coupled cavity drift tube linac (CCDTL) [1]. The experimental data were then compared to numerical models to derive relationships between heat transfer rates, Reynold's number, and Prandtl number, over a range of flow rates. Data reduction consisted of axisymmetric finite element modeling where the heat transfer coefficients were modified to match the experimental data. Unfortunately, the derived relationship is valid only for this specific geometry of the test drift tube. Fortunately, the heat transfer rates were much better (approximately 2.5 times) than expected.

1 INTRODUCTION

The objective of this experiment was to use experimental results combined with numerical simulation to measure heat transfer rates in drift tube coolant passages for the cavities in the Accelerator Production of Tritium (APT) [2], Low Energy Demonstration Accelerator (LEDA) [3] CCDTL Hot Model. The hot model is a full scale, copper brazed structure that will be exposed to full RF fields, but will not have beam through it. A goal of the experiment is to refine the design of the cooling passages and coolant systems for the LEDA CCDTL. The results of this experiment were used to give a better estimate of the heat transfer rates within the drift tube coolant passages and are just a first look at the drift tube thermal problem. Since the experiment is not error-free, the Nusselt equation coefficients determined are probably not an exact representation of all the physics of the problem, but a match with this empirical data using the specific geometry of the test item.

In the CCDTL, the drift tube is located within an RF cavity and provides a region of no electric field which shields the beam when the electric field would decelerate the beam (for an in-depth description, see [4]). A great deal of RF power is dissipated on the outer surface of the APT drift tubes. A method was developed to form an elaborate network of cooling passages within the body of each drift tube [5]. The coolant passages within the drift tubes are rectangular, short, and curved, a situation which is not well covered in the literature.

In the literature [6], the heat transfer coefficient in long, straight, circular passages is given as

$$\overline{h}_c = \frac{k_{water}}{D_{Tube}} * 0.023 * Re^{0.8} * Pr^{0.4} \qquad (1)$$

where k_{water} is the thermal conductivity of water, Re is the Reynold's number, Pr is the Prandtl number. Since these drift tube passages are not round, the convention is to use the equivalent hydraulic diameter for a rectangular cross section which is given by

$$D_h = \frac{4A}{P}$$

where A is the flow area and P is the wetted perimeter. It is much more difficult to account for the passages being short and curved. The complex three dimensional geometry of the drift tube coolant passages make it difficult to determine an effective heat transfer coefficient directly from published data. It was necessary to use a finite element, thermal/structural model to extract an approximate value for the heat transfer coefficient. From that data, an approximate relationship between the Nusselt number, the Reynold's number, and the Prandtl number for this geometry was derived.

2 SETUP

The test setup consisted of a water chiller, approximately 5 gallon reservoir, a flow meter with range of 0 to 2 gpm, water filter, 17 heater cartridges, rheostat, 100X amplifier, a modified drift tube slug placed on a styrofoam base with styrofoam "popcorn" completely over it, tubing to connect these components together, two thermocouples to measure drift tube temperatures, another thermocouple to measure coolant inlet temperature, a two pass thermopile to measure the coolant temperature rise through the drift tube, and a data acquisition system to record the data. Figure 1 shows a schematic of the setup. For data reduction purposes, the flow rate was determined from the heater power and the temperature rise within the coolant from inlet to outlet. Much depends on this measurement, so a two pass thermopile was used to increase the sensitivity of the measurement and lessen the effect of noise in the data.

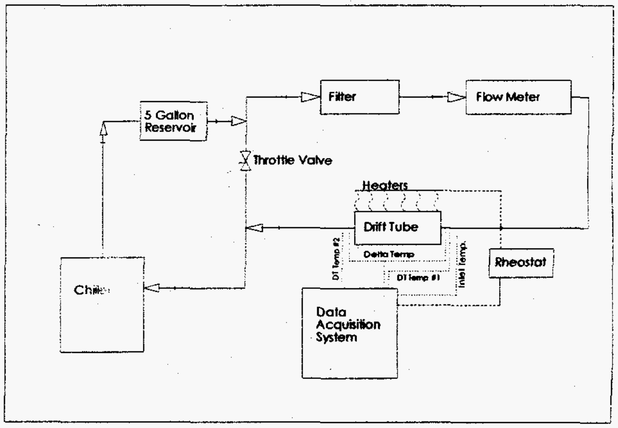

Figure 1. Schematic of the experiment.

The drift tube slug used in the heat transfer experiments was a three passage drift tube that is identical to those from which CCDTL hot model drift tubes were made. Figure 2 shows the cross sectional drawing showing the 3

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concentric cylinders and the 3 coolant passages. The center, longitudinal holes were drilled in 1 inch each side (not shown in Figure 2) with a type T thermocouple inserted into each. These thermocouples were labeled Body Temp 1 and Body Temp 2. Because the water temperature increases with each successive passage, the *area of the drift tube that dissipates the most power and/or* most affects the cavity frequency needs to be cooled first. So the placement of the drift tube coolant passages is not arbitrary and the experimental drift tube passages closely resembled an actual drift tube.

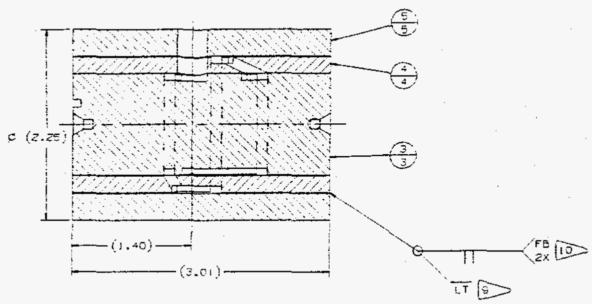

Figure 2. Drift Tube Slug Showing Cross Sections of Cooling Passages.

The heater cartridges were placed in longitudinal slots cut into the drift tube slug with thermal conducting grease, copper shim stock wrapped around, and hose clamps to keep them in place. Figure 3 shows the assembly in the styrofoam box with associated hardware prior to filling the box with styrofoam "popcorn" and topping it with a foam pad. The inlet thermocouple and the thermopile were installed into the hardware at the ends of the copper tubes protruding from the drift tube shown in Figure 3.

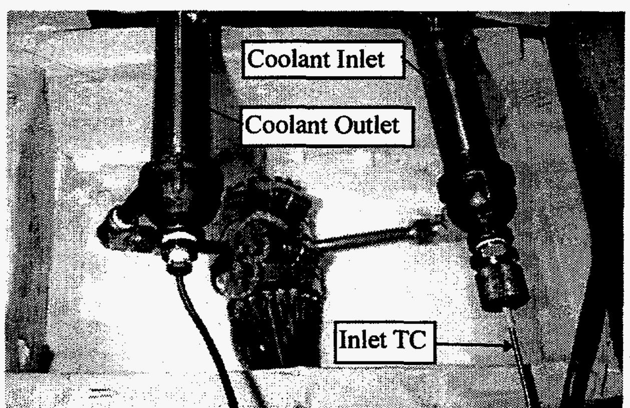

Figure 3. Drift Tube in Styrofoam box.

The reservoir was used to increase the thermal mass of the system so each time step was more closely steady state. The rheostat was used to control the power to the heaters. The data acquisition system consisted of a computer running Labview software, two Keithley 2002 Multimeters, three type T thermocouples with electronic "ice point" and one type T thermopile, and the data was recorded into text files for easy transfer to other data manipulation software. A thermistor was used to monitor the outlet temperature, but it was not recorded.

3 THERMOCOUPLE CALIBRATION

The process began by identifying the offset inherent in the system. This was done by setting the thermostat on the chiller to its lowest point of 42° F. Once the system reached this temperature, the flow rate was throttled way down (~0.30 to 0.35 gpm) to minimize heating due to pressure loss across the drift tube, the thermostat on the chiller was raised to approximately 110° F, and the data acquisition system began recording the data. The heaters and rheostat were turned off and unplugged from the wall. It took the system 2 to 2 ½ hours to reach approximately 75° F and much longer if the desired system temperature was near 100° F. When the system reached ~75° F, the data recording was stopped, the flow rate was turned up, and the temperature on the chiller was set down to its lowest point.

4 TEST PROCEDURE

Each measurement was assumed to be steady state due to the high thermal conductivity of copper and the 5 gallon reservoir added to increase the thermal mass of the system. The chiller system's compressor cycled too much to hold the temperature constant so it was used only as a pump and as a means to cool the entire system to an initial <45° F condition. Quasi-steady state data was then taken as the drift tube heaters gradually drove the system temperature upward.

When the system temperature was 42° F, the heaters and rheostat were turned on, near 1000 watts, and the desired flow rate was set. Once the system reached quasi-steady state, the chiller thermostat was set to 100° F and the data was then recorded as the system temperature gradually climbed to >100° F. The process took 1 ½ to 2 hours, depending on the flow rate. Figure 4 shows a plot of the data collected at one specific flow rate. The thermocouple data was smooth with very little noise; however, the thermopile data (Delta temperature) had a 3% noise range throughout due to the sensitivity of the measurement. This is believed to be stray electrical noise, not variations in the flow rate. Once the inlet temperature reached approximately 100° F, the data collection was stopped, the heaters and rheostat were turned off, and the chiller thermostat was turned down to its lowest setting. It took approximately 1 hour to return to the 42° F starting point. The procedure was then repeated using a different flow rate.

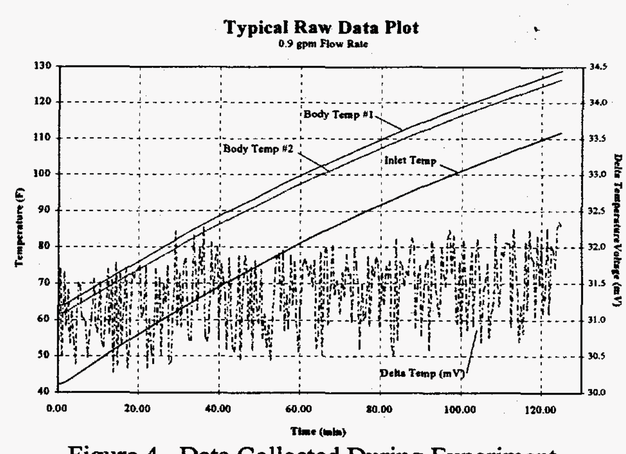

Figure 4. Data Collected During Experiment.

5 ANALYSIS

The analysis was performed in two sections. The first was to analyze the data collected to determine the delta temperature between the outlet and inlet. The thermocouple calibration was "backed out" and the resulting measurements converted to temperatures. As can be seen in Figure 4, there was a significant range of data for the voltage measurement which translates over to the temperature measurement. To smooth out this data, a weighted time average was taken from the surrounding data points. The weighted average equation is

$$\overline{X}_N = \frac{X_{N-2} + 3X_{N-1} + 5X_N + 3X_{N+1} + X_{N+2}}{13} \qquad (3)$$

where X_{N-x} was a data point x steps before or after the N^{th} data point. All quantities (temperatures and heater power) were averaged this way. This smoothed data was then entered into a spreadsheet [7] that calculated the average heat transfer coefficients and temperatures for each passage on the drift tube.

The second part of the analysis used numerical simulation, specifically COSMOS/M finite element analysis (FEA) software [8]. An axisymmetric model was generated within COSMOS where the boundary conditions were taken from the spreadsheet [7] which calculated the average heat transfer coefficients, average coolant temperatures, and the heat flux from the heaters. The model was then thermally analyzed to determine the nodal temperatures in the model. The comparison between the measured data and the numerical data was made by averaging the two measured body temperatures and comparing them to the average nodal temperatures that correspond to those thermocouple's location. One iteration required modifying the heat transfer coefficient, applying the calculated boundary conditions to the FEA model, running the thermal analysis, averaging the temperatures at the nodes corresponding to the thermocouple's location, and comparing it to the averaged measured body temperature and repeating until the two averages were within ±0.03° F (even though the accuracy of the measurement was much worse). These iterations were performed at 4 to 5 temperatures within the measured temperature range for each flow rate.

6 RESULTS

Four flow rates were analyzed in this experiment: 0.9, 0.7, 0.5, 0.3 gallons per minute (gpm) which corresponded to approximately 15, 12, 9, and 6 feet per second (fps) (4.57, 3.66, 2.74, and 1.83 m/s) flow velocity in the coolant channels, respectively. The Reynold's numbers corresponding to these flow rates are 15,549; 11,159; 8,275; and 5,291, respectively. The data can be fairly well described by the following equation.

$$Nu = .0862 * Re^{.75} * Pr^{.42} \qquad (5)$$

Figure 5 compares the data to this equation graphically.

The design flow velocity within the passages of the drift tube is a critical factor due to erosion of the copper passages at higher flow velocities. A rule of thumb is to keep flow velocities within copper coolant channels below 15 fps (4.57 m/s) to minimize this erosion. Therefore, there was no need to test a flow rate higher than 0.9 gpm.

Note that, for this geometry, the heat transfer rates are ~2.5 times greater than predicted by Eqn (1) (long, straight, circular passages). For the design on the APT/LEDA CCDTL Low Beta Hot Model drift tubes, a value of 1.5 times better was used to offset any experimental errors that may have influenced the data and to stay on the conservative side of the design. Thorough tests of the Low Beta Hot Model will be done to verify and refine these results.

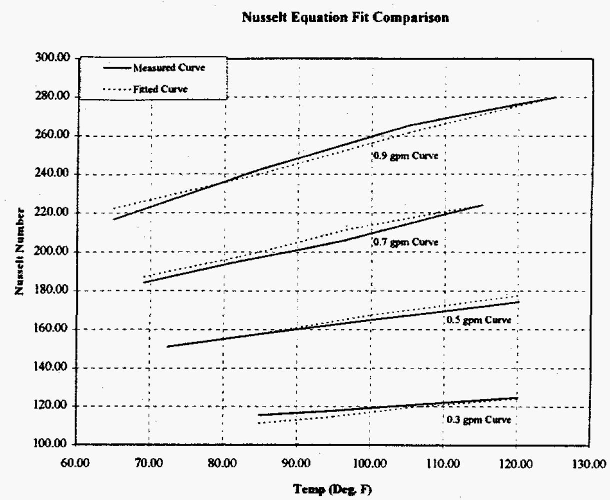

Figure 5. Data fit comparison.

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