LOCAL AND SPATIALLY AVERAGED HEAT TRANSFER DISTRIBUTIONS IN A CURVED CHANNEL WITH 40 TO 1 ASPECT RATIO FOR DEAN NUMBERS FROM 50 TO 200

by
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March 1990

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### Abstract
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Locally and Spacially Averaged Heat Transfer Distributions in a Curved Channel with 40 to 1 Aspect Ratio for Dean Numbers from 50 to 200

by

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Submitted in partial fulfillment of the requirements for the degree of

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The effects of curvature and the resulting centrifugal instabilities on local heat transfer distributions are studied in a curved channel at Dean numbers ranging from 50 to 200. The channel has a rectangular cross section of 1.27 cm by 50.1 cm giving an aspect ratio of 40 to 1. Flow is heated in a straight portion of the channel prior to the curved portion in order to obtain flow which is hydrodynamically and thermally fully developed. All baseline tests confirm techniques employed and qualify flow behavior. These consist of energy balance checks and comparison of results from the straight section to numerical and analytic solutions. Nusselt numbers in the curved section initially show an abrupt decrease after the imposition of the stabilizing influences of convex curvature. These are followed by a gradual increase as centrifugal instabilities and Dean vortices form and develop. Spacially resolved results also show significant surface Nusselt number variations across the span of a vortex pair, especially on the concave surface. On the convex surface, local Nusselt numbers are much more apt to be spanwise uniform.
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I. INTRODUCTION

A. DESCRIPTION OF THE PROBLEM

The effects of curvature and the resulting centrifugal instabilities on local heat transfer distributions are studied using a curved channel with an internal aspect ratio of 40 to 1. Results are presented for Dean numbers ranging from 50 to 200, resulting in laminar flow which sometimes contains pairs of Dean vortices. Other work at the Naval Postgraduate School in this area addresses the development of vortex flows due to centrifugal instabilities and the effects of these secondary flows on transition from laminar to turbulent behavior without considering heat transfer. Ligrani and Niver [Ref. 1] describe some results from this study which are also obtained in a curved channel with a similar aspect ratio of 40 to 1. These authors also review other non-thermal studies.

Several other studies focus on heat transfer in channels with turbulent flow. Johnson and Launder [Ref. 2] present local heat transfer results from a channel with a 180 degree bend and square cross section. They emphasise that instabilities from curvature produce very complex examples of turbulent flows due to substantial secondary flows. Additional studies are described by Chang, Humphrey, Johnson and Launder [Ref. 3], who employ a channel with a 180 degree bend of square cross section, and by Brinch and Graham [Ref. 4], who use a curved channel with an aspect ratio of 6. Studies dealing with the effects of heat transfer on turbulent boundary layers include ones described by Gibson, Verriopoulos and Nagano [Ref. 5], Simon and Moffat [Ref. 6] and Chang, Humphrey and Modavi [Ref. 7].
No studies, other than the present one are known to the present authors, which consider the effects of centrifugal instabilities and vortex flows on local heat transfer distributions in laminar and transitional flow in a channel with 40 to 1 aspect ratio. Understanding such curved flows is important to the design of devices such as heat exchangers and gas turbine blades.

B. OBJECTIVES

The objective of this thesis is to provide experimental evidence as to how curvature affects local heat transfer distributions on the surfaces of a curved channel for Dean numbers from 50 to 200. This is accomplished by:

1. Qualifying experimental procedures, measurement techniques and channel facilities by conducting several baseline comparisons on the straight portion of the channel.

2. Provide accurate Nusselt number data for both surfaces of the 180 degree curved channel, thus providing information as to how centrifugal instabilities and the resulting vortex pairs affect local heat transfer distributions.

C. ORGANIZATION

Subsequent to this introduction, Chapter II details the design and construction of the curved heated channel used in this study. Chapter III discusses the experimental procedures and measurement techniques. Chapter IV provides a series of baseline qualification comparisons and then presents the results of the Nusselt number measurements. Chapter V presents conclusions and recommendations.
II EXPERIMENTAL FACILITIES

A. CURVED HEATED CHANNEL

The apparatus used in this study is a rectangular curved channel shown in Figure 1. Within this facility, local surface temperatures and other quantities are measured so that the effects of curvature on local Nusselt number distributions can be studied. To coincide with work done on a transparent channel used for flow visualization, shown in Figure 2, identical internal dimensions are used for the heat transfer channel. The channel has an interior rectangular cross section of 1.27 cm by 50.1 cm, giving an aspect ratio of 40 to 1. The heat transfer channel is different from the transparent channel since it allows for thermal expansion which results from heating. This is accomplished by non-rigid securing of channel surfaces and by constructing much of the channel on skids to allow for longitudinal expansion. A detailed description of the design and construction of the channel is presented by Hughes [Ref. 8]. A brief summary of design and construction details is also given here.

1. Channel Construction

A photograph of the channel for heat transfer measurements is shown in Figure 1. Details of the inlet section are shown in Figure 3. The rectangular duct inlet section contains an aluminum honeycomb and three wire screens all placed normal to the flow direction. These devices reduce spacial non-uniformities in the flow. As the air exits the 25.4 cm by 50.8 cm rectangular duct inlet, it enters a 20 to 1 contraction ratio nozzle. The nozzle is constructed from the same two continuous pieces of lexan as used for the straight and curved test section walls, thus eliminating a seam between the nozzle and the channel. Lexan is the
commercial name of a polycarbonate material. The shape of the nozzle is designed using a fifth order polynomial with respect to streamwise distance to ensure that the flow remains laminar and unseparated as it enters the channel.

After exiting the inlet nozzle, air first enters a straight section, 244 cm (96 inches) in length with interior dimensions of 1.27 cm by 50.8 cm (.5 inches by 20 inches). The straight section allows hydrodynamically fully developed channel flow to develop before the flows enter the curved section. The fluid then enters a 180 degree curved channel section with a convex surface radius of 59.69 cm (23.5 inches) and a concave surface radius of 60.96 cm (24 inches). Upon exiting the curved section, air then enters a second straight section with a length of 244 cm (96 inches).

On the convex surface of the second straight section, a 5.08 cm (2.0 inches) wide slot is present which allows insertion of a probe into the channel to measure outlet mixed mean temperatures. The slot is located approximately 57.1 cm (22.5 inches) from the trailing edge of the heater on the curved convex wall, or 19.05 cm (7.5 inches) downstream of the end of curvature. A support block was constructed to support the area around the slot. Brass inserts are placed just inside the slot to allow insertion of two thin foam strips which adhere to the brass to provide an air tight seal between the channel interior and exterior as the probe is traversed. As flow leaves the second straight portion it passes through four screens, a honeycomb, a diffuser with a total angle of 3 degrees, and finally into Plenum #1.

The channel is designed to allow for thermal expansion in longitudinal and transverse directions which occurs as test surfaces are heated. As is illustrated in Figure 7 [Ref. 8], the surfaces of the channel are not rigidly secured, but are clamped onto the sidewalls to allow for such motion. In addition, both the inlet and outlet sections, including Plenum #1,
are constructed on two plexiglass skids to allow for longitudinal expansion.

2. Blower Assembly

The blower and plumbing connecting it to the channel are shown schematically in Figure 4 [Ref. 8]. The device itself is an ICG Industries type 10P blower, capable of producing 10.2 cm of water vacuum at $4.82 \, m^3/min$ volumetric flowrate. It is used to depressurize outlet Plenum #2 to pressures just below atmospheric pressure. In doing so, air is then drawn from Plenum #1 through a globe valve and a 5.08 cm inside diameter PVC pipe. With this system, flowrates can be varied to produce Dean numbers from 10 to 435. Flowrates are measured by means of a pressure drop across a 3.51 cm (1.5 inch) orifice plate. Details concerning the orifice plate are given in Hughes [Ref. 8] and also discussed in Chapter 3.

To reduce vibrations from the blower to the test section and to minimize disturbances to the flow, a damper is used to connect the suction side of the blower to Plenum #2 and a second damper is used to connect Plenum #2 to the orifice plate. Both dampers consist of thin sheets of plastic wrapped around the piping with air tight seals. The plastic is attached at both ends by double sided tape and then tightly secured with standard hose clamps in order to prevent any leaks. In addition, rubber mounts are used to mount the blower and blower frame to further reduce vibrations. A photograph of this assembly is shown in Figure 5.

3. Heaters

Four etched foil heaters, manufactured by the Electrofilm Corporation, are installed on each of the two lexan surfaces, as shown in Figure 6 [Ref. 8]. In the present study, these heaters provide a constant heat flux to both the concave and convex surfaces of the channel, all at the same heat rates, to produce uniform boundary conditions at the channel walls. The
dimensions of each heater are 38.1 cm by 152.4 cm (15.0 inches by 60.0 inches), and power capacity is 2 KW. Drawings indicating how the heaters are attached are shown in Figure 7 and Figure 8 [Ref. 8] and are discussed further by Schwartz [Ref. 9]. The leading edge of the CC1 and CV1 heaters are 88.4 cm from the channel inlet. The trailing edges of the CC2 and CV2 heater ends approximately 38.1 cm (15.0 inches) from the completion of channel curvature. This is about 57.15 cm (22.5 inches) upstream of the location of the temperature traverse probe. CC1 and CV1 refer to the two heaters on the concave and convex surfaces of the straight section, respectively, and CC2 and CV2 refer to the heaters on the concave and convex surfaces of the curved section, respectively.

Each heater is powered by a Superior Electric type 136B variac as shown in Figure 9 [Ref. 8]. With each variac, the voltage to a heater may be adjusted between 0 and 115 volts, and the current may be adjusted from 0 to 10 amps. A detailed schematic of the heater circuitry is given by Hughes [Ref. 8] and also shown in Figure 10. Power inputs to each heater are determined by measuring voltage drops across 50 mv, 10 amp shunt resistors, as well as across each heater. The shunt resistance measurement allows for the heater current to be determined. Details in adjusting desired heat input levels are given in Chapter 3.

4. Insulation

To minimize the loss of conducted heat to the surroundings, and maximize the amount of heat into each heater which is convected away, the outside of the entire heated portion of the channel is insulated with black foam insulation manufactured by the Halstead Company. A detailed drawing showing how various layers of insulation are attached to the channel to produce a total thickness of 6.0 cm (2.36 inches) on each side is illustrated in Figure 7.
For this study, an additional row of 2.5 cm (1.0 inch) insulation is attached along all edges of the heated channel to further minimize conduction from heated channel walls. Photographs of the insulation are shown in Figure 11 and Figure 12.

5. Thermocouples

200 copper-constantan thermocouples are placed on channel surfaces to allow detailed spatially resolved surface temperature measurements to be made. Figure 8 [Ref. 8] shows the approximate locations of each thermocouple. The exact locations are listed in Appendix A, as determined by measurements conducted as a part of this study. Measurements are based on a coordinate system shown in Figures 13 and 14. Here X is in the streamwise direction measured from the inlet of the straight section, Z is in the spanwise direction and measured from the centerline, and Y is normal to channel surfaces in the vertical direction measured from the channel midplane.

On the straight section, 15 thermocouples are placed on each of the CC1 and CV1 surfaces in 5 spanwise rows of 3 per row. The first row is located 103.64 cm (40.8 inches) from the inlet or 15.24 cm (6 inches) from the leading edge of the heater. Each additional row is spaced 30.48 cm (12 inches) apart in the X direction. The remaining 170 thermocouples are located on the curved section walls in 5 spanwise rows of 17 per row. Again, each row is spaced 30.48 cm (12 inches) apart in the X direction. Of the 17 thermocouples in each spanwise row, two are located near the edges of the channel with the remaining 15 placed over a spanwise length of approximately 5.08 cm (2.0 inches) on each side of the centerline. The purpose of these thermocouples is to provide an accurate average local Nusselt number, as well as a means of determining local Nusselt numbers in the presence of two pairs of Dean vortices [Ref. 8].
Four additional thermocouples are used to measure ambient temperatures around the outside of the channel, mixed mean temperature at the air inlet and mixed mean temperature at the channel outlet. Thermocouple (TC) #200 is placed just in front of the inlet and provides the mixed mean inlet temperature. TC #201 and TC #202 are placed respectively on the outsides of the CC2 and CV2 surfaces to provide local ambient temperatures. Finally, TC #203 is used in the temperature traverse probe to provide measurement of local temperatures used to calculate outlet mixed mean temperatures.

Each of the 204 thermocouples is attached to a data acquisition channel and the corresponding channel numbers are also listed in Appendix A. Briefly, thermocouples 1 to 30 correspond to acquisition channels 0 to 29, thermocouples 31 to 115 to channels 30 to 79 and 100 to 134, and finally, thermocouples 116 to 204 correspond to channels 135 to 179 and 200 to 243.

B. DATA ACQUISITION

1. Surface Temperature Measurement

Voltages from the 204 thermocouples are read by Hewlett-Packard type T20 relay multiplexer card assemblies for T type thermocouples. These assemblies are installed in a HP3497A low-speed Data Acquisition /Control Unit and a HP3498A Extender. This system provides thermocouple compensation electronically such that voltages for type T thermocouples are given relative to $0^\circ$ C. This system is connected to a Hewlett-Packard 9836S computer which processes voltages from the 204 thermocouples which are then recorded into data files along with corresponding temperatures. A schematic of this system is shown in Figure 15 [Ref. 8].
2. Outlet Mixed Mean Temperature Measurement

The acquisition system used for exit mixed-mean temperature measurements is illustrated in Figure 16 [Ref. 8]. The thermocouple probe used to measure the mixed mean temperature at the exit of the heated test section consists of a thermocouple attached to a metal support. This support is then secured to a traversing block. An automated two-dimensional traversing device, which allows both spanwise and radial movement, is used to traverse the temperature probe and shown in Figure 17. The movements of the motors on the traversing devise are controled by signals from the HP9836S computer, which are sent to a Modulynx Mitas PMS085-CZAR Drive Controller and then to a Modulynx Mitas PMS085-D0J0 Motion Drive. The motor Drive sends signals to each of the Superior Electric M092-FD310 stepping motors, which rotate drive screws which then move two traversing blocks. The threads of the drive screws are such that one rotation gives a vertical or lateral probe movement of .127 cm (.05 inches). For each traverse, 320 local temperatures are recorded over a channel cross section area of 5.08 cm (2.0 inches) in the spanwise direction by 1.02 cm (.4 inches) in the radial direction in increments of 0.127 cm (.05 inches). Upon completion of the traverse, the controller is programmed to return the probe to its original position. Voltage readings from the traversing thermocouple are processed in the same manner as the channel surface thermocouples.
III. EXPERIMENTAL PROCEDURES

To obtain heat transfer distributions, flow is first induced in the channel, heater power levels are adjusted to provide a constant surface heat flux boundary condition, voltages from thermocouples are read, surface temperatures are determined, and finally Nusselt numbers are calculated. Flow is laminar for all tests. Pairs of counter rotating Dean vortices are present over a large range of the Dean numbers studied, especially at farther downstream locations.

Experimental procedures are discussed in three parts. In part one, means to set flowrate and Dean number are discussed along with details on the adjustment of thermal boundary conditions. Part two, processing of surface temperature measurements and the determination of local Nusselt numbers. Finally, part three discusses the use of the temperature traverse probe, and method used to determine local and exit mixed-mean temperatures.

A. CHANNEL PREPARATION

1. Flowrate and Dean Number Setting

Local Nusselt numbers are determined at Dean numbers ranging from 50 to 200 in increments of 25 using the software program Nuscurv. The first step in each run is to set the flowrate to obtain the desired Dean number. The mass flowrate, $\dot{m}$, is determined from the equation:

$$\dot{m} = De\left[ \rho A_{ch} \frac{v}{d} \right] \sqrt{\frac{r_i}{d}}$$
Where:

$De = \text{Dean Number}$

$\rho = \text{Air density}$

$A_{ch} = \text{Channel area}$

$r_i = \text{Inside radius of channel curvature}$

d = \text{Channel height}$

$v = \text{Kinematic viscosity}$

The flowrate is set by regulating the pressure drop across a 3.81 cm (1.5 in) standard ASME orifice plate, located in piping between the two exit plenums. The initial value of this pressure drop is estimated using results such as the ones presented in Figure 18. The pressure drop, in inches of water, is then also measured using a digital monometer. Afterwards, an iteration process is used to determine the true Dean number.

As the iteration process is implemented, the pipe Reynolds number, $Re_p$, is calculated from the mass flowrate, $\dot{m}$, using the equation:

$$Re_p = \frac{\dot{m}d_p}{\rho v A_p}$$

Where:

$d_p = \text{Pipe diameter}$

$A_p = \text{Pipe cross-sectional area}$

A corrected mass flowrate is then calculated using the equation:
\[ m = KA_{on}Y\sqrt{2\rho \Delta P} \]

Where:

\( K \) = Flow coefficient (A function of Reynolds number which is interpolated from the ASME Tables [Ref. 10])

\( Y \) = Expansion coefficient (Calculated from Holman and Gajda [Ref. 11])

\( A_{on} \) = Orifice Area

\( \Delta P \) = Pressure drop across orifice

A corrected Reynolds number is then determined and the iterative process is continued until the mass flowrate converges to within one percent for two successive calculations. The final Dean number is then determined using the equation:

\[ De = \frac{m}{\rho A_{on} \left[ \frac{d}{r} \right]} \sqrt{\frac{d}{r}} \]

2. Constant Heat Flux Setting

The four heaters, CC1, CC2, CV1, and CV2 are designed to provide a constant heat flux over both the concave, convex, and straight channel surfaces. The magnitude of the heat flux is set based on two constraints. First, heat fluxes must be large enough to obtain significant differences between surface temperatures and local mixed mean fluid temperatures. The second constraint limits maximum surface temperatures for any location in the channel to 60 Deg. C. This is necessary because the bonding agent used to attach lexan sheets to cross beams is not dependable at temperatures above 65 Deg. C. For most runs, the maximum temperature occurs at TC #199, which is located in the final spanwise row of the convex wall of the curved section. A curve showing heat flux required per heater.
to reach a maximum temperature of 60 Deg. C, obtained from experimental tests, is shown in Figure 19 for Dean numbers from 50 to 200.

To determine the heat flux, the voltage drops across each shunt and each heater are measured using a simple volt-ohm meter and are entered into a program called Temcurv. Figure 20 shows voltage drops across shunt resistors which are needed to produce desired power levels in each heater. Equations relating these voltage drops to power are given by:

\[ I = \frac{\Delta V_S}{R} \]

and

\[ P = \Delta V_H \times I \]

Where:

- \( I \) = Current (amps)
- \( P \) = Power (watts)
- \( \Delta V_S \) = Voltage drop across the shunt (volts)
- \( \Delta V_H \) = Voltage drop across the heater (volts)
- \( R \) = Shunt resistance (ohms)

### B. SURFACE HEAT TRANSFER DISTRIBUTION MEASUREMENTS

#### 1. Surface Temperature Measurements

The system is heated for approximately 8-9 hours prior to data acquisition to reach steady state and to insure that thermal equilibrium exists. Great care is taken to insure that maximum surface temperatures never exceed 60 Deg. C. by routinely checking temperatures using the temperature acquisition program, Tcheck. Figure 21 provides a
curve from Run #120689.2125 which illustrates how temperature tends to increase with time. When steady state conditions are reached, voltage readings from each thermocouple are recorded using Temcurv and converted into Deg. C using calibration equations determined by Hughes [Ref. 8]. For thermocouples 1 through 200, 201, 203 and 204, voltages are converted into Deg. C using the equation:

\[ T = -1.988321 + 29436.53V - 2622880 V^2 + 36932000V^3 \]

For thermocouple 202 the equation is:

\[ T = -1.816347 + 2928.91V - 2622880V^2 + 373752000V^3 \]

Where:

\[ T = \text{Temperature Deg. C} \]
\[ V = \text{Thermocouple voltage (volts)} \]

These equations are utilized in the software program Nuscurv3B.

The surface thermocouples are attached behind Lexan surfaces exposed to the flow field. Consequently, there is a thermal contact resistance between thermocouples and convective Lexan surfaces. The temperature difference, \( \Delta T_{\text{wcorr}} \), between the actual surface and that measured by the thermocouple is given by the equation:

\[ \Delta T_{\text{wcorr}} = q''_{\text{conv}} \] (CR)

Where:

\[ q''_{\text{conv}} = \text{Convected heat flux per heater area} \]
CR = Thermal contact resistance (°C m²/watt)

Experimental procedures used to determine CR are described by Bella [Ref. 12]. From his work, CR equals $3.4 \times 10^{-3}$ °C m²/watt.

2. Nusselt Number Calculations

The program Nuscurv3B is used to determine local Nusselt numbers. The process begins with determination of the convective power transfer using:

$$q_{\text{conv}} = q_{\text{in}} - q_{\text{cond}} - q_{\text{rad}}$$

Where:

$q_{\text{conv}} = \text{Power convected to the channel air flow}$

$q_{\text{in}} = \text{Power to the heaters (V x I)}$

$q_{\text{cond}} = \text{Power loss by conduction}$

$q_{\text{rad}} = \text{Power loss by radiation, set equal to zero}$

Power loss by conduction is given by Bella [Ref. 12], who conducted tests on a full scale prototype model of the curved heated section. His results show that conduction losses are given by:

$$q_{\text{cond}} = -0.0294 + 0.4222(T_{\text{corr}} - T_{\text{ambient}}) - 0.0015(T_{\text{corr}} - T_{\text{ambient}})^2$$

Where:

$T_{\text{corr}} = \text{Corrected local surface temperature}$

$T_{\text{ambient}} = \text{Ambient temperature for surroundings}$
The convective heat flux is then determined using:

\[ \dot{q}''_{\text{conv}} = \dot{q}_{\text{conv}} / A_{\text{sec}} \]

Where:

\( A_{\text{sec}} \) = Surface area for each of the four heater surfaces, CC1, CC2, CV1 and CV2

With the convective heat flux known, the local mixed mean temperature at any streamwise channel location is determined using:

\[ t_{\text{mm}}(x) = t_{\text{mmin}} + [ \dot{q}''_{\text{conv}}(x) b x ] / C_p (\dot{m}) \]

Where:

\( t_{\text{mm}}(x) \) = Local mixed mean Temperature
\( t_{\text{mmin}} \) = Inlet mixed mean temperature
\( b \) = Spanwise width
\( x \) = Streamwise distance
\( \dot{q}''_{\text{conv}} \) = Heat flux over streamwise distance \( x \)
\( C_p \) = Specific heat of air

Local heat transfer coefficients are then determined using the equation:

\[ h = \dot{q}_{\text{conv}} / (T_{\text{wcorr}} - t_{\text{mm}}) \]
Finally, the local Nusselt number, $Nu$, is calculated from:

$$Nu = \frac{h(D_H)}{k}$$

Where:

$D_H = \text{Hydraulic diameter (2 x channel width = .0254 m)}$

$k = \text{Thermal conductivity of air at 200 Celcius (0.02563 W/m K)}$

C. EXIT MIXED MEAN TEMPERATURE MEASUREMENTS AND DETERMINATION

1. EXIT Mixed-Mean Temperature Determination

A two step procedure is used to determine the local mixed-mean temperature from the convective heat flux. This temperature is determined at the location of the two-dimensional traverse slot, which is 57.1 cm from the trailing edge of the heater on the convexly curved wall. First, the mixed-mean temperature is estimated at the downstream edge of the heaters using procedures described in the previous section. This temperature is noted as, $t_{mni}$. In the second step, an energy balance is then used to estimate the heat loss from the flow to the unheated portion of the channel, which extends from the trailing edge of the heater to the traversing slot. This energy balance is given by:

$$\dot{m} C_p t_{mni} = \dot{m} C_p t_{numo} + A_{tot} h[(t_{mni} + t_{mno})/2 - T_W]$$

Where:

$t_{mnr} = \text{Local mixed-mean temperature at the traverse slot.}$

$T_W = \text{Temperature of the unheated channel surface (approximately equal to ambient temperature).}$
\[ A_{\text{tot}} = \text{Concave and convex surface area for this portion of the channel.} \]

\[ h = \text{Heat transfer coefficient at last spanwise row of thermocouples} \]

This equation is employed in the software plotting program Tmxdmean, in the form where slot temperature \( t_{\text{mmo}} \) is given by:

\[
t_{\text{mmo}} = \frac{\dot{m}C_p t_{\text{ffi}} - hA_{\text{tot}} \left( \frac{t_{\text{mmo}}}{2} - T_w \right)}{\dot{m}C_p + \frac{hA_{\text{tot}}}{2}}
\]

2. Exit Mixed-Mean Temperature Measurement

After the channel is thermally steady state, TC #204 is set in place into the temperature traversing carriage. Using the program Temtrav, 320 voltage readings are taken over a 5.08 cm (2.0 inch) by 1.02 cm (0.4 inch) area as the probe moves in the traverse slot. Voltage readings are transformed into Deg. C using the equation described in the previous section. The mixed mean temperature is then given by:

\[
t_m = \frac{1}{A_{\text{ch}} U} \int u t dA
\]

Where:

\[ t_m = \text{Mixed mean temperature} \]

\[ A_{\text{ch}} = \text{Channel traversed cross section area} \]

\[ U = \text{Bulk velocity (spatially averaged)} \]

\[ u = \text{Local mean streamwise velocity} \]
The software program Tmxdmean is used to determine the mixed-mean temperature. This is accomplished using an approach which assumes that the area corresponding to each temperature measurement is the same:

\[ t_m = \frac{1}{320} \sum_{i=1}^{320} \frac{u_i}{\nu} t_i \]

Velocity data, \( U \) and \( u_i \), were obtained using a miniature five-hole pressure probe described by Baun [Ref. 13]. The velocity data used to determine mixed-mean temperatures in the present study are the same as given by Hughes [Ref. 8] and Tuzzolo [Ref. 14].
IV. RESULTS AND DISCUSSION

The objective of the present channel study is to examine the effects of curvature and the resulting centrifugal instabilities on local heat transfer distributions. Flows at Dean numbers ranging from 50 to 200 are investigated. The results of these tests are discussed in four parts. First, two types of baseline tests are discussed, which qualify measuring procedures and apparatus. This section includes discussion of spanwise-averaged Nusselt numbers from the straight portion of the channel as well as comparison of measured exit mixed-mean temperatures with values determined from energy balances. Second, spanwise-averaged Nusselt number distributions from concave and convex surfaces are presented. Third, the effects of counter-rotating Dean vortex pairs on the local Nusselt number distributions are detailed. Finally, in the fourth section, surveys of local temperatures used to determine the mixed mean temperature are presented in conjunction with mean velocity data obtained by Hughes [Ref. 8] and Tozzolo [Ref. 14].

A. BASELINE RESULTS


The walls of the straight portion of the channel are heated to insure that flow is thermally and hydrodynamically fully developed prior to entering the curved section. Because this portion of the channel is straight, much information is available on the behavior and distributions of local Nusselt numbers. First, because the flow is symmetric in the direction normal to the walls. Nusselt numbers should be the same on the top and bottom walls at each streamwise location. Second, the analytical solution for flow between two infinite parallel plates with constant heat flux boundary conditions gives the fully
developed Nusselt number value. Third, a numerical solution is used to predict thermal entry length behavior. All three of these are now discussed in regard to measured Nusselt number distributions.

a. Comparison of Results from the Straight CC1 and CV1 Surfaces

Figure 22 and Figure 23 show spanwise-averaged Nusselt numbers for Dean numbers of 100 and 125 for the CC1 and CV1 surfaces, respectively. The graphs show that the Nusselt number distributions are nearly identical over the entire lengths of the straight sections, from x/d of 82.1 cm, which is the location of the first spanwise row of thermocouples, to x/d of 178.6 cm, the beginning of curvature.

b. Comparison of Experimental Data with the Analytic Solution for Thermally Fully Developed Flow

Kays and Crawford [Ref. 15], indicate that the average Nusselt number is 8.24 for fully developed laminar flow between two infinite parallel plates with constant heat flux boundary conditions. Figures 22 and 23 show measured Nusselt numbers ranging from 8.45 to 8.11, which agree excellently with the value of 8.24, for x/d from 130.4 to 178.6. As indicated earlier, these data are obtained for Dean numbers of 100 and 125 and are approximately constant with x/d only after thermal entry length effects disappear and the flow becomes thermally fully developed.

c. Comparison of Thermal Development Effects with Numerical Solutions

Spanwise-averaged Nusselt numbers for Dean Numbers of 100 and 125 are compared to results from numerical predictions in Figures 24 and 25. The numerical prediction results are obtained using the STAN5 boundary layer program, described by Kays and Crawford [Ref. 15]. Exact experimental conditions are used as initial conditions.
and boundary conditions, and include the effects of the unheated starting length between
the nozzle exit and heater leading edges. The data and predictions show excellent agreement
along the entire length of the straight portion of the channel, from \( x/d \) of 70.0 to 178.64,
which validates measurements made in the thermal entry length region of the channel.

Predicted Nusselt numbers for the straight portion of the channel for all Dean
Numbers studied are given in Figure 26. At each Dean number, Nusselt numbers are
infinite at the initial heating point. With streamwise development, Nusselt numbers then
decrease. At each particular streamwise location, spanwise-averaged Nusselt numbers
increase with Dean number because the entry length needed for flow to become thermally
fully developed increases. The same Nusselt numbers are non-dimensionalized in Figures
27 and 28. Predicted and measured non-dimensional Nusselt numbers for the concave
surfaces are compared in Figures 29 and 30. Figures 31 and 32 then give the same type of
information for the convex surface.

2. Mixed-Mean Temperature Comparison

Figures 33 through 39 present mixed-mean temperature distributions along the
length of the channel, which are determined from the convected heat flux into the flow.
These figures also compare these distributions to mixed-mean temperatures determined
from local measurements of temperature and velocity obtained from probes placed in a slot
located downstream of the last heater. In figures 33-39, results are given for Dean
numbers of 58.5, 73.7, 98.1, 126.0, 149.8, 173.5 and 198.4. In all cases, mixed-mean
temperatures measured at the exit of the curved portion closely match values at \( x/d \) of 356.6
from the energy balance. This further validates experimental procedures, experimental
apparatus, and the energy balances employed, and confirms that the conduction loss
estimates used in determining the convected heat fluxes are accurate. Small differences in
the two exit mixed-mean temperatures sometimes result from small uncertainties in measured temperatures and velocities. Mean temperature distributions, mean velocity distributions and distributions of the mean temperature/mean velocity product used to determine the local mixed-mean temperature are presented in Part D. of Chapter 4.

3. Baseline Data Summary

All baseline test results validate experimental procedures, the measurement approach, the energy balances employed, and the facilities used in this study, and provide confidence that the measurements taken in the curved portion of the channel, located just downstream of the straight portion, represent actual flow behavior.

B. SPANWISE-AVERAGED NUSSELT NUMBER DISTRIBUTIONS

In this section, spanwise-averaged Nusselt numbers are presented, which are non-dimensionalized using the spanwise-averaged Nusselt number at x/d of 130.4. This is done to minimize some of the scatter in dimensional Nusselt numbers measured at Dean numbers lower than 100 and higher than 125.

Figures 40, 41, 42 and 43 present Nusselt number ratios for concave and convex surfaces at seven Dean Numbers. At x/d values between 0 and 82.1, no Nusselt numbers are plotted because this portion of the channel is unheated. Between x/d of 82.1 and approximately 110, Nusselt numbers increase with Dean number, with trends consistent with predictions which account for thermal entry length effects. At x/d of about 110 to 130, flow is hydrodynamically and thermally fully developed and Nusselt numbers are approximately constant with streamwise distance. As shown in Figure 22, the spanwise-averaged Nusselt number is about 8.24 for Dean numbers of 100 and 125, which is in agreement with the theory for the same flow. The same behavior exists for Dean numbers up to about 150, whenever the flow is fully laminar and no transition phenomenon are
present. As x/d approaches 160-180, spanwise-averaged Nusselt number ratios increase for Dean numbers of 175 and 200, as a result of initiation of transition phenomena. Peak values exist at x/d of about 178.6, which is just at the beginning of curvature. Here, some mixing of flow is present that does not occur otherwise. This trend occurs on both concave and convex surfaces, but it is more pronounced on the convex surface.

As x/d becomes greater than 180, effects of curvature are evident in measurements from both the concave and convex surfaces. These effects are complex and vary not only with the Dean number, but also with the surface considered. On the concave surface at Dean numbers of 50, 75, 100 and 125, Nusselt number ratio distributions are the same for x/d of 178.6 to approximately 226.9. For each case, the Nusselt numbers decrease drastically as a result of the stabilizing influence of convex curvature, which causes a stabilizing effect initially near the convex surface before propagating across the width of the channel.

At x/d greater than 226.9 for both surfaces, the presence of Dean vortices is evident in local Nusselt number ratios, which gradually increase with streamwise development. These vortices develop because of the imposition of centrifugal instabilities. For x/d from 226.9 to 299.4, the lowest Nusselt number ratios correspond to a Dean numbers of 50, and the highest correspond to Dean numbers of 100 and 125. At these higher Dean numbers, centrifugal instabilities and the accompanying Dean vortices are more intense, and thus, Nusselt numbers are also higher.

At Dean numbers of 150, 175 and 200, results from both the concave and convex surfaces show evidence of transition phenomena resulting in the onset of some chaotic motions. This is particularly so for x/d ranging from 154.5 to 178.6, just prior to curvature, where Nusselt number ratios are locally higher. At x/d from 178.6 to 300, ratios
from the convex surface are generally lower than from the concave surface. This occurs because of the stabilizing and destabilizing influences of the convex and concave surfaces, respectively. Flow visualization results for the same experimental conditions show considerable mixing.

C. EFFECTS OF COUNTER-ROTATING DEAN VORTEX PAIRS ON LOCAL NUSSLELT NUMBER DISTRIBUTIONS

Local Nusselt numbers versus non-dimensional spanwise length, z/d, for both concave and convex surfaces at Dean numbers of 58.5, 73.7, 98.1, 126.0, 149.8, 173.5 and 198.4, are given in Figures 44-57. These data, especially those from the concave surface, illustrate the effects of pairs of counter-rotating Dean vortices on local Nusselt number distributions.

In general, local Nusselt numbers on the convex surface are generally spanwise uniform, regardless of Dean number. Exceptions occur at Dean numbers of 149.8, 173.5 and 198.4, where some slight variations are present probably due to vortex pairs. On the concave surface, local Nusselt numbers develop significant variations in the spanwise direction as the flows develop downstream as a result of vortex upwash and downwash regions. Local Nusselt number values are augmented near downwash regions and diminished in upwash regions. Such spanwise variations coincide with increases of spanwise-averaged Nusselt numbers, and are most significant for Dean numbers of 98.6 and greater. Such behavior is also consistent with flow visualization results which show counter-rotating vortex pairs for the same experimental conditions.

Focussing on Nusselt number results for a Dean number of 98.6 and comparing them to the spanwise averaged Nusselt number data presented in section B, interesting conclusions are made. Data at x/d of 202.8 to 226.9 show that local Nusselt numbers are
spanwise uniform. These locations coincide with regions where spanwise averaged Nusselt numbers are lower than values at other x/d, as a result of the imposition of centrifugal instabilities. As x/d increases to 251.1, 275.2 and 299.4, spanwise-averaged Nusselt numbers increase and local Nusselt numbers show considerable spanwise variations due to the formation of Dean vortex pairs.

Such trends are present at all Dean numbers greater than or equal to 98.6. However, at Dean numbers of 173.5 and 198.4, local Nusselt numbers show spanwise variations at x/d as low as 202.8 to 226.9.

D. LOCAL MIXED-MEAN TEMPERATURE SURVEY DATA

1. Local Mean Temperature Distributions

Mean temperature surveys are presented for Dean numbers of 58.5, 73.7, 98.1, 126.0, 149.8, 173.5 and 198.4 in Figures 58-64. Here, mean refers to a time-average. For Dean numbers of 58.5 and 73.7, spanwise periodicities result due to vortex pairs. At Dean numbers from 98.1 to 198.4, the local mean temperature distributions are fairly spanwise uniform, with only slight variations due to vortex pairs. Such variations are present at lower Dean numbers because Dean vortices form only after a considerable amount of streamwise development. Such vortices continue to exist at higher Dean numbers, but they are unsteady over spanwise distances greater than the spacing between vortex pairs, which makes them appear spanwise uniform in the time-averaged flow field.

2. Local Mean Velocity Distributions

Mean velocity distributions are given by Hughes [Ref. 8] and Tuzzolo [Ref. 14]. Results for Dean numbers of 50, 75, 100 and 150 are shown in Figures 65-68. Qualitative trends of the velocity contours coincide nicely with local mean temperature contours; velocity profiles at low Dean numbers show some spanwise variations, while at higher
Dean numbers the profiles are fairly spanwise uniform.

3. Distributions of the Mean Temperature-Mean Velocity Product

Mean temperature-mean velocity product distributions are presented in Figures 69-75 for Dean numbers of 58.5, 73.7, 98.1, 126.0, 149.8, 173.5 and 198.4. The graphs show good spanwise uniformity for all Dean numbers, except for some of the lower values, as expected. As mentioned in Part C of Chapter 3, these temperature-velocity product data are used to compute the local mixed mean temperatures discussed in Chapter 4, Part A., Section 2.
V. CONCLUSIONS AND RECOMMENDATIONS

A. CONCLUSIONS

The effects of curvature and the resulting centrifugal instabilities on local heat transfer distributions are studied in a curved channel at Dean numbers ranging from 50 to 200. The channel is designed with a rectangular cross section of 1.27 cm by 50.1 cm giving an aspect ratio of 40 to 1. These interior dimensions coincide with those of another channel which is transparent and used for flow visualization. In the channel used for thermal studies, the flow is heated in a 2.44 m straight portion, located just prior to the curved portion to obtain flow which is hydrodynamically and thermally developed. The flow then enters a 180 degree heated curved section with a convex surface radius of 59.69 cm and a concave surface radius of 60.96 cm. Surface temperatures and ambient temperatures are measured using 204 thermocouples, whose outputs are used to determine local Nusselt numbers as constant heat flux boundary conditions are maintained on all heated surfaces.

Several baseline comparisons qualify flow behavior as well as the experimental procedures and measurement techniques. In the first of these, spanwise-averaged Nusselt numbers from the two walls of the straight portion of the channel match with each other very well. As thermal entry length effects disappear, dimensional Nusselt numbers approach 8.24 as expected from the theory for thermally fully developed flow between two parallel infinite plates with constant heat flux boundary conditions. In the thermally developing region, measured Nusselt numbers match results from numerical predictions. Finally, measured mixed-mean temperatures at the exit of the curved section match values determined from energy balances. These baseline comparisons thus validate experimental procedures, the measurement approach and the facilities used, and provide confidence of
the accuracy of measurements from the curved portion of the channel.

Spanwise-averaged Nusselt numbers are non-dimensionalized with respect to the Nusselt number determined at x/d of 130.4. For Dean numbers of 50, 75, 100 and 125, these ratios approach constant values on the straight portion of the channel for x/d from 110 to 180. At smaller x/d, Nusselt number ratios increase with Dean number as a result of thermal entry length effects. For Dean Numbers of 150, 175 and 200 increases in Nusselt number ratios are present for 160 ≤ x/d ≤ 180 as a result of transition phenomena.

The stabilizing influences of convex curvature are evidenced at the onset of curvature by drastic decreases in Nusselt number ratios for both surfaces at Dean numbers of 50, 75, 100 and 125. These effects initially produce a stabilizing effect, which occurs primarily near the convex surface and then propagates across the width of the channel. As x/d increases, the formation of Dean vortices as a result of centrifugal instabilities is evidenced by gradual increases of the local Nusselt number ratios. Here, Nusselt number ratios are generally lower on the concave surface, where the lowest Nusselt number ratios are present at a Dean number of 50 and the highest values exist at Dean Numbers of 100 and 125. Higher Nusselt numbers are expected at higher Dean numbers because centrifugal instabilities and the resulting Dean vortices are more intense. Higher Nusselt numbers are expected near the concave surface because of the intensity of secondary flows from counter rotating vortex pairs, which are much less intense near convex surfaces.

At Dean numbers of 150, 175 and 200, effects of transition phenomena are considerably more apparent. Nusselt number ratios show local increases on the straight portion of the channel just prior to curvature. After curvature is imposed, trends are generally similar ones observed at the lower Dean numbers, except that Nusselt number ratios are higher probably because of increased flow mixing associated with transition.
phenomena.

Local Nusselt numbers from the convex surface are fairly spanwise uniform, regardless of Dean number. On the concave surface, however, significant spanwise variations are evident, especially at Dean numbers of 98.6 or greater, which result from upwash and downwash regions in vortex pairs. Such spanwise variations correspond to increases in spanwise-averaged Nusselt numbers which occur with streamwise development.

Mean temperature and mean velocity survey data show some spanwise periodicity at the lower Dean numbers due to upwash and downwash regions formed by vortex pairs. As the Dean number increases, these surveys show distributions which are fairly spanwise uniform.

B. RECOMMENDATIONS

Recommendations are, first, repeat the present study at Dean numbers from 200 to 450 to obtain additional information on the connections between centrifugal instabilities, Dean vortices and transition from laminar flow to turbulent flow. Second, Nusselt numbers should be measured in situations where the heat flux differs on the two surfaces of the channel.
## APPENDIX A
### THERMOCOUPLE PLACEMENT TABLES

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APPENDIX B
FIGURES

The following pages contain the figures used in the development of this thesis.
Figure 1. Curved Heat Transfer Channel
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NUSSELT NUMBER DISTRIBUTION

Figure 46. Spanwise Nusselt Number Distributions vs. z/d For Curved Concave Surface, D = 74.2

RUN #11890.1945 CONCAVE SURF.  DE = 74.18

1 X/D=282.79
2 X/D=226.93
3 X/D=251.07
4 X/D=275.21
5 X/D=299.35

Nu

4.0  6.0  8.0  10.0

Z/D

-2.0 -1.5 -1.0 -0.5  0.0  0.5  1.0  1.5  2.0
Figure 47. Spanwise Nusselt Number Distributions vs $z/d$ For Curved Convex Surface, $De = 74.2$
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Figure 58. Measured Exit Local Mean Temperature Contours.

Temp. (deg.C) RANGES

0 : 10.53 - 11.62
1 : 11.62 - 12.71
2 : 12.71 - 13.88
3 : 13.88 - 14.89
4 : 14.89 - 15.98
5 : 15.98 - 17.07
6 : 17.07 - 18.16
7 : 18.16 - 19.25
8 : 19.25 - 20.34
9 : 20.34 - 21.43
Figure 59. Measured Exit Local Mean Temperature Contours.
De = 73.7
Figure 60. Measured Exit Local Mean Temperature Contours.

Temp. (deg. C) RANGES

0 : 12.52 - 13.69
1 : 13.69 - 14.88
2 : 14.87 - 16.05
3 : 16.05 - 17.22
4 : 17.22 - 18.4
5 : 18.4 - 19.58
6 : 19.58 - 20.77
7 : 20.75 - 21.93
8 : 21.93 - 23.11
9 : 23.11 - 24.28

RUN #11990.1925  DF = 98.11
Figure 61. Measured Exit Local Mean Temperature Contours.
De = 126.0
Figure 62. Measured Exit Local Mean Temperature Contours. De = 149.8
Figure 63. Measured Exit Local Mean Temperature Contours.
De = 173.5
Figure 6.4. Measured Exit Local Mean Temperature Contours.

Temp. (deg.C) RANGES

Z/d

0 : 11.95 - 13.32
1 : 13.32 - 14.69
2 : 14.69 - 16.07
3 : 16.07 - 17.44
4 : 17.44 - 18.81
5 : 18.81 - 20.18
6 : 20.18 - 21.55
7 : 21.55 - 22.93
8 : 22.93 - 24.3
9 : 24.3 - 25.67

RUN #12390.19 DE = 198.4
RUN #50289.1759  DEAN = 50

Ux

Figure 65. Mean Velocity Contours, De = 50

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<thead>
<tr>
<th>Range</th>
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<tbody>
<tr>
<td>0</td>
<td>&lt;0.06298</td>
</tr>
<tr>
<td>1</td>
<td>&lt;0.1141</td>
</tr>
<tr>
<td>2</td>
<td>&lt;0.08571</td>
</tr>
<tr>
<td>3</td>
<td>&lt;0.16</td>
</tr>
<tr>
<td>4</td>
<td>&lt;0.2343</td>
</tr>
<tr>
<td>5</td>
<td>&lt;0.3086</td>
</tr>
<tr>
<td>6</td>
<td>&lt;0.3829</td>
</tr>
<tr>
<td>7</td>
<td>&lt;0.4572</td>
</tr>
<tr>
<td>8</td>
<td>&lt;0.5315</td>
</tr>
<tr>
<td>9</td>
<td>&lt;0.6058</td>
</tr>
<tr>
<td>10</td>
<td>&lt;0.6801</td>
</tr>
</tbody>
</table>
Figure 66. Mean Velocity Contours, De = 75

RUN #42989_0055  DEAN = 73

$U_x$ (m/s) RANGES

0 : $0.3925 \leq 0.4629$
1 : $0.4639 \leq 0.5352$
2 : $0.5352 \leq 0.6066$
3 : $0.6066 \leq 0.6779$
4 : $0.6779 \leq 0.7493$
5 : $0.7493 \leq 0.8206$
6 : $0.8206 \leq 0.8919$
7 : $0.8919 \leq 0.9633$
8 : $0.9633 \leq 1.035$
9 : $1.035 \leq 1.106$
Figure 67. Mean Velocity Contours, De = 100
Figure 68. Mean Velocity Contours, De = 150

RUN #43089.1253  DEAN = 150

Ux

Ux (m/s) RANGES

0 : 0.5341  <  0.5919
1 : 0.5919  <  0.6299
2 : 0.6299  <  0.6776
3 : 0.6776  <  1.125
4 : 1.125   <  1.273
5 : 1.273   <  1.421
6 : 1.421   <  1.559
7 : 1.559   <  1.717
8 : 1.717   <  1.965
9 : 1.965   <  2.012
RUN #11790.2015  De = 59.47

Figure 69. Mean Temperature/Mean Velocity Product Contours.

Temp. * Velocity (deg. C * m/s) RANGES
0 : 10.65 - 13.56  5 : 25.2 - 28.11
1 : 13.56 - 16.47  6 : 28.11 - 31.82
2 : 16.47 - 19.38  7 : 31.82 - 33.93
3 : 19.38 - 22.29  8 : 33.93 - 36.84
4 : 22.29 - 25.2   9 : 36.84 - 39.75
Figure 70. Mean Temperature/Mean Velocity Product Contours.

RUN #11890.1945  De = 73.66

Temp. * Velocity (deg.C * m/s) RANGES
0 : 10.69 - 13.71  5 : 25.81 - 28.83
2 : 16.74 - 19.76  7 : 31.86 - 34.88
3 : 19.76 - 22.78  8 : 34.88 - 37.9
4 : 22.78 - 25.81  9 : 37.9 - 40.93

z/d
Figure 71: Mean Temperature/Mean Velocity Product Contours, De = 98.1

RUN #11990.1925  De = 98.11

Temp. * Velocity (deg.C * m/s) RANGES
0 : 11.39 - 14.44  5 : 26.64 - 29.69
1 : 14.44 - 17.49  6 : 29.69 - 32.74
2 : 17.49 - 20.54  7 : 32.74 - 35.79
3 : 20.54 - 23.59  8 : 35.79 - 38.84
4 : 23.59 - 26.64  9 : 38.84 - 41.89
RUN #12090.1435   De = 126

Temp. * Velocity (deg.C * m/s) RANGES
0 : 14.69 - 18.71   5 : 34.78 - 38.79
1 : 18.71 - 22.73   6 : 38.79 - 42.81
2 : 22.73 - 26.74   7 : 42.81 - 46.83
3 : 26.74 - 30.76   8 : 46.83 - 50.84
4 : 30.76 - 34.78   9 : 50.84 - 54.86
Figure 73. Mean Temperature/Mean Velocity Product Contours,
De = 149.8
Figure 74. Mean Temperature/Mean Velocity Product Contours.

RUN #12290.193  De = 173.5

Temp. * Velocity (deg.C * m/s) RANGES
0 :  19.46 - 25.36  5 :  40.86 - 54.86
1 :  25.36 - 31.26  6 :  54.86 - 60.76
2 :  31.26 - 37.15  7 :  60.76 - 66.67
3 :  37.16 - 43.06  8 :  66.67 - 72.57
4 :  43.06 - 48.96  9 :  72.57 - 78.47
Figure 75. Mean Temperature/Mean Velocity Product Contours,
De = 198.4
APPENDIX C
SOFTWARE DIRECTORY

This Appendix gives a listing of the various programs used in this thesis. Each program listing contains a summary of how the program is used, user inputs, program outputs and any additional features. Programs are listed in two groups; Nusselt number distributions and exit mixed-mean temperature measurement. All programs were written in BASIC 4.0 for use on the HP9836S computer.

I. NUSSLET NUMBER DISTRIBUTION PROGRAMS

A. TCHECK:
This program acquires multiple channel thermocouple data and performs temperature calibration equations to provide measurements in Degrees Celsius.
user inputs: thermocouple voltage readings, X
program output: provides monitor summary of thermocouple number, data acquisition channel number, voltage reading and corresponding temperature reading in Deg. C.

B. TEMCURV:
This program acquires multiple channel thermocouple data and creates two files to be used in other programs for the curved heating channel. Primary purpose is to supply data for Nusselt number calculations.
user inputs: a) thermocouple voltage readings.
b) run number.
c) voltages across shunts for all four heaters, mvolts.
d) voltages across heaters, volts.
e) ambient pressure, inches of mercury.

f) pressure drop across orifice plate, inches of water.

program outputs: a) data file, Idata, provides data required to determine heat input, conduction loss and Dean number; Runno, Amp1, Amp2, Amp3, Amp4, Vh1, Vh2, Vh3, Vh4, Pamb, Deltap, Tmi, Tccamb, Tcvamb, Ro.

b) data file, Tdata, provides temperature data in Deg. C for determination of local Nusselt numbers.

C. NUCURV3B:

This program determines heat transfer coefficients and Nusselt numbers by conducting surface energy balance. Heat flux inputs and conducted heat losses are determined.

user inputs: a) reads data files Tdata and Idata from program TEMCURV.

b) estimated Dean number.

program outputs: a) data file, Hdata, of heat transfer coefficients, local Nusselt numbers streamwise and spanwise locations, measured and corrected temperatures as follows; H, Nus, X2, Z, T1 and T.

additional features: a) provides spanwise-averaged Nusselt numbers.

b) using energy balance, calculates local mean temperatures.

c) with estimated Dean number, performs iteration process to determine actual Dean number.

D. NUSCCPL2

This program reads data file and plots spanwise variations of Nusselt numbers by rows for concave section of curved heating channel utilizing thermocouples 31-115.
user inputs: local Nusselt numbers along with corresponding streamwise and spanwise distances (output data file, Hdata, from program NUCURV3B).

program outputs: Nusselt number distribution contours for all rows of concave curved section.

E. NUSCVPL2

This program reads data file and plots spanwise variations of Nusselt numbers by rows for convex section of curved heating channel utilizing thermocouples 116-120.

user inputs: local Nusselt numbers along with corresponding streamwise and spanwise distances (output data file, Hdata, from program NUCURV3B).

program outputs: Nusselt number distribution contours for all rows of convex curved section.

II. EXIT MIXED-MEAN TEMPERATURE MEASUREMENT

A. TEMTRAV

Program for control of temperature traverse probe positioning and data acquisition in the curved heating channel. Interfaces with MITAS controller to control the probe position. Provides temperature difference between exit temperature and ambient inlet temperature.

user inputs: a) estimate of Dean number
   b) initial position of pressure probe.
   c) number of y and z points to take data at.
   d) resolution of data.
   e) ambient inlet temperature channel number.
   f) temperature probe channel number.
   g) ambient pressure.
h) pressure drop across orifice plate.
i) delay time to start the experiment.

**program outputs:**
a) data file, Temp, provides position and temperature data as follows: Y, Z, temperature, inlet temperature, temperature difference.
b) provides actual Dean number through iteration process.

**B. TMXDMPLOTB**
Program to plot time averaged traverses of temperature probe measurements. Will plot up to ten radial traverses for different spanwise locations.

**user input:**
a) temperature data file, TEMP (output from program TEMTRAV).
b) spanwise positions to plot.

**program output:** temperature traverse plot

**C. TMXDMEAN**
Modified version of TMXDMEAN to plot traverse of mean temperature, mean velocity products. Will plot up to ten radial traverses, for different spanwise locations. Program has been modified to determine exit mixed-mean temperature using process described in Chapter 3, Section C, part 2.

**user inputs:**
a) temperature data file, TEMP (output from program TEMTRAV).
b) velocity data file, VEL (output from program VELOCITY).
c) spanwise positions to plot.

**program output:**
a) mean temperature, mean velocity traverse plot.
b) monitor output of exit mixed-mean temperature.
APPENDIX D
DATA FILE LISTING

A Summary of the data files and corresponding experimental parameters is given in two parts, the first of which involves the determination of all Nusselt number results and the second with the calculation of the measured exit mixed-mean temperature results.

A. NUSSELT NUMBER RESULTS.

The following lists of data were used for the determination of all Nusselt number data presented in this thesis. This includes both the spanwise-averaged Nusselt data and the local spanwise Nusselt number distributions.

**Experimental parameters:**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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<tbody>
<tr>
<td>Dean number</td>
<td>58.9</td>
</tr>
<tr>
<td></td>
<td>74.2</td>
</tr>
<tr>
<td></td>
<td>98.6</td>
</tr>
<tr>
<td></td>
<td>126.4</td>
</tr>
<tr>
<td></td>
<td>150.9</td>
</tr>
<tr>
<td></td>
<td>174.4</td>
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<tr>
<td></td>
<td>198.9</td>
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<td>Pressure drop (in water)</td>
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<tr>
<td></td>
<td>0.05</td>
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<tr>
<td></td>
<td>0.09</td>
</tr>
<tr>
<td></td>
<td>0.15</td>
</tr>
<tr>
<td></td>
<td>0.215</td>
</tr>
<tr>
<td></td>
<td>0.29</td>
</tr>
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<td></td>
<td>0.38</td>
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<tr>
<td>Heat flux/heater (watts)</td>
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</tr>
<tr>
<td></td>
<td>37.8</td>
</tr>
<tr>
<td></td>
<td>53.75</td>
</tr>
<tr>
<td></td>
<td>52.9</td>
</tr>
<tr>
<td></td>
<td>73.6</td>
</tr>
<tr>
<td></td>
<td>72.2</td>
</tr>
<tr>
<td></td>
<td>76.0</td>
</tr>
<tr>
<td>Max. surface temp. (TC#199, °C)</td>
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<td></td>
<td>48.6</td>
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<td></td>
<td>54.9</td>
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<td></td>
<td>51.5</td>
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<td></td>
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**Data file listings:**

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<thead>
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<td>10</td>
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<td>Hdata # (file #19)</td>
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<td>75</td>
<td>100</td>
<td>125</td>
<td>150</td>
<td>175</td>
<td>200</td>
</tr>
</tbody>
</table>
B. MEASURED MIXED-MEAN TEMPERATURE RESULTS

The following list of data provides the data information used in calculating the measured exit mixed-mean temperature and the plotting of the temperature traverse contours. Also included is the velocity file number, which is presented by Hughes [Ref. 8] and Tuzzolo [Ref. 14].

**Experimental parameters:**

- **Dean number**: 58.5, 73.7, 98.1, 126.0, 149.8, 173.5, 198.4
- **Pressure drop**: 0.031, 0.05, 0.09, 0.150, 0.215, 0.29, 0.38 (in water)
- **Heat flux/heater**: 40.0, 47.2, 56.7, 63.3, 73.2, 81.9, 92.2 (watts)
- **Max. surface Temp.**: 60.4, 60.3, 59.2, 56.8, 57.6, 58.8, 59.7 (°C)

**Data file listing:**

- **Tdata/Idata #**: 3, 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 1, 2, 3
- **File number**: 15, 15, 15, 15, 15, 15, 15, 15, 15, 15, 15, 17, 17
- **Temp #**: 50, 75, 100, 125, 150, 175, 200
- **File number**: 15, 15, 15, 15, 15, 15, 15, 15, 15, 15, 15, 17, 17
- **Vel #**: 1, 1, 3, 4, 4, 5, 51
LIST OF REFERENCES


10. ASME Power Test Committee, ASME Power Test Codes (Supplement on Instruments and Apparatus), part 5,Chapter 4, p. 25, American Society of Mechanical Engineers, 1959.


# INITIAL DISTRIBUTION LIST

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<th>Name and Address</th>
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<td>6</td>
<td>Professor Phillip M. Ligrani, Code 069Li, Department of Mechanical Engineering, Naval Postgraduate School, Monterey CA 93943-5000</td>
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<td>1</td>
<td>Professor Chelakara S Subramanian, Code 069Su, Department of Mechanical Engineering, Naval Postgraduate School, Monterey CA 93943-5000</td>
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<td>1</td>
<td>Department Chairman, Code 69, Department of Mechanical Engineering, Naval Postgraduate School, Monterey CA 93943-5004</td>
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<td>Lt Paul E. Skogerboe, 1281 Spruance Rd., Monterey, CA 93940</td>
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<tr>
<td>1</td>
<td>Naval Engineering Curricular Officer, Code 34, Department of Mechanical Engineering, Naval Postgraduate School, Monterey CA 93943-5004</td>
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<td>8</td>
<td>Dr. K. Civinskas, Propulsion Directorate, U.S. Army Aviation Res. and Technology Activity, AVSCOM, NASA-Lewis Research Center, Cleveland, OH 45433</td>
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