EXPERIMENTAL AND ANALYTIC EVALUATION OF GAS-COOLED REACTOR
CAVITY COOLING SYSTEM PERFORMANCE

by

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(November 1984)

SUBMITTED TO THE DEPARTMENT OF
NUCLEAR ENGINEERING IN PARTIAL
FULFILLMENT OF THE
REQUIREMENTS FOR THE
DEGREE OF

DOCTOR OF SCIENCE

at the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

February 1991

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JUL 12 1991
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Submitted to the Department of Nuclear Engineering on January 22, 1991 in partial fulfillment of the requirements for the Degree of Doctor of Science in Nuclear Engineering

ABSTRACT

An experiment was carried out using air in a heated upflow pipe in the forced and mixed convection regimes to determine Nusselt numbers and friction factors over a wide range of conditions, which include projected Modular High Temperature Gas Cooled Reactor (MHTGR) Reactor Cavity Cooling System (RCCS) operating and off-normal conditions. Large (maximum of 50%) decreases in Nusselt number and a slight increase in friction factor compared with forced convection values, were found under the most severe mixed convection conditions. Results were compared with literature data, theory and correlation, and calculations using an in-house k-ε computer program. Consideration of air property changes as a function of temperature both in the analysis and in analytic simulation of the experiments was found to be important.

The correlated experimental results were used in a computer code, RECENT, to calculate the overall heat transfer from the reactor vessel to the ambient air, to evaluate the impact of design parameter and other changes on RCCS performance, and to optimize the RCCS riser design. The calculated results show that: 1) no single parameter dominates the RCCS performance, nor is performance sensitive to changes in design or operating conditions. Therefore, once the system is built, it is difficult to either worsen or improve its performance. 2) the present riser design will provide adequate cooling to the reactor vessel under even severe off-normal conditions.

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ACKNOWLEDGEMENTS

The author would like to express his sincere gratitude to Professor Michael Driscoll and Professor Neil Todreas for their most helpful advice, encouragement and guidance throughout the course of this work. This study was sponsored by the DOE and conducted under the technical direction of Bechtel National, Inc. The author is grateful for the technical assistance provided during the course of this work by Dr. S. Ghose of Bechtel and others at GA Technologies, Inc.

The author would also like to thank his associates Serhat Yesilyurt and Yuksel Parlatan for their most valuable comments on this project. The timeless assistance of Ms. Paula Cornelio is also acknowledged.

Finally, many thanks to my wife, Han, for her support and encouragement during my years as a graduate student at MIT.
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NOMENCLATURE

English Letters

A — area, aspect ratio
c_f — skin friction factor
c_p — heat capacity
D — pipe diameter
D_e — hydraulic diameter
d — vessel-riser distance
E — uncertainty
F — view factor
f — friction factor
G — mass flow velocity
g — gravitational acceleration
H — height
h — heat transfer coefficient
I — current
K — flow resistance coefficient
k — thermal conductivity
L — length, riser length in perpendicular to vessel direction
\dot{m} — mass flow rate
p — pressure
P — power
P_h — heated perimeter
P_w — wetted perimeter
Q — total heat
q'' — heat flux
q_i — incident heat flux
q_o — radiosity
q''' — heat generation rate
R — pipe radius
r — radial direction
T — temperature
t — time
V — voltage
v — velocity
W — riser width
w — velocity in flow direction
z — flow direction

Dimensionless Numbers

\[ \text{Bo} = \frac{8 \times 10^4 \text{Gr}_d}{\text{Re}^{3.425} \text{Pr}^{0.8}} \] — Buoyancy number
\[ \text{Gr}_A = \frac{g \beta D^4}{16 \mu^2} \frac{dT_a}{dz} \quad \text{Grashof number} \]

\[ \text{Gr}_q = \frac{g \beta q'' D^4}{k \mu^2} \quad \text{Grashof number} \]

\[ \text{Gr}_{\Delta T,D} = \frac{g \beta \Delta TD^3}{\mu^2} \quad \text{Grashof number} \]

\[ \text{Gr}_{\Delta T,L} = \frac{g \beta \Delta TL^3}{\mu^2} \quad \text{Grashof number} \]

\[ \text{Nu} = \frac{h D}{k} \quad \text{Nusselt number} \]

\[ \text{Pr} = \frac{c_p \mu}{k} \quad \text{Prandtl number} \]

\[ \text{Ra} = \text{GrPr} \quad \text{Rayleigh number} \]

\[ \text{Ra}_A = \text{Gr}_A \text{Pr} \quad \text{Rayleigh number} \]

\[ \text{Re} = \frac{\rho v D}{\mu} \quad \text{Reynolds number} \]

\[ \text{St} = \frac{h}{\rho c_p v} = \frac{\text{Nu}}{\text{RePr}} \quad \text{Stanton number} \]

**Greek Letters**

- \( \beta \) — thermal expansion factor
- \( \delta \) — boundary layer thickness
- \( \varepsilon \) — emissivity
- \( \mu \) — viscosity
- \( \mu_t \) — turbulent viscosity
- \( \rho \) — density, reflectivity
- \( \sigma \) — Stefan-Boltzmann constant
- \( \tau \) — shear stress
- \( \phi \) — dissipation function

**Subscripts**

- \( a \) — air
- \( b \) — bulk
- \( \text{ave} \) — average
- \( \text{acc} \) — acceleration
- \( \text{b} \) — buoyancy
- \( c \) — cross section, chimney
- \( \text{DB} \) — Dittus-Boelter
- \( e \) — exit
- \( f \) — friction, flow
in — inlet
iso — isothermal
HT — heat transfer
loss — heat loss
meas — measured
nom — nominal
o — reference condition; or outgoing (radiation)
out — outlet
r — riser
t — total, turbulent
v — vessel
w — wall

Superscripts

* — dimensionless, reference condition
— — average
CHAPTER 1
INTRODUCTION

1.1 Introduction

The Modular High Temperature Gas-Cooled Reactor (MHTGR) is one of the next generation power reactors currently being developed in the United States under DOE sponsorship. A major emphasis in the development of these reactors, and explicitly the MHTGR, is on improvement of safety margins by taking advantage of basic physical laws. Thus, completely-passive safety systems will be used wherever possible in the overall design process. A feature of the MHTGR design, shared in principle by the liquid metal cooled PRISM reactor design is, the use of a non-insulated steel reactor vessel and an array of riser tubes in the reactor cavity to provide a passive decay heat removal path from the core to the ambient air.

The design of this Reactor Cavity Cooling System (RCCS) requires attention, not only to its engineering aspects, but also to the fundamental heat transfer and fluid flow processes involved. One question is what geometry and arrangement will result in the best performance. This requires characterization of the heat transfer coefficient and the friction factor inside the riser tube in the operating regime, which has been predicted to involve a condition intermediate between forced and free (i.e. mixed) convection — a field not yet well understood, either qualitatively or quantitatively.

Accordingly, the objective of the present work was to first determine the heat transfer coefficient and friction factor inside the riser under its projected operating conditions, both experimentally and theoretically. The second objective was to incorporate these results into an overall model of RCCS performance, developed for this purpose, to find the best geometric configuration for the riser.
1.2 MHTGR RCCS Function and Description

The Modular High Temperature Gas-Cooled Reactor is a small version of a High Temperature Gas-Cooled Reactor (HTGR). The MHTGR has many of the same advantages as the HTGR since its fuel element design is virtually identical. These key advantages are high heat capacity and high threshold fuel damage temperature (1600°C), which will allow passive dissipation of decay heat. The MHTGR is being designed to achieve passive heat removal in a loss of coolant accident; the decay heat can be dissipated via the graphite moderator, reactor vessel wall, and other structures, to the surrounding environment without exceeding the fuel damage limiting temperature, 1600 °C. Hence, many feel that the MHTGR will reduce technical and financial risk and have the potential to facilitate the licensing procedure, a key obstacle to nuclear industry expansion in the U.S.

A schematic of an MHTGR design is shown in Figure 1.1 and Figure 1.2. The reactor core and the steam supply systems are mounted in two separate cavities (not shown in the figure), which are connected through a concentric cross-duct. The prismatic core consists of an annular active core made up of columns of stacked hexagonal section graphite blocks, upper and lower end reflectors, and central and side reflectors. The fuel and reflector are made up of essentially identical graphite blocks. The cooled helium, at about 260 °C from the circulator, goes through the outer annulus of the cross-duct, then flows upwards along the passages adjoining the core barrel at the side of the reactor vessel to an upper plenum at the top of the core. From the upper plenum it flows downwards through the coolant channels in the fuel blocks, where it is heated to about 690 °C when it leaves the core, entering the lower plenum. This hot helium then flows through the insulated inner channel of the cross-duct to the steam generator, and then to the circulator, to be pumped back into the core.

In the MHTGR design there are several ways to remove the decay heat during accident conditions, especially during loss of coolant accidents, such as using the shutdown circulator at the bottom of the reactor vessel to transport the decay heat out of the
Figure 1.1 A Schematic of a U.S. MHTGR Design (from [S-2])
Figure 1.2 MHTGR Primary Coolant Flow Path (from [S-2])
reactor core. However, in case all active means of providing emergency cooling fail, the Reactor Cavity Cooling System (RCCS) is incorporated in the MHTGR design to provide a passive way to remove the decay heat from the reactor core. The system, as shown in Figure 1.3, relies on natural circulation to effect heat removal by ambient air. The design has no valves or active components. It consists of a chimney section and an inside-cavity section. In the chimney, cross-connected coaxial rectangular ducts provide passages to allow air flow in and out, with cold air flowing down the outer-duct annulus and hot air, having been heated in the cavity, flowing up the inner-duct. The inner-ducts are insulated from the outer-ducts to prevent heat transfer between the hot air and the cold air. The entire chimney section rises about 27 meters above the reactor core. Thus, it will provide an extra driving force to maximize the natural convection effect. The cross-connection between ducts at the plenum will ensure that the blockage of one or more flow passages will not prevent the cooling air from reaching all the risers in the cavity.

Inside the reactor cavity, the cold air flows within the downcomers to a cold plenum; it then flows upwards in more than 200 separate rectangular risers situated around the walls of the reactor vessel cavity, where it absorbs the decay heat radiated from the reactor vessel. The hot air leaving the risers goes into a hot plenum, which is at the top of the riser, and connects risers and chimney, and then to the chimney. A cross-section top view of the inside-cavity region is shown in Figure 1.4. The cavity has a nearly square cross-section. The downcomers form the cavity wall. The typical design riser has a rectangular cross-section of about 2"x10". The risers are located several feet from the reactor vessel wall and near the cavity wall. The space between adjacent risers is approximately one riser width. This enables more surface area to be exposed to the vessel wall for better radiation heat transfer. However, this space also allows energy from the vessel to reach the cavity wall. Hence, the space between the risers and the cavity wall will also let the risers absorb the reflected radiation from the cavity wall, and make the riser wall temperature more evenly distributed circumferentially, (although conduction in the riser
Figure 1.3  Reactor Cavity Cooling System (RCCS) (from [G-1])
Figure 1.4 Cross-Section View of the Reactor Cavity (from [G-1])
walls also makes a major contribution to smoothing the distribution of the riser wall temperature).

1.3 Contributions of the Present Work

In the present thesis, work is concentrated on heat transfer from the outer surface of the vessel wall to the ambient air, given a vessel wall heat flux distribution. Inside the cavity, the heat transfer by radiation and convection from the vessel to the riser array has been studied by selecting a representative unit cell consisting of vessel wall, cavity wall and one riser tube. To determine the heat transfer coefficient and the friction factor inside the riser tube, which are two key parameters governing RCCS performance, an experiment was carried out which covered all possible RCCS operating conditions and extended well into the mixed convection regime. The experimental results were used to test existing correlations and develop new correlations for these two parameters. The overall heat transfer from the reactor vessel to the ambient air was then determined by an analysis coded as the computer code RECENT (Reactor Cavity Energy Transfer). The RECENT code was also used to optimize the configuration and arrangement of the riser array to make the vessel wall temperature as low as possible.

In this thesis, Chapter 2 reviews the existing literature on the mixed convection heat transfer coefficient and friction factor, particularly the limited number of studies dealing with air in up-flow.

Chapter 3 uses the existing theories and experiment results to predict the heat transfer coefficient and friction factor under RCCS operating conditions. It also models the overall heat transport ability within the risers.

Chapter 4 describes the MHTGR RCCS mockup experiment and its Nusselt number and friction factor results.

Chapter 5 compares the experimental results with the existing theories.
Chapter 6 presents the models for radiation heat transfer and convective heat transfer inside the cavity.

Chapter 7 puts the experimental Nusselt number and friction factor results from Chapter 4 and the radiation and convection heat transfer model from Chapter 6 together and calculates the overall heat transfer from the vessel to the ambient air, and the maximum vessel wall temperature. This calculation is coded as the RECENT code.

Chapter 8 analyzes the impact of the present experiments and calculations on RCCS performance and MHTGR safety.

Chapter 9 summarizes the work, and makes recommendations for future work.

The appendices contain several detail derivations of correlations used in the text, such as the temperature limit on the experimental range, the experiment data reduction procedure, and the error analysis process. The detailed experimental results for each run are also given. Finally a sample problem and its input and output for the RECENT code are included.
CHAPTER 2
LITERATURE REVIEW:
VERTICAL MIXED CONVECTION IN A HEATED DUCT

2.1 Introduction

Recently, mixed convection has received increasing attention because it has been encountered in potential applications in passive decay heat removal from nuclear reactors, solar power development, and electronic device cooling. The term "mixed convection" is used to describe the flow condition where both external forces and the gravitational body force have obvious effects on the velocity and temperature profiles of the flow. Mixed convection can be classified under two categories: laminar flow and turbulent flow, where each can be further divided into aiding flow (heated upflow and cooled downward flow) and opposing flow (heated downward flow and cooled upflow). Many studies have been carried out both experimentally and analytically on mixed convection phenomena. Most of these studies focus on Nusselt number behavior; only a few deal with friction factors. On one hand, these studies lead to qualitative conclusions and some empirical correlations. On the other hand, some results are still contradictory and controversial. However, reviewing previous studies on mixed convection will certainly shed light on ongoing experiments and theoretical research on this subject.

In this literature review, attention will be focused on aiding mixed convection flow, particularly on heated upflow. Since the current design of the MHTGR RCCS operates well into the turbulent regime, the review will be restricted to the turbulent flow mixed convection regime. Laminar mixed convection will be mentioned only when it is useful to make comparisons. Furthermore, mixed convection flow using gases as flow media will be given a closer look than water and other liquids. The review will be subdivided into three parts: mixed convection criteria (Sections 2.3 and 2.4), the Nusselt number (heat transfer coefficient) (Section 2.5) and the friction factor (Section 2.6).
2.2 Dimensionless Groups Used in Mixed Convection

The general governing equations for determining the velocity and temperature fields in a mixed convection flow are given as:

\[
\frac{Dp}{Dt} + \rho(\nabla \cdot v) = 0
\]

(2.1)

\[
\rho \frac{Dv}{Dt} = -\nabla p - \nabla \left[ \mu \nabla \cdot v \right] + \nabla \left[ \frac{4}{3} \mu \nabla \cdot v \right] + \rho g
\]

(2.2)

\[
\rho_c p \frac{D\theta}{Dt} = -\nabla \cdot q'' + q''' + \epsilon \frac{Dp}{Dt} + \phi
\]

(2.3)

where \( D/ Dt \) is the substantial derivative, \( x \) is the vector cross product and \( g \) is the acceleration due to the gravity. The meaning of the rest of the parameters can be found in the Nomenclature. The above equation set is very difficult to solve. Empirical correlations are usually employed to summarize the experimental results. These correlations use dimensionless parameters, which are derived by non-dimensionalizing the above equations. To obtain dimensionless parameter groups for the purpose of applying them to mixed convection conditions, some simplifications can be made to the above equations.

For the cases we are interested in, steady state is always assumed. In addition, the following assumptions can be made.

1) The Boussinesq approximation is employed, namely, the density variations are neglected everywhere but in the gravity term of the momentum equation.

2) The fluid is incompressible, i.e. \( \nabla \cdot v = 0 \).

3) The viscosity is assumed to be a constant.

4) The energy term due to the thermal expansion of the fluid is small compared to the other terms, and it is negligible.

5) There is no heat generation within the fluid, i.e. \( q''' = 0 \).
With the above assumptions Eqs.(2.1) to (2.3) become

\[
\frac{\partial v_i}{\partial x_i} = 0
\]  \hspace{1cm} (2.4)

\[
\rho v_i \frac{\partial v_i}{\partial x_i} = \rho_o g \beta (T - T_o) - \frac{\partial \tilde{p}}{\partial x_i} + \mu \frac{\partial^2 v_i}{\partial x_i^2} \quad (j = 1,2,3)
\]  \hspace{1cm} (2.5)

\[
\rho c_p v_i \frac{\partial T}{\partial x_i} = k \frac{\partial^2 T}{\partial x_i^2} + \phi
\]  \hspace{1cm} (2.6)

where

\[
\tilde{p} = p + \rho_o g x_i
\]  \hspace{1cm} (2.7)

x is a general coordinate direction and i = 1, 2, 3.

To non-dimensionalize Eqs.(2.4) to (2.6), let

\( v_o \) be the characteristic velocity,

\( D \) be the reference length,

\( \rho_o \) be the reference density,

\( T_w \) and \( T_o \) be the wall and environment temperatures, respectively. Then, one has the dimensionless parameters (with the superscript \( * \)) as follows:

\[
v^* = \frac{v}{v_o}
\]  \hspace{1cm} (2.8)

\[
x_i^* = \frac{x_i}{D}
\]  \hspace{1cm} (2.9)

\[
\tilde{p}^* = \frac{\tilde{p}}{\rho_o v_o^2}
\]  \hspace{1cm} (2.10)

\[
t^* = \frac{t}{(D/v_o)}
\]  \hspace{1cm} (2.11)
\[ T^* = \frac{T - T_0}{T_w - T_0} \quad (2.12) \]

Substituting the above expressions into Eqs.(2.5) and (2.6), one has

\[ v_i \frac{\partial v}{\partial x_i} = \frac{G_{\Delta T} T^*}{Re^2} - \frac{\bar{p}^*}{Re} + \frac{1}{Re} \frac{\partial^2 v}{\partial (x_i^*)^2} \quad (2.13) \]

\[ \frac{\partial T^*}{\partial x_i^*} = \frac{1}{Pr Re} \frac{\partial^2 T^*}{\partial (x_i^*)^2} \quad (2.14) \]

where

\[ Re = \frac{\rho_o v_o D}{\mu} \quad (2.15) \]

\[ Pr = \frac{\mu c_p}{k} \quad (2.16) \]

\[ Gr_{\Delta T} = \frac{\rho^2 g \beta (T_w - T_0) D^3}{\mu^2} \quad (2.17) \]

are the most commonly used dimensionless parameter groups in mixed convection analysis. In addition to the Grashof number based on the temperature difference as defined in Eq.(2.17), other Grashof number definitions, such as one based on the wall heat flux, are also widely used in mixed convection analyses. The definitions of these different Grashof numbers, their relations, as well as other dimensionless parameter groups which will be encountered in this literature review are summarized in Table 2.1.
Table 2.1  Definitions of Dimensionless Parameter Groups
Encountered in the Mixed Convection Analysis

<table>
<thead>
<tr>
<th>Name</th>
<th>Dimensionless Group</th>
<th>Note</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reynolds Number</td>
<td>( \text{Re} = \frac{\rho_0 v_0 D}{\mu} )</td>
<td></td>
</tr>
<tr>
<td>Prandtl Number</td>
<td>( \text{Pr} = \frac{\mu c_p}{k} )</td>
<td></td>
</tr>
<tr>
<td>Grashof Number</td>
<td>( \text{Gr}^{\dagger}_{\Delta T,D} = \frac{\rho^2 g \beta \Delta T D^3}{\mu^2} )</td>
<td></td>
</tr>
<tr>
<td></td>
<td>( \text{Gr}^{\dagger}_{\Delta T,L} = \frac{\rho^2 g \beta \Delta T L^3}{\mu^2} )</td>
<td></td>
</tr>
<tr>
<td></td>
<td>( \text{Gr}_{\text{q}''} = \frac{\rho^2 g \beta q'' D^4}{k \mu^2} )</td>
<td></td>
</tr>
<tr>
<td></td>
<td>( \text{Gr}_A = \frac{\rho^2 g \beta D^4}{16 \mu^2} \frac{dT_a}{dz} )</td>
<td></td>
</tr>
<tr>
<td></td>
<td>( \text{Gr}<em>A = \frac{\text{Gr}</em>{\text{q}''}}{4 \text{RePr}} )</td>
<td></td>
</tr>
<tr>
<td>Peclet Number</td>
<td>( \text{Pe} = \text{Re Pr} )</td>
<td></td>
</tr>
<tr>
<td>Rayleigh Number</td>
<td>( \text{Ra} = \text{Gr Pr} )</td>
<td></td>
</tr>
<tr>
<td>Stanton Number</td>
<td>( \text{St} = \frac{h}{(pc_p v)} )</td>
<td></td>
</tr>
<tr>
<td></td>
<td>( = \frac{\text{Nu}}{(\text{RePr})} )</td>
<td></td>
</tr>
<tr>
<td>Nusselt Number</td>
<td>( \text{Nu} = hD/k )</td>
<td></td>
</tr>
</tbody>
</table>

\( \dagger \) The specification of \( \Delta T \) varies among different investigators. What \( \Delta T \) should be used will be noted when dealing with specific problems.

2.3 Phenomenological View

In a mixed convection flow in a vertical pipe, the presence of the buoyancy force will change the heat transfer and pressure drop from their isothermal forced convection values by affecting the velocity and shear stress distributions. Although the way buoyancy affects the flow is similar for laminar and turbulent flows, the outcomes on the heat transfer and pressure drop are quite different for the laminar and turbulent flows. For a laminar
aiding flow, buoyancy increases the heat transfer rate compared to the forced convection value, while for a turbulent aiding flow, buoyancy decreases the heat transfer rate compared to the value if there were no buoyancy present. Both effects reverse for an opposing flow. This behavior can be explained qualitatively in terms of how buoyancy affects the velocity and shear stress distributions.

The velocity and shear stress distributions in a vertical tube for both aiding and opposing flows are qualitatively illustrated in Figure 2.1, along with the isothermal forced convection flow profiles. The convection heat transfer is in general determined by two factors: the velocity distribution near the surface, and the heat diffusivity in the fluid. In a laminar flow, the fluid heat diffusivity is constant, equal to the molecular diffusivity, since there is no turbulent diffusivity generated. Therefore, variation in the heat transfer rate is solely determined by changes in the velocity distribution near the surface. The higher the velocity near the surface, the higher the heat transfer rate. Figure 2.1 shows that for aiding flow the buoyancy will increase the velocity near the wall and increase the heat transfer rate. For an opposing flow the buoyancy will decrease the velocity near the wall, and, therefore, will decrease the heat transfer rate. For the friction factor, a similar conclusion can be obtained since the friction factor is a function of the wall shear stress, \( \tau_w \), which in turn is determined by the fluid velocity gradient at the wall.

In a turbulent flow, the buoyancy effect on the velocity profile is the same, but the fluid heat diffusivity is no longer a constant, since the turbulent heat diffusivity is non-zero. As a matter of fact, the turbulent heat diffusivity is several orders of magnitude greater than the molecular diffusivity and is a dominant factor in turbulent convection heat transfer. Because of the complexity of predicting the turbulent heat diffusivity, it is difficult to accurately predict the turbulent mixed convection heat transfer coefficient. However, its qualitative behavior can still be explained as follows. In a turbulent pipe flow, the turbulent heat diffusivity, \( \kappa_t \), increases (decreases) with the level of turbulent kinetic energy, which is mainly generated by turbulent shear production. It is known that the turbulent shear
Figure 2.1 Qualitative Illustration of Buoyancy Effects (from [A-3])
production is equal to the turbulent shear stress times the velocity gradient. The high turbulence production region is at $Y^+ = 16$ to $30$ [H-2] which is close to the wall. From Figure 2.1, an aiding flow produces less turbulent energy than a forced convection flow at the same Reynolds number, which leads to a reduction in the heat transfer. For an opposing flow, the opposite will occur.

2.4 Mixed Convection Criteria

The principle upon which to establish a criterion for mixed convection is clear, i.e. when both forced convection effects and buoyancy effects have a measurable influence on the flow field and the temperature field. However, where the boundaries should be drawn between forced convection and mixed convection, as well as between natural convection and mixed convection is far less clear. Different investigators have their own criterion based on how they define the importance of the buoyancy effect on the flow field. For example, some define the boundary between forced and mixed convection [M-1] when the heat transfer coefficient exhibits a $10\%$ deviation from the forced convection value. Others define the boundary as a $1\%$ deviation [P-5].

The flow regime maps of Metais and Eckert [M-1] were among the earliest. The flow regime map for vertical tube flow is given in Figure 2.2. The ordinate is the Reynolds number; the abscissa is $GrPrD/L$ and it is valid for $10^{-2} < PrD/L < 1$. The boundaries dividing mixed convection from forced convection, and mixed convection from natural convection are drawn using the criterion that the actual heat flux under the combined influences of both forces does not deviate by more than $10\%$ from that which would be caused by a single force (either an external force or a body force). The primary basis of this flow regime map is the experimental results available at that time. It should be pointed out that the $D/L$ dependency on the abscissa has not appeared in subsequent researchers’ criteria. An increase of $L/D$ may decrease buoyancy effects for a constant wall temperature boundary condition. However, an increase of $L/D$ will certainly increase the buoyancy
Fig. 2.2  Regimes of Free, Forced, and Mixed Convection for Flow in a Vertical Tubes 
\((10^{-2} < \Pr \frac{D}{L} < 1, \text{ From } [M-1])\)
effect for a constant heat flux boundary condition. Therefore, the parameter on the abscissa is questionable, since the map is for both boundary conditions.

Figure 2.3 shows the flow regime map presented by Tanaka et al [T-1]. The map is based on numerical calculations for nitrogen gas using a k-ԑ turbulent flow model proposed by Jones and Launder [J-5, J-6], and its modified version by Kawamura [K-1]. The map indicates that when increasing Gr for a given Re number in turbulent flow, at the point where the calculated Nusselt number becomes 80% of the full forced convection Nusselt number, the flow enters mixed convection. For further increase of Gr, a point will be reached where the calculated Nusselt number recovers from its lowest value to a value equal to the full forced convection value. The flow is considered as entering the natural convection regime at that point. The two boundaries can be expressed approximately as

\[
Re = 50 \text{Gr}^{8/21} \quad (2.18-1)
\]

\[
Re = 16.5 \text{Gr}^{8/21} \quad (2.18-2)
\]

This flow regime map has been supported by experimental data from Tanaka et al [T-1] (not shown in Figure 2.3) obtained in a 2.3 cm I.D. stainless steel pipe using nitrogen gas in the range of 3000 < Re < 5000 and a relatively large variation of Gr.

The criterion given by Jackson and Hall [J-2] is on the basis of buoyancy reducing the heat transfer coefficient from the corresponding forced convection value by more than 5%. The criterion predicts that when

\[
\frac{\text{Gr} \left( \frac{\mu_w}{\mu_b} \right) \left( \frac{\rho_b}{\rho_w} \right)^{0.5}}{(\text{Re}^{2.7})(\text{Pr}^{0.5})} > 10^{-5} \quad \text{for } 0.7<\text{Pr}<5 \quad (2.19)
\]
Fig. 2.3  Flow Regime Map for Combined Forced and Natural Convection in a Vertical Tube (From [T-1])
the flow is in mixed convection. This criterion is derived from the shear stress reduction caused by the buoyancy force and its influence on heat transfer impairment. Empirical correlations were also used in the derivation. Therefore, the value of $10^{-5}$ is somewhat speculative.

As mentioned earlier, Polyakov's criterion \[ P-5 \] is based on when $Nu$ deviates from its forced convection value by more than 1%, rather than 5%. However, the form of the correlation is quite different from that of Jackson and Hall.

\[
Gr_q^* > \frac{1.3 \times 10^{-4} Re^{2.75} Pr^{[Re^{1/8} + 2.4(Pr^{2/3} - 1)]}}{\log_{10} Re + 1.15 \log_{10} (5Pr + 1)+0.5Pr-1.8} \quad \text{for } Pr \geq 0.5
\]  

(2.20)

Polyakov also gives a criterion based on when the friction factor deviates from its forced convection value by more than 1%. This occurs when

\[
Gr_q^* > 2 \times 10^{-4} Pr Re^{2.75}
\]  

(2.21)

Other criteria are given by Hall \[ H-1 \] and Byrne and Ejiogu \[ B-3 \]. Hall derived an expression showing that when the shear stress at $Y^+ = 30$ is approximately zero, the heat transfer is significantly impaired by the buoyancy. This occurs at

\[
\frac{Gr_D}{Re^{2.7}} = 0.7 \times 10^{-4}
\]  

(2.22)

However, since this is a condition for significant impairment of heat transfer, it seems that this is a criterion for minimum heat transfer rather than onset of mixed convection.

The criterion given by Byrne and Ejiogu is also based on the existence of a significant buoyancy effect on heat transfer, and it predicts that the boundary between forced or mixed convection is at
\[ \frac{Gr_p}{Re^{1.8}} = 0.05 \]  \hspace{1cm} (2.23)

It also predicts that the minimum heat transfer occurs at

\[ \frac{Gr_p}{Re^{2.8}} = 10^{-4} \]  \hspace{1cm} (2.24)

based on their air flow mixed convection experiments. Comparing Hall's criterion with the Byrne and Ejiogu minimum heat transfer criterion, one finds that they are very close to each other.

In general, all criteria rely on the dimensionless numbers, Gr, Re and Pr. Several criteria using the dimensionless group

\[ \frac{Gr^m}{Re^nPr^s} \]  \hspace{1cm} (2.25)

with m, n and s larger than zero, which is consistent with the non-dimensionalizing analysis in Section 2.2.

2.5 Nusselt Number

The Nusselt number behavior in mixed convection has been more widely studied, and is relatively better understood than other parameters in mixed convection, such as friction factor, either from a theoretical or experimental point of view. Some simple theories have been proposed, and a number of experiments have been conducted in the past. Recently, several numerical calculations based on k-\( \epsilon \) models have been reported. In this section, the representative theories, experiments and numerical calculation results will be reviewed.
In an aiding mixed convection flow, two types of velocity profiles can occur. As shown in Figure 2.4, type I occurs when the velocity near the wall has increased due to the buoyancy force, but the maximum flow velocity is still at the pipe center, (lines 1-3 in Fig. 2.4). The type II profile occurs when the buoyancy effect is so large that the maximum velocity is no longer at the center of the pipe but at some location between the wall and the center, (line 4 in Fig. 2.4).

Figure 2.4 also shows that as the buoyancy effect increases, the total shear stress, which is dominated by turbulent effects, decreases, and the Nusselt number decreases (ref. Section 2.3) until a point where turbulent shear stress production is close to zero. At that point, laminarization occurs and the Nusselt number reaches its minimum value. Further increase of the buoyancy effect will make the velocity profile change from type I to type II, and the turbulent shear stress becomes negative. However, the absolute values of this turbulent shear stress production start increasing, with the result that the heat transfer also starts improving from its minimum value.

Based on the turbulent shear stress reduction caused by the buoyancy force, Jackson and his co-workers [J-1] derived a semi-empirical correlation for turbulent mixed convection. They assumed the buoyancy layer and thermal boundary layer are identical. Expressing the shear stress reduction, $\Delta \tau_{\delta B}$, across the buoyancy layer, as

$$\Delta \tau_{\delta B} = \int_0^{s_f} (\rho_b - \rho)gdy$$  \hspace{1cm} (2.26)

They found that

$$\frac{\Delta \tau_{\delta B}}{\tau_w} = \frac{Gr_{gb}^*}{Re^2 f Nu^2}$$  \hspace{1cm} (2.27)
Figure 2.4  Effect of Buoyancy on Velocity, Shear Stress and Heat Transfer for Aiding Turbulent Flow (from [S-6])
By making use of a friction coefficient–Reynolds number relationship of the form \( f = C R e^n \), combined with the facts that \( \tau_w \propto f R e^2 \) and \( \text{Nu} = C' R e^m P r^s \), a correlation resulted having the form

\[
\left( \frac{\text{Nu}}{\text{Nu}_0} \right) = \left( 1 - \frac{K G r_q}{R e^{m'} P r^s} \left( \frac{\text{Nu}}{\text{Nu}_0} \right)^{-2} \right)^{n'}
\]

(2.28)

In general the coefficients \( K, m', n', \) and \( s' \) depend on the surface conditions, and can be found from experiment. For a smooth surface, Jackson gives the following values

\[
m' = 3.35, \quad n' = 0.46
\]

\[
s' = 0.9, \quad K = 2.4 \times 10^4
\]

Later, they [J-3] have revised these value to

\[
m' = 3.425, \quad n' = 0.46
\]

\[
s' = 0.8, \quad K = 8 \times 10^4
\]

and define a dimensionless buoyancy number as follows

\[
\text{Bo} = \frac{8 \times 10^4 G r_q}{R e^{3.425} P r^{0.8}}
\]

(2.29)

Utilizing this \( \text{Bo} \) number, the correlation of Eq.(2.28) becomes

\[
\left( \frac{\text{Nu}}{\text{Nu}_0} \right) = \left( 1 - \frac{\text{Bo}}{\left( \frac{\text{Nu}}{\text{Nu}_0} \right)^2} \right)^{0.46}
\]

(2.30)

Note that the term in the brackets on the right hand side of Eq. (2.28) has been taken as the absolute value, which makes the correlation capable of describing both type I and type II flows in a single equation. The correlation was also compared with a few experimental results, as shown in Figure 2.5. The data of Steiner [S-5] in the figure were taken from the
\[
\frac{Nu}{Nu_0} = \left[ 1 \pm \frac{Bo}{[Nu/Nu_0]^2} \right]^{0.46}
\]

Figure 2.5  Comparison of Jackson's correlation, Eq.(2.30), and a Turbulent k-ε Model with Experimental Data From Carr et al [C-1], Steiner [S-5] and Easby [E-1] (from [C-3])
fully developed section in his air mixed convection experiment. The Carr et al [C-1] data in the figure are also in the fully developed region in their air flow mixed convection experiment.

An experimental study of mixed convection in a vertical tube using water was performed by Petukhov [P-3] and Petukhov and Strigin [P-2]. The upflow results will be given here. For type I flows (i.e. no double-hump velocity profile occurs), the Nusselt number is given as

$$\frac{N_u}{N_{u_0}} = (1 + B \frac{Ra_A}{Re^2})^{-1}$$

(2.31)

where $B$ is a coefficient which depends on the experimental range and inlet length as follows

If $\left[ \frac{L}{D_{in}} \right] = 0$ and $\frac{Ra_A}{Re^2} < 3 \times 10^{-4}$, $B=0.15 \times 10^4$

If $\left[ \frac{L}{D_{in}} \right] = 26$ and $\frac{Ra_A}{Re^2} < 10^{-4}$, $B=1.15 \times 10^4$

For type II flows, the Nusselt number is independent of the inlet length and has the following form:

$$\frac{N_u}{N_{u_0}} = 10 \left( \frac{Ra_A}{Re^2} \right)^{1/3}$$

$$\frac{Ra_A}{Re^2} > 3 \times 10^{-4} \text{ if } \left[ \frac{L}{D_{in}} \right] = 0$$

$$\frac{Ra_A}{Re^2} > 10^{-4} \text{ if } \left[ \frac{L}{D_{in}} \right] = 26$$

(2.32)

where $N_{u_0}$ is the forced convection Nusselt number calculated from the Dittus-Boelter correlation under the same flow conditions. The experiments covered large ranges of $Re$.
and Gr numbers. However, the Prandtl number range is from 2 to 6, which is out of the air application range. The experimental results and a comparison with the correlation are shown in Figure 2.6. Petukhov and Polyakov [P-1] recently publish another correlation which applies to a wide range, namely Pr > 0.6, Re > 3000, 0 < Gr_q < 10^9 and L/D > 40,

\[
\frac{f/8}{St} = \frac{1 + 0.83e^2}{1 + 0.042e^2[E^{1/4} \log_{10}(Re/8)]^{-1}} + 12.74 \sqrt{\frac{1 + 0.72e^3(1+0.28e)}{1 + 0.43e^4}} \cdot \frac{1 + 0.58e^2}{1 + 0.83e^2}
\]

(2.33)

where

\[
e = \frac{Gr_q^{0.7} \cdot 10^3}{Pr \cdot Re^{2.75}}
\]

\[
E = \frac{Gr_q}{Pr \cdot Re^4}
\]

and the friction factor f is

\[
f = \left[ \frac{1 + 0.83e^2}{1.82 \log_{10}(Re/8) + 0.076e^2} \right]^2
\]

(2.34)

both St number and friction factor are shown in Figure 2.7.

Connor and Carr [C-2] also performed an experiment using air in aiding flow in a vertical tube. The buoyancy effects on the velocity and turbulent shear stress profiles are given in Figure 2.8, along with the fully developed Nusselt number corresponding to the profiles calculated from the measured parameters. Connor and Carr also summarized experimental results using water and mercury from other experiments and gave a general correlation for type II flows

\[
\frac{Nu}{Nu_0} = 8.84 (\frac{Gr_A}{Re^2})^{0.263}
\]

(2.35)
Figure 2.6  Comparison of Eqs.(2.31) and (2.32) (solid line) with Experimental Data (point)
(Solid points for (D/L)_{in}=0, (L/D)_H=80; open points for (L/D)_{in}=26, (L/D)_H=99;
from [P-2])
Figure 2.7 Stanton Number and Friction Factor Versus Parameter E (from [P-1])

1) Pr=0.7, Re=4000-6000, experimental data from [S-5], [P-6] and [C-1]; 2) Re=5000, calculated using (2.33); 3) Re=5000, calculated using (2.34); 4) Pr=0.07, Re=8000-12,000, experiment data from [P-6]; 5) Re=10,000, experiment data from [C-1] and [S-5]; 6) Re=10,000 calculated using (2.33); 7) Re=10,000 calculated using (2.34); 8) Pr=0.6, Re=10,000, experiment data from [S-1]; 9) Pr=0.6, Re=10,000, calculated using (2.33); 10) Pr=0.7, Re=500, experiment data from [S-5]
Figure 2.8  Velocity, Turbulent Shear Stress and Nusselt Number Distributions Obtained in the Connor et al [C-2] Experiment
The correlation and the corresponding experiment data are shown in Figure 2.9. It is interesting to note the continuously decreasing trend of the air data beyond the Gr\(_A/\text{Re}^2\) value at which it was predicted that type I flow should occur and the heat transfer rates should improve. No explanation was given for this behavior.

An experimental and theoretical study of the aiding flow with air and argon in a vertical tube was performed by Grief [G-2]. His turbulent measurements are in the range of 10,000 < Re < 19,500. The measured point was located at L/D = 108, where the flow is fully developed. However, his Gr numbers are too low to see the buoyancy effects in the turbulent flow. The measured Nusselt numbers were very close to the forced convection values.

Figure 2.10 shows experimental results of Byrne and Ejiogu [B-3] for air flow in a vertical pipe of 8.86 cm diameter with a uniform wall temperature boundary condition. Figure 2.11 shows the same data with a plot of constant Nu on a Gr-Re plane. The effect of the buoyancy force on the Nusselt number is obvious.

In addition to relatively simple theoretical analyses and the experimental results, numerical calculations have also been developed based upon more complicated theories. The k-\(\varepsilon\) turbulence models are the most common models used in such calculations. Abdelmeguid and Spalding [A-1] use a two equation turbulent model, calculating the heat transfer coefficient impairment due to the laminarization caused by the buoyancy effect, Figure 2.12. The calculated results were also compared with the Buhr [B-2] experimental data of mercury. Satisfactory agreement was obtained. Cotton [C-3] and Yesilyurt [Y-1] use different k-\(\varepsilon\) models in their calculations, but their results are almost identical. Their results are very close to the Jackson correlation, Eq.(2.30), but have better agreement with the experimental data of Steiner [S-5] and Carr et al [C-1], Figure 2.5.

In summary, the review shows that the theories and the experiments all predict similar qualitative Nusselt number behavior: an initial decrease compared with the forced
Figure 2.9  Mixed Convection Nusselt Number Correlation (only Type II data are shown; from [C-2])

- Equation (2.35)
- Estimated line for water data of Petukhov and Strigin [P-2]
- Curve 1: Type I data from [K-2]
- ▲ Air data [C-2]
- ○ Water data [K-2]
- ▲ Mercury data [H-4]
- + Mercury data [L-2]
- × Water data [A-2]
- • Mercury data [J-4]
Figure 2.10  Effect of Grashof Number on the Heat Transfer Coefficient (from [B-3])
Fig. 2.12 Variation of Nusselt Number with Heat Input in an Ascending Flow
(Re=25,000, From [A-1])
convection value until a minimum point is reached, beyond which it starts to improve continuously.

2.6 Friction Factor in Turbulent Mixed Convection

Compared to the heat transfer behavior in turbulent mixed convection, the friction factor has received much less attention. A key reason for this is the difficulty in measuring the friction factor, either from the velocity gradient at the wall or from direct friction pressure drop measurements. Consequently, neither a satisfactory theory nor a modest data set exist for the friction factor in mixed convection.

In principle, the friction factor is proportional to the shear stress at the wall between the fluid and the surface. Therefore, an increase of that shear stress will increase the friction factor. However, some researchers do not base friction factor prediction on wall shear stress behavior. Petukhov and Polyakov [P-1] employed a theory similar to the Reynold’s analogy principle, in which they predict similar behavior between the Nusselt number and the friction factor in a turbulent aiding flow. Their correlation for calculating the friction factors in a turbulent aiding flow has been given in Eq. (2.34) and is repeated here.

\[ f = \left( \frac{1 + 0.83e^2}{1.82\log_{10}(Re/8) + 0.076e^2E^{-0.25}} \right)^2 \]  \hspace{1cm} (2.34)

where

\[ e = \frac{Gr_q^{2.5} \cdot 10^3}{Pr \cdot Re^{2.75}} \]

\[ E = \frac{Gr_q^{7.5}}{Pr \cdot Re^4} \]
Their isothermal forced convection friction factor correlation is

\[ f_{iso} = \left[ 1.82 \log_{10}(Re/8) \right]^{-2} \]  \hfill (2.36)

Yesilyurt [Y-1] numerically calculates the wall shear stress and friction factor in air in aiding flow, using a k-\( \varepsilon \) turbulent flow model, with his code CONDOR. His results show that for air flow the friction factor does experience a small decrease and then will continuously increase as the buoyancy effect increases. However, Abdelmeguid and Spalding [A-1] predict a continuous increase of the friction factor in their numerical calculation results using laminarization theory, Figure 2.13.

There are limited experimental results reported for friction factor behavior in a turbulent mixed convection aiding flow. In general, there are two fundamental ways to measure the friction factors. One is through the measurement of wall shear stress. The other is to measure the friction pressure drop. In the case of wall shear stress measurement, the friction factor can be found as [P-7]

\[ f = \frac{4\tau_w}{\rho V^2} \]  \hfill (2.37)

where fluid density \( \rho \) and mean flow velocity \( V \) can be evaluated at the bulk temperature of the measuring point. \( \tau_w \) is wall shear stress.

When measurement of the friction factor is made by measuring the friction pressure drop, the friction factor is obtained as

\[ f = \frac{\Delta P / 2D}{\Delta z / \rho V^2} \]  \hfill (2.38)
Fig. 2.13 Friction Factor Variation with Heat Input for an Ascending Flow (from [A-1])
where $\Delta p_f$ is the friction pressure drop, $\Delta z$ is the length over which $\Delta p_f$ was measured, $D$ is hydraulic diameter, $\rho$ and $v$ are fluid density and mean velocity, respectively. Carr et al [C-1] measured the velocity gradient at Reynolds numbers around 5000 and $Gr_A$ from $1.11 \times 10^4$ to $2.54 \times 10^4$. Since it is very difficult to measure the fluid velocity gradient at the wall without disturbing the fluid velocity profiles and causing a large error, the closest point they measured was 0.05R from the wall in a 3.886 in. diameter pipe. Their friction factor results calculated from the measured shear stresses, as shown in Figure 2.14, show that the friction factor exhibits a small decrease, and then increases to values higher than the isothermal friction factor. It should be noted that their experimental Reynolds number range is relatively narrow and the initial decrease of the friction factor is also relatively small: less than 10%, compared with the isothermal value. Considering the errors in the measurements, which are not reported, and the uncertainty in the correlation used to calculate the forced convection friction factors, this small observed decrease can hardly be considered to be conclusive. On the other hand, the friction factor is 20% higher than the forced convection value, with a continuously increasing trend, at high $Gr_A$ numbers. Thus, the increase of the friction factor under mixed convection is thus considered to be a valid overall observation.

Petukhov and Strigin [P-2], in their earlier series of measurements of pressure drops on their 49.66 mm diameter loop, show that there is no decrease in the friction factor as the buoyancy force increases, Figure 2.15. They have given an expression to describe their experimental data as

$$\frac{f}{f_{iso}} = \left[ 1 + \frac{56(\frac{Ra_A}{Re^{3/2}})^2}{10.4} \right]$$

(2.39)

where $f_{iso}$ is the forced convection friction factor calculated from the Blasius correlation.
Figure 2.14  Effect of Grashof Number, Gr\textsubscript{A}, on Friction Factor (from [C-1])
Fig. 2.15 Experimental Data for Friction Factors Subject to Buoyancy Effects (from [P-21])
\[ f_{iso} = 0.316 \text{Re}^{-0.25} \quad (2.40) \]

and \( f \) is the fully developed friction factor under mixed convection conditions. The range of the equation is

\[
5 \times 10^3 < \text{Ra}_A < 4 \times 10^5 \\
300 < \text{Re} < 3 \times 10^4 \\
2 < \text{Pr} < 6
\]

As can be seen from the figure, the friction factors increase substantially when the buoyancy force increases. The highest value in the figure is more than 100 times greater than that of the isothermal forced convection value. In a later paper by Petukhov [P-3], the correlation was given with another set of coefficients as follows,

\[
\frac{f}{f_{iso}} = \left[ 1 + 3.5 \left( \frac{\text{Ra}_A}{\text{Re}^{3/2}} \right) \right]^{0.4} \quad (2.41)
\]

However, neither Eq.(2.39) nor Eq.(2.41) gives the line drawn in the figure, which the authors believed represented the data. Finally, the inclusion of data with Reynolds numbers down to 300, which should be well into the laminar flow regime, is not understood. It should be noted that the Prandtl range excludes the application of these results to air flow without further justification.

2.7 Chapter Summary

This brief review of turbulent aiding flow mixed convection shows that uncertainties exist in the predictions of thermal and hydraulic parameters in this regime. Flow regime maps and the criteria for the onset of mixed convection are still incomplete.
The existing criteria are based on somewhat arbitrary definitions of the importance of buoyancy effects. Even these limited criteria are contradictory.

The Nusselt number in turbulent mixed convection is relatively better understood compared to other phenomena. The qualitative behavior is in general agreement with both theory and experiment. However, quantitative predictions under particular conditions still exhibit large uncertainties.

The friction factor has received little attention so far. The limited work dealing with it is rather preliminary. Significant research work is needed, both theoretically and experimentally, to gain a better understanding of this phenomenon.
CHAPTER 3
CONVECTIVE HEAT TRANSFER IN A RCCS RISER TUBE

3.1 Overview

In the MHTGR RCCS design, the decay heat will be removed from the core to the ambient air via a series of heat transfer processes, namely: heat conduction through the graphite moderator to the outer surface of the graphite, heat radiation from the outer surface of the graphite to the inner surface of the reactor vessel, heat conduction through the reactor vessel, heat radiation from the reactor vessel outer surface to the riser tube, and then convection from the inside of the riser wall to the flowing air, which transports the energy to the ambient. The series nature of this process makes each step important. Therefore, each heat transfer process must be studied carefully.

Among all of the processes employed in the RCCS design, convective heat transfer inside the risers deserves priority attention because there is a potential of operating in a turbulent mixed convection condition, which is less understood than other modes of heat transfer. Two key parameters governing the heat transport ability in mixed convection are the heat transfer coefficient and the friction factor. Given the heat transfer coefficient and the friction factor, the convective heat transfer inside the risers can be easily determined by solving the continuity, momentum and energy equations.

In this chapter, the riser tube design is first reviewed. The operating conditions of a representative riser tube are then evaluated and compared with the existing flow regime criteria given in Chapter 2 to define the operating regime. After the operating regimes is identified, Nusselt number and friction factor predictions are made based on existing theories and experimental data. Finally, the equations for solving this heat convection problem and the scheme for their solution are given.
3.2 Riser Operation Regime

3.2.1 Riser Description

A riser flow path consists of three sections: an unheated inlet section, a heated section and an insulated chimney section. The heated section is located inside the reactor cavity where it receives heat from the reactor vessel by radiation and transfers it to the flowing air inside the riser. The chimney is about 27 m high, and located above the heated section to provide extra-driving force for the air flow.

A riser tube is made up of commercial carbon steel. Typically it has a height of about 60 ft. with a cross-section of 2″×10″, as shown in Figure 3.1. Except at the inlet plenum and exit plenum, there are no other cross connections between the risers, and therefore no cross flow between risers.

The hydraulic diameter, \( D_e \), of a single riser is easily obtained as 0.232 ft. from the dimensions given in Fig. 3.1, since \( D_e \) is defined as

\[
D_e = \frac{4A_f}{P_w} \tag{3.1}
\]

where \( A_f \) is air flow area, and \( P_w \) is the wetted perimeter.

3.2.2 Operating Conditions

The RCCS system will be active whenever the vessel wall temperature exceeds the ambient air temperature. The reactor vessel wall is not insulated; under normal reactor operating conditions the core center temperature is relatively low and the heat loss through the RCCS is small, about 0.8 MW out of a total thermal power of about 350 MW. However, under accident conditions, the heat transfer through the RCCS could be substantial if the other decay heat removal systems fail to activate.
Figure 3.1  Schematic of Riser Flow Channel (Not to Scale)
Since the RCCS is designed as the last barrier to prevent reactor core and vessel damage, we will always assume that there are no other safety systems in operation in the remaining analyses of this thesis.

The cases we are interested in here are both depressurized and pressurized loss of coolant accidents. They are among the most severe accidents in terms of the maximum core and reactor vessel temperatures attained. In a loss of coolant accident, the decay heat generated inside the core will all be either retained in the graphite moderator or transported to the ambient air through the RCCS. To predict the riser operating conditions, namely, mass flow rate, surface heat flux and air outlet temperatures one must know the decay heat generation rate as a function of time and what portion of that decay heat has been deposited in the graphite moderator. One must also know the heat transfer coefficient and the friction factor inside the riser. Unfortunately, the latter two are the parameters we are looking for. To simplify the analysis for this initial iteration, typical data will be assumed which utilizes several simplifications. The most important are a forced convection friction factor inside the riser; a heat transfer coefficient inside the riser calculated as 2/3 of the Dittus-Boelter correlation result, which is applicable to forced convection; and representation of all riser tubes by a few coarse nodes having equally effective radiation surface areas. Even with the above approximations, the calculated results will still provide a rough idea of the riser operating range. The design calculation results are a function of time. We will use the data at about 100 hours after the reactor has been shut down as the reference case, since this is when the maximum core, reactor vessel, riser wall, riser air exit temperatures, and the maximum air flow rate occur. The reference data of Table 3.1 are all the maximum values of the relevant parameters. They do not necessarily occur at the same time.

Table 3.2 shows the dimensionless parameters which will be used in determining the riser flow regimes calculated from the reference case data. It should be noted that the air properties change significantly in the operating range, and thereby affect the dimensionless parameters. The air bulk average temperature is used in obtaining the
Table 3.1  Reference Case Data to Determine Riser Operating Conditions

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Pressurized Conduction Cooldown</th>
<th>Depressurized Conduction Cooldown</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum Riser Wall Temperature (°C)</td>
<td>220</td>
<td>200</td>
</tr>
<tr>
<td>Air Inlet Temperature (°C)</td>
<td>43</td>
<td>43</td>
</tr>
<tr>
<td>Air Exit Temperature (°C)</td>
<td>183</td>
<td>163</td>
</tr>
<tr>
<td>Air Mass Flow Rate (kg/sec)</td>
<td>12.45</td>
<td>12.20</td>
</tr>
<tr>
<td>Total Heat Load (MW)</td>
<td>1.74</td>
<td>1.50</td>
</tr>
<tr>
<td>Power Peaking Factor</td>
<td>1.1</td>
<td>1.4</td>
</tr>
<tr>
<td>Average Heat Flux (W/m²)</td>
<td>834</td>
<td>719</td>
</tr>
<tr>
<td>Total Air Flow Area (m²)</td>
<td>2.291</td>
<td>2.291</td>
</tr>
<tr>
<td>Effective Convective Wall Area (m²)</td>
<td>2085.7</td>
<td>2085.7</td>
</tr>
<tr>
<td>Hydraulic Diameter (m)</td>
<td>0.0706</td>
<td>0.0706</td>
</tr>
</tbody>
</table>

parameters in Table 3.2. Should other temperatures be used, the Re and Gr numbers may differ considerably.

It should be pointed out that the use of these maximum values will provide an optimistic estimation in terms of the mixed convection prediction, namely: if the data from the reference case indicates the riser is already operating in the mixed convection regime, the entire riser operation will be in the mixed convection regime. The opposite is not true, however. Even if the reference data is not in the mixed convection regime, operating conditions other than the reference case may still be in the mixed convection regime. In other words, the reference case gives the minimum buoyancy effect over the entire range of potential RCCS potential operating conditions, which can be explained briefly as follows.
Table 3.2  Dimensionless Parameters for Determining Riser Operating Conditions Calculated From the Reference Case

<table>
<thead>
<tr>
<th>Cases</th>
<th>Pressurized Conduction Cooldown</th>
<th>Depressurized Conduction Cooldown</th>
</tr>
</thead>
<tbody>
<tr>
<td>((T_{air})_{ave} \hspace{0.5cm} (\text{C}))</td>
<td>113</td>
<td>103</td>
</tr>
<tr>
<td>(\rho \hspace{0.5cm} \text{(kg/m}^3)</td>
<td>0.915</td>
<td>0.939</td>
</tr>
<tr>
<td>(k \hspace{0.5cm} \text{(W/m} \cdot \text{C)})</td>
<td>0.0322</td>
<td>0.0315</td>
</tr>
<tr>
<td>(\mu \hspace{0.5cm} \text{(kg/m} \cdot \text{sec)})</td>
<td>2.193x10^{-5}</td>
<td>2.153x10^{-5}</td>
</tr>
<tr>
<td>(Re = \frac{\rho v D}{\mu} \hspace{0.5cm} )</td>
<td>1.75x10^{4}</td>
<td>1.75x10^{4}</td>
</tr>
<tr>
<td>(Gr_q'' = \frac{\rho^2 g \beta q'' D^4}{k \mu^2} \hspace{0.5cm} )</td>
<td>3.66x10^{7}</td>
<td>3.53x10^{7}</td>
</tr>
<tr>
<td>(Gr_A = \frac{Gr_q''}{4RePr} \hspace{0.5cm} )</td>
<td>765</td>
<td>736</td>
</tr>
<tr>
<td>(Pr \hspace{0.5cm} )</td>
<td>0.709</td>
<td>0.709</td>
</tr>
<tr>
<td>(Ra_A = Gr_A Pr \hspace{0.5cm} )</td>
<td>542</td>
<td>522</td>
</tr>
<tr>
<td>(Bo = \frac{8 \times 10^4 Gr_q''}{Re^{3.425} Pr^{0.8}} \hspace{0.5cm} )</td>
<td>0.0113</td>
<td>0.0110</td>
</tr>
<tr>
<td>(\frac{Ra_A}{Re^2} \hspace{0.5cm} )</td>
<td>1.77x10^{-6}</td>
<td>1.70x10^{-6}</td>
</tr>
</tbody>
</table>

First of all, the buoyancy effect is usually evaluated by the parameter \(Gr/Re^n\), where \(n \geq 1\). The Reynolds and Grashof number are proportional to the air flow rate and the heat removal rate. However, 100 hours after the reactor has been shut down, both the decay heat removal rate and the flow rate decrease at almost the same rate. Therefore, the ratio of \(Gr\) to \(Re^n\) is a minimum when the heat removal rate and the flow rate are at their maximum values. Second, the use of the smooth pipe forced convection friction factor in the reference case calculation will make the results optimistic, i.e. underestimate the buoyancy effects. Nearly all likely changes in the friction factor increase it: mixed convection may increase the friction factor (as noted in the review in last chapter), and
surface roughness and channel blockage will also cause it to increase. The result of a friction factor increase is a flow rate decrease, hence buoyancy effects increase.

There are several reasons for choosing the minimum buoyancy effect case as the reference case. First, as mentioned above, this case is the most optimistic one. If this case is already in the mixed convection regime, the other possible RCCS operating conditions will definitely be in the mixed convection regime. Second, the maximum reactor vessel wall and core center temperatures occur at this time, and they are the parameters which are of most concern. Third, it is difficult to determine the most pessimistic (i.e., the maximum buoyancy effect) case since the heat transfer coefficients and the friction factors are not known, yet.

Although the reference case will be used in the flow regime determination, a reasonable boundary envelope also should be given to indicate the potential RCCS operating range. This range is shown schematically in Figure 3.2. A three times increase of normal friction factor will be used for a lower limit on the flow rate. This gives a Reynolds number around 10,000. As can be seen from the figure, the maximum Reynolds number is 22,000 (instead of the value of 17,500 quoted for the reference case). The reason is that in this maximum range calculation the riser inlet condition is used, while in the reference case the riser average bulk temperature is used as the reference temperature to evaluate the properties. The heat flux on the riser wall could be from 0 to the reference case value, 1,000 W/m², with a peaking factor 1.4. However, very low power operation is not of interest, and therefore, the minimum heat flux on the riser wall was arbitrarily chosen as 2/5 of the maximum average heat flux, which occurs at about 500 hours after the reactor has been shut down. Note that in Figure 3.2, the maximum Reynolds numbers at different total powers are different (represented by a slanted line instead of a vertical line) because at lower power the buoyancy force will be smaller than at full power. The corresponding maximum flow rate will also decrease.
Figure 3.2  Schematic of the MHTGR RCCS Operating Range
3.3 Comparisons of Typical MHTGR RCCS Operating Conditions with Existing Mixed Convection Criteria

With the reference data in Tables 3.1 and 3.2, comparisons can be made with the mixed convection criteria reviewed in Chapter 2. The results are given in Table 3.3.

Figure 2.2 shows the flow regime map presented by Metais and Eckert [M-1] and the RCCS operating conditions; the operating regime is not in the mixed convection condition because of its high Reynolds number. In Chapter 2, the discussion has indicated that the parameter on the abscissa is questionable for a fixed wall heat flux case. Some well known mixed convection experimental results, such as those of Buhr et al [B-2] using mercury, are located close to the upper left hand corner. As can be seen from the figure, although the RCCS operating regime is not in the mixed convection regime, it is very close to it. Therefore, the conclusion that the RCCS operating regime is not in mixed convection can not be drawn conclusively.

Compared with the flow regime map provided by Tanaka et al [T-1], shown in Figure 2.3, the RCCS operating regime is well into the mixed convection regime. In fact, part of the normal operating condition is even in the natural convection regime. In the Jackson and Hall criterion [J-2], although the constant $10^{-5}$ is a speculative value, the reference case data valve is more than an order of magnitude larger; therefore the reference case is considered as being in the mixed convection regime according to their criterion. From the Polyakov [P-5] criterion based on Nusselt number deviation, the reference data is just inside the mixed convection regime. However, his criterion based on friction factor deviation predicts that the reference case is not in the mixed convection regime. Hall's criterion [H-1] has been considered to indicate a condition well into the mixed convection regime. The reference case data just matches his criterion. Finally, the reference data is well beyond Byrne and Ejiogu's criterion for onset of mixed convection.

Based on the above comparisons, except for the Polyakov criterion based on friction factor deviation, and Metais and Eckert's criterion, which is considered as having an incorrect D/L effect on the abscissa, all other criteria predict that the RCCS riser
### Table 3.3 Comparisons Between the Reference Case and Mixed Convection Criteria

<table>
<thead>
<tr>
<th>Authors of criterion</th>
<th>Criteria</th>
<th>Reference data</th>
<th>Conclusions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Metais and Eckert [M-1]</td>
<td>Figure 2.2; as a function of Re and GrPrD/L</td>
<td>Re = 1.75 \times 10^4, GrPrD/L = 6 \times 10^4</td>
<td>Forced Convection</td>
</tr>
<tr>
<td>Tanaka et al [T-1]</td>
<td>Figure 2.3, as a function of Re and Gr</td>
<td>Re = 1.75 \times 10^4, Gr = 3.6 \times 10^7</td>
<td>Mixed Convection</td>
</tr>
<tr>
<td>Jackson and Hall [J-2]</td>
<td>( \frac{\text{Gr}_D}{(\text{Re}^{2.7})(\text{Pr}^{0.5})} &gt; 10^{-5} )</td>
<td>( \frac{\text{Gr}_D}{(\text{Re}^{2.7})(\text{Pr}^{0.5})} = 2.5 \times 10^{-4} )</td>
<td>Mixed Convection</td>
</tr>
<tr>
<td>Polyakov [P-5] based on</td>
<td>( \text{Gr}<em>q'' &gt; \frac{1.3 \times 10^{-4} \text{Re}^{2.75} \text{Pr}^{1/8 + 2.4(Pr^{2/3} - 1)}}{\log</em>{10} \text{Re} + 1.15 \log_{10}(5 \text{Pr} + 1) + 0.5 \text{Pr} - 1.8} )</td>
<td>( \text{Gr}<em>q'' = 3.53 ) to 3.66 \times 10^7; ( 1.3 \times 10^{-4} \text{Re}^{2.75} \text{Pr}^{1/8 + 2.4(Pr^{2/3} - 1)} ) ( \log</em>{10} \text{Re} + 1.15 \log_{10}(5 \text{Pr} + 1) + 0.5 \text{Pr} - 1.8 ) = 3.5 \times 10^7</td>
<td>Mixed Convection</td>
</tr>
<tr>
<td>Nusselt number deviation</td>
<td>( \text{Gr}_q'' &gt; 2 \times 10^{-4} \text{Pr Re}^{2.75} )</td>
<td>( \text{Gr}_q'' = 3.5 \times 10^7; ) 2 \times 10^{-4} \text{Pr Re}^{2.75} = 6.6 \times 10^7</td>
<td>Forced Convection</td>
</tr>
<tr>
<td>Polyakov [P-5] based on</td>
<td>( \text{Gr}_D = 0.7 \times 10^{-4} )</td>
<td>( \text{Gr}_D = 0.76 \times 10^{-4} )</td>
<td>Mixed Convection</td>
</tr>
<tr>
<td>friction factor deviation</td>
<td>( \frac{\text{Gr}_D}{\text{Re}^{2.7}} = 0.05 )</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Hall [H-1]</td>
<td>( \frac{\text{Gr}_D}{\text{Re}^{1.8}} = 0.05 )</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Byrne and Ejiogu [B-3]</td>
<td>( \frac{\text{Gr}_D}{\text{Re}^{1.8}} = 0.8 )</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>


reference case is in a mixed convection condition, although the extent differs: some predict it is just inside the border, others predict it is well into mixed convection.

As mentioned in the previous section, the reference case is an optimistic one in terms of the onset of mixed convection. In addition to the reasons given in the previous section, some other considerations will also support this "optimistic" designation. One such point is the fact that the real heat flux imposed on the riser wall is not uniformly distributed, as assumed in the reference case. Rather, it peaks at the mid-height position. This will make the local buoyancy effect higher than that obtained from the reference case.

It should be pointed out that, as analyzed, larger buoyancy effects, hence mixed convection, may possibly occur after the maximum flow rate and decay power values have been passed. However, the resulting decrease in Nusselt number and the increase in friction factor at that time, if any, may not impose any danger to the reactor core and reactor vessel because the maximum vessel and core temperatures have also been passed. Therefore, from the reactor safety point of view, the reference case is of most concern; while from the mixed convection point of view, RCCS operation under the maximum buoyancy effect is of most interest.

The above comparisons and discussions indicate that the reference case, as well as the potential RCCS operating regime, are in mixed convection. How deep the RCCS moves into the mixed convection regime can not be known until the friction factor and the heat transfer coefficient are known. Since the heat transfer and friction factor depend on mixed convection conditions, the mixed convection condition and the heat transfer coefficient and the friction factor can only be obtained from an iterative process.

3.4 Nusselt Number and Friction Factor Predictions for Risers under Their Possible Operating Conditions

Since the RCCS riser operation regimes are predicted to be in mixed convection, it is important to know the heat transfer coefficient (Nusselt number) and the friction factor
under such operating conditions. In considering the design requirements, some questions arise. First, is the current design near the minimum heat transfer point, and what is the minimum heat transfer coefficient? Second, if the current design is not operating at the minimum heat transfer rate, can the operating condition move into the worst heat transfer region, and what would cause the operating condition to shift to this minimum heat transfer region? Third, the same questions can be asked for the friction factor. What is the friction factor for the current design, what is the worst case, and how could the worst scenario be encountered? Finally, how does the combination of the heat transfer coefficient and friction factor affect RCCS operation. The last question will be addressed in Chapter 7. To answer the first two questions, an estimation of the Nusselt number and the friction factor over the entire RCCS operating range is helpful.

3.4.1 Nusselt Number

Nusselt number predictions under possible RCCS operating conditions can be made using existing theories and correlations. In particular, the Jackson correlation, Eq.(2.30), and the k–ɛ turbulence model will be used for the estimations. The reference case will be calculated first. A potential worst case estimate will be made following that.

Analyzing the Jackson correlation, Eq.(2.30), one finds that the dividing boundary between type I flow and type II flow is at about Bo = 0.265, which is also the point where the minimum heat transfer occurs. In type I flow, the expression

\[
\frac{\text{Nu}}{\text{Nu}_0} = \left( 1 - \frac{\text{Bo}}{\left( \frac{\text{Nu}}{\text{Nu}_0} \right)^2} \right)^{0.46}
\]

for Bo < 0.265

should be used to make the calculation. In type II flow,
\[ \frac{\text{Nu}}{\text{Nu}_0} = \left( 1 - \frac{\text{Bo}}{\left( \frac{\text{Nu}}{\text{Nu}_0} \right)^2} \right)^{0.46} \quad \text{for } \text{Bo} > 0.265 \quad (2.30) \]

should be used, since the term inside the brackets on the right side of the equation is a negative number. Substituting the reference case parameters from Tables 3.1 and 3.2 into the Jackson correlation, this operating condition exhibits only a 1% heat transfer coefficient decrease. To achieve the minimum heat transfer rate, which is about a 50% decrease of the Nusselt number according to the Jackson correlation, the reference case Bo number has to increase 25 times. To have a 10% decrease of heat transfer, it has to increase by a factor of 5. If the increase of Bo comes from Grashof number changes, the same order of Gr change is required. However, if the increase of Bo comes from a Reynolds number decrease, only a 60% to 55% decrease in Re is required.

For the cases of current interest the Gr will not change significantly since the decay power is fixed at a given time, and varies only slowly over a long period. The change of Gr as the air temperature changes is also not significant. However, the air flow rate can be reduced significantly by an increase in flow resistance. This will greatly increase the buoyancy effect, and has the potential of shifting the operating condition toward the vicinity of the minimum Nusselt number. The increase in flow resistance can be due to underestimation of the friction factor by the correlation being used, surface roughness (the reference case uses a smooth tube turbulent forced convection friction factor correlation), and the potential that channels or ducts are partially or entirely blocked. A changes in design parameters, such as flow path, is another potential to cause the RCCS to operate in the minimum heat transfer region. A three times increase of the friction factor will result in a more than 40% flow rate decrease and a 118 °F air exit temperature increase. The combined effect will result in a 55% decrease of the Reynolds number. A forced convection friction factor correlation was used in the above estimation. In reality, when the buoyancy effect increases, it will further trigger an increase in the friction factor, until a
new steady state is reached where the system flow resistance will be more than tripled. What this value is, depends on the particular conditions. Since the buoyancy effect is usually proportional to $Re^{-n}$, where $n \geq 1$, this effect itself will suffice to shift RCCS operation toward the minimum heat transfer point.

3.4.2 Friction Factor

With little experimental data and limited theoretical interpretation of friction factor behavior in the mixed convection region, it is difficult to make a confident prediction. The experimental data by Polyakov and Strigin [P-6] cover a large range in terms of $Gr$ and $Re$. However, the Prandtl number ranges from 2 to 6, which excludes application to air. The Carr et al experimental results [C-1] are for air. However, its range is too narrow to apply to the present case. The Petukhov [P-1] correlation is the only one applicable to the present case, although it is still open to question. The k-ε turbulent model is another potential candidate for this purpose.

Substituting the reference case data into the Petukhov correlation,

$$f = \left[ \frac{1 + 0.83e^2}{1.82 \log_{10}(Re/8) + 0.076e^{2E^{-0.25}}} \right]^2$$ (2.34)

where

$$e = \frac{Gr^{0.001 \cdot 10^3}}{Pr \cdot Re^{2.75}}$$

$$E = \frac{Gr^{0.001}}{Pr \cdot Re^4}$$

one finds that the friction factor in the reference case has a 4% decrease compared to the forced convection friction factor calculated using the following correlation.
\[ f_{iso} = [1.82 \log_{10}(Re/8)]^2 \]  

(2.36)

CONDOR calculation results which are based on the k-ε model show a friction factor increase of only a few percent compared with forced convection results. Even with a 50% decrease in flow rate, the Petukhov correlation shows a 30% decrease in friction factor, while CONDOR predicts a small increase. The conflict between Petukhov results and CONDOR results makes predictions of friction factor behavior even more problematic. To be conservative, a maximum factor of 3 increase in the friction factor will be used for assessment purposes.

3.5 Overall Heat Transfer of the Riser

The overall heat transfer of the riser can be solved for, using the mass, momentum and energy equations.

1) mass equation

\[ \rho v A_f = \text{constant.} \]  

(3.3)

2) momentum equation (pressure balance between the buoyancy head \( \Delta p_b \) and the friction loss \( \Delta p_f \), and the acceleration loss \( \Delta p_a \): \( \Delta p_b = \Delta p_f + \Delta p_a \))

\[ \int_0^L g\beta (\rho_0 - \rho(z)) u_0 dz + g\beta (\rho_0 - \rho_e) h_c = \sum_{i=1}^n \frac{L_i G_i^2}{\rho_0 D_e^2} + K \frac{G^2}{2pD_e} \]  

(3.4)

where the acceleration loss has been neglected, since it is relatively small compared with other terms.

3) energy equation

\[ \dot{m} c_p [T_a(z) - T_{in}] = \int_0^L P_h q''(z) \, dz \]  

(3.5)
At the riser exit, \( z = H_r \), the energy equation becomes

\[
\dot{m}c_p[T_e - T_{in}] = \dot{Q}
\]  
(3.6)

In the above equations,

- \( g \) = acceleration due to gravity, m/s\(^2\)
- \( \beta \) = Thermal expansion coefficient, 1/K
- \( \rho \) = air density, kg/m\(^3\)
- \( H_r, H_c \) = riser and chimney heights, respectively, m
- \( G \) = air mass flow velocity, kg/m\(^2\)s
- \( f \) = friction factor, dimensionless
- \( T_a \) = air bulk temperature, K
- \( \dot{m} \) = air mass flow rate, kg/s
- \( c_p \) = air heat capacity at constant pressure, J/kg
- \( q'' \) = heat flux at the riser wall, W/m\(^2\)

The subscripts "e" and "in" refer to the riser exit and inlet conditions.

To solve the above equations, some constitutive correlations must be provided: air properties as a function of air pressures and temperatures, a friction factor correlation and a heat transfer coefficient correlation. Since the operating pressure of the RCCS is atmospheric, and does not change very much, it can be assumed that a constant pressure condition exists. Then, air properties are only a function of temperature, and can be obtained from air property tables. In the current calculations, air properties given by the National Bureau of Standard [H-3] are used. The heat transfer coefficient and friction factor are not known, yet. The selection of suitable correlations will be discussed in the next two chapters. With all constitutive correlations needed available, the above equation set can be solved by assuming a flow rate first; this flow rate will be used to solve the energy equation. After obtaining the temperature fields from the energy equation, the
momentum equation will be solve for a new flow rate. If the new flow rate does not match
the pre-assumed flow rate, this new flow rate will be used as the up-dated pre-assumed
flow rate to solve the energy equation again, to obtain a new temperature field. The
iteration continues until the new flow rate obtained, and the pre-assumed flow rate agree
within a certain error. The detailed solution process will be presented in Chapter 7,
together with that for the radiation heat transfer from the reactor vessel to the riser outer
surface.

3.6 Chapter Summary

In this chapter, the possible range of RCCS operating conditions has been defined
and compared with existing mixed convection criteria. The results show that even the
reference case, which is considered to have the least buoyancy effect, is inside the mixed
convection regime. However, the reference case has a small mixed convection effect,
because the Nusselt number only has a 4% decrease compared to the forced convection
value.

One potential way to shift the designated operating condition to a minimum heat
transfer coefficient scenario is to decrease the flow rate by increasing flow resistance. A
friction factor three times larger than the currently used in the reference case calculation will
suffice to make the shift.

The Nusselt Number prediction is made using the Jackson correlation. A maximum
decrease in Nusselt number of 50% in the worst case is predicted by the correlation.

To predict the friction factor is not easy because few theories and experimental
results exist. Moreover the validity of these theories and experimental results are somewhat
controversial. Therefore, a friction factor of 3 times higher than the corresponding forced
convection value is employed.

Finally, to calculate the heat transport inside the riser, mass, momentum and energy
equations, along with some constitutive equations, need to be solved. Since the
constitutive equations include the friction factor and heat transfer coefficient, an iteration is required.
CHAPTER 4

MHTGR RISER MIXED CONVECTION EXPERIMENT

4.1 Introduction

The MHTGR Reactor Cavity Cooling System (RCCS) is designed as a passive safety system. The important feature of this system is that heat rejection to ambient air happens naturally under accident conditions (under normal operation a small amount of heat will also be lost through the RCCS). This system is not subject to the types of failure modes normally associated with active cooling systems. However, in the current design of the MHTGR RCCS, because of the added driving force produced by a chimney, it is predicted that mixed convection air flow (an intermediate situation between pure natural and forced convection) will be generated inside the riser tube. This flow regime is not yet well understood. Since the risers account for a large fraction of RCCS pressure drop, one quarter of the buoyancy head and all heat removal, the heat transfer coefficient and friction factor in the RCCS operating regime are key parameters. In view of the lack of completely satisfactory correlations for the heat transfer coefficient and friction factor of air undergoing mixed convection in the heat transfer literature, an experiment was undertaken to obtain heat transfer and friction factor data in, but not only limited to, the RCCS operational regime.

The experimental data will be useful in support of RCCS design, and also in the development of more general heat transfer coefficient and friction factor correlations in the mixed convection regime.

4.2 Experimental Apparatus

For investigations into the effect of turbulent mixed convection on the heat transfer coefficient and pressure drop, a small scale apparatus will not suffice in the case of a gas at
atmospheric pressure, because of its low density, small pressure drop, and difficult-to-cover large experimental range. However, a large scale experiment will be subject to laboratory restrictions, such as space and power supplies, and cost. Therefore, the design of an experiment is a compromise among the need to represent key features of the real system, the desire to create a readily interpretable experiment, and practical restrictions on cost and size. Thus, as shown in Figure 4.1, the "riser" in the present investigation is a 2.5 inch ID commercial seamless cylindrical carbon steel pipe (instead of a rectangular slot, as in the present MHTGR design). It has a 28.25 foot (135.6L/D) heated section, which is heated uniformly by helically-wound wrap-around electrical heating tapes at a pitch of 2.75", and a 5.5 foot (26L/D) inlet unheated section incorporating a flow straightener, see Figure 4.1. The total height is approximately half that of a real riser. The overall apparatus is a once-through system with ambient air as the flow medium. The pipe is insulated by an inner layer of 0.5 in. thick fiber glass insulation, a middle layer of 2.5 inch thick thermo-12 calcium silicate insulation, and an outer layer of 1.0 inch thick fiber glass insulation. The insulation is covered by a PVC pipe to prevent rain damage, since the pipe is located out-of-doors. The heater power is provided by a 208V single phase power line. A variac is used to control the input heating power in the range of 0 - 240V and 0 - 17A, hence 0 - 4 kW. Air flow is provided by a centrifugal blower. The flow rate is controlled and adjusted using an in-line control valve.

The directly measured parameters are air flow rate, total pressure drop, heating power (from current and voltage) and temperatures. The air flow rate is measured using an ASME standard orifice [L-1]. A pitot-tube is used as a secondary monitor. The total pressure drop is measured between two wall taps using a commercial differential pressure transducer manufactured by MKS Instrument, Inc. based on the measurement of electric capacity. This pressure transducer has a full scale range of 1 mmHg (133 Pa); the minimum measurable differential pressure drop depends upon the system stability (see Section 4.4). One of the pressure taps is located at the beginning of the heated section and
10 rope heaters: wound helically in opposite direction; at a coil pitch of 2.75 inches

2.5" thick calcium silicate and 1.5" fiber glass insulation covered with PVC pipe

HP-3852A Data Acquisition System

Pressure transducer

wall thermocouples every two feet at pipe outer surface

riser pipe, 2.5" ID

flow straighteners

flow control valve

ASME orifice

Figure 4.1 Schematic of Riser Mockup (Not to Scale)
the other is 6" from the end of the heated section. The heater power measurement is simple: an ammeter and voltmeter pair gives \( P = I \times V \). However, the actual power input to the flowing air is the difference between total input power and heat loss through the insulation. This total heat loss has been determined by running the experiment with the entrance and exit blocked and insulated. See Section 4.6 for a detailed discussion of this determination.

There are 14 T-type thermocouples imbedded in the outer wall of the pipe to monitor the wall axial temperature profile. The glass insulation on the thermocouples restricts the maximum wall temperature to 400 °C. The thermocouples start where heating begins; successive thermocouples are positioned every two feet thereafter, each rotated 90° circumferentially from the preceding one.

The temperatures and pressure drop are collected using a computerized data acquisition system, the Hewlett Packard HP-3852A.

Finally, a temperature trip emergency shut off device is placed on the heater power line to prevent overheating. The signal is taken from thermocouple No. 3, and the trip level set to keep the temperature at this location below 325 °C.

4.3 Range of the Experiment

The current experiment is subject to certain limitations. The range that can be covered is shown in Table 4.1 and Figure 4.2. Figure 4.2 also shows the approximate RCCS operating regime. The determination of this operating regime has been described in Chapter 3. The extreme of the operating regime of RCCS system can be characterized by an inlet Reynolds of 2.2x10^4 and a riser Grashof number, \( Gr_q \), of 5x10^7. Note that the minimum Grashof number, corresponding to the minimum heat flux, in Table 4.1 and Figure 4.2 is zero, which is different from that in Chapter 3, where it arbitrarily gives the value at 500 hours after the reactor has been shut down. The boundaries of the experiment operating regime are as follows: Reynolds number from 5,000 to 33,000, pipe wall
temperature up to 400 °C, and Grq" up to a maximum value of 2.5×10^8, which is corresponding to a maximum riser heat flux of 2.4 kW/m², or a maximum total heat input 4 kW, about half of the value a riser received. Since the test section is about half of the riser height this allows the experiment achieve at least the same air temperature rise, about 140 °C, as the design case experiences, so that air property changes effects will be reflected in the test results. As can be seen from Figure 4.2, the experimental Re and Grq" range covers the full range of design interest, and extends into the mixed convection regime to study conditions which lead to maximum heat transfer impairment.

In Table 4.1, a lower limit on the inlet Reynolds number, 5000, is given. Below this lower limit the measured air flow rate will be subject to a large error because of the accuracy with which the pressure drop across the ASME orifice can be measured. The maximum Reynolds number limit for this experiment is set by the maximum Δp between the two pressure taps which can be measured by the pressure transducer (This limit can be extended by using other available Δp measurement devices, such as a slant tube manometer. However, the maximum Reynolds number obtainable using the transducer already exceeds the maximum possible RCCS operational Reynolds number). The system total pressure drop, Δp_t, is determined using a commercial transducer, which can measure

<table>
<thead>
<tr>
<th>Table 4.1</th>
<th>Range of The MHTGR RCCS Riser Experiment</th>
</tr>
</thead>
<tbody>
<tr>
<td>Re_{in}</td>
<td>5,000 to 33,000</td>
</tr>
<tr>
<td>Q</td>
<td>0 to 4 kW</td>
</tr>
<tr>
<td>q&quot;</td>
<td>0 to 2.4 kW/m²</td>
</tr>
<tr>
<td>Δp_t</td>
<td>1 to 133 Pa</td>
</tr>
<tr>
<td>(T_w)_{max}</td>
<td>400 °C</td>
</tr>
<tr>
<td>Grq&quot;</td>
<td>0 to 2.5×10^8</td>
</tr>
</tbody>
</table>
Figure 4.2 Experiment Range Compared to MHTGR Operating Regime
from a few Pascals to a maximum of 133 Pascals. The maximum power limitation is due to the available power line capacity. Finally, the maximum wall temperature limitation is due to the type of thermocouples employed (T-type with glass insulation) in the current mockup design. In Figure 4.2 a family of curves at different Bo (Buoyancy) number is plotted, where Bo is defined in Eq.(2.9) as

$$Bo \equiv 8 \times 10^4 \frac{Gr_q^n}{Re^{3.425} Pr^{0.8}}$$

(2.9)

The Bo number is used as an indicator of the extent of mixed convection. When Bo<0.01 it is considered that the flow is in the fully forced convection regime. When Bo>0.1 mixed convection will have a non-negligible effect on both the heat transfer coefficient and the total pressure drop. The key limitation on the experimental range is the maximum wall temperature, 400 °C, which is constrained by the insulation of the thermocouples. This wall temperature limitation will restrict the wall maximum heat flux, which in turn limits the $Gr_q^n$ to a maximum value of about $2.5 \times 10^8$. The details of the calculation of these Bo number lines and $T_w$ limits can be found in Appendix B.

4.4 Procedure For the Experiment

The experimental runs are rather simple to conduct. Heater power is adjusted to the level designated using a variac, and air flow is adjusted using the in-line valve. The wall temperatures and the pressure drops are monitored at an IBM-XT computer using self-programmed software through the data acquisition system HP-3852A. The steady state situation is considered adequately approached when wall temperature drift with time is less than 1 °C per 30 minutes. This condition generally takes 10 hours to achieve from the time experimental conditions are reset to new levels from an existing steady state condition.
4.5 Instrument Calibration

The instruments used in the experiments and relevant information about them are shown in Table 4.2. All the instruments used in the experiment were calibrated. The ammeter, voltmeter, HP-3852A data acquisition system and its 5-1/2 digit voltmeter were calibrated against National Institute of Standard and Technology (NIST) standards by certified laboratories. The pressure transducer was calibrated, both by the manufacturer, and by the author using a commercial slant tube manometer. For flow measurement an ASME orifice having a diameter of 1-3/64" was fabricated according to the relevant ASME standard [L-1].

The input power measurement is determined from total power measured with an ammeter and a voltmeter, minus the computed heat loss through the insulation at the measured wall temperature (see Section 4.6). The two meters were calibrated against NIST standards by Calibrion Instrument, Inc. located at 220 Grove Street, Waltham, Mass 02154. The certification of the calibration is in the project's files (report No. C34831).

The HP-3852A data acquisition system was calibrated by General Electric Co., Electronic Services located at 215 Salem Street, Woburn, MA 01801. As mentioned earlier, two items were calibrated — the HP-3852A data acquisition system (main frame, series No. A633137) and the HP-44701A 5-1/2 digit voltmeter (series No. 2851A06004). Both items have been calibrated against NIST standards. Both instruments were within their manufactures' specified accuracy/uncertainty, which are far beyond the requirements of this experiment.

The pressure drop measurement was made using a commercial differential pressure transducer based on the electric capacity principle. It was calibrated against a commercial slant tube manometer (STM) purchased from Mariam Instrument located at 10920 Madison Avenue, Cleveland, Ohio 44102. The transducer has a full range of 1 mmHg (133.32 Pa). The slant tube manometer has a full range of 1 in.H2O (249 Pa). The transducer and the slant tube manometer are connected in parallel, as shown in Figure 4.3. The calibration
<table>
<thead>
<tr>
<th>Name of Instruments</th>
<th>Maker</th>
<th>Model Number</th>
<th>Calibration Agency</th>
<th>Dealer</th>
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<tr>
<td>Pressure Transducer</td>
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<td>398H</td>
<td>1) MKS Instrument, Inc. 2) MIT Using Slant Tube Manometer</td>
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<td>40AM10-E111</td>
<td>NA</td>
<td>Bordwieck Engineering Sales Co., Inc</td>
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<tr>
<td>Data Acquisition Unit</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Main frame</td>
<td>Hewlett Packard</td>
<td>HP3852A</td>
<td>GE Electric Service</td>
<td>Hewlett Packard</td>
</tr>
<tr>
<td>Digital voltmeter</td>
<td>Hewlett Packard</td>
<td>HP-44701</td>
<td>GE Electric Service</td>
<td>Hewlett Packard</td>
</tr>
<tr>
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<td>HP-44720</td>
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<td>Hewlett Packard</td>
</tr>
<tr>
<td>Temperature Control Unit</td>
<td>Omega Technologies</td>
<td>CN-371-KC2</td>
<td>NA</td>
<td>Omega Technologies</td>
</tr>
<tr>
<td>Pitot Tube</td>
<td>Meriam Instruments and Accessories</td>
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</tr>
<tr>
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<td>Variac</td>
<td>General Radio Company</td>
<td>W50HM</td>
<td>NA</td>
<td>General Radio Company</td>
</tr>
</tbody>
</table>

Note: NA = Not Available
Figure 4.3  Schematic of Transducer and Manometer Arrangement Used to Calibrate the Transducer
procedure is as follows: a $\Delta p$ is applied as shown in Fig. 4.3, which will produce the same pressure drop across the slant tube and the transducer. The two readings, from the slant tube and the transducer, are then compared and plotted as shown in Figure 4.4. The slant tube readings from the indicating scale are considered to be accurate. The maximum error of the STM is 0.005 inch H$_2$O (1.25 Pa), which is half of the minimum scale interval. Because of electronic fluctuations, the $\Delta p$ values from the transducer are integrated using the HP-3852A data acquisition system with an integrating time of approximately 0.25 seconds. A large number of these readings are collected for each measurement (in Figure 4.4 there are 300 readings at each point). The average of these readings is taken as the measured $\Delta p$, and the square root of their variation ($\sigma^2$) as the "error band". As can be seen from Figure 4.4, the transducer is very accurate in the range of 12.5 to 133 Pa. However, in the range from 0 to 12.5 Pa there is no appropriate manometer available to calibrate the transducer. Therefore, linearity is assumed for the transducer over its full range. From its linearity in the range between 12.5 and 133 Pa, its good agreement with the manometer over that range, and the fact that the correlation line extrapolates back through (0,0), we can infer that in the range 0 to 12.5 Pa the transducer will also respond linearly and accurately.

The thermocouples used in the measurement were calibrated against boiling water. It was found that the maximum discrepancy between these thermocouples in measuring the temperature of boiling water is less than 0.4 °C.

4.6 Heat Loss Measurement

The heat generated by the heater does not all pass through the riser pipe and heat the air inside it. Part is lost through the insulation, and a small fraction is lost through conduction at the end of the pipe. To obtain good experimental results for both the Nu number and the friction factor, the axial variation of this heat loss must be determined. A measurement was therefore undertaken by inputting energy to the pipe with the inlet and
Figure 4.4 Pressure Transducer Calibration Results
exit blocked using insulation. When the steady state was reached, the riser wall and the PVC pipe wall temperatures were then measured. The wall temperatures are fairly uniform axially, Figure 4.5, which indicates that there is no substantial internal convection. Since there is no flow through the inlet and the outlet of the pipe, the heat input is considered to be lost through the insulation. The measured heat loss versus wall temperature differences are shown in Figure 4.6. As can be seen from the figure, heat loss versus temperature difference is close to linear. A simple linear curve based on a least square fit can be used to represent this heat loss characteristic:

\[ Q_{\text{loss}} = a + b \Delta T \quad (W) \]  

(4.1)

where it is found that

\[ a = -87.84 \]

\[ b = 3.35 \]

or

\[ q''_{\text{loss}} = \frac{1}{\pi DL} (a + b \Delta T) \]

\[ = -51.59 + 1.97 \Delta T(\text{oC}), \quad (W/m^2) \]  

(4.2)

As shown in Figure 4.5, the axial wall temperature distribution is fairly uniform except at the inlet and exit ends, where axial conduction is important. The axial conduction effect can be estimated using Fourier's equation

\[ Q = A k \Delta T/\Delta z \]  

(4.3)

In the present case, the pipe wall cross-section area is 0.0044 m², and a typical value of thermal conductivity for carbon steel is 45 W/(m.°C). This will give
Figure 4.5 Wall Temperature Profile in Heat Loss Measurement (Q=735 W)
Figure 4.6 Heat Loss Measurement Points and Correlation Line
\[ Q = 0.198 \left( \frac{\Delta T}{\Delta z} \right), \quad (W) \]

In the heat loss measurement data at an input power equal to 924 W, the first thermocouple reading, which is 2 ft above the point at which heating starts, is 269.3 °C. This gives a total heat loss by axial heat conduction of 78 W under the assumption that the temperature of the environment is 30 °C. This heat loss is less than 10% of the total heat loss through the insulation. At the exit, the temperatures at thermocouple No. 13 and No. 14 are 343.3 and 312.1 °C, respectively. Since they are located 2 ft apart, the axial heat conduction is only about 10 W, which is negligible compared with the total heat input of 924 W.

Although total axial heat conduction at the entrance and exit ends is about 10% of the total heat loss in this zero-flow experiment, in an actual run much more power is needed to reach this wall temperature. Furthermore, some of the axial-conducted energy at the inlet can be transferred to the incoming air below the nominal start of the heated section. Therefore, the heat loss by axial heat conduction at the entrance and exit will only account for a very small fraction of the total power. Therefore, neglect of these heat losses will lead to a negligible effect in the data reduction procedure.

4.7 Experimental Data Reduction Procedure

The major parameters we are seeking are the friction factor and the Nusselt number. Since the measured parameters are the air flow rate, the wall and inlet air temperatures and the total pressure drop, the friction factor and Nusselt number must be calculated from these measured parameters.

4.7.1 Friction Factor

As discussed in Chapter 2, there are two ways to obtain the friction factor through experiments. One is to measure the wall shear stress. The other is to measure the friction
pressure drop. The current experiment is the latter case, since the total pressure drop across the pipe was measured. Defining the channel average friction factor as

\[ \bar{f}_1 = \frac{\int_0^L \frac{f_0}{\rho} dz}{\int_0^L \frac{dz}{\rho}} \]  

(4.4)

from the relation

\[ \Delta p_f = \int_0^L \frac{fG^2}{2D\rho} dz \]  

(4.5)

one has

\[ \bar{f}_1 = \frac{2D\Delta p_f}{G^2 \int_0^L \frac{dz}{\rho}} \]  

(4.6)

Since the total pressure \( \Delta p_t \) was measured instead of \( \Delta p_f \), Eq.(4.6) becomes

\[ \bar{f}_1 = \frac{2D(\Delta p_t + \Delta p_b - \Delta p_a)}{G^2 \int_0^L \frac{dz}{\rho}} \]  

(4.7)

where

\( \Delta p_t = \) measured total pressure drop, Pa

\( \Delta p_b = \) computed buoyancy pressure gain, Pa
\[ \Delta p_a = \text{computed acceleration pressure drop, Pa} \]

\[ D = \text{hydraulic diameter, m} \]

\[ L = \text{length of heated region over which } \Delta p_t \text{ is measured, m} \]

\[ G = \text{air mass flow velocity, kg/(s m^2)} \]

\[ \rho = \text{air density, a function of air temperature, kg/m}^3 \]

\[ \bar{f} = \text{channel average friction factor obtained from the experiment} \]

The subscript 1 is to distinguish the above usage from other friction factor definitions which will be given later.

The buoyancy head \( \Delta p_b \) can be evaluated as follows:

\[ \Delta p_b = \int_{0}^{L} [\rho_{in}(T_{in}) - \rho_{a}(T_{a})] \ g \ dz \]  \hspace{1cm} (4.8)

where \( T_{in} \) and \( T_{a} \) are the ambient air temperature and air stream temperature, respectively.

The air density, \( \rho \), can be expressed, by treating air as a perfect gas, as:

\[ \rho = \frac{353.12}{T(K)} \text{, kg/m}^3 \]  \hspace{1cm} (4.9)

The acceleration pressure drop is calculated as follows

\[ \Delta p_{acc} = G^2 \left[ \frac{1}{\rho_e(T_e)} - \frac{1}{\rho_{in}(T_{in})} \right] \]  \hspace{1cm} (4.10)

where \( G \) is the mass flow velocity, kg/(m\(^2\)-sec), and subscripts in and e indicate inlet and exit, respectively.

In Eq.(4.4) the definition of the friction factor involves integrals. The fluid density as a function of location is not known before the flow rate is calculated. However, the
flow rate depends on the friction factor. This makes the definition in Eq.(4.4) difficult to use in engineering applications.

In engineering applications, the friction pressure drop is calculated as follows,

\[ \Delta p_f = \bar{f} \frac{L}{D} \frac{G^2}{2\rho^*} \]  
(4.11)

Therefore, a channel average friction factor can be defined as

\[ \bar{f}_2 = \frac{2\rho^* D \Delta p_f}{LG^2} \]  
(4.12)

or

\[ \bar{f}_2 = \frac{2\rho^* D (\Delta p_t + \Delta p_b - \Delta p_a)}{LG^2} \]  
(4.13)

in terms of the present experiment's measured parameters, where \( \rho^* \) is a reference fluid density. The rest of the parameters are the same as in Eq.(4.7). Note that \( \bar{f}_2 \) is very similar to the apparent friction factor in an isothermal flow. And, also, if we let \( \frac{L}{\rho^*} = \int_0^L \frac{dz}{\rho} \), then \( \bar{f}_1 = \bar{f}_2 \). Which density should be used as the reference density in Eq.(4.13) will be discussed later, when the experimental results are presented, and in Chapter 5 where the experimental results are compared with theory.

For the case of isothermal flow, the above defined friction factors will be equal to the isothermal friction factor, such as that calculated from the Petukhov correlation, Eq.(2.36), or from the Blasius correlation

\[ f_{ISO} = 0.316 \text{ Re}^{-0.25} \]  
(4.14)
4.7.2 Nusselt Number

The Nusselt number is also not a directly measured parameter. It must be calculated from the measured parameters, such as mass flow rate, wall temperature and heat input. Since the wall heat flux can be found from the difference between the total heat input and the heat loss through the insulation, the local Nusselt number can be obtained easily as

\[ \text{Nu} = \frac{(q''_\text{nom} - q''_\text{loss}) \ D}{k \ (T_w - T_a)} \] (4.15)

where \( q''_\text{nom} \) is the nominal wall heat flux calculated from heat input. \( q''_\text{loss} \) is the heat flux loss through the insulation. \( D \) is the riser tube diameter. \( k \) is the air thermal conductivity evaluated at local bulk temperature. \( T_w \) is the measured wall temperature. The air stream bulk temperature, \( T_a \), is calculated from the energy balance as follows:

\[ T_a(z) = \frac{1}{\dot{m}c_p} \int_0^z (q''_\text{nom} - q''_\text{loss}) \ dz + T_{in} \] (4.16)

where

- \( T_{in} \) is the air inlet temperature, °C.
- \( c_p \) is air heat capacity, taken to be constant, 1018 J/kg
- \( \dot{m} \) is the air mass flow rate, kg/sec.
- \( z \) is the channel height measured from the start of heating, m.

This Nu number will be compared with the forced convection Nu number calculated using the Dittus-Boelter correlation

\[ \text{Nu} = 0.023 \ (\text{Re})^{0.8}(\text{Pr})^{0.4} \] (4.17)
where Re will be evaluated at the local air bulk temperature and Pr is taken as constant, 0.7, since it is virtually unchanged in the temperature range of the experiment.

4.8 Error Analysis

The results, for both Nu and f, obtained from the experiment are subject to certain errors. These errors are associated with the uncertainties in the measured parameters: in the present experiment principally from the uncertainties in the temperature, pressure drop, and heat flux measurements. The errors also stem from the process of finding the indirectly measured parameters, such as air stream temperatures, using the measured parameters. In the following error analysis it is assumed that the dimensional measurements are error free except for the orifice diameter. In this section the error analysis principles and the uncertainties in Nu and f will be presented. The detailed derivations can be found in Appendix D.

The principle for doing error analysis is simple. For any function

\[ y = f(x_1, x_2, ..., x_i, ..., x_n) \]  \hspace{1cm} (4.18)

the uncertainty in \( y \), \( E_y \), can be expressed as \([B-1]\)

\[ E_y^2 = \sum_{i=1}^{n} \left( \frac{\partial f}{\partial x_i} \right)^2 E_{x_i}^2 \]  \hspace{1cm} (4.19)

where \( x_i, i=1, 2, ..., n \), are the directly measured parameters, \( y \) is the derived result from directly measured parameters, and \( E \) represents the uncertainty.

In our experiment, two parameters, Nu and f, are derived from measured parameters. For the Nu calculations from the measured parameters, Eq.(4.15) is used. Applying Eq.(4.19) to Eq.(4.15), one has
\[
\left(\frac{\Delta N_u}{\text{Nu}}\right)^2 = \left(\frac{\Delta k_a}{k_a}\right)^2 + \left(\frac{\Delta q''_{\text{nom}}}{q''_{\text{nom}}} + \frac{\Delta q''_{\text{loss}}}{q''_{\text{loss}}}\right) + \frac{(\Delta T_w)^2 + (\Delta T_a)^2}{(T_w - T_a)^2}
\]  

(4.20)

where the first term on the right hand side is the uncertainty from the air thermal conductivity, which depends on the air stream temperatures. Given the air temperature range in the current experiments, the uncertainty in obtaining \(k_a\) from the property table is less than 1.5%. The second term on the right hand side is the uncertainty associated with the heat flux measurements. The uncertainties in measuring the nominal heat flux input, \(\Delta q''_{\text{nom}}\), and the heat flux loss through the insulation, \(\Delta q''_{\text{loss}}\), are as follows

\[
\Delta q''_{\text{nom}} = \left(\frac{\Delta Q_{\text{nom}}}{\pi DL}\right) = \frac{1}{\pi DL} \sqrt{(\Delta I \cdot V)^2 + (\Delta V \cdot I)^2}
\]  

(4.21)

\[
\Delta q''_{\text{loss}} = \sqrt{(\Delta q''_{\text{corr}})^2 + \frac{1}{(\pi DL)^2}[(\Delta I \cdot V)^2_{\text{loss}} + (\Delta V \cdot I)^2_{\text{loss}}]}
\]  

(4.22)

where \(I\) and \(V\) are the current and the voltage in each measurement, and \(\Delta I\) and \(\Delta V\) are the uncertainties associated with these measurements, which are half of the minimum scale on the meters. \(\Delta q''_{\text{corr}}\) is the uncertainty of using Eq.(4.2) to calculate the heat loss instead of using directly measured data in determining the heat loss. \(\Delta q''_{\text{corr}}\) is taken as the mean value of the uncertainties at each heat loss measurement point. The last term on the right hand side of Eq.(4.20) is the uncertainty from the wall and air temperature measurements. The wall temperature uncertainty, \(\Delta T_w\), is taken as 3 °C, about the maximum detected undulation in the wall temperature ripple measurements. The uncertainty of the air stream temperature at \(n\)-th node, \((\Delta T_a)_n\), is calculated from the following correlation

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\[
[(\Delta T_a)_n]^2 = [\Delta T_{in}]^2 + [(T_a)_{n}-T_{in}]^2 \left( \frac{\Delta G}{G}^2 + \frac{\Delta c_p}{c_p}^2 + \frac{n(\Delta Q_{nom})^2 + (\pi DL\Delta q^{"corr})^2 + (\Delta Q_{loss})^2}{N \sum_{i}^n (Q_{nom}-Q_{loss})_i^2} \right)
\]

where \(\Delta T_{in}\) is the uncertainty in the inlet air temperature measurement, 2 °C. The air heat capacity uncertainty, \(\frac{\Delta c_p}{c_p}\), is 1.5%, the maximum possible uncertainty in the experimental range. The uncertainties in the flow rate measurement, \(\frac{\Delta G}{G}\), have been estimated in Appendix D, and range from less than 2% for high Re number runs, to about 4% at Re equals 10,000, and finally about 7% at Re equals 7,000. The heat input term can be estimated using a similar approach as in the heat flux input case. Calculations show that the air stream temperatures have an uncertainty below 5% at each point for every run (For details see Appendix D). With these uncertainties in the air properties, heat flux and the temperatures available, the uncertainty in the Nusselt number can be obtained. The calculated results show that the Nusselt number at the exit has the largest uncertainty, about 10%, with the largest value being 14.7% in Run 920; the smallest uncertainty occurs at the channel inlet, about 5%, with the largest value being 7.9% in Run 929.

There were two kinds of friction factors measured: isothermal and non-isothermal. For the isothermal friction factor there is no buoyancy head. To simplify the analysis it is also assumed that the acceleration pressure drop is very small and can be neglected since it originates only from velocity profile changes. The isothermal friction factor, \(f_{iso}\), can be expressed as

\[
f_{iso} = \frac{\Delta p_{meas}}{L \frac{G^2}{D} \frac{2\rho}}
\]

where \(\Delta p_{meas}\) = measured pressure drop (Pa)
\( G = \) air mass velocity \((\text{kg/m}^2\text{s})\)
\( \rho = \) air density \((\text{kg/m}^3)\)
\( L = \) length over which \( \Delta p_{\text{meas}} \) is measured, m
\( D = \) hydraulic diameter, m

For mixed convection, friction factors have been given in Eqs. (4.7) and (4.13).

They are rewritten here

\[
\bar{f}_1 = \frac{2D(\Delta p_t + \Delta p_b - \Delta p_a)}{G^2 \int_0^L \frac{dz}{\rho}}
\]  \hspace{1cm} (4.7)

\[
\bar{f}_2 = \frac{2\rho^*D(\Delta p_t + \Delta p_b - \Delta p_a)}{LG^2}
\]  \hspace{1cm} (4.13)

Applying Eq (4.19) to both the isothermal and the mixed convection friction factors, one has, for the isothermal friction factor,

\[
\left( \frac{\Delta f_{\text{iso}}}{f_{\text{iso}}} \right)^2 = \left( \frac{\delta \Delta p_{\text{meas}}}{\Delta p_{\text{meas}}} \right)^2 + \left( 2\frac{\Delta G}{G} \right)^2 + \left( \frac{\Delta T_{\text{in}}}{T_{\text{in}}} \right)^2
\]  \hspace{1cm} (4.25)

where \( \Delta p_{\text{meas}} \) and \( \delta \Delta p_{\text{meas}} \) are measured pressure drop and the uncertainty associated with that measurement, respectively. The relative error from the first term depends on the total pressure drop measured. In the isothermal measurements this term is very small. The largest one is 3% at \( \text{Re} \) equals 5700. The third term on the right hand side of the equation is the uncertainty associated with the inlet air temperature measurement. \( \Delta T_{\text{in}} \) is taken as 2 \( ^\circ \text{C} \) in all the calculations. The third term has little contribution to the uncertainty of the isothermal friction factor. The largest uncertainty of the isothermal friction factor comes from the second term on the right hand side of Eq.(4.24): the uncertainty of the flow rate
measurement. At Reynolds numbers larger than 10,000 the flow rate uncertainty ranges from less than 2% to about 3.3%, which result in uncertainty in the isothermal friction factor of 3.5% to less than 7%. When the Reynolds numbers are lower than 10,000, the uncertainties in the flow rate measurement increase rapidly: 5% at Re equals 8,200 and 10% at Re equals 5,700. The resulting uncertainty of the isothermal friction factor is 11% and 25%, respectively.

For the mixed convection friction factor, only uncertainties in $\tilde{f}_1$ will be estimated here because it depends upon one more variable, $T_a$, and hence has larger uncertainty than does $\tilde{f}_2$. The correlation to calculate the uncertainty in $\tilde{f}_1$ is

$$
\left( \frac{\Delta f}{f} \right)^2 = \frac{(\delta \Delta p_t)^2 + (\delta \Delta p_b)^2 + (\delta \Delta p_{acc})^2}{(\Delta p_t + \Delta p_b - \Delta p_{acc})^2} + \left( \frac{2\Delta G}{G} \right)^2 + \frac{\sum_{i}^{N} (\Delta T_{a_i})^2}{\left[ \sum_{i}^{N} (\Delta T_{a_i})^2 \right]^2}
$$

(4.26)

The last term on the right hand side of the equation has a relatively smaller contribution than that of the other two terms. At a Reynolds number larger than 15,000 the second term is dominant, and the uncertainties of the friction factor reflect the uncertainty in the accuracy of the flow rate measurement. The uncertainties on the friction factors are lower than 6%.

When Reynolds numbers are about 10,000, the first two terms have approximately the same contributions. The resulting uncertainties are close to 10%. Upon further decrease of the Reynolds number, the uncertainty in the flow rate increases rapidly. So does the uncertainty in the pressure drop measurement, because its denominator becomes much smaller than for the high Reynolds number cases. The resulting uncertainty in the friction factor increases to about 15% at Reynolds numbers equal to 7,000.
4.9 Experimental Results

The friction factors and the Nusselt numbers are the major results sought. The wall temperatures are also of interest. In addition, the changes of the dimensionless numbers, Re, Gr and Bo, along the channel height will be presented, since their behavior is related to the local Nusselt number behavior.

4.9.1 Re, Gr and Bo numbers

The use of air as the flow medium in the experiments leads to a significant increase in temperatures over the test length. Since gas properties are much more sensitive to temperature change than are those of liquids, the property changes of the flowing air must be considered in organizing the experimental results. For example, the air density will experience a decrease of about 40% when the temperature increases from the ambient temperature, 20 °C, to 200 °C, a temperature a number of runs have reached. The air viscosity and thermal conductivity will have a 40% and 50% increase, respectively, as it experiences the same temperature rise. These property changes will affect the dimensionless numbers. Typical Re and Gr with changes in a heated channel are shown in Figs. 4.7 and 4.8. As can be seen, the Re decreases at almost a constant rate from 10,000 at channel inlet to a little below 7,000 at the channel exit, which has a temperature of 250 °C. The Gr with exhibits a rapid decrease in the first half of the channel, then decreases at a much slower rate. The channel exit Gr with is less than 4% of the channel inlet Gr with number. The changes of the Re and Gr with numbers also affect the Bo number. As shown in Fig. 4.9, the Bo number also decreases because of its minus 3.425 exponential power dependence on the Reynolds number. The rate of decrease of the Bo number is between those of the Re and Gr with numbers.

4.9.2 Wall Temperatures

Three typical wall temperature profiles are shown in Figs. 4.10, 4.11 and 4.12. (the plots start at 2 ft because the thermocouple at location zero was broken after the test section was assembled). They represent the low, middle and high inlet Bo number cases.
Figure 4.7  Re Number Variation Along Channel Axial Direction
Figure 4.8  Gr Number Variation Along Channel Axial Direction
Figure 4.9 Bo Number Variation Along Channel Axial Direction
Figure 4.10  Wall and Air Temperatures at Relatively High Re and Low Bo (Bo=22,000, Bo=0.01)
Figure 4.11  Wall and Air Temperatures at Intermediate Reynolds and Buoyancy Numbers
(Re=10,000 and Bo=0.29)

Run 916:
Inlet Re = 10,000
Inlet Gr = 1.54E8
Inlet Bo = 0.29
Heat Input = 2.97 kW
Figure 4.12  Wall and Air Temperatures at Low Reynolds Number and High Bo Number (Re=7,000 and Bo=0.5)
The air stream temperatures computed from the energy balance are also shown in the figures. As can be seen, when Bo number increases, buoyancy effects cause progressively larger temperature distortions. The wall temperature variation in Fig. 4.10 is very close to linear for the inlet Bo equal to 0.02 case. Figure 4.11 shows that the wall temperature exhibits a distortion about half way up the channel for the inlet Bo = 0.29 case. Figure 4.12 corresponds to the inlet Bo equals 0.50 case. It shows that the wall temperature distortion starts earlier than that of Figure 4.11.

4.9.3 Friction Factors

The measured isothermal friction factors (at no heat input, i.e. cold run conditions) are given in Fig. 4.13, along with the uncertainties estimated in Section 4.8. The solid line is from the correlation given by Petukhov et al [P-1].

\[ f_{iso} = (1.82 \log_{10} Re - 1.64)^{-2} \quad (2.36) \]

As can be seen, the measured friction factors are very close to the calculated friction factor from Eq.(2.36), except at Re less than 10,000, where Eq.(2.36) is no longer valid. Agreement is within the typical accuracy of such correlations. It should be noted that the consistently higher values of the measured friction factor compared to the calculated values from Eq.(2.36) (about 3 - 5%) may be attributed to flow development (i.e., "entrance") effects. However, as can be seen later, this will not affect the mixed convection friction factor results significantly.

The non-isothermal (mixed convection) friction factor measured in the experiments is shown in Fig. 4.14 as a function of inlet Bo number. In Figure 4.14a, \( \bar{f}_2 \) is plotted at \( \rho^* = \rho_{in} \), \( \rho^* = \rho_e \), and \( \rho^* = \bar{\rho} \), the density evaluated at the average bulk temperature, \( \bar{T}_b = 0.5(T_{in} + T_e) \). As can be seen from the figure, the density has a large effect on the magnitude of the computed friction factor. By plotting \( \bar{f}_2 \) evaluated at both inlet and exit
Figure 4.13  Comparison of Measured Isothermal Friction Factors and the Petukhov Correlation
Figure 4.14a  Experimental Friction Factor from Eq.(4.13) as a Function of Inlet Bo Number
Figure 4.14b  Comparison of Two Friction Factor Definitions from Eqs.(4.7) and (4.13)
density all friction factor values are enveloped. Therefore, these conventions provide upper and lower limits; \( \bar{f}_2 \), based on inlet conditions is convenient for engineering use. A friction factor based on channel average density is more appropriate, but involves an iteration process in the flow rate and heat transfer calculations, because the down-stream air temperatures are not known when the calculations start. Figure 4.14b shows the comparisons between \( \bar{f}_1 \) and \( \bar{f}_2 \) at \( \rho^* = \bar{\rho} \). They are very close to each other. Therefore, in the following analyses only \( \bar{f}_2 \) will be used. Note that \( \bar{f}_2(\bar{\rho}) \) or \( \bar{f}_1 \) are insensitive to buoyancy effects, within our experimental uncertainty; in other words, a single correlation, such as Eq.(2.36) will suffice.

4.9.4 Nusselt Numbers

In the plots which follow, the local Nusselt number results are given as functions of axial location and local Bo number. Several results, at low inlet Bo number, intermediate inlet Bo number, and high inlet Bo number are shown here. They represent different Nusselt number variation trends. The Dittus-Boelter correlation, based on the local bulk temperature, is also shown in each plot. Figure 4.15 is at high Reynolds number. Since the inlet Bo number is close to 0.01, the run is considered in the forced convection regime. The measured Nusselt number is very close to the calculated values from the Dittus-Boelter correlation. Figure 4.16 shows the Nusselt number at a Re lower than that in Fig. 4.15, but the heat inputs are about the same. Its inlet Bo number is about 0.1. This case is considered to be just inside the mixed convection regime. The results show that the measured Nusselt numbers do not exhibit an obvious decrease, yet. When the inlet Re decreases to about 10,000, the buoyancy force shows its effect. Figures 4.17 to 4.18 give Nusselt numbers for these results. They show a gradual decrease from the inlet until the middle of the channel, then a gradual recovery. At the channel exit the Nusselt number reaches or exceeds the forced convection values. For further decreases in the Re number (increasing Bo number), the Nusselt numbers stop recovering from their minimum values,
Fig. 4.15 Comparison of Measured Nu with Dittus-Boelter Correlation, for Run 101
Fig. 4.16  Comparison of Measured Nu with Dittus-Boelter Correlation, for Run 925
Fig. 4.17 Comparison of Measured Nu with Dittus-Boelter Correlation, for Run 914
Fig. 4.18 Comparison of Measured Nu with Dittus-Boelter Correlation, for Run 916
as shown in Figs. 4.19 to 4.20. The Nusselt results are flat at about 70% to 50% of the forced convection Nusselt number values, depending on how large the buoyancy force is. Figure 4.21 summarizes the buoyancy effect on the Nusselt numbers. It should be pointed out that at high Reynolds number the measured Nu numbers exhibit ripple – periodic undulations every 3 or 4 measured points. The ripples disappear at lower Re numbers. The ripples in Nusselt number are due to corresponding ripples in the wall temperature distributions. However, careful examination of the manner in which the heater coils were wrapped and the thermocouple locations does not show a direct connection with the observed ripples (see Appendix F for details). Figure 4.22 plots the Nusselt number as a function of Re number. The Dittus-Boelter correlation results and 67% and 50% of the correlation values are also shown in the figure. A value equal to 67% of the Dittus-Boelter correlation envelops all the experimental data for Reynolds number larger than 10,000. At Re lower than 10,000, the experimental Nusselt number is lower than 67% of the Dittus-Boelter value. However, 50% of the Dittus-Boelter correlation will envelope all of our experimental data.

4.10 Chapter Summary

In this chapter, the riser mockup experiment was discussed. The experiment range covers almost all possible RCCS operating conditions. The major results of the experiment are Nusselt numbers and friction factors. They are presented in this chapter as functions of Bo number and channel axial location.

For the friction factors it is shown that inferred values are sensitive to the reference temperature employed. A friction factor inferred based on channel average density is most appropriate since, as Figure 4.14b demonstrated, it closely approximates \( \tilde{f}_1 \). However, use of a channel average density involves an iteration process in the flow rate and heat transfer calculations, because the down-stream air temperatures are not known when the calculations start. Nevertheless, since designers generally can closely estimate the channel
Fig. 4.19 Comparison of Measured Nu with Dittus-Boelter Correlation, for Run 102

Run 102:
Inlet Re = 7,000
Inlet Gr = 7.03E7
Inlet Bo = 0.50
Heat Input = 0.75 kW
Fig. 4.20 Comparison of Measured Nu with Dittus-Boelter correlation, for Run 103

Run 103:
- Inlet Re = 7,100
- Inlet Gr = 1.30e8
- Inlet Bo = 0.87
- Heat Input = 1.32 kW
Fig. 4.21 Measured Nusselt Number as a Function of Inlet Bo Number
Figure 4.22  Comparison of Present Experimental Results with Dittus-Boelter Correlation
average temperature, use of $\bar{T}_2$ with $\rho^*$ evaluated at channel average bulk temperature is recommended.

The Nusselt number results show the same trend as theoretical predictions, and what other investigators have found. At low Bo number they are very close to the results calculated from the Dittus-Boelter correlation. Increasing the Bo number will cause a Nusselt number decrease. A maximum 50% decrease has been measured for our experimental range. It should be pointed out that the Nusselt number may change substantially along the flow direction. It may have a minimum value between the inlet and exit, such as in Runs 914 and 916.

Measured Nusselt numbers have not reached values below 50% of the Dittus-Boelter correlation. The Jackson correlation, Eq.(2.30) and the Petukhov correlation, Eq.(2.33) all also show that the maximum decrease of heat transfer is about 50%. Therefore, using the Dittus-Boelter correlation with a safety factor of 2 will be sufficient to account for mixed convection effects in RCCS design calculations, although this leads to a larger than actual decrease under most circumstances.

Finally, it should be pointed out that the real riser has a rectangular cross section, while the mockup experiment uses a circular cross section. Although they have almost the same hydraulic diameter, the real riser has about three times more surface area and four times more flow area than the mockup experiment. In another words, for the same Reynolds number, the real riser surface will have $1/3$ the heat flux of the present experiment (at the same the total heat input). Therefore, the real riser buoyancy effect, in terms of $Gr_q^{\nu}/Re^n$, will be 3 times lower than that obtained in this circular riser experiments. As a matter of fact, for a given hydraulic diameter the circular cross section has the minimum surface area. Any non-circular cross section will have a lower buoyancy effect in terms of $Gr_q^{\nu}/Re^n$ than that in the circular cross section. Hence this conservative feature is another reason to choose a circular tube as the test section instead of a rectangular one. However, it should be noted that the conclusion drawn above is only suitable for
buoyancy effects expressed as a function of $\text{Gr}_q^*/\text{Re}^n$. Should the Grashof number
definition $\text{Gr}_A$ be used to correlate data, the above conclusion will no longer be valid since
$\text{Gr}_A$ is based on the total heat input. It should also be pointed out that for a given hydraulic
diameter, the same Reynolds number does not mean the same mass flow rate. The real
riser only has to have about 1/3 the mass flow rate of the present experiment to keep the
Reynolds number the same. The impact of this much lower mass flow rate on the energy
balance in the heat transport process must also be considered when applying the
experimental results to a real riser design.
CHAPTER 5

COMPARISON OF EXPERIMENTAL DATA WITH THEORETICAL PREDICTIONS

5.1 Introduction

In this chapter, the experiment results presented in Chapter 4 will be compared with the results of other experiments and existing theoretical predictions. Wall temperature distributions will be compared with other similar measurements and with numerical predictions by the CONDOR code, which employs a turbulent k-ε model [Y-1]. Friction factor and Nusselt number results will also be compared with the numerical calculation results from CONDOR and with Petukhov’s correlations [P-1]. In addition, the Nusselt number results and Jackson’s correlation [C-3] are also compared.

Turbulent k-ε models are chosen for the comparison because they have been successful in fluid mechanics and heat transfer calculations in the past, and because of their increasing use in the mixed convection field. Petukhov’s correlations and the Jackson correlation will be used because they represent two different theoretical approaches to the mixed convection problem. The former focuses on the effects of the gravitational force on the boundary layer, while the latter considers the changes in turbulent shear stress production caused by the buoyancy force.

5.2 Wall Temperature

As shown in Chapter 4, when flow is in forced convection, the measured wall temperatures are close to a linear distribution in the axial direction. However, the measured wall temperatures do not remain close to a linear distribution as wall heat flux increases. The wall temperature distortions are believed to be caused by the buoyancy effect, since the axial position at which the distortions start depends on the heat flux and Reynolds number. The higher the ratio of heat flux to Reynolds number, the earlier the wall temperature
distortions start. This phenomenon was also found by other investigators, such as the
group at Argonne National Laboratory whose results were reported by Sohn et al [S-2],
Fig. 5.1, by Miyamoto et al from their vertical parallel plate experiments [M-2], Fig. 5.2,
and by Tanaka et al [T-1] in their nitrogen gas upflow experiments, Fig. 5.3. In Figure
5.3, lines (a) and (b) correspond to Gr=4.5x10^3 and Gr=9.7x10^4, respectively. The flow
Reynolds number is about 3,000. Therefore, the experimental conditions are considered to
be in the forced convection regime. The resulting wall temperatures profiles are near to
linear. At Gr=7.4x10^5, the wall temperature starts showing distortion, line (c).

Comparisons of the experimental wall temperature with CONDOR results are
shown in Figures 5.4 to 5.6. Run 930 is considered to be close to forced convection
conditions, inlet Bo = 0.0037. The measured and calculated wall temperatures are nearly
linear distributions, Figure 5.4. Figures 5.5 and 5.6 are the results at moderate (Bo =
0.29) and high (Bo = 0.50) inlet Bo numbers, respectively. The wall temperature
distortions are obvious. The CONDOR predictions and the experimental results agree well
except at L/D beyond 50 for Run 916 in Figure 5.5, where CONDOR predicts a much
higher wall temperature. The reason for this is that CONDOR fails to make the transition
between the type I and type II flows (for details see Section 5.4).

5.3 Friction Factor

As mentioned earlier, there are two ways to measure friction factor. One is through
wall shear stress measurements. For a measured wall shear stress \( \tau_w \) at location \( z \), one has

\[
f_z = \frac{4\tau_w}{\frac{\rho v^2}{2}}
\]

(5.1)
Fig. 5.1 Comparison of the Predicted Wall Temperature with Experimental Data for $q''_{w}=4.25 \text{ kW/m}^2$
(from [S-3])
Fig. 5.2 Comparison of the Predicted Wall Temperature with Experimental Data of Miyamoto et al (from [S-3])
Fig. 5.3 Measured Wall Temperature Distributions at Three Grashof Numbers under a Constant Reynolds Number of 3000 (from [T-1])
Figure 5.4  Comparison of Experimental Wall Temperatures with CONDOR Predictions
Figure 5.5  Comparison of Experimental Wall Temperatures with CONDOR Predictions
Figure 5.6  Comparison of Experimental Wall Temperatures with CONDOR Predictions
where \( \rho \) is the fluid density evaluated at the local bulk temperature, and \( v \) is local mean flow velocity. If the flow is isothermal and \( z \) is in the fully developed region, this friction factor is the isothermal friction factor, \( f_{\text{iso}} \), i.e.

\[
f_{\text{iso}} = f_z = \frac{4 \tau_w}{\rho v^2} \frac{1}{2}
\]

Another way to measure the friction factor is through the total friction pressure drop, \( \Delta p_f \), measurement, as defined in Eqs. (4.6) and (4.12),

\[
\bar{f}_1 = \frac{2D \Delta p_f}{G^2 \int_0^l \frac{dz}{\rho}}
\]

and

\[
\bar{f}_2 = \frac{2 \rho^* D \Delta p_f}{L G^2}
\]

For an isothermal flow, \( \rho = \rho^* = \rho(T_{\text{iso}}) \), then, \( \bar{f}_1 = \bar{f}_2 = f_{\text{iso}} \). Most empirical correlations were developed for isothermal flow conditions, such as Blasius correlation, Eq.(4.14), Petukhov correlation, Eq.(2.36), and the Moody diagram. For non-isothermal flows, if the fluid properties such as density and viscosity do not change very much, using either inlet density or average density (i.e., evaluated at average bulk temperature) will not cause \( \bar{f} \) to deviate from \( f_{\text{iso}} \) significantly. However, if the fluid properties change significantly along the flow direction, the use of the different reference fluid properties will significantly affect the resulting friction factor, which is the present case, as illustrated in Chapter 4.

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Strictly speaking, for comparison with the empirical correlations, only friction factors measured from the wall shear stress and measured from the fully developed isothermal flow friction pressure drop are eligible. To compare $\bar{f}_1$ and $\bar{f}_2$ with empirical correlations, the relations between $\bar{f}_1$ or $\bar{f}_2$ and $f_{te}=f_{iso}$ must be developed to distinguish between the non-isothermal effect and the mixed convection effect (i.e. the effect due to the velocity profile changes and turbulent shear production).

However, in practice engineers use $f_{iso}$ from empirical correlations as the friction factor in non-isothermal flow. Therefore, it is still meaningful to compare $\bar{f}_2$ with $f_{iso}$ calculated from the empirical correlations for engineering application purposes. Since the definition of $\bar{f}_2$ is more close to engineering application and its values have a wider variations than that of $\bar{f}_1$, only $\bar{f}_2$ will be used in the following comparisons. Figure 5.7 shows the ratio of $\bar{f}_2/f_{iso}$ as a function of inlet Bo number from the experiments, where isothermal friction factor, $f_{iso}$, is calculated using Petukhov’s forced convection correlation

$$f_{iso} = (1.82\log_{10}Re - 1.64)^2$$  \hfill (2.36)

The solid circles represent $\bar{f}_2$ and $f_{iso}$ calculated using experimental inlet conditions, while the open circles represent $\bar{f}_2$ and $f_{iso}$ calculated using air properties evaluated at the experiment's average bulk temperature. It should be noted that the use of different temperatures to evaluate the air properties in calculating friction factors from the empirical correlations will only cause a small change (a few percent in the present case of a 200 °C air temperature increase) in the final friction factor results, because of their weak dependence on Reynolds number, which changes as temperature changes through its viscosity term. However, it will make a big difference when calculating friction factors from experimental data using Eq.(4.12). The friction factor in Eq.(4.12) is proportional to the air density, and therefore is inversely proportional to the air temperature. As can be seen in Figure 5.7, $\bar{f}_2$ calculated using inlet conditions shows an increasing trend with inlet
Figure 5.7  Friction Factor Ratio, $\frac{f_2}{f_{iso}}$, as a Function of Inlet Bo Number
Bo number, as compared to $f_{iso}$. The maximum ratio of $\bar{f}_{2}/f_{iso}$ is about 1.6. In contrast to $\bar{f}_{2}/f_{iso}$ evaluated at the inlet condition, the $\bar{f}_{2}/f_{iso}$ calculated at the average bulk temperature for the lower inlet Bo numbers is in the range 1.0 to 1.1. At higher inlet Bo numbers, the ratio shows a small decrease. In the experimental results, there is one point at an inlet Bo number of 0.5 which is much lower than all others for unexplained reasons.

In Figures 5.8 and 5.9, the experimental friction factor results from Eq.(4.12) are compared with Petukhov's mixed convection friction factor correlation, Eq.(2.34). As can be seen, the friction factor predicted using Petukhov's correlation starts with a decreasing trend and then increases from Bo = 0.1 onward, while the experimental results show no such decreasing trend. They either increase as Bo increases ($\rho^{*}=\rho_{in}$), or do not change very much ($\rho^{*}=\bar{\rho}$).

Figures 5.10 and 5.11 present the comparisons of the experimental friction factors with CONDOR predictions. The CONDOR results were obtained by inputting the experiment conditions into the code; the output friction pressure drop values were then used in Eq.(4.12) to obtain the friction factors. The results show that a reasonable agreement has been achieved, except for the point at inlet Bo of 0.5, as noted above, and two at inlet Bo numbers around 0.3. The latter are suspected due to the CONDOR incapable switching type II velocity profile back to type I velocity profile (see Chapter 2 and next section for detailed discussion).

5.4 Nusselt Numbers

The Nusselt numbers obtained from the experiments are the local Nusselt number. Since the Bo numbers change substantially along the flow direction, and the velocity profiles also keep changing along the flow direction in the mixed convection regime, no fully developed Nusselt numbers can be given. Nevertheless, these local Nusselt numbers are still compared with the results from the Jackson correlation [C-3], in Figure 5.12, with
Figure 5.8  Comparison of Experimental Friction Factors with Petukhov's Correlation  
(Air Properties were Evaluated at Inlet Temperature)
Figure 5.9  Comparison of Experimental Friction Factors with Petukhov's Correlation
(Air Properties were Evaluated at Average Bulk Temperature)
Figure 5.10  Comparison of Experimental Friction Factors with Petukhov's Correlation
(Air Properties were Evaluated at Inlet Temperature)
Figure 5.11  Comparison of Experimental Friction Factors with Petukhov's Correlation
(Air Properties were Evaluated at Average Bulk Temperature)
Fig. 5.12 Comparison of MIT Experimental Nu Results with Jackson's Correlation
Jackson correlation and CONDOR results, in Figure 5.13, and the Petukhov correlation [P-1], in Figures 5.14 and 5.15.

In Figure 5.12, Nu/Nu₀ as a function of the local Bo number is shown, where Nu₀ is the forced convection Nusselt number from the Dittus-Boelter correlation. Figure 5.13 shows three run results and predictions by the CONDOR and Jackson correlation. These three runs represent low, middle and high inlet Bo number cases. As can be seen from the figures, the local Nusselt numbers are in qualitative agreement with the Jackson correlation. Figure 5.13 also shows that the Nusselt number does appear to recover after a precipitous decrease in the moderate mixed convection region. The measured extent of the decrease in Nusselt numbers agree with the prediction of Jackson’s correlation; between 30% to 55%, as compared with the forced convection values. However, the measured Nusselt numbers show that the minimum Nusselt numbers occur at Bo = 0.1, while Jackson’s correlation predicts that it occurs at Bo = 0.25. Therefore, Jackson’s correlation can only be used to give a rough estimation of the Nusselt number for the conditions of the present experiments.

Figures 5.14 and 5.15 show comparisons of the experimental Nusselt numbers and the results from Petukhov’s correlation. It is obvious from the figures that Petukhov’s correlation fails to predict correctly the local Nusselt number behavior. However, the Nusselt numbers of Petukhov’s correlation based on channel average properties are close to the arithmetical average of the measured Nusselt numbers. Therefore, Petukhov’s correlation gives a rough estimate of the riser experiment “average” Nusselt number behavior.

The lack of success of using the buoyancy parameter, Bo, or the parameter E as proposed by Petukhov, to represent the current experiment’s data suggests that a more representative dimensionless parameter group needs to be developed. Among the possible approaches it appears that presenting the current experiment’s data as a conventional Nu vs Re plot, as in Figure 4.22, is a useful approach, at least for engineering application
Figure 5.13  Evaluation of Jackson Nu Number Correlation Against Test Results and CONDOR
Figure 5.14  Comparison of Experimental Nu Results (Run 916) with Petukhov's Correlation

Run 916:
Inlet Re = 10,000
Inlet Gr = 1.54E8
Inlet Bo = 0.29
Heat Input = 2.97 kW
Figure 5.15  Comparison of Experimental Nu Results (Run 103) with Petukhov's Correlation
purposes. However, since the buoyancy effect does cause Nusselt numbers to decrease at low Reynolds numbers, a modification to the forced convection Nusselt number correlation is needed to better express the experimental results. Analysis shows that (see Appendix C) the buoyancy effect on the ratio of mixed convection Nusselt number to forced convection Nusselt number can be expressed as

\[
\frac{Nu}{Nu_0} = \frac{1}{[1 + (\frac{C}{Re})^n]} \tag{5.3}
\]

where C and n are coefficients to be determined from the experiment, and \(Nu_0\) is the forced convection Nusselt number calculated using the Dittus-Boelter correlation. Choosing C = 4200 and n = 3, and plotting current experimental data as \(Nu[1 + (\frac{4200}{Re})^n]\) versus Re gives the results shown in Figure 5.16. As can be seen from the figure, the results fall close to a linear distribution on a log-log plot; all data are effectively lying within ±30% of the modified Dittus-Boelter correlation. Therefore, the correlation

\[
Nu = \frac{0.023Re^{0.8Pr^{0.4}}}{[1 + (\frac{4200}{Re})^3]} \tag{5.4}
\]

for \(5,000 \leq Re \leq 33,000\) and \(1 \times 10^7 \leq Gr_q \leq 2.5 \times 10^8\), is a simple and easily applied one for engineering applications, with reasonable accuracy.

The comparisons of the experimental local Nusselt numbers with CONDOR predictions and Dittus-Boelter results based on the local properties are summarized in Fig. 5.17. The four results represent four different inlet Bo number, hence different Nusselt number behavior. The inlet Bo number of Run 101 is close to 0.037 and hence the data are considered to be in the forced convection regime. The measured Nusselt number is very close to the calculated values from the Dittus-Boelter correlation for large \(z/D\). The CONDOR results indicate the extent of the developing region (to \(z/D = 30\)). Run 925 with
Figure 5.16 Comparison of Dittus-Boelter Correlation with Experimental Nu Data with Correction Factor \([1 + (4200/\text{Re})^3]\)
Figure 5.17  Heat Transfer Behavior for the Range of Bo Number Tested
an inlet Bo number of about 0.10 is considered to be just inside the mixed convection regime. The results show that the measured Nusselt numbers do not yet exhibit an obvious decrease. For Bo increased to 0.29 as in Run 916, the effect of the buoyancy force is reflected by a gradual decrease in Nusselt number from the inlet until the middle of the channel and then a gradual recovery, while the CONDOR results show a continuous decrease of the Nusselt numbers. The discrepancies stem from the fact that as flow moves along the channel, the Bo number decreases; at about L/D = 50, a transition point is reached based on behavior predicted considering a local condition hypothesis, where the velocity field changes from type II flow (maximum velocity is close to the wall) to type I flow (maximum velocity is at the pipe center). In the experiments, this transition apparently physically did occur which caused the heat transfer coefficients to improve. In CONDOR calculations, the code locks the velocity field into type II flow, and no transition occurs. Therefore, CONDOR results show a continuous Nusselt number decrease. Finally, for further increase of Bo number, as represented by Run 102 in Figure 5.17, flow is well into the mixed convection regime. The velocity fields over the entire channel are of type II flow and no Nusselt number recovery occurred. The experimental results and the CONDOR results are in good agreement in regard to the Nusselt number's decreasing trend. However, the CONDOR results show more heat transfer impairment than found in the experiments.

In general, although the CONDOR predictions of the local Nusselt numbers overestimate the mixed convection effects, and therefore the amount of heat transfer impairment, they are still in good agreement with the riser experiment results, except for the situations where the transition from type II flow to type I flow occurs.

5.5 Conclusions

The above experimental results and the comparisons with theoretical predictions show that the current experiment has covered both forced and mixed convection flows.
From the experimental results and the comparisons, the following conclusions can be drawn.

1) The friction factor behavior under the present experiment's conditions depends on which properties are used in evaluating the friction factor, because the air temperature, and its properties, change substantially along the channel. For the condition where the inlet temperature is used as the reference temperature, a maximum friction factor increase of 50% over isothermal correlation values is inferred. For the condition where the channel average bulk temperature is used, an initial 10% friction factor increase up to Bo = 0.1 and then a same order of decrease were detected. The reason for this difference is the much larger effect of density variations on the friction factor evaluated from the friction pressure drop measurements is much larger than the effect of viscosity variation on the friction factor obtained from empirical correlations. Therefore, attention should be paid to the choice of the reference temperature when applying the present experimental results to engineering applications.

2) The measured Nusselt numbers are in qualitative agreement with existing theoretical predictions. The present experiments provide an opportunity to examine the local Nusselt number behavior along the flow direction under mixed convection conditions since the air properties change substantially. The Jackson correlation is better for describing the local Nusselt behavior in the current experiment than is Petukhov's correlation, which only qualitatively agrees with experimental results on a channel average basis. The Jackson correlation is also in good agreement with the experimental results on the extent of the heat transfer impairment. Both show a maximum decrease in heat transfer of about 50%. This bound is very useful in RCCS integral heat removal calculations, since the limiting "worst case" is of most concern from a reactor safety point of view.

3) Finally, the measured Nusselt number results show the effects of Grashof number on the Nusselt numbers in a Nu vs. Bo plot. Jackson's correlation is too simple to predict this behavior. Petukhov's correlation also fails to predict this dependence.
Therefore, a better correlation or even a better dimensionless parameter group, instead of Bo, should be developed in the future. For present purposes the most useful approach would appear to be to present the data as expressed in Eq.(5.4)

$$\text{Nu} = \frac{0.023 \text{Re}^{0.8} \text{Pr}^{0.4}}{[1 + \left(\frac{4200}{\text{Re}}\right)^3]}$$

(5.4)

for \(5,000 \leq \text{Re} \leq 33,000\) and \(1 \times 10^7 \leq \text{Gr}_{q''} \leq 2.5 \times 10^8\), and plotted in Figure 5.16, where at low Reynolds number the buoyancy effects on the Nusselt numbers can be correctly reflected through the term on the denominator.
CHAPTER 6
HEAT TRANSFER FROM THE REACTOR VESSEL WALL
TO THE MHTGR RCCS RISER ARRAY

6.1 Introduction

In the MHTGR RCCS design, the dominant heat transfer mode from the reactor vessel to the riser tubes is radiation; convection in the reactor cavity is a small contribution (on the order of 10%). In view of the importance of radiation heat transfer to RCCS performance, it deserves careful study. The convective heat transfer inside the cavity between the reactor vessel and the riser array will also be evaluated, but at a lower level of detail.

To simplify the study, a unit cell consisting of vessel wall, cavity wall and one riser tube was defined and studied. A computer code, RECENT (Reactor Cavity Energy Transfer), was written to perform not only the radiation and inside-cavity convective calculation, but also to couple it to riser tube internal convection, and hence provide the capability for integral heat removal calculations from the reactor vessel wall to the ambient air. In this chapter, mathematical models of the radiation and convective heat transfer inside the cavity are described. The integral heat removal model and the calculated results will be presented in the next chapter, Chapter 7.

6.2 Modeling of Radiation Heat Transfer from the Vessel Wall to the Riser Array

6.2.1 Geometric Configuration

A cross section view of the vessel, risers and cavity wall configuration is shown in Figure 1.3. As can be seen, the cylindrical vessel is located at the center of the rectangular cavity. More than 200 riser tubes are placed between the vessel and the cavity wall. The
vessel has a diameter on the order of 20 ft, while the rectangular riser tubes are 2"x10", with the narrow side toward the vessel. The distance from the riser front wall to the vessel depends on where the riser is located; in general, this distance is on the order of a few feet. Because of the complexity of the real vessel-riser array geometric configuration, a simplified unit cell is used for the calculation. As shown in Fig. 6.1, the cell consists of a riser tube, vessel wall section and cavity wall section. The boundaries of the cell are the vessel wall, cavity wall and two imaginary diffusive walls located at the half distance between the riser and its adjacent neighbors. The cell is symmetric about its center line.

6.2.2 Assumptions

Radiation calculations involving complicated geometry usually require excessive long computation times and/or are otherwise difficult to accomplish. Even with the unit cell in the present case the three dimensional geometry will still make the calculation complicated. For example, the view factor matrix will be an order of magnitude larger than in the two dimensional case. The danger of complicated calculations is that the effect of changes in physical parameters on the results could be lost. Therefore, although the riser has a height of about 19 meters, the radiation calculation is based on a two dimensional geometry. Namely, it is assumed that the vertical direction is infinitely long (only for radiation calculation purposes). The error of using two dimensional geometry to calculate the three dimension radiation problem is from the view factor calculation because of the end effects. However, as discussed later in the view factor calculation section, this error in present case is very small and can be neglected. On the other hand, this assumption makes the radiation calculation much simpler and parametric studies much easier, while retaining sufficient accuracy.

As is conventional, it is assumed that the radiation heat transfer takes place inside an enclosure. In the present case there are two enclosures in one unit cell, as shown in Figure 6.1. One enclosure is the inside-riser part consisting of four riser inside walls. The second
Figure 6.1  Unit Cell of Riser Layout
enclosure consists of the vessel wall, cavity wall, riser outside walls and the boundaries of
the unit cell with its adjacent unit cells. Although there is no physical wall at these cell
boundaries, because of the assumed symmetry the radiation from the adjacent cells makes
these boundaries act as if there are walls located there reflecting the radiation incident upon
them. Although this "reflection" is actually specular (i.e., mirror-like), it is approximated
here as diffuse. The validation of this diffusive wall assumption will be provided later in
this chapter.

For the surfaces forming both enclosures, it is assumed that all surfaces (including
the imaginary surfaces at the boundaries with the adjacent cells) are diffuse and gray (i.e.,
spectral emissivity and absorptivity do not depend on wavelength). In the design, carbon
steel is used in the vessel wall and riser tubes and concrete (perhaps with a steel liner) is
used for the cavity wall. Since their emissivities and absorptivities depend only very
weakly on the wavelength in the effective heat radiation range, this assumption is very
close to reality.

In the calculations, the walls of the first enclosure which make up the inside of the
riser are divided into six surfaces. Each narrow inside wall is one surface. Each side wall
is equally divided into two surfaces. The walls of the second enclosure have been divided
into 16 surfaces, as shown in Figure 6.2. The vessel wall and cavity wall are each one
surface. Each diffusively reflective imaginary wall is divided into 4 surface units. The riser
outer walls are divided into 6 surfaces, as was done for the corresponding inside walls.
The emissivity/absorptivity of each surface is assumed independent of wall temperature.
There is no heat conduction between walls except for the riser walls (when specified), but
there is infinite heat conductivity within each wall. The infinite heat conductivity of a wall
implies that its surface temperature is uniform. Notice that infinite heat conductivity also
implies that the connected inside and outer riser walls will have the same temperature. For
representative riser heat fluxes, the actual ΔT is only about 0.01 °F, and hence is

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Figure 6.2  Nodalization of Enclosure Surfaces (Not to Scale)
negligible. The other assumption for each wall is that the incident and reflected heat fluxes are uniformly distributed over each surface.

6.2.3 Mathematical Treatment

The method used in the vessel-riser radiation calculation is the so-called Net-Radiation Method [S-2]. As shown in Figure 6.3, consider the k-th surface of an enclosure with area $A_k$, and let $\tilde{q}_i$ and $\tilde{q}_o$ be the rates of incoming and outgoing radiation heat flux, respectively. The quantities $q''$ and $Q$ are the net heat flux and total heat supplied or absorbed by the surface. A heat balance at the surface provides the relation

$$Q_k = q''_k A_k = (q''_o - q''_i) A_k$$

(6.1)

Another relation can be found from the fact that the heat flux leaving the surface is composed of the directly emitted heat flux and the reflected heat flux

$$q''_o = \varepsilon_k \sigma T_k^4 + \rho_k q''_i$$

(6.2)

where $\sigma = 5.67 \times 10^{-8}$ W/(m$^2$·K$^4$) is the Stefan-Boltzmann constant, $\rho_k$ is the k-th surface reflectivity, $\varepsilon_k$ is its emissivity and $T_k$ is the surface temperature, K. The reflectivity $\rho_k$ and the emissivity $\varepsilon_k$ are related for opaque gray surfaces by the following relation

$$\rho_k + \varepsilon_k = 1$$

(6.3)

Hence Eq.(6.2) can be written as

$$q''_o = \varepsilon_k \sigma T_k^4 + (1-\varepsilon_k)q''_i$$

(6.4)
(a) Enclosure Composed of N Surfaces with Typical Surfaces j and k

\[ Q_{i,k} = (q_i^A)_{k} \quad (\rho q_i^A)_{k} \quad \sigma (\varepsilon T^4 A)_{k} \]

\[ Q_{o,k} = (q_o^A)_{k} = (\rho q_i^A)_{k} + \sigma (\varepsilon T^4 A)_{k} \]

(b) Energy Quantities Incident Upon and Leaving Surface k of an Enclosure

\[ Q_k = (q_k^A)_{k} \]

k-th surface with area \( A_k \)

Figure 6.3   Schematic of an Enclosure of N Surfaces and the Energy Components on a Surface
Because the radiation heat transfer under consideration occurs inside an enclosure (both inside and outside enclosures described earlier), the incident energy is equal to

\[ A_k q_{i,k} = A_1 q_{o,1} F_{1-k} + A_2 q_{o,2} F_{2-k} + \ldots + A_N q_{o,N} F_{N-k} \]  
(6.5)

where \( F_{j-k} \) is the view factor from surface \( j \) to surface \( k \), and \( N \) is the total number of surfaces. From the reciprocity relation

\[
\begin{align*}
A_1 F_{1-k} &= A_k F_{k-1} \\
A_2 F_{2-k} &= A_k F_{k-2} \\
\ldots & \quad \ldots \\
\ldots & \quad \ldots \\
\end{align*}
\] (6.6)

One has, by substituting Eq.(6.6) into Eq.(6.5),

\[ q_{i,k} = \sum_{j=1}^{N} (F_{k-j} q_{o,j}) \] 
(6.7)

Eliminating parameter \( q_{i,k} \) between Equations (6.1), (6.4) and (6.7) yields two equations as follows

\[ Q_k = \frac{A_k \varepsilon_k}{1 - \varepsilon_k} (\sigma T_{i,k}^4 - q_{o,k}) \] 
(6.8)

and

\[ Q_k = A_k \left[ q_{o,k} - \sum_{j=1}^{N} (F_{k-j} q_{o,j}) \right] \] 
(6.9)

By eliminating \( Q_k \) from Equations (6.8) and (6.9) one has

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\[
q_{o,k} - (1 - \varepsilon_k) \sum_{j=1}^{N} (F_{k,j} q_{o,j}) = \varepsilon_k \sigma T_k^4
\]

(6.10)

or

\[
\sum_{j=1}^{N} \left[ \delta_{kj} - (1 - \varepsilon_k) F_{k,j} \right] q_{o,j} = \varepsilon_k \sigma T_k^4
\]

(6.11)

where \( \delta_{kj} \) is the Kronecker delta defined as

\[
\text{For } \delta_{kj} = 1 \quad \text{if } k = j
\]

\[
\text{For } \delta_{kj} = 0 \quad \text{if } k \neq j
\]

(6.12)

Both Eq.(6.9) and Eq.(6.11) can be used to calculate the \( k \)-th surface radiosity \( q_{o,k} \). Which one should be used depends on surface boundary conditions. There are three kinds of boundary conditions which could be imposed on a surface: the surface can be held at a specified temperature, have a specified heat input, or be perfectly insulated (i.e., specified heat input is zero). When temperature is specified, Eq.(6.11) is used. When heat input is specified, Eq.(6.9) is used. In a more general form, for an \( N \)-surface enclosure having surfaces 1, 2, 3, \ldots, \( n \) with specified temperature boundary conditions and surfaces \( n+1, n+2, n+3, \ldots, N \) with specified heat input boundary conditions, then the system of equations are

\[
\sum_{j=1}^{N} \left[ \delta_{kj} - (1 - \varepsilon_k) F_{k,j} \right] q_{o,j} = \varepsilon_k \sigma T_k^4 \quad 1 \leq k \leq n
\]

(6.13a)

\[
\begin{bmatrix}
q_{o,k} - \sum_{j=1}^{N} (F_{k,j} q_{o,j})
\end{bmatrix} = \begin{bmatrix}
Q_k \\
A_k
\end{bmatrix} \quad n+1 \leq k \leq N
\]

(6.13b)
With these N equations, the N unknown radiosities, $q_{o,k}$ can be solved for. After obtaining $q_{o,k}$, Eq.(6.8) can then be used to get the total heat transferred, $Q$, for the surfaces having a specified temperature as the boundary condition, or to get the surface temperatures for the surfaces having specified heat input as the boundary condition.

6.2.4 View Factors

The view factor is a key parameter in the above radiation heat transfer calculation method. View factor calculations are usually very complicated. However, in a two dimensional geometry, which is what we have assumed in the present calculation, a simple method, the Hottel string method [H-4], can be used in the calculations. As shown in Fig. 6.4, for any two surfaces 1 and 2, which have an infinite length in the direction perpendicular to the paper, the view factor from 1 to 2 can be calculated as

$$F_{1-2} = \frac{(ad + bc) - (ac + bd)}{2ab}$$  \hspace{1cm} (6.14)

where $\overline{ab}$, $\overline{cd}$, ..., are the lengths of the corresponding surfaces in the plane of the paper.

The use of a two dimensional method to calculate the RCCS cavity three dimensional geometry will cause a small error. This occurs because the dimension in the direction perpendicular to the paper, on the order of 20 meters, is much larger than the distance between the vessel and riser, which is about 0.7 meters. Hence the view factor from the vessel wall to the bottom or top end sections is only about 0.017. Therefore only 1.7% of the vessel energy does not intercept the riser. Further most energy radiated from the vessel wall to these end sections will be reflected back to the riser by the actual structures present. Therefore, the errors in using two dimensional assumptions to calculate view factors, and hence radiation from the vessel to the riser are very small and will be neglected here.
\[ F_{1-2} = \frac{(ad+bc) - (ac+bd)}{2ab} \]

Figure 6.4  Schematic of Hottel String Method to Calculate View Factors
6.2.5 Boundary Conditions for Radiation Heat Transfer Analysis

To solve the vessel-riser radiation heat transfer problem using the Net-Radiation Method, the boundary conditions on each surface must be assigned: either a given temperature or a given heat flux. The boundary conditions for this particular application are as follows

(1) Specified heat flux at the vessel wall, since the total decay heat is fixed for each scenario.

(2) The cavity wall is perfectly insulated, i.e. the net heat flux through the cavity wall is zero.

(3) There is no net heat generation or absorption on the imaginary side walls. Therefore, the heat fluxes on the imaginary walls are also zero.

(4) The boundary conditions on all riser walls are given temperatures. Since the temperatures on these wall are the parameters we are interested in, and are not known before the calculations, a set of guessed values must be given to start the calculations. The new boundary conditions are determined from the previous calculation. The final riser wall temperatures must satisfy an energy balance from the vessel to the riser and from the riser to the ambient air. The detailed iteration process for determining these wall temperatures is described in the next chapter.

6.3 Test of Model Validity

The radiation model presented above employed some assumptions and simplifications. Two important assumptions are the two dimensional geometry and the diffusive reflective imaginary walls. One of the simplification is that dividing the riser walls into limited numbers of nodes, six in the present calculation. The two dimensional geometry assumption has been discussed in preceding section. It is shown that it only introduce a small error in the calculation.
To quantify the effect of using the diffuse wall assumption instead of a specular one for the imaginary wall between cells, and to check the correctness of our cell selection, an alternative cell configuration, as shown in Figure 6.5, was used to do the same calculation as that in the Figure 6.1 configuration. This cell differs from the previous one in that the outer boundary intersects the riser. As a result, the effective surface area of the imaginary wall has been decreased. If the diffusive wall approximation is acceptable, the results should be the same for both wall configurations. RECENT calculated results show that the two results are very close (less than 1 K difference in vessel temperature). Thus, the conclusion that the use of a diffusive boundary is valid in this calculation can be drawn.

It is known that in the differential calculation, the more the number of surface nodes the more accurate the calculated results. However, the more the number of the surface nodes the more complicated the calculation because the view factor calculation is an N x N matrix, where N is the number of nodes. Since the riser surface involves radiation, conduction and convection calculations, the number of nodes utilized is more important. In the present calculation six nodes were used. The question arises whether these six nodes are enough to give accurate results. Since the node selection on the riser walls affects the vessel temperature through the radiation calculation and also through its own wall temperature distributions (conduction is a major contribution to this distribution) within the riser — the finer the distribution the more accurate the results. To validate the present selection of the riser wall nodes, the radiation effect and the conduction effect have been assessed as follows.

For the radiation calculation, different numbers of nodes were selected to do the calculations (with conduction in the riser wall being suppressed) and the resulted vessel wall temperatures were compared. For a typical riser operating condition, one node, four nodes (each side of riser surface is a node) and current six nodes were used in the calculation. The results show that vessel wall temperature in the four node case is about 10 K higher than that of the one node case, while the six node case is about 3.5 K higher than
Figure 6.5 Alternative Unit Cell for Riser Layout
that of the four node case. From this trend, further increasing the node numbers can only result in a small gain in the vessel wall temperature. Therefore, the present six node selection will yield reasonable accuracy in the calculated results from the radiation calculation point of view.

Conduction within the riser wall will smooth the wall temperature distribution. To quantify the effect on the vessel wall temperature, the same calculation as above was carried out but with conduction within the riser walls. Note that the one node case is a limiting case since it implies an infinite heat conductivity. The results show the same trend as that in the radiation case. The vessel wall temperature in the four node case is only about 6 K higher than that of the one node case, while the vessel temperature in the six node case is only about 2 K higher than that of the four node case. Therefore, selecting six nodes for the riser walls will yield reasonable accuracy in the calculated results from the conduction point of view.

In general, the above discussion has shown that the present radiation model is a reasonable accurate model. Considering the typical vessel wall temperature is in the range of 700 K, the overall error is less than 1%.

6.4 Convective Heat Transfer inside the Reactor Cavity

The convective heat transfer from the reactor vessel wall to the riser array can be considered as convection in an enclosure. Although natural convection heat transfer in an enclosure has been extensively studied both experimentally and numerically, the geometries of most studies are relatively simple compared with the RCCS configuration. For the application to RCCS cavity convection problem, the results for vertical rectangular enclosures with both ends closed will be used. In a vertical rectangular enclosure, as shown in Figure 6.6, it is usually assumed that the two opposing walls are held at different temperatures, \( T_H \) and \( T_C \). The remaining walls are assumed to be insulated. The parameter of interested is the heat transfer coefficient from the hot wall at a temperature of
Figure 6.6  Schematic of a Vertical Rectangular Enclosure Showing Dimensions and Temperatures
T_H to the cold wall at a temperature of T_C through the fluid filling the cavity. Since the velocity and temperature fields under such boundary conditions are usually complicated, empirical or semiempirical correlations based upon experiments are used to determine the heat transfer coefficient. The correlations in general take the dimensionless form

$$\text{Nu} = a \text{Gr}^b_{\Delta T} A^c$$  \hspace{1cm} (6.15)

where the Grashof number, \( \text{Gr}_{\Delta T} \), is defined as

$$\text{Gr}_{\Delta T} = \frac{g \beta \rho^2 (T_H - T_C) d^3}{\mu^2}$$  \hspace{1cm} (6.16)

and the aspect ratio A is

$$A = \frac{H}{d}$$  \hspace{1cm} (6.17)

a, b and c are constants determined from the experiments. Equation (6.15) is applicable to laminar flow in the range \( 2 < H/d < 20 \), which usually requires the condition that \( \text{Gr}_{\Delta T} < 10^9 \).

The set of constants (a, b and c) suggested by Eckert and Carlson [E-2] from their air experimental data was used in the present RCCS cavity convection calculations, namely \( a = 0.119 \), \( b = 0.3 \) and \( c = -0.1 \). With these constants, Eq.(6.15) becomes

$$\text{Nu} = 0.119 \text{Gr}^{0.3}_{\Delta T} (\frac{H}{d})^{-0.1}$$  \hspace{1cm} (6.18)

Eq.(6.18) shows that the Nusselt number depends very weakly on the aspect ratio \( H/d \). Since the heat transfer coefficient \( h \propto (\text{Nu}/d) \) and \( \text{Gr} \propto d^3 \), the heat transfer
coefficient \( h \) is also independent of the wall-to-wall spacing \( d \). Hence we could also have treated this configuration as isolated non-interacting vertical plates in heat upflow or cold downflow.

It should be noted that in the Eckert and Carlson experiments the maximum Rayleigh number measured was \( 2 \times 10^5 \), while the upper-limit of the Rayleigh number for laminar flow is around \( 10^9 \). The calculated RCCS operating conditions show that the RCCS cavity convection Rayleigh number is about \( 10^9 \), i.e. it operates on the border between laminar and turbulent natural convection. Nevertheless, the use of the laminar flow heat transfer coefficient in a turbulent flow situation is conservative, since the turbulent Nusselt number has a higher power dependence on Grashof number, \( \text{Gr}_{AT} \), than the laminar case. As shown schematically in Figure 6.7, the continuation of the laminar Nusselt number correlation into the turbulent region will underestimate the Nusselt number.

6.5 Chapter Summary

In this chapter the models for analysis of heat transfer from the reactor vessel wall to the riser array were presented. Radiation heat transfer is the dominant contributor to this heat transfer process. In the radiation heat transfer calculation, a two dimensional geometry was assumed and the so-called Net-Radiation Method [S-2] was employed. The Hottel string method was used for view factor calculations. The assumptions and simplifications used in the model have been proved that they will only introduce small error (less than 1%) in the vessel wall temperature.

For convection heat transfer calculation within the reactor cavity, a simple laminar correlation for the heat transfer coefficient was adopted over the entire RCCS cavity convection regime, which could be either laminar flow or turbulent flow natural convection. Since convection heat transfer makes a relatively small contribution to the overall heat transfer from the reactor vessel wall to the riser array, this simplification will not introduce significant error in the final results. Also, by using a laminar flow
Figure 6.7 Schematic of Nusselt Number Dependence on Grashof Number in Different Flow Regimes (from [P-4])
correlation, the results are conservative: convective heat transfer is underestimated, and hence vessel wall temperature will be slightly overestimated.
CHAPTER 7
CALCULATIONS OF THE OVERALL HEAT TRANSFER FROM THE
REACTOR VESSEL TO AMBIENT AIR

7.1 Introduction

The overall heat transfer from the reactor vessel to the ambient air involves several
heat transfer modes. As can be seen schematically in Figure 7.1, from the reactor vessel to
the riser tube radiation is dominant, although cavity convection has a non-negligible
contribution. The radiation from the insulated cavity wall and the imaginary diffuse
reflective surfaces to the riser tube is only re-radiation of the heat they received from the
reactor vessel wall. Conduction through the riser wall transports the energy, received as
radiation, from its outer-surface to its inner-surface. The heat conduction within the riser
wall will smooth its temperature distribution. The heat will finally be transferred to the
ambient air by convective heat transfer from the inner riser surface to the flowing air,
drawn from ambient air; after absorbing the energy, the air transports the energy to the
atmosphere.

The radiation heat transfer from the reactor vessel to the riser tube has been
discussed in Chapter 6, along with a simple model of inside-cavity convection. Chapters 2
to 5 have discussed the convection heat transfer inside the riser tube, especially the
behavior of two key governing parameters, friction factor and Nusselt number, in detail.
Hence, an overall heat transfer calculation can now be made to find the importance of each
design parameter, and the sensitivity of the vessel wall temperature to variation of these
parameters.

In this chapter, an overall heat transfer model will be developed and the scheme for
solution will be given. After defining the boundary conditions and the geometric
configuration of the system, the results of calculations using the computer code, RECENT,
Figure 7.1  Heat Transfer Modes from Reactor Vessel Wall to Flowing Air Inside Riser
will be presented. From the calculated results, criteria for selection of an optimal riser configuration are proposed.

7.2 System Description

The system under consideration consists of the reactor vessel, cavity, riser tube and chimney. The geometric arrangement of these components is shown in Figure 7.2. Note that the riser tube does not completely intercept the radiation from the vessel wall to the cavity wall. A cross section view of the entire reactor vessel cavity can be found in Figure 1.3.

The actual RCCS system consists of parallel flow channels connected at the plenums in the chimney section and cold flow paths, and more than 200 riser tubes parallel to each other inside the cavity. The calculations of overall heat transfer based on this typical system must be accomplished by employing a large scale computer code, such as COMMIX. However, the calculation nodes in that computation are necessarily fairly coarse, such that detailed information is lost. On the other hand, a detailed analysis focused on local points of interest or phenomena easily losses sight of the "big picture" — system performance. The present overall heat transfer calculations are a compromise, combining selected aspects of system performance and local details. For the region outside the cavity, one chimney and one cold flow path will be used, which have similar fluid flow characteristics to a typical system, such as flow resistance. Inside the cavity, a unit cell consisting of a single riser tube, vessel wall and cavity wall, as described in Chapter 6, will be used. This unit cell represents an average among the riser tubes, i.e. if there are total of N riser tubes, it has one N-th of the total flow rate and removes one N-th of the total decay power.

In the calculation, the chimney and cold flow path are each taken as a node, since they are only related to the flow calculations. Inside the cavity all surfaces have been
Figure 7.2   Schematic of RCCS System Considered in RECENT
divided into several nodes. The surface node division and their corresponding node numbers are shown in Figure 6.2.

7.3 Assumptions and Boundary Conditions

Since the decay power changes very slowly and the RCCS response time is very short, a steady state will always be assumed in the calculations. In addition, several other assumptions, and the boundary conditions, will be given or repeated (for those previously specified) as follows.

The assumptions with regard to radiation heat transfer have been given in Chapter 6. Several will be repeated here because of their importance.

1) The radiation calculation is based on a two dimensional geometry, i.e. the vertical direction is assumed to be infinite.

2) All surfaces are diffuse and gray.

3) All surface emissivities are independent of surface temperature.

4) The temperature difference between the inner-surface and outer-surface of a riser wall node is neglected, i.e. the riser inner-surface and the corresponding outer-surface have the same temperature. However, the lateral heat conduction between the nodes is still finite; which will be computed using a constant conductivity of 45 W/(m °C).

The assumptions on riser/chimney convection are as follows:

1) the chimney is perfectly insulated, i.e., no heat loss through the chimney wall occurs.

2) The acceleration pressure drop is small compared to the buoyancy and friction terms, and can be neglected.

3) Ambient air density changes with height are neglected, since we have a relatively small elevation change. This density is equal to the air density at the riser inlet.

The boundary conditions for this system are simple.
1) On the reactor vessel wall the total decay heat $Q$ is given, and the heat flux is assumed to be uniform on that surface.

2) The ambient temperature, hence air temperature at the riser inlet, is given as a constant.

With the above assumptions and boundary conditions, the overall heat transfer model can now be developed.

### 7.4 Mathematical Model

The equations governing the radiation heat transfer have been given in Eqs. (16.13), where Eq. (6.13a) is for the surfaces with specified temperature as the boundary condition, and Eq. (6.13b) is for the surfaces with specified heat flux as the boundary condition. For inside-riser radiation all walls will have the temperature boundary condition. Therefore, for inside-riser radiation, Eq. (6.13a) will be used

\[
\sum_{j=1}^{N} \left[ \delta_{kj} - (1-\varepsilon_k)F_{k,j} \right] q^o_{o,j} = \varepsilon_k \sigma T_k^4 \quad 1 \leq k \leq 6 \quad (7.1)
\]

For radiation in the cavity, surfaces 1 to 10 have heat flux specified as the boundary condition, while surfaces 11 to 16 have temperature specified as the boundary condition (see Figure 6.2). Then, the equations for radiation in the cavity are

\[
\sum_{j=1}^{N} \left[ \delta_{kj} - (1-\varepsilon_k)F_{k,j} \right] q^o_{o,j} = \varepsilon_k \sigma T_k^4 \quad 1 \leq k \leq 10 \quad (7.2)
\]

\[
q^o_{o,k} - \sum_{j=1}^{N} \left( F_{k,j} q^o_{o,j} \right) = \frac{Q_k}{A_k} \quad 11 \leq k \leq 16 \quad (7.3)
\]

The cavity convection heat transfer can be calculated as
\[ q_{c,v} = h_{c,v} (T_v - \overline{T}_r) \]  

(7.4)

where \( q''_{c,v} \) is the heat flux leaving the vessel wall and \( h_{c,v} \) is the heat transfer coefficient, calculated using Eq. (6.18)

\[ h_{c,v} = \frac{k}{d} \text{Nu} = 0.119 \ \frac{k}{d} \ Gr_{\Delta T}^{0.3} (\frac{H_r}{d})^{-0.1} \]  

(7.5)

Note that the Grashof number, \( Gr_{\Delta T} \), is defined, using the temperature difference between the vessel and riser walls,

\[ Gr_{\Delta T} = \frac{g\beta \rho^2 (T_v - \overline{T}_r) d^3}{\mu^2} \]  

(7.6)

In the above equations, \( H_r \) is the height of the riser, \( d \) is the distance from the vessel wall to the riser wall, \( T_v \) is vessel wall temperature, and \( \overline{T}_r \) is riser wall circumferential average temperature calculated as

\[ \overline{T}_r = \frac{\sum_{i=1}^{N} A_i T_i}{\sum_{i=1}^{N} A_i} \]  

(7.7)

where \( A_i \) and \( T_i \) are the i-th surface area and temperature.

The total heat flux leaving the vessel surface, \( q''_{v} \), is equal to the sum of radiation heat flux, \( q''_{r,v} \), and convective heat flux, \( q''_{c,v} \),

\[ q''_{v} = q''_{r,v} + q''_{c,v} \]  

(7.8)
The thermal conduction within the riser wall is calculated using Fourier's equation

\[ q_{cd} = -k \frac{\Delta T}{\Delta x} \quad (7.9) \]

The inside-riser convection heat transfer is governed by mass, momentum and energy equations, as presented in Chapter 3. The momentum and energy equations need to be developed specifically for the present system application. The momentum equation is the pressure balance between the buoyancy gain and friction loss, \( \Delta p_b = \Delta p_f \), with acceleration loss being neglected. The buoyancy gain is

\[ \Delta p_b = \int_0^H g \beta (\rho_o - \rho(z)) dz + g \beta (\rho_o - \rho_e) H_e \quad (7.10) \]

which can be written simply as

\[ \Delta p_b = [(\rho_o - \rho_{ave}) H_r + (\rho_o - \rho_e) H_e] g \quad (7.11) \]

where \( \rho_{ave} \) and \( \rho_e \) are air densities evaluated at riser average bulk temperature and exit temperature, respectively. \( \rho_o \) is inlet air density, which is equal to atmospheric air density.

The friction pressure loss is from friction inside the riser, friction in the chimney and cold flow path, and form losses such as that due to bends, fittings, expansions and contractions.

\[ \Delta p_f = \sum_i (K_c + K_f) \frac{G_i^2}{2 \rho_i} + \bar{f} \frac{H_r}{2 \rho^*} \frac{G_r^2}{(D_{ek})^2} \quad (7.12) \]
where \( K_c \) = resistance coefficient associated with changes in flow path: bends, fittings, etc.

\( K_f \) = friction loss coefficient = \( f(L/D) \): in which \( f \) = friction factor, \( L \) = Length, \( D \) = hydraulic diameter (4 times flow area divided by "wetted" perimeter)

Using mass flow rate instead of mass velocity, Eq. (7.12) can be written in the form

\[
\Delta p_f = K_c \frac{\dot{m}_r^2}{2 \rho_e} + \bar{f} \frac{H_r}{(D_e)^k} \frac{\dot{m}_r^2}{2A_{fl} \rho^*}
\]

(7.13)

where \( K = \sum_i \frac{(K_c + K_f)_i}{(N \cdot A)_i^2} \)

\( N \) = number of flow branches

\( A \) = riser flow area

\( \dot{m}_r \) = mass flow rate in a single riser

\( \dot{m}_t \) = total mass flow rate of the RCCS system

Subscript \( fl \) refers to flow and subscripts \( r \) and \( t \) refer to riser and total, respectively.

Combining Eqs.(7.11) and (7.13), the final momentum equation for the riser convection calculation is

\[
[(\rho_o - \rho_{ave})H_r + (\rho_o - \rho_c)H_c]g = K_c \frac{\dot{m}_r^2}{2 \rho_e} + \bar{f} \frac{H_r}{(D_e)^k} \frac{\dot{m}_r^2}{2A_{fl} \rho^*}
\]

(7.14)

It should be noted that the use of \( \rho_e \) in the first term on the RHS of above equation may cause some errors in the calculation because this term represents both chimney and cold channel flow resistance, and the air density has different values at different locations. However, the total flow resistance coefficient \( K \) is an estimated value in this calculation,
and taken as a constant; at a higher level of sophistication it would be a function of flow rate. The uncertainty in the total flow resistance coefficient will be much larger than that arising from use of a constant density. On the other hand, using a single density will make the calculation process much simpler.

The energy equation is much simpler than the momentum equation, i.e.

$$m_r c_p (T_e - T_{in}) = Q$$  \hspace{1cm} (7.15)

where $c_p$ is air heat capacity, $T_e$ and $T_{in}$ are air bulk temperature at riser exit and inlet, respectively, and $Q$ is the total energy transferred to a single riser. Because this total energy transferred to the air can also be expressed as

$$Q = A_{r,HT} \bar{h} (T_r - T_{ave})$$  \hspace{1cm} (7.16)

where $A_{r,HT}$ is riser total heat transfer area, $\bar{h}$ is channel circumferential and axial average heat transfer coefficient, $\bar{T}_r$ is riser wall average temperature, as defined in Eq.(7.7), and $T_{ave}$ is flowing air average temperature. For a uniform heat flux, $T_{ave}$ can be approximated as

$$T_{ave} = \frac{1}{2} (T_{in} + T_e)$$  \hspace{1cm} (7.17)

Then, the energy equation can be written as

$$m_r c_p (T_e - T_{in}) = A_{r,HT} \bar{h} (\bar{T}_w - T_{ave})$$  \hspace{1cm} (7.18)

To close the above equation set, the constitutive correlations for friction factor and heat transfer coefficient inside the riser must be provided. From the discussion in Chapters 2 to
5, one finds that the behavior of the friction factor and Nusselt number can not easily be predicted. However, this evaluation does suggest that Eq.(5.4) can be used for the Nusselt number based on the experiment results discussed in Chapters 4 and 5. For the friction factor the fully developed forced convection Blasius correlation will be used:

\[
f = 0.316 \text{Re}^{-0.25}
\] (7.19)

Table 7.1 The unknowns and equations for the overall heat transfer problem

<table>
<thead>
<tr>
<th>Unknowns</th>
<th>Equations</th>
<th>Eqn. #</th>
</tr>
</thead>
<tbody>
<tr>
<td>Riser surface heat fluxes, total = 6</td>
<td>( q'' )</td>
<td>6.13a</td>
</tr>
<tr>
<td>Temperatures on each surface, total = 16</td>
<td>( T )</td>
<td>6.13a, 6.13b</td>
</tr>
<tr>
<td>Lateral conduction heat fluxes, total = 6</td>
<td>( q''_{cd} )</td>
<td>7.9</td>
</tr>
<tr>
<td>Radiation heat flux at vessel wall</td>
<td>( q''_{r,v} )</td>
<td>7.4</td>
</tr>
<tr>
<td>Convection heat flux at vessel wall</td>
<td>( q''_{c,v} )</td>
<td>7.8</td>
</tr>
<tr>
<td>Mass flow rate of riser</td>
<td>( \dot{m} )</td>
<td>7.14</td>
</tr>
<tr>
<td>Riser exit temperature</td>
<td>( T_e )</td>
<td>7.18</td>
</tr>
<tr>
<td>Riser average temperature</td>
<td>( \bar{T}_{ave} )</td>
<td>7.17</td>
</tr>
<tr>
<td>Air densities, total = 4 ( \rho )</td>
<td>( \bar{\rho}, \rho_e, \rho_o )</td>
<td>4.9</td>
</tr>
<tr>
<td>Cavity convective heat transfer coefficient</td>
<td>( h_{c,v} )</td>
<td>7.4</td>
</tr>
<tr>
<td>Riser average friction factor</td>
<td>( \bar{f}_r )</td>
<td>7.19</td>
</tr>
<tr>
<td>Riser average heat transfer coefficient</td>
<td>( \bar{h} )</td>
<td>5.4</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>total = 37</th>
</tr>
</thead>
</table>

183
where properties are evaluated at the riser air average bulk temperature. Variation of the friction factor in mixed convection will be studied in the calculations by changing the multiplicative constant coefficient in Eq.(7.19).

With the above two constitutive correlations, the above equation set is closed, as shown in Table 7.1, and the unknowns can be solved for.

7.5 Scheme for Solution

As shown in Table 7.1, the 37 unknowns can be solved using the 37 equations given and appropriate boundary conditions. The parameters sought are vessel wall and riser wall temperatures. Heat flux distributions on riser walls are also of interest. To solve this overall heat transfer problem, a computer code, RECENT, has been written for this purpose (see Appendix A).

Since the only fixed boundary conditions are the total heat flux on the vessel wall and the air temperature at the riser inlet, which is equal to the ambient air temperature, several iterations must be employed to solve this problem. The key parameters in the solving process are the riser wall temperatures (or heat fluxes), because they connect the radiation inside the cavity and the convection inside the riser. Unfortunately, these values are not known. As a matter of fact, they are among the parameters sought. Therefore, in the code calculation a set of initial-guess riser wall temperatures will be furnished to start the calculation. These temperatures are updated in the subsequent iterations, until the total heat transferred from the vessel to the riser equals that transferred from the riser to the ambient air. A flow chart of the iterating process in the RFCENT code is given in Figure 7.3.

As can be seen from the flow chart for the RECENT code, after reading in all the parameters needed to start the calculation, the first iteration — the radiation calculation — will be performed first. When the relative errors of the successive riser wall temperatures and energy balance are less than 0.5% (i.e., ${\varepsilon_1}=0.5\%$ and ${\varepsilon_2}=0.5\%$) based on the estimated
Start-read input: geometry, emissivities, b.c., air inlet temp., Q at vessel wall

Read initial guess wall temp., radiation heat flux at each wall, and outlet air temp.

Calculate view factor

Calculate mass flow rate and riser average air temp.

Cavity radiation

Adjust riser wall temp.

Inside riser radiation

Riser conduction calculation?

Yes

Riser conduction

No

$(\Delta T)_{\text{min}} < \varepsilon_1$?

Yes

No

$(\Delta Q)_{\text{min}} < \varepsilon_2$?

Yes

(continue from next page) (continue to next page) (continue from next page)

Figure 7.3  RECENT Code Flow Chart
Figure 7.3 (continued)  RECENT Code Flow Chart
riser air outlet temperature, this iteration is satisfied. The new riser outlet temperature and flow rate are then calculated using the new riser wall temperatures. Usually this new riser air outlet temperature does not match the previous one. Then the code returns to the radiation calculation again, and then to re-determination of the riser air temperature and mass flow rate. This second loop of iterations continues until both the error on the riser wall temperature and the riser air outlet temperature from the successive calculations are less than the preset error levels (typically, $\epsilon_1 = 0.5\%$ and $\epsilon_3 = 0.02$ K). Finally, the cavity convection computation, based on the vessel wall and riser wall temperatures calculated above, is executed. The computed convective heat flux leaving the vessel surface will be used to adjust the total radiation heat flux leaving the reactor vessel. After the adjustment, the code repeats the radiation and riser convection calculations. The process continues until the error of consecutive radiation heat fluxes leaving the reactor vessel wall are less than $\epsilon_4$, 0.5%. At this stage, the system is in steady state, namely pressure is balanced and heat transfer from the vessel to the riser by radiation and cavity convection is equal to the heat transfer from the riser to the ambient air.

7.6 Parametric Studies

7.6.1 Nominal Case

The primary parameters of interest in the present study are the vessel wall temperature, riser wall temperature distributions, and heat flux distributions. Therefore, any parameters having considerable impact on the vessel and riser wall temperatures will be studied, such as wall emissivities, decay power level, heat transfer coefficient and friction factor inside the riser.

For the convenience of the study and comparisons, a nominal case is defined based on "typical" design parameters. The related dimensions and other parameters used in the nominal case are shown in Table 7.2. This nominal case is used as a reference case in the following calculations. All other cases are variations from the nominal case.
Table 7.2 Parameters Used in Nominal Case

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat flux at vessel wall (uniform)</td>
<td>5000 W/m²</td>
</tr>
<tr>
<td>Chimney height</td>
<td>27.4 m</td>
</tr>
<tr>
<td>Riser height, H</td>
<td>19.2 m</td>
</tr>
<tr>
<td>Riser W, L, Spacing</td>
<td>0.0508, 0.254 and 0.1016 m</td>
</tr>
<tr>
<td>Distance between riser front wall to vessel wall</td>
<td>0.7 m</td>
</tr>
<tr>
<td>Distance between riser back wall to cavity wall</td>
<td>0.18 m</td>
</tr>
<tr>
<td>Wall emissivity (all walls)</td>
<td>0.8</td>
</tr>
<tr>
<td>Flow resistance coefficient, K</td>
<td>0.0064</td>
</tr>
<tr>
<td>Air inlet temperature.</td>
<td>110 °F</td>
</tr>
<tr>
<td>Heat transfer coefficient, h</td>
<td>Eq.(5.4)</td>
</tr>
<tr>
<td>Friction factor, f</td>
<td>Blasius correlation, Eq.(7.19)</td>
</tr>
<tr>
<td>Conduction in riser</td>
<td>yes</td>
</tr>
<tr>
<td>Convection in cavity</td>
<td>yes</td>
</tr>
<tr>
<td>Inside riser radiation</td>
<td>yes</td>
</tr>
</tbody>
</table>

The calculated results using the RECENT code for the nominal case are given in Table 7.3, where all temperature are at mid-height, except for the air exit temperature. As can be seen from Table 7.3, in this nominal case the vessel wall temperature is far less than the limiting temperature, 811 K (1000 °F).

In the RECENT code, vessel wall and riser wall temperatures at any given height can be calculated. This calculation is made by specifying the heat flux at that location and the air temperature at that height calculated from the energy balance equation, Eq.(3.3) using the total integrated heat flux up to the point of interest. The code can also calculate cases having a non-uniform heat flux on the vessel wall by specifying its axial distribution. However, in the following results, all temperatures presented are at mid-height and calculated for uniform heat flux, except where otherwise specified.
Table 7.3 RECENT Results for Nominal Case

<table>
<thead>
<tr>
<th>Parameters</th>
<th>RECENT Result</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vessel Wall Temperature</td>
<td>679.62 K(763.7 °F)</td>
</tr>
<tr>
<td>Riser Front Wall Temperature</td>
<td>455.86 K(360.9 °F)</td>
</tr>
<tr>
<td>Riser Side Wall Temperature:</td>
<td></td>
</tr>
<tr>
<td>Front Node</td>
<td>434.92 K(323.2 °F)</td>
</tr>
<tr>
<td>Back Node</td>
<td>417.88 K(292.5 °F)</td>
</tr>
<tr>
<td>Riser Back Wall Temperature</td>
<td>416.85 K(290.7 °F)</td>
</tr>
<tr>
<td>Cavity Wall</td>
<td>438.03 K(328.8 °F)</td>
</tr>
<tr>
<td>Riser Air Exit Temperature</td>
<td>469.50 K(385.4 °F)</td>
</tr>
<tr>
<td>Riser Air Mass Flow Rate</td>
<td>0.05876 kg/sec</td>
</tr>
</tbody>
</table>

7.6.2 The Effect of Surface Emissivity

Surface emissivity has been identified as a parameter having a strong effect on RCCS performance. To quantify this effect, different combinations of surface emissivities have been input in the present calculations. The results are shown in Figure 7.4. As can be seen from Figure 7.4, the riser emissivity itself has little effect on the vessel temperature until its emissivity decreases to a very small value. The slope of the curve is fairly flat. The vessel emissivity, however, has a stronger effect on the vessel temperature than does the riser emissivity. For example, when riser emissivity decreases from 0.8 to 0.4 the vessel temperature only increases about 25 K, while when the vessel emissivity decreases from 0.8 to 0.4 the vessel temperature increase is more than 60 K. The reason for this is that decreasing vessel emissivity is equal to an increase of its surface thermal resistance; since
Figure 7.4   Vessel Temperatures as a Function of Emissivities
the heat flux is fixed the surface temperature must rise to overcome the higher resistance and transfer the fixed heat flux out. It should be noted that the value of the reference emissivity is also important. A 0.1 change in vessel emissivity at ~ 0.6 results in a change of only about 5 K in the vessel temperature; but the same change at lower emissivity could result in a much larger change in vessel temperature. It should be pointed out that a decrease in riser emissivity only results in a very small increase in the riser maximum wall temperature (front wall), as shown in Figure 7.5. This happens because, for a given power level, air inlet temperature, flow channel resistance, and heat transfer coefficient characteristics, the air flow rate and exit temperature are fixed. When all riser wall emissivities become lower, the only result is to redistribute the heat flux among riser surfaces, because energy has to reflect back and forth more times than for high emissivity cases. The difference among view factors is responsible for this small energy redistribution.

7.6.3 The Effect of Vessel Surface Heat Flux

The decay power level is another factor having a strong effect on the vessel and riser wall temperatures. Since the vessel surface heat flux is proportional to the decay power level, a series of vessel surface heat fluxes were used in the calculation to quantify its effect. The resulting vessel temperatures and riser maximum wall temperatures are shown in Figure 7.6. Although the power rejection capability is proportional to the 4th power of the vessel temperature, the temperature increases resulting from a heat flux increase are still significant. When heat flux increases from 5000 W/m² to 6000 W/m², a 20% increase, the vessel wall temperature increases by about 34 K, and the riser front surface temperature increases by more than 22 K. Figure 7.6 also shows that at lower power level the vessel wall temperature curve has a steeper slope than at higher power level, while the riser front wall temperature increases at an almost constant rate.
Figure 7.5  Riser Maximum Temperatures vs. Riser Emissivity
Figure 7.6  Vessel and Riser Temperatures vs. Vessel Heat Flux
The variations of vessel wall heat flux also affect the mass flow rate inside the riser, Figure 7.7. Mass flow rate increases as vessel wall heat flux increases. The mass flow rate is determined by the pressure balance between the buoyancy gain and friction loss. At lower heat flux, the buoyancy force increase is dominant, therefore mass flow rate increases at a higher rate. As mass flow rate increases, the influence of the friction force will become more important because it has an about 2nd power dependence on flow rate. Consequently, the increase in mass flow rate becomes less pronounced.

It should be noted that the vessel wall temperature behavior is connected with the riser mass flow rate behavior. As vessel wall heat flux increases, the increase of mass flow rate will result in an increase of heat transfer coefficient, as well as a lower air exit temperature. This will help to slow the riser wall temperature increase, which further slows the rate of vessel wall temperature increase.

7.6.4 The Effects of Riser Wall Conduction, Inside-Riser Radiation and Cavity Convection

Different riser surfaces receive different amounts of radiation energy from the vessel wall because of their different locations. The front wall receives much more energy than any other surface. Heat conduction within the riser walls and radiation between riser walls will redistribute the energy received. As a result of the conduction and radiation, the riser surface temperature distribution is smoothed significantly. As shown in Table 7.4, in an otherwise nominal case, the riser front wall temperature without heat conduction within the riser wall and internal radiation between riser walls is more than 60 K higher than with conduction and internal radiation. Since the other surfaces do not directly face the vessel wall, their temperatures show a moderate decrease without conduction and radiation. The heat conduction within the riser wall is dominant in terms of smoothing the riser surface temperature distribution, while the internal radiation only has a small contribution: in an otherwise nominal case, but without radiation, the riser front wall temperature is only about
Figure 7.7  Riser Mass Flow Rate vs. Vessel Heat Flux
Table 7.4 Comparisons of Temperature Results with and without Different Heat Transfer Mode Effects (K)

<table>
<thead>
<tr>
<th></th>
<th>Nominal</th>
<th>Nominal+ No-Conduction</th>
<th>Nominal+ No-Radiation</th>
<th>Nominal+ No-Conduction +No-Radiation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Riser front wall</td>
<td>455.9</td>
<td>490.2</td>
<td>458.7</td>
<td>516.7</td>
</tr>
<tr>
<td>Riser side wall:</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>front node</td>
<td>434.9</td>
<td>433.1</td>
<td>435.0</td>
<td>431.6</td>
</tr>
<tr>
<td>back node</td>
<td>417.9</td>
<td>411.7</td>
<td>416.1</td>
<td>408.9</td>
</tr>
<tr>
<td>Riser back wall</td>
<td>416.9</td>
<td>413.7</td>
<td>414.6</td>
<td>409.7</td>
</tr>
<tr>
<td>Vessel wall</td>
<td>679.6</td>
<td>684.2</td>
<td>679.9</td>
<td>688.5</td>
</tr>
</tbody>
</table>

3 K higher than the nominal case. In another case, with only heat conduction absent, the riser front wall temperature is about 35 K higher than the nominal case result.

The riser wall temperature differences between the nominal case and other cases also affect the vessel wall temperature moderately. The without-internal-radiation and without-heat-conduction case results in a 9 K higher vessel wall temperature than nominal.

The inside-cavity convection transfers about 10 -15% of the vessel heat flux to the riser wall. Since the total heat flux at the vessel wall is fixed at a given decay power level, the inclusion of cavity convection will lower the heat flux transferred by radiation, and hence lower the vessel wall temperature. Calculations of the nominal case and cases without cavity convection show, Figure 7.4, that in the former the case vessel wall temperature is about 20 K lower — a considerable decrease.

7.6.5 The Effects of Riser Heat Transfer Coefficient and Friction Factor

The possible variations of the friction factor and heat transfer coefficient in the RCCS operating regime have been assessed in Chapters 2 to 5. The impacts of variations
around their nominal values on the vessel wall temperature will be discussed in this section. The range of variations in this calculation are consistent with that discussed in earlier chapters, \( \frac{\text{Nu}}{\text{Nu}} < \text{Nu} < \text{Nu}_\text{DB} \) and \( f < f < 3f_\text{iso} \), where \( \text{Nu}_\text{DB} \) is the Nusselt number calculated from the Dittus-Boelter correlation, and \( f_\text{iso} \) is the isothermal friction factor from the Blasius correlation.

Although both a decrease in Nusselt number and an increase in friction factor will result in an increase in riser wall temperature, and an increase in vessel wall temperature, the mechanisms are different. In an increasing friction factor case, the air flow rate will decrease. Since the total decay power does not change, the air temperatures inside the riser must increase to maintain energy conservation. From the heat transfer equation,

\[
q'' = h (T_r - T_a)
\]  

(7.20)

the increase in \( T_a \) will result in an increase in riser wall temperature, \( T_r \), and hence vessel wall temperature. In a decreasing Nusselt number case, however, the air mass flow rate and air exit temperature are not changed, since the total decay power and flow resistance are the same. The air temperature inside the riser will also not be changed. From Eq.(7.20), the extent of the increase in \( T_r \) is approximately proportional to the extent of the decrease in Nusselt number.

The variations of vessel wall temperature, riser wall temperature, and other parameters as riser friction factor and heat transfer coefficient change are shown in Table 7.5. It should be noted that the Re number in the nominal case is at the higher end of the Nusselt number equation, Eq.(5.4), range. Therefore, the Nusselt number calculated from Eq.(5.4) is almost identical to the Nusselt number calculated from the Dittus-Boelter correlation. Even with tripling of the isothermal friction factor the Re still has a significant value, i.e., 15,000. At this Re number, the Nusselt number from Eq.(5.4) is still close to Dittus-Boelter value. To find the vessel wall temperature response to heat transfer
Table 7.5 Major Parameter Variations as Riser Friction Factor Increases and/or Heat Transfer Coefficient Decreases

<table>
<thead>
<tr>
<th></th>
<th>Nominal</th>
<th>Nominal + $f = 2f_{iso}$</th>
<th>Nominal + $f = 3f_{iso}$</th>
<th>Nominal + 0.7h</th>
<th>Nominal + 0.5h</th>
<th>Nominal + 0.5h + $f = 3f_{iso}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vessel Temp. (K)</td>
<td>679.4</td>
<td>693.1</td>
<td>705.8</td>
<td>684.9</td>
<td>692.5</td>
<td>731.2</td>
</tr>
<tr>
<td>Riser Front Wall Temp. (K)</td>
<td>455.3</td>
<td>494.0</td>
<td>525.3</td>
<td>472.2</td>
<td>493.4</td>
<td>580.2</td>
</tr>
<tr>
<td>Riser Air Exit Temp. (K)</td>
<td>468.0</td>
<td>523.6</td>
<td>568.8</td>
<td>467.8</td>
<td>467.7</td>
<td>568.8</td>
</tr>
<tr>
<td>Riser Mass Flow Rate (kg/sec)</td>
<td>0.0633</td>
<td>0.0460</td>
<td>0.0378</td>
<td>0.0633</td>
<td>0.0633</td>
<td>0.0378</td>
</tr>
</tbody>
</table>

Coefficient changes, a 30 to 50% percent heat transfer coefficient decrease was imposed in the calculation. As can be seen from the table, a doubling of riser friction factor will result in an about 30% decrease in riser air flow rate, and an about 14 K increase in vessel wall temperature. A tripling of $f_{iso}$ results in a more than a 40% decrease in air flow rate inside the riser, and a more than 26 K increase in vessel wall temperature. The heat transfer coefficient decrease has the same order of effect on the vessel wall temperature. A 50% decrease in heat transfer coefficient alone will result in a 13 K increase of vessel wall temperature. In the worst combined case, triple the $f_{iso}$ and half the heat transfer coefficient, the vessel wall temperature increases more than 52 K. Although the final vessel wall temperature, 731 K, does not exceed the limiting temperature, 811 K, the safety margin has been considerably reduced. It should be pointed out that the increases in friction factor discussed above all refer to the riser friction factor. Should the chimney and cold flow path flow resistance increase the same order as in the riser almost no effect on the vessel wall temperature is incurred, because the riser flow resistance is dominant. This is very important, because blockage of flow channels would most likely occur in the chimney and cold flow path, and not inside the risers.
Intentional or unintentional changes in riser wall roughness are also of interest. An increase in surface roughness not only causes the friction factor to increase, but also causes the heat transfer coefficient to increase. Analysis [W-1] for repeated-rib roughness shows that, as a rule of thumb, the effects of surface roughness will, for a tripling of the friction factor, approximately double the heat transfer coefficient. RECENT calculations for this extreme case (i.e., 3xf in Eq.(4.14), 2xNu in Eq.(5.4)) show that the vessel wall temperature increases 14 K. Hence, intentional internal wall roughening to increase heat transfer coefficient is counterproductive.

7.7 Configuration Studies

In these studies four scenarios were calculated based on variations in the nominal case. The purpose of these calculations was to seek the best geometric configuration for the riser. "Best" means to achieve a vessel temperature as low as possible. The first scenario is to keep the riser geometry the same but change its location, measured by the distance from the vessel to the riser front wall. It should be noted that when risers move closer to the vessel, fewer total riser tubes can be installed because of reduced perimeter, and each riser must receive more energy. Consequently in the analysis to follow both cases riser number changed and unchanged, are presented. In a case that riser numbers are not changed when riser location changes, the resulting trend is clear: as shown in Figure 7.8, the closer the riser to the vessel, the lower the vessel wall temperature. The vessel wall temperature is not very sensitive to the location of the riser in the region of real interest (distance between vessel and riser front wall larger than 12 inches). When the riser is close to the cavity wall, the riser location has almost no effect on the vessel wall temperature. When the riser moves a foot closer to the vessel wall than the nominal case, the vessel wall temperature decreases only about 5 K. The vessel wall temperature becomes sensitive to the distance when the riser is much closer to the vessel. In more realistic case, as mentioned above, the change in riser location should be accompanied by a change in the
Figure 7.8  Vessel Temperature vs. Riser Distance from the Vessel
total numbers of riser and the total heat transferred to a single riser. In this case, as shown in Figure 7.8, when the riser is moved from very close to the cavity wall towards the vessel, the vessel wall temperature increases slowly until the riser is at about central location, where vessel wall temperature reaches its maximum value. For further movement of the riser towards the vessel, the vessel wall temperature decreases at an accelerated rate. There are two major factors affecting the vessel temperature, total heat transferred to a single riser and the view factors between the vessel and riser surface (especially its front surface). When the riser moves towards the vessel, both total heat transferred to a riser and the view factors between vessel and riser increase. However, increasing total heat transfer that a given riser tube must extract will increase the vessel wall temperature, while increasing the view factors will decrease the vessel wall temperature. When riser is at far from the vessel wall, the effect of increase in total heat transferred is larger than that of increase in the view factors. Hence the vessel temperature shows small increase when riser moves towards the vessel. When the riser location is close to the vessel, the view factors between the riser and the vessel wall are dominant. Therefore, the vessel wall temperature exhibits a sharp decrease when the riser moves towards the vessel. However, this region is not of interest because of the need to provide room to inspect the reactor vessel (larger cavities are also not very interesting because of the higher plant construction cost incurred).

The second scenario is to fix the riser flow area. By changing the ratio of width to length, W/L, different vessel and riser maximum temperatures were obtained. It should be noted that when W/L changes at fixed flow area, the riser hydraulic diameter and its total heat transfer area also changes. The riser flow area $A_f$, total heat transfer area, $A_{HT}$, and hydraulic diameter $D_e$ can be calculated as follows for the riser, when the riser wall thickness is neglected,

$$A_f = W L$$  \hspace{1cm} (7.21)
\[ A_{HT} = 2L (W/L + 1) \]  \hspace{1cm} (7.22)

and

\[ D_e = 2W/(W/L + 1) \]  \hspace{1cm} (7.23)

Since \( W/L \) is always less than 1, decreasing \( W/L \) will cause the hydraulic diameter to decrease and the total heat transfer area to increase in the range of interest. The calculated results are shown in Figure 7.9. Although the vessel wall temperature is not very sensitive to the \( W/L \) ratio (an overall 10 K difference), it does show a general trend. Starting on the right at \( W = 3.75 \) inches, decreasing \( W/L \) causes the vessel wall temperature to decrease, because the effect of increasing the riser heat transfer area is larger than the effect of decreasing hydraulic diameter since \( A_{HT}/D_e = (W/L + 1)^2/(W/L) \). The vessel wall temperature continuously decreases until \( W = 1.5 \) inches, where the vessel wall temperature starts to increase. The reason for this increase is that the effect of a further decrease in hydraulic diameter causes a sharp reduction in the air flow rate. Since the total power does not change, the air temperature inside the riser increases. This effect is larger than the effect of increasing the riser total heat transfer area. Therefore, the vessel wall temperature increases.

The third scenario is to fix the riser perimeter (hence heat transfer area) while changing the ratio of the width to the length of the riser, \( W/L \). It should be noted that when \( W/L \) changes at fixed perimeter, the riser flow area and hydraulic diameter increase. The calculated results are shown in Figure 7.10. As can be seen from the figure, at \( W > 2.5 \) inches, the vessel wall temperature is not sensitive to the changes of \( W/L \): it only changes 1 - 2 K. When \( W < 2.5 \), the vessel wall temperature becomes more and more sensitive to the \( W/L \) ratio. The smaller the \( W/L \), the higher the vessel wall temperature. The reason for this rapid increase in vessel wall temperature is the decrease in the hydraulic diameter and flow area as \( W/L \) decreases.
Figure 7.9  Vessel Temperature vs. W/L at Fixed Riser Flow Area
Figure 7.10  Vessel Temperature vs. W/L at Fixed Riser Perimeter (Heat Transfer Area)
The second and third scenario calculation results suggest that the larger the total flow area and heat transfer area, the better the riser design. To quantify the effect of increases in flow area and heat transfer area, a fourth scenario was examined: fixed width and variable length. The results, as shown in Figure 7.11, confirm that the larger the flow area and heat transfer area, the better the riser design, since the corresponding change in hydraulic diameter is smaller than the changes in flow area and heat transfer area.

From the above four scenario calculations, one can conclude that the riser location does not affect riser performance very much; and that the best riser width is between 1.5 to 2.5 inches, with the largest flow area, hydraulic diameter and total heat transfer area practicable. Since the effect of the hydraulic diameter is only significant when it is small, the preferable choice would be to keep it reasonably large (e.g. larger than 0.05 m), and choosing the best combination of flow area and total heat transfer area. This can be achieved by making the riser length perpendicular to the vessel wall as large as practicable.

7.8 Chapter Summary

The major objective of the current calculations was to identify the parameters which have a strong effect on RCCS performance, so that the optimal RCCS system design can eventually be achieved by adjusting these important parameters. These parameters can be summarized as follows:

1) The decay power level has a strong effect on the vessel wall temperature. The vessel wall emissivity has a moderate effect on the vessel wall temperature as long as its values is above 0.6. The effects becomes stronger at values lower than 0.6. Compared with vessel wall emissivity, the riser wall emissivity has a small effect on both vessel and riser wall temperatures.

2) Heat conduction within the riser wall and the inside-riser radiation between the riser walls can significantly smooth the riser wall temperature distribution. It also affects the vessel wall temperature indirectly. Among these two heat transfer modes, heat
Figure 7.11  Vessel Temperature as a Function of Riser Length at Fixed Width
conduction within the riser wall has a much greater effect than radiation in smoothing the riser wall temperature distribution.

3) Larger variations of the riser heat transfer coefficient and friction factor within the possible riser operating regime have a strong effect on the vessel wall temperature, and hence RCCS performance; likely variations have only moderate effects.

4) The riser location inside the cavity has little effect on the vessel wall temperature for distances of real design interest. However, the shape of the riser has a moderate effect on the vessel temperature, as long as extreme variations are avoided. The best choice appears to be a hydraulic diameter larger than 0.05 m, riser width between 1.5 to 2.5 inches, and a combination having the largest air flow area and total heat transfer area. This can be achieved by increasing the riser length perpendicular to the vessel wall. The nominal riser design (\(D_e = 0.0706\) m, \(W = 0.0508\) m and \(L = 0.254\) m) studied here satisfies the first two criteria, and it is not far from the last.
CHAPTER 8
IMPACT OF THE PRESENT STUDIES ON MHTGR RCCS DESIGN

8.1 Introduction

The function of the MHTGR RCCS design is to protect the public by averting the release of radioactivity into the environment under any circumstances by limiting core fuel temperatures, and to protect owner investment by limiting the peak reactor vessel wall temperature. In a well designed system, meeting the vessel wall temperature limit (1000°F) will also insure that the core center line temperature as within its limit. Consequently, the vessel wall temperature was focused on as the criterion to judge the importance of variations in parameters. There are three group of parameters which have potential impacts on the RCCS performance based on this criterion: the first group is the parameters governing the convective heat transfer inside the riser, such as flow resistance and heat transfer coefficient; the second group is the parameters governing the heat transfer from the vessel wall to the riser, such as vessel wall emissivity; the third group is the parameters which affect both the convective heat transfer inside the riser and the heat transfer from the vessel wall to the riser. The third group includes the riser configuration and geometric parameters. The importance of these parameters to RCCS performance will be assessed in the following section.

8.2 Riser Heat Transfer Coefficient and Air Flow Resistance

The riser heat transfer coefficient and friction factor behavior within possible RCCS operating regimes have been experimentally studied and discussed in Chapters 2 to 5. Their impact on the vessel wall temperature was discussed in Chapter 7.
For the heat transfer coefficient, the experiment conducted has covered and even exceeded the likely RCCS operating range in terms of Reynolds number and Grashof number, Grq^n. The important result is embodied in Eq. (5.4), which is repeated here,

$$\text{Nu} = \frac{\text{Nu}_{DB}}{\left[ 1 + \left( \frac{4200}{\text{Re}} \right)^3 \right]} = 0.023 \frac{\text{Re}^{0.8}}{\left[ 1 + \left( \frac{4200}{\text{Re}} \right)^3 \right]} \text{Pr}^{0.4}$$

(5.4)

The above equation shows that in its applicable range 5000 < Re < 30,000, the biggest Nu reduction is by about half of the Nu number calculated from the Dittus-Boelter correlation, which occurs at the lower end of the Reynolds number. Since the nominal operating condition of the RCCS system is at the middle and upper part of the Reynolds number range (Re ≥ 10,000), the use of a forced convection Nusselt number correlation, such as the Dittus-Boelter correlation, will not cause a significant overestimate of the heat transfer coefficient. However, as shown in Figure 5.16, which plots Eq. (5.4) and the Dittus-Boelter correlation, a 30% uncertainty in the use of the Dittus-Boelter correlation must be considered. Even when the Reynolds number decreases to the lower end of the range (how it can be decreased will be discussed later), the attendant 50% or so decrease in the heat transfer coefficient will only affect the vessel wall temperature moderately, as already shown in Chapter 7. Therefore, the potential change in the riser heat transfer capability due to mixed convection will affect RCCS performance, but it seems pose no great threat to the RCCS functioning. Flow resistance changes along the air flow path will change the RCCS Reynolds number. An increase in flow resistance will cause a Reynolds number decrease. Large increases in flow resistance may cause the RCCS to operate in the mixed convection regime with the accompanying heat transfer coefficient decrease. Flow resistance increases may arise from several causes, such as channel blockage, increased surface roughness, and mixed convection effects. With respect to mixed convection effects the riser mock up
experiment results shows that they will not change the friction factor appreciably within the RCCS operating regime. For example, an increase of only 10% has been observed when channel average properties are used in the computation. Surface roughness may have a bigger effect on the friction factor than that of mixed convection effects in the RCCS operating regime. However, unlike the mixed convection effect, the increase in friction factor due to surface roughness is accompanied by an increase in heat transfer coefficient, rather than a decrease. Therefore, surface roughness may even be beneficial. Further study on the connection between surface roughness and the friction factor and heat transfer coefficient is needed to reach a decisive conclusion.

Flow channel blockage is a potential cause of large increases in channel flow resistance, and hence can shift the RCCS operating point towards mixed convection. Consequently, the RCCS may operate at a smaller flow rate and lower heat transfer coefficient, and therefore much higher vessel wall temperature. However, in the current RCCS design, namely several parallel chimneys and cold flow paths and more than 200 riser tubes, the flow resistance of the riser tubes is dominant. On the other hand, from the design configuration, blockage would most likely occur in the cold flow paths or chimneys. The calculation results in Chapter 6 show that even when the flow resistance of the cold flow paths and chimneys increases by a factor of 5, it has almost no effect on the vessel wall temperature (less than 1 K). Therefore, unless all parallel flow paths are blocked, the blockage outside the riser section will have little impact on RCCS performance. Conversely, an increase in riser flow resistance can have a significant impact on RCCS performance. Since the riser tubes are located inside the cavity, in the middle of the flow path and consisting of more than 200 parallel channels, severe blockage is not considered to be credible. Even with blockage sufficient to triple the riser flow resistance, the calculations in chapter 6 shows that the vessel wall temperature is still well below the limiting temperature. The lack of sensitivity to the number of risers evident in Figure 7.8
of Chapter 7, also applies against any important effect of total blockage of a number of risers.

In the above discussions, the riser mockup experiment has been used as a guideline for specifying the Nusselt number and friction factor variations. The Reynolds number, Re, and Grashof number, Grq", have been used to justify the coverage by the current experiment on possible RCCS operating regimes. Although Re and Grq" are two parameters widely used for such characterization, there are other aspects that must be considered, particular in the current case, before a conclusive assertion that the experiment does indeed convey all possible variations of potential RCCS operating conditions. As mentioned earlier, the experiment has been designed to have at least the same total power input and temperature rise. The latter is necessary because of the large variation in air properties and their effect on the heat transfer coefficient and friction factor. In addition, the geometric difference between the real riser and its "mockup" must be addressed. The choice of a cylindrical 2.5 " ID pipe for the experiment is based on achievement of a similar hydraulic diameter. However, the flow areas and total heat transfer areas between the experiment and RCCS design are different. The real riser has about 3 times larger flow area and heat transfer area than the experimental mockup. The larger heat transfer area means a lower heat flux, and therefore a lower Grq" on the riser wall. This will make the mockup experiment results conservative. Since a circular tube has the smallest surface area given the same hydraulic diameter, to keep the Reynolds number,

\[ \text{Re} = \frac{GD_e}{\mu}, \]

the same, the experiment only needs to supply about 1/3 of the mass flow rate as the real riser, since it only has about 1/3 of a riser's flow area. Because the current experiment has covered the entire RCCS operating Re range, from forced convection to possible mixed
convection, and the purpose of this experiment is to explore the Nusselt number and friction factor behavior in the mixed convection regime, further increasing the experiment's mass flow rate, hence its Re number, will push the experiment further into forced convection, and is not necessary. It turns out that the riser mockup experiment is also conservative from the air temperature rise point of view. With full power input, but only 1/3 of mass flow rate, the air temperature rise in the experiment exceeds the real riser design values, therefore, the large air property change effect due to temperature changes will be fully covered.

In conclusion the riser mockup experiment has covered every thermal-hydraulic aspect of possible RCCS riser operations and has revealed the corresponding heat transfer and friction factor behavior. The analysis using the experiment's results, and design information, shows that there seems to be no easy way to shift the current RCCS into the deep mixed convection regime, unless certain design changes are made (which will be discussed later in Section 8.4). Based on the current design, the riser air convection heat transfer will provide adequate cooling to maintain the reactor safety and investment protection, with a large margin.

8.3 Heat Transfer from the Vessel Wall to the Risers

The heat transfer from the vessel to the risers is through radiation and cavity convection. Since the cavity convection only has a small contribution, it will not have a significant impact on the vessel wall temperature, and hence the RCCS performance. The analysis of the governing parameters in the radiation heat transfer from the vessel to the riser is straightforward. The radiation heat flux is proportional to the vessel emissivity, the view factor from the vessel to the riser, and the 4th power of vessel wall temperature and riser temperature difference. At a given heat flux, the riser wall temperature is also fixed, and so the vessel wall temperature is only determined by the emissivity and the view factor. The change in view factor between the vessel and the risers is accompanied by changing
riser geometry and location, which will be discussed in the next section. The effect of changing vessel emissivity on the vessel wall temperature has been assessed in Chapter 7, which shows that it only has a moderate effect as long as its value is higher than 0.6. In practice, it should be easy to fabricate a steel reactor vessel having an emissivity higher than 0.6. Therefore variations of vessel emissivity within design allowances will not have a significant impact on RCCS performance.

It is interesting to note that the 4th power dependence of the radiation heat transfer rate on the vessel wall temperature makes this process, in the series of decay heat transfer processes, act as a buffer. When significant variations of other parameters interact with this buffer, their effect on the vessel wall temperature is greatly reduced. The significance of this buffer region on the design performance of the RCCS is that once the system is built, it is difficult to either worsen or improve it.

8.4 Riser Geometry and RCCS Configuration

A change in riser geometry or its location will affect both convective heat transfer inside the riser and the radiation heat transfer within the cavity. Any design change must consider the effects on both sides of the risers walls carefully.

Inside the cavity, the riser geometry or location variation will affect the radiation heat transfer though the changes in view factors, and in total heat transferred to a single riser if the number of riser tubes is changed. Because of the limited space inside the cavity, changing the number of risers is limited to a rather narrow range, and hence the change in the total heat transferred to a single riser will not be significant. Their impacts on the vessel wall temperature is limited. The effect of view factor changes on the vessel wall temperature, as presented in Chapter 7, is only significant when the risers are moved much closer to the vessel. However, this region is not available in the real design because it is reserved for the reactor vessel inspection. Otherwise we could elect the optimizing case of direct riser/vessel contact. Increasing the width of the riser will shadow the riser sidewalls,
and does not yield much benefit as can be seen in Figures 7.9 and 7.10. Reducing riser width to a value less than 1.5 inches may sharply reduce the front wall contribution (which usually is dominant), hence cause the vessel wall temperature to sharply increase, Fig. 7.10. Therefore, from an inside-the-cavity radiation heat transfer point of view, as long as extremes are avoided, no parameters have a significant impact on RCCS performance.

Riser geometry changes affect the inside-riser convective heat transfer through the changes in total heat transfer area, flow area and riser hydraulic diameter. An increase in total heat transfer area will lower the riser surface heat flux, and therefore lower the riser wall temperature, which in turn decreases vessel wall temperature. This effect has been shown in Figure 7.11. The changes in flow area and hydraulic diameter, unlike the change in total heat transfer area, affect the air flow rate through changes in flow resistance. If the flow rate is greatly reduced, it may cause the RCCS operating condition shift into the mixed convection regime. However, the hydraulic diameter is only significant when it is small (< 0.05 m), so also the flow area. Therefore, selection of a riser having a reasonable width and flow area can avoid this extreme. Increases in flow area and hydraulic diameter, on the other hand, will increase the air mass flow rate, decrease air temperature, and hence reduce vessel wall temperature. However, this effect is an indirect one as regards vessel wall temperature, and not likely to be a significant effect.

In general, when the extreme conditions are avoided such as a very narrow riser (W<0.05m) or very small flow area, geometric changes in riser design have no significant impact on RCCS performance. Nevertheless, by increasing the riser total heat transfer area and flow area, a moderate vessel wall temperature decrease can be achieved.

8.5 Chapter Summary

The analysis above and in earlier chapters show that the riser convection heat transfer coefficient and flow resistance have the potential affect RCCS performance significantly. A mixed convection regime exists where the heat transfer coefficient is
considerably reduced. To shift the present RCCS operating condition into the deep mixed convection regime, the riser flow resistance has to be increased by at least 3-fold. It is this increase in riser flow resistance, and consequently the decrease in heat transfer coefficient, that causes vessel wall temperature to sharply increase. The effect of an independent increase in flow resistance, or a decrease of heat transfer coefficient inside the riser on the vessel wall temperature is far less than their combined effect. However, evaluation of the present RCCS design does not suggest a credible way to cause an increase in riser flow resistance of the magnitude required to cause significant performance degradation. Nevertheless, the above trends should be kept in mind in making design changes.

The other parameters, such as vessel surface emissivity and riser geometry, do not have a significant impact on RCCS performance as long as extreme cases are avoided. However, carefully choosing these parameters can moderately reduce the vessel wall temperature, and therefore is beneficial.
CHAPTER 9

SUMMARY, CONCLUSIONS AND RECOMMENDATIONS

9.1 Introduction

The Modular High Temperature Gas-Cooled Reactor (MHTGR) is one of the next generation power reactors currently being developed in the United States under Department of Energy sponsorship. A major emphasis in the development of these reactors is improvement of safety margins by use of passive safety systems. A feature of the MHTGR design, shared in principle by the liquid metal cooled PRISM reactor design, is the use of a non-insulated steel reactor vessel and an array of riser tubes in the reactor cavity to provide a passive decay heat removal path from the core to the ambient air as illustrated in Figure 9.1.

The design of this Reactor Cavity Cooling System (RCCS) requires attention, not only to its engineering aspects, but also to the fundamental heat transfer and fluid flow processes involved. This requires characterization of the heat transfer coefficient and the friction factor inside the riser tube in the operating regime, which involves a condition intermediate between forced and free convection, i.e., mixed convection—a regime not yet well understood. Accordingly, one major objective of the present work was to determine the heat transfer coefficient and friction factor inside the riser under its projected operating conditions. The second major task was to carry out a system performance analysis and optimization. The organization of the present work is shown in Figure 9.2. The work can be summarized under the following two general headings.

PART I  HEAT TRANSFER AND FRICTION FACTOR BEHAVIOR IN THE MIXED CONVECTION REGIME FOR AIR UP-FLOW IN A HEATED VERTICAL PIPE
Figure 9.1 Reactor Cavity Cooling System (RCCS)
Figure 9.2 Summary of Present Work
9.2 General Characterization of Heated Duct Flow

A useful characterization of riser performance can be developed by examining the momentum balance for upward flow in a vertical heated duct.

\[
\frac{\partial (\rho w^2)}{\partial z} = -\frac{dp}{dz} + \frac{1}{r} \frac{\partial}{\partial r} \left( r (\mu + \mu_i) \frac{\partial w}{\partial r} \right) + \rho_o \beta g (T - T_o) \tag{9.1}
\]

Averaging over the cross section area, \( A_c \), and assuming \( dp/dz \) is constant across this cross section, one gets

\[
\left( \frac{dp}{dz} \right)_{acc} = -\left( \frac{dp}{dz} \right)_t + \left( \frac{dp}{dz} \right)_b + \left( \frac{dp}{dz} \right)_f , \tag{9.2}
\]

where

\[
\left( \frac{dp}{dz} \right)_{acc} = \frac{1}{A_c} \int_A \frac{\partial (\rho w^2)}{\partial z} \, dA , \quad \left( \frac{dp}{dz} \right)_t = \frac{1}{A_c} \int_A \frac{dp}{dz} \, dA ,
\]

\[
\left( \frac{dp}{dz} \right)_b = \frac{1}{A_c} \int_A \rho_o \beta g (T - T_o) \, dA .
\]

and

\[
\left( \frac{dp}{dz} \right)_f = \frac{1}{A_c} \int_A \frac{1}{r} \frac{\partial}{\partial r} \left( r (\mu + \mu_i) \frac{\partial w}{\partial r} \right) \, dA
\]

\[
= -\frac{1}{A_c} \left( 2\pi R \right) \mu_w \left( \frac{\partial w}{\partial y} \right)_w = -\frac{2\pi R}{A_c} \tau_w ,
\]

since \((\mu_i)_R = 0\) and \( \left( \frac{\partial w}{\partial r} \right)_R = -\left( \frac{\partial w}{\partial y} \right)_w \).

Assuming that the fluid properties are constant (and taken at reference temperature, \( T_o \)), except for density in the buoyancy term, that the flow is fully developed (the acceleration term is neglected) and applying the continuity condition, integration of Equation (9.1) over the length of the duct, \( L \), yields:

\[
0 = \Delta p_t - \frac{2\pi R}{A_c} \tau_w L + \beta \rho_o g \int_0^L \Delta \bar{T} \, dz , \tag{9.3}
\]
where
\[ \Delta p_T = p_{in} - p_{out} \text{ and } \Delta \bar{T} = \frac{1}{A_e} \int_A (T - T_o) \, dA. \]

Now applying the definition of the skin friction factor and the Blasius correlation,
\[ \tau_w = c_f \frac{1}{2} \rho_o w_0^2 = \frac{L}{8} \rho_o w_0^2 = 0.316 \text{Re}^{-1/4} \frac{\rho_o w_0^2}{8}, \]
and evaluating \( \Delta \bar{T} \) from the energy equation as
\[ \Delta \bar{T} = \frac{q'' \pi D z}{c_p \dot{m}} = \frac{4q''}{c_p \rho_o w_0} \left( \frac{z}{D} \right), \]
Equation (9.3) can be written in the following dimensionless form:
\[ \Delta \bar{p}_t = \frac{2 \text{Gr}_q \left( \frac{L}{D} \right)^2}{\text{Re}^3 \text{Pr}} + \frac{0.158 \left( \frac{L}{D} \right)}{\text{Re}^{1/4}}, \quad (9.4) \]
where
\( \Delta \bar{p}_t \) is \( \Delta p_t \) normalized by \( \rho_o w_0^2 \).

Equation (9.4) can be used to demonstrate several possible riser configurations. If both ends of the duct are open to ambient air and the slight atmospheric static head difference is neglected, \( \Delta \bar{p}_t = 0 \) and the riser creates its own flow, thereby operating in what can be termed the "pure" natural convection regime. In this condition the operating \( \text{Re} \) number from Equation (9.4) is
\[ \text{Re} = \left[ \frac{\text{Gr}_q \left( \frac{L}{D} \right)^2}{0.039 \text{Pr}} \right]^{4/11}. \quad (9.5) \]
Our heated riser of length, \( L \), can be modified by the addition of a pump or a throttling valve. If the duct flow is "pumped", \( \Delta \bar{p}_t > 0 \), and the \( \text{Re} \) number will be greater than that of Equation (9.5). On the other hand, if the duct is throttled by a valve, \( \Delta \bar{p}_t < 0 \), the \( \text{Re} \) number will be less than that of Equation (9.5).

The heated riser can also be pumped by an unheated chimney of length, \( L_c \). In this case, Equation (9.4) becomes:
\[ \Delta \bar{p}_t = 0 = \frac{2 \text{Gr}_q}{\text{Re}^3 \text{Pr}} \left( \frac{L}{D} \right) \left( \frac{L + 2L_c}{D} \right) + \frac{0.158 \left( \frac{L}{D} \right)}{\text{Re}^{1/4} \left( \frac{L + L_c}{D} \right)}, \quad (9.6) \]
which accounts for the buoyant and friction components of the pressure change in the chimney. Figure 9.3 illustrates these modes of riser operation.

The reactor cooling system of Figure 9.1 utilizes an unheated inlet section, an unheated chimney section and a heated riser whose performance will be considered next.

9.3 Performance Range of the Reactor Riser Tube

The RCCS is designed as the ultimate barrier to prevent reactor core and vessel damage for severe accidents. The loss of coolant accident in which the reactor system can be either depressurized or pressurized is one of the most severe accidents in terms of the maximum core and reactor vessel temperature attained. In a loss of coolant accident, the decay heat generated inside the core will all be either deposited in the graphite moderator or transported to the ambient air through the RCCS. To predict the riser operation conditions, namely mass flow rate and outlet air temperature, one must know the decay heat generation rate as a function of the time and what portion of that decay heat is deposited in the graphite moderator. We will use the data at about 100 hours after the reactor has been shut down as the reference case, to approximate the maximum core, reactor vessel, riser wall temperature, riser outlet air temperature and air flow rate, although these maxima do not in reality all occur at exactly the same time.

The extreme of the operating regime of the reactor system can be characterized by an inlet Re of $2.2 \times 10^4$ and a riser Grashof number, $Gr_q$, of $5 \times 10^7$. The regime can be conservatively bounded by noting that, for lower powers, this self-driven riser system cannot achieve higher Re, while, for this maximum power, lower Re can only result from increased flow resistance. Such resistance could arise from an unintentional channel blockage. Assuming an increase in friction factor of about a factor of 3 yields a lower Re bound of $10^4$. This bounded operating regime is illustrated on Figure 9.4.

The determination of actual riser operating conditions within this operating regime requires simultaneous solution of the conservation equations, with assumptions for the heat
A) "Pure" natural convection
B) Pumped
C) Throttled
D) With an adiabatic chimney working as a pump.

Figure 9.3  Modes of Riser Operation
Figure 9.4  Experiment Range compared to MHTGR Operating Regime
transfer coefficients and fraction factor. Figure 9.4 also illustrates values of the
dimensionless buoyancy number, Bo, which Jackson and his co-workers [C-3] have
defined and utilize as a measure of degree of mixed convection:
\[ Bo = \frac{8 \times 10^4 Gr_q}{Re^{3.425} Pr^{0.8}} \]
Taking the value of Bo = 0.01 as a conservative criteria for onset of significant mixed
convection effects, we observe that a portion of the riser operating regime may be in mixed
convection. This is of design significance, since heat transfer coefficients and friction
factors can vary significantly from forced convection values under mixed convection
conditions.

We next assess the relevant mixed convection correlations and define the
experimental conditions for which new design data is needed.

9.4 Relevant Correlations in the Literature

A thorough review of the mixed convection literature has been made in Chapter 2.
Correlations for Nusselt number proposed by Jackson and co-workers [C-3], Petukhov
and Polyakov [P-1] and Connor and Carr [C-2] have been assessed. All reflect the fact that
in an aiding mixed convection flow, two types of velocity profiles can occur. As shown in
Figure 9.5, for the Type I profile, the velocity near the wall has increased due to the
buoyancy force, but the maximum flow velocity is still at the pipe center (lines 1-3 in
Figure 9.5). For the Type II profile, the buoyancy effect is so large that the maximum
velocity is no longer at the center of the pipe, but at some location between the wall and the
\[ \text{stem (line 4 in Figure 9.5).} \]

Figure 9.5 illustrates the mechanism for heat transfer behavior proposed by Hall and
Jackson [H-2]. Specifically, as the buoyancy effect increases, the turbulent shear stress
decreases and the Nusselt number decreases until a point where turbulent shear stress
production is close to zero. At that point laminarization occurs and the Nusselt number

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Figure 9.5 Effect of Buoyancy on Velocity, Shear Stress and Heat Transfer for Aiding Turbulent Flow (from Symolon [S-3])
reaches its minimum value. Further increase of the buoyancy effect will make the velocity profile change from Type I to Type II and the turbulent shear stress become negative. However, the absolute value of this turbulent shear stress production starts increasing, causing the heat transfer rate to also start improving from its minimum value.

Jackson's correlation as revised in [C-3] was selected for focus since it is based on the most comprehensive study currently available. The correlation is

\[
\frac{Nu}{Nu_{fe}} = \left(1 - \frac{Bo}{\left(\frac{Nu}{Nu_{fe}}\right)^2}\right)^{0.46}
\]  

(9.7)

Note that the term in the brackets on the right hand side of Equation (9.7) has been taken as the absolute value, which makes the correlation capable of describing both Type I and Type II flows. The correlation was also compared by Jackson with available air experiment results by Steiner [S-5] and Carr, et al. [C-1] in the fully developed region.

Compared to the heat transfer behavior in turbulent mixed convection, the friction factor has received much less attention. Neither a satisfactory theory nor a modest data set exist for friction factor in mixed convection. A key reason for this is the difficulty in measuring the friction factor either from the velocity gradient at the wall or from direct friction pressure drop measurements.

Petukhov and Polyakov [P-1] employed a theory similar to the Reynold's analogy principle in which they predict similar behavior between the Nusselt number and the friction factor in a turbulent aiding flow. Their correlation is presented since it is recently published and presumably is based on the extensive long term effort in this field of the authors. The correlation for calculating the friction factors in a turbulent aiding flow is

\[
f = \left[\frac{1 + 0.83e^2}{1.82\log_{10}(Re/8) + 0.076e^2E^{-0.25}}\right]^2
\]  

(9.8)

where

\[
e = \frac{Grq \times 10^3}{Pr \cdot Re^{2.75}} \quad \text{and} \quad E = \frac{Grq}{Pr \cdot Re^4} \, .
\]
Their forced convection friction factor correlation is

\[ f = \left[ 1.82 \log_{10} \left( \frac{Re}{\delta} \right) \right]^2 \]  

(9.9)

In this assessment heat transfer and friction results from a companion project is also presented, which has developed a code, CONDOR, for mixed convection analysis based on the \( k-\epsilon \) model [Y-1].

These literature results for both the Nusselt number and friction factor show deleterious trends for the mixed convection regime. Specifically, for modest Bo (0.1 to 1.0) the flow laminarizes and the Nusselt number decreases to values which approach 50% of the comparable forced convection values. For friction factor, while the evidence is less extensive, after a modest decrease for conditions below Bo \( \sim 0.3 \), friction factors can increase to levels a factor of 3 above comparable forced convection values for Bo reaching values around 5.

9.5 Experimental Design and Operating Procedures

The uncertainties in mixed convection performance suggested that experiments should be performed to bracket the expected regime of reactor riser operation. An experiment was designed for this purpose, which covers the experimental range illustrated in Figure 9.4. The experiment also achieves at least the same air temperature rise, about 140°C, as the design case experiences, so that air property change effects will be reflected in our test results. The experimental constraints limiting operation of the current apparatus are:

- Lower Re of 5000 due to error in measured air flow rate using the standard ASME orifice.
- Upper Re of 33000 due to the maximum \( \Delta p \) measurable by the pressure transducer.
- Maximum wall temperature of 400°C due to limitation of the thermocouple insulation.
- Maximum power of 4000 W, consistent with the wall temperature limitation imposed by the thermocouples and the available laboratory power supply.

The design of the experiment is a compromise among the need to represent key features of the real system, the desire to create a readily interpretable experiment, and practical restrictions on cost and size. The overall apparatus is a once-through system with ambient air as the flow medium. The "riser" in the present investigation is a 2.5 inch ID commercial seamless cylindrical carbon steel pipe (instead of a rectangular slot, as in the present MHTGR design). It has a 28-3/12 foot (135.6 \( L/D \)) heated section, which is heated uniformly by helically-wound wrap-around electrical heating tapes at a pitch of 2.75\(^\text{"} \), and a 5.5 foot (26 \( L/D \)) inlet unheated section incorporating a flow straightener, see Figure 9.6. The total height is approximately half that of a real riser. The pipe is insulated by an inner layer of 0.5 inch thick fiber glass insulation, a 2.5 inch thick thermo-12 calcium silicate insulation at the middle and an outer layer of 1.0 inch thick fiber glass insulation. The insulation is covered by a PVC pipe to prevent rain damage, since the pipe is located out-of-doors. The heater power is provided by a 208 V single phase power line. A variac is used to control the input heating power in the range of 0 - 240 V and 0 - 17 A, hence 0 - 4 kW. Air flow is provided by a centrifugal blower. The flow rate is controlled and adjusted using an in-line control valve.

The directly measured parameters are air flow rate, total pressure drop, heating power (from current and voltage) and temperatures. The air flow rate is measured using an ASME standard orifice [L-1]. A pitot-tube is used as a secondary monitor. The total pressure drop is measured between two wall taps using a commercial differential pressure transducer manufactured by MKS Instrument, Inc., based on the measurement of electric capacity. This pressure transducer has a full scale range of 1 mm Hg (133 Pa); the minimum measurable differential pressure drop depends upon the system stability. One of the pressure taps is located at the beginning of the heated section and the other is six inches from the end of the heated section. The heater power measurement is simple: an ammeter
Figure 9.6  Schematic of Riser Mockup
and voltmeter pair gives P=I*V. However, the actual power input to the flowing air is the
difference between total input power and heat loss through the insulation. This total heat
loss has been determined by running the experiment with the entrance and exit blocked and
insulated. There are 14 T-type thermocouples imbedded in the outer wall of the pipe to
monitor the wall axial temperature profile. The glass insulation on the thermocouples
restricts the maximum wall temperature to 400°C. The thermocouples start where heating
begins; successive thermocouples are positioned every two feet thereafter, each rotated 90°
circumferentially from the preceding one.

The temperatures and pressure drop are collected using a computerized data
acquisition system, the Hewlett Packard HP-3852A. All instruments used in this
experiment were calibrated.

The experimental runs are rather simple to conduct. Heater power is adjusted to the
level designated using a variac, and air flow is adjusted using the in-line valve. The wall
temperatures and the pressure drops are monitored at an IBM-XT computer using self-
programmed software through the data acquisition system HP-3852A. The steady state
situation is considered adequately approached when wall temperature drift with time is less
than 1°C per 30 minutes.

The major parameters we are seeking are the friction factor and the Nusselt number.
Since the measured parameters are the air flow rate, the wall and inlet air temperatures and
the total pressure drop, the Nusselt number and friction factor must be calculated from
these measured parameters.

**Nusselt Number**

Since the wall heat flux can be found from the difference between the total heat
input and the heat loss through the insulation, the local Nusselt number can be obtained
easily as
\[ \text{Nu} = \frac{(q^\text{nom} - q^\text{loss}) D}{k (T_w - T_{a})}. \quad (9.10) \]

The air stream bulk temperature, \( T_a \), is calculated from the energy balance as follows:

\[ T_a(z) = \frac{1}{mc_p} \int_0^z (q^\text{nom} - q^\text{loss}) \, dz + T_{in}. \quad (9.11) \]

This Nu number will be compared with the forced convection Nu number calculated using the Dittus-Boelter correlation

\[ \text{Nu} = 0.023 \, (\text{Re})^{0.8}(\text{Pr})^{0.4}, \quad (9.12) \]

where Re will be evaluated at the local air bulk temperature and Pr is taken as constant, 0.7, since it is virtually unchanged in the temperature range of the experiment.

**Friction Factor**

In the current experiment the friction factor is determined from measurement of the total pressure drop. Therefore, the results will unavoidably combine developing and fully developed contributions, yielding a channel-averaged friction factor. Define this channel-average friction factor as

\[ \bar{f}_1 = \frac{\int_0^L \frac{f}{\rho} \, dz}{\int_0^L \frac{dz}{\rho}}. \quad (9.13) \]

From the relation

\[ \Delta p_f = \int_0^L \frac{fG^2}{2D\rho} \, dz, \quad (9.14) \]

for constant \( G \) one has

\[ \bar{f}_1 = 2D \Delta p_f \left/ \left( G^2 \int_0^L \frac{dz}{\rho} \right) \right. \quad (9.15) \]

Since the total pressure \( \Delta p_t \) was measured instead of \( \Delta p_f \), Equation (9.15) becomes

\[ \bar{f}_1 = 2D \left( \Delta p_t + \Delta p_b - \Delta p_{acc} \right) \left/ \left( G^2 \int_0^L \frac{dz}{\rho} \right) \right. \quad (9.16) \]
The buoyancy head, $\Delta p_b$, can be evaluated as follows:

$$\Delta p_b = \int_0^L [\rho_{in}(T_{in}) - \rho(T_a)] \ g \ dz . \quad (9.17)$$

The air density, $\rho$, can be expressed by treating air as a perfect gas, as:

$$\rho = \frac{353.12}{T(K)} \text{, kg/m}^3 . \quad (9.18)$$

The acceleration pressure drop is calculated as follows:

$$\Delta p_{acc} = G^2 \left[ \frac{1}{\rho_e(T_e)} - \frac{1}{\rho_{in}(T_{in})} \right] . \quad (9.19)$$

In Equation (9.13) the definition of the friction factor involves integrals. The fluid density as a function of location is not known before the flow rate is calculated. However, the flow rate depends on the friction factor. This makes the Equation (9.13) definition difficult to use in engineering applications.

Consequently, in engineering applications the channel average friction factor is typically defined in terms of a reference density $\rho_0$, such that Equation (9.16) becomes:

$$\overline{f_2} = \frac{2\rho_0 D (\Delta p_t + \Delta p_b - \Delta p_a)}{L G^2} \quad (9.20)$$

Note that $\overline{f_2}$ is very similar to the apparent friction factor in an isothermal flow. The evaluation of the reference density in Equation (9.20) will be discussed subsequently, when the experimental results are presented.

9.6 Experimental Results

The friction factors and the Nusselt numbers are the major results sought. The wall temperatures are also of interest. In addition, the changes of the dimensionless numbers, Re, Gr and Bo, along the channel height will be presented, since their behavior is related to the local Nusselt number behavior. Table 9.1 summarizes the experimental conditions investigated, which are illustrated in Figure 9.4.
<table>
<thead>
<tr>
<th>Run</th>
<th>Bo (Inlet/Exit)</th>
<th>$T_{\text{air}}$ (°C) (Inlet/Exit)</th>
<th>$T_{\text{w}}$ (°C) (Inlet/Exit)*</th>
<th>Re (Inlet/Exit)</th>
<th>Gr$_{q}$ (Inlet/Exit)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.0092/0.0051</td>
<td>18.2/68.7</td>
<td>53.8/93.6</td>
<td>21800/19600</td>
<td>6.22E7/2.41E7</td>
</tr>
<tr>
<td>2</td>
<td>0.031/0.014</td>
<td>17.3/88.3</td>
<td>60.2/115.1</td>
<td>15600/13500</td>
<td>6.71E7/1.84E7</td>
</tr>
<tr>
<td>3</td>
<td>0.037/0.0080</td>
<td>14.2/195.5</td>
<td>136.6/270.9</td>
<td>21800/15500</td>
<td>2.49E8/1.64E7</td>
</tr>
<tr>
<td>4</td>
<td>0.020/0.0073</td>
<td>16.8/120.5</td>
<td>91.5/174.4</td>
<td>21900/17800</td>
<td>1.37E8/2.46E7</td>
</tr>
<tr>
<td>5</td>
<td>0.060/0.017</td>
<td>16.0/148.1</td>
<td>98.5/196.5</td>
<td>15700/12100</td>
<td>1.32E8/1.52E7</td>
</tr>
<tr>
<td>6</td>
<td>0.10/0.017</td>
<td>21.5/258.1</td>
<td>169.9/332.3</td>
<td>15400/10100</td>
<td>2.14E8/0.79E7</td>
</tr>
<tr>
<td>7</td>
<td>0.15/0.042</td>
<td>25.8/153.3</td>
<td>107.0/204.8</td>
<td>10200/8000</td>
<td>7.80E7/0.91E7</td>
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<tr>
<td>8</td>
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<td>26.6/212.4</td>
<td>151.5/286.5</td>
<td>10200/7300</td>
<td>1.15E8/0.69E7</td>
</tr>
<tr>
<td>9</td>
<td>0.29/0.045</td>
<td>22.6/249.7</td>
<td>173.8/336.6</td>
<td>10400/6900</td>
<td>1.54E8/0.58E7</td>
</tr>
<tr>
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<td>17.4/133.1</td>
<td>111.4/242.3</td>
<td>7000/5600</td>
<td>7.03E7/0.79E7</td>
</tr>
<tr>
<td>11</td>
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<td>12.9/216.8</td>
<td>167.9/254.8</td>
<td>7200/4900</td>
<td>1.30E8/0.52E7</td>
</tr>
</tbody>
</table>

* Inlet at 2 feet from start of heated length and exit at 2 feet from end of heated length

Re, Gr and Bo Number

The use of air as the flow medium in the experiments leads to significant temperature changes over the test length. Since gas properties are much more sensitive to temperature change than are those of liquids, the property changes of the flowing air must be considered in interpreting the experimental results. For example the air density will decrease about 40% when the temperature increases from the ambient temperature, 20°C, to 200°C, a temperature a number of runs have reached. These property changes will affect the dimensionless numbers. Typically for these conditions, the Re decreases at almost a constant rate: for example, from 10,000 at channel inlet to a little below 7,000 at the channel exit, a 30% decrease. The Gr$_{q}$ exhibits a rapid decrease in the first half of the channel, then decreases at a much slower rate. The channel exit Gr$_{q}$ is less than 4% of the channel inlet Gr$_{q}$ number. The rate of decrease of the Bo number is between those of the Re and Gr$_{q}$ numbers; overall, an 85% decrease takes place.
Wall Temperatures

Two typical wall temperature profiles are shown in Figure 9.7. They represent low and high inlet Bo number cases. The air stream temperatures computed from the energy balance are also shown in the figure. The wall temperature variation is very close to linear for the inlet Bo equal to 0.037 case, whereas for the high (0.29) Bo case, the buoyancy effects cause a noticeable distortion in the wall temperature over the entire duct length. This phenomena of wall temperature distortion has also been observed by other investigators, namely Argonne National Laboratory investigators, and Miyamoto, as reported by Sohn, et al. [S-4], and Tanaka, et al. [T-1].

Nusselt Numbers

Local Nusselt number results are given as functions of axial location and local Bo number in Figure 9.8. They span the inlet Bo number range investigated and represent different Nusselt number variation trends. Dittus-Boelter correlation and CONDOR predictions, based on the local bulk temperature, are also shown for each case. The inlet Bo number of Run 3 is close to 0.037 and hence the data are considered to be in the forced convection regime. The measured Nusselt number is very close to the calculated values from the Dittus-Boelter correlation for large $Z/D$. The CONDOR results indicate the extent of the developing region (to $Z/D = 30$). Run 6 with an inlet Bo number of about 0.10 is considered to be just inside the mixed convection regime. The results show that the measured Nusselt numbers do not yet exhibit an obvious decrease. For Bo increased to 0.29 as in Run 9, the effect of the buoyancy force is reflected by a gradual decrease in Nusselt number from the inlet until the middle of the channel and then a gradual recovery. Note that the associated CONDOR predictions do not recover as $Z/D$ increases. The suspected reason for this behavior is presented in the next section. At the channel exit, the Nusselt number reaches or exceeds the forced convection values. For further increase of Bo number, the Nusselt numbers stop recovering from their minimum values, as shown by
Figure 9.7  Wall Temperature Behavior for Low and High Bo Number Cases
Figure 9.8  Heat Transfer Behavior for the Range of Bo Number Tested
Run 10 (Bo = 0.50). The Nusselt results are flat at about 70% to 50% of the forced convection Nusselt number values, depending on how large the buoyancy force is. It should be pointed out that at high Reynolds number the measured Nu numbers exhibit ripple – periodic undulations every 3 or 4 measured points. The ripples disappear at lower Re numbers. The ripples in Nusselt number are due to corresponding ripples in the wall temperature distributions. However, careful examination of the manner in which the heater coils were wrapped, and the thermocouple locations, does not show a direct connection with the observed ripples.

**Friction Factors**

The measured isothermal friction factors (at no heat input, i.e., cold run conditions), while within typical error range of forced convection friction factors, were consistently higher than the nominal values by about 3-5%. This may be attributed to flow development (i.e., "entrance") effects.

The non-isothermal (mixed convection) friction factor measured in the experiments is shown in Figure 9.9 as a function of inlet Bo numbers. Both $\bar{f}_1$ and $\bar{f}_2$ are shown in the figure, where $\bar{f}_2$ is plotted at $\rho_o = \rho_{in}$, and $\rho_o = \rho$, the density evaluated at the average bulk temperature, $\bar{T}_b = 0.5(T_{in} + T_e)$. As can be seen from the figure, $\bar{f}_2$ at $\rho_o = \rho$ is very close to $\bar{f}_1$. It is clear from the figure that the density has a large effect on the magnitude of the friction factor. In fact, $\bar{f}_1$ decreases with Bo number while $\bar{f}_2$ with $\rho_o = \rho_{in}$ increases with Bo number.

9.7 Assessment of Predictive Correlations Using the Experimental Results

In this section, it assesses and recommends predictive approaches for Nusselt number and friction factor for design applications using our experimental results.
Figure 9.9  Experimental Friction Factor Results Interpreted with Various Air Density Assumptions
Nusselt Number

For Nusselt number the Jackson correlation and the $k$-$\varepsilon$ model in the CONDOR code are considered. Figure 9.10 presents the Jackson correlation and the CONDOR predictions for three experimental cases of Figure 9.8, which span the Bo range of interest. Note that due to the large property changes along the duct, each experimental case appears as a trajectory of local conditions, with Bo decreasing from inlet to outlet. Also, recall that the Jackson correlation is applicable to fully developed flow and was compared only with data that had small property changes. On the other hand, the CONDOR predictions and the experimental results presented here represent developing, local property dependent behavior.

Figure 9.10 illustrates that for each case, as thermal and hydrodynamic behavior develops, the CONDOR predicted trajectory approaches the Jackson correlation. The predicted trajectories of Runs 9 and 10 suggest disagreement with Jackson regarding the minimum Nusselt number and the Bo at which it occurs. The Run 9 trajectory also undershoots the correlation, reflecting the fact that the code does not allow the velocity field to transition from a Type II profile (maximum velocity close to wall) to the Type I profile (maximum velocity at pipe center). This occurs even though the local Bo decreases to the necessary range based on the local condition transition hypothesis, consistent with our interpretation of the Jackson correlation in terms of local conditions.

Figure 9.10 also includes the experimental data for these three cases. We observe three characteristics:

1. The local conditions are fairly well represented by the Jackson correlation in the low mixed convection regime (Run 3).
2. The Nusselt number does appear to recover after a precipitous decrease in the moderate mixed convection region (Run 9 and to some extent Run 10).
Figure 9.10  Evaluation of Jackson Nu Number Correlation Against Test Results and CONDOR
3. The local Bo value for recovery (~0.1) is lower than that predicted (<0.25) by the Jackson correlation (Run 9) and reasonably represented by the CONDOR prediction (Run 10).

For design use both the Jackson correlation and the CONDOR code have current limitations — large local condition variations and apparent error in the predicted Bo number for minimum Nusselt number for the Jackson correlation — and for the CONDOR code the inability to switch back to a Type I velocity profile (and the corresponding Nu (Bo) behavior), once a Type II velocity profile has been established. While design needs are typically for low Bo values, i.e., less than 0.1 where both approaches are reasonably satisfactory, it is preferred to present a modified Dittus-Boelter correlation for design use. Our modification accommodates the observed decrease in Nusselt number (up to about 50%) due to the large property changes, and the progression into the mixed convection region, by a multiplier. The proposed correlation is:

\[ \text{Nu} = \text{Nu}_{\text{DB}} \left/ \left[ 1 + \left( \frac{4200}{\text{Re}} \right)^3 \right] \right. \]  

(9.21)

for Re \( \geq 5000, 1 \times 10^7 \leq \text{Gr}_q \leq 2.5 \times 10^8 \),

which is plotted with all the data measured in this study, in Figure 9.11. Observe that all the data effectively lies within about ±30% of this proposed correlation.

**Friction Factor**

Figure 9.12 compares Petukhov's correlation, presented in Equation (9.8), with experimental results of \( \overline{f}_1 \) and \( \overline{f}_2 \) with \( \rho_o = \rho_\text{in} \). For each case the friction factor has been normalized by the isothermal friction factor from Petukhov's forced convection correlation, Equation (9.9). The experimental results for \( \overline{f}_1 \) are generally larger than that predicted by Petukhov. Further, they do not exhibit the recovery and increasing trend predicted by Petukhov for Bo numbers greater than about 0.1. A friction factor calculation based on channel average density is most appropriate since, as Figure 9.12 demonstrates, it closely approximates \( \overline{f}_1 \). However, use of a channel average density involves an iteration process.
Figure 9.11 Proposed Correlation for Heat Transfer Results
Fig. 9.12 Evaluation of Petukhov Friction Factor Correlation with Test Results
in the flow rate and heat transfer calculations, because the down-stream air temperatures are not known when the calculations start. Nevertheless, since designers generally can closely estimate the channel average temperature, use of \( \bar{f}_2 \) with \( \bar{\rho}_0 = \bar{\rho} \) is recommended. From Figure 9.12 (since \( \bar{f}_1 \equiv \bar{f}_2 \) with \( \bar{\rho}_0 = \bar{\rho} \)) it is recommended that one use

\[
\frac{f}{f_0} = 1.1 \quad \text{for } Bo < 0.1.
\]

For larger Bo, sufficient results are not available to make a definitive recommendation. However, \( \frac{f}{f_0} = 1.1 \) for Bo up to 1.0 appears justifiable and conservative, pending completion of further experiments.

**PART II  SYSTEM PERFORMANCE EVALUATION**

9.8 Mathematical Model

The objective of the system performance evaluation is to analyze the heat transfer from the reactor vessel to the ambient air. As shown in Figure 9.13, the heat removal process involves radiation heat transfer and convection from the reactor vessel to the riser, and convection heat transfer inside the riser to the ambient air. The radiation heat transfer can be described using the following equations

\[
\sum_{j=1}^{N} [\delta_{kj} - (1 - \varepsilon_k) F_{k,j}] q''_{o,j} = \varepsilon_k \sigma T_k^4 \tag{9.22}
\]

\[
\left[ q''_{o,k} - \sum_{j=1}^{N} (F_{k,j} q''_{o,j}) \right] = \frac{Q_k}{A_k} \tag{9.23}
\]

where \( q''_{o} \) is radiosity; \( F \) is view factor; \( \varepsilon \) is surface emissivity; \( T \) is temperature, \( K \); \( A \) is surface area; \( Q \) is total heat transferred; \( \delta \) is the Kronecker delta and \( \sigma = 5.67 \times 10^8 \) \( \text{W/}(\text{m}^2 \cdot \text{K}^4) \) is the Stefan-Boltzmann constant. The subscripts \( j \) and \( k \) refer to the different
Figure 9.13  Major Heat Transfer Modes and Parameters Involved in the System Performance Evaluation
surfaces. Equation (9.22) is for surfaces with specified temperature as the boundary condition, and Equation (9.23) is for surfaces with specified the heat flux as the boundary condition.

The cavity convection heat transfer can be calculated as

$$q''_{c,v} = h_{c,v} (T_v - T_r)$$ \hspace{1cm} (9.24)

where \(q''_{c,v}\) is the heat flux leaving the vessel wall and \(h_{c,v}\) is the heat transfer coefficient, calculated using the correlation

$$h_{c,v} = \frac{k}{d} N_u = 0.119 \frac{k}{d} \text{Gr}_{\Delta T}^{0.3} \left(\frac{H_r}{d}\right)^{-0.1}$$ \hspace{1cm} (9.25)

Note that the Grashof number, \(\text{Gr}_{\Delta T}\), is defined using the temperature difference between the vessel and riser walls,

$$\text{Gr}_{\Delta T} = \frac{g \beta \rho^2 (T_v - T_r) d^3}{\mu^2}$$ \hspace{1cm} (9.26)

In the above equations, \(H_r\) is the height of the riser, \(d\) is the distance from the vessel wall to the riser wall, \(T_v\) is vessel wall temperature, and \(T_r\) is riser wall average temperature.

For riser convection heat transfer, the momentum equation has the form

$$[(\rho_o - \rho_{ave}) H_r + (\rho_o - \rho_e) H_c] g = \frac{k m_i^2}{2 \rho_e} + \frac{H_r}{(D_{ch}^{hr} 2 A_{fr} \rho^*})$$ \hspace{1cm} (9.27)

with acceleration pressure drop being neglected. The energy equation is
\[ \dot{m}_t c_p(T_e - T_{in}) = A_{r,HT} \bar{h}(T_w - T_{ave}) \]  

(9.28)

where \( K = \sum_i \frac{(K_c + K_f)_i}{(N \cdot A)_i^2} \) is flow resistance of the chimney and cold flow path.

\( N \) = number of flow branches

\( A \) = riser flow area

\( \dot{m}_r \) = mass flow rate in a single riser

\( \dot{m}_t \) = total mass flow rate of the RCCS system

Subscript HT refers to heat transfer; subscripts \( r \) and \( t \) refer to riser and total, respectively.

9.9 Geometric Configuration

Because of the complexity of the real RCCS system, a simpler geometry is used in the system performance evaluation. As shown schematically in Figure 9.14, the system consists of a chimney, a cold flow path (not shown in the figure) and a single riser. Inside the cavity, a unit cell was selected, Figure 9.15, which consists of a riser tube, vessel wall section and cavity wall section.

For the convenience of the study and comparisons, a nominal case is defined based on "typical" design parameters. The related dimensions and other parameters used in the nominal case are shown in Table 9.2. This nominal case is used as a reference case in the following calculations. All other cases are variations from the nominal case.

9.10 Calculation Results

The model presented above has been incorporated into a computer code, RECENT (Reactor Cavity Energy Transfer). The major parameters of interest are vessel wall and riser wall temperature, air exit temperature and air mass flow rate. The variations of the
Figure 9.14  Schematic of RCCS System Considered in RECENT
Figure 9.15  Unit Cell of Riser Layout
Table 9.2 Parameters Used in Nominal Case

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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</thead>
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<tr>
<td>Heat flux at vessel wall (uniform)</td>
<td>5000 W/m²</td>
</tr>
<tr>
<td>Chimney height</td>
<td>27.4 m</td>
</tr>
<tr>
<td>Riser height</td>
<td>19.2 m</td>
</tr>
<tr>
<td>Riser W, L and spacing</td>
<td>0.0508, 0.254 and 0.1016 m</td>
</tr>
<tr>
<td>Distance between riser front wall to vessel wall</td>
<td>0.7 m</td>
</tr>
<tr>
<td>Distance between riser back wall to cavity wall</td>
<td>0.18 m</td>
</tr>
<tr>
<td>Wall emissivity (all walls)</td>
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</tr>
<tr>
<td>Flow resistance coefficient, K</td>
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</tr>
<tr>
<td>Air inlet temperature.</td>
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</tr>
<tr>
<td>Heat transfer coefficient, h</td>
<td>Eq.(9.21)</td>
</tr>
<tr>
<td>Friction factor, f</td>
<td>Blasius correlation</td>
</tr>
<tr>
<td>Conduction in riser</td>
<td>yes</td>
</tr>
<tr>
<td>Convection in cavity</td>
<td>yes</td>
</tr>
<tr>
<td>Inside riser radiation</td>
<td>yes</td>
</tr>
</tbody>
</table>

Table 9.3 Major Parameter Variations as Riser Friction Factor Increases and/or Heat Transfer Coefficient Decreases

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Nominal</th>
<th>Nominal + ε_w = 0.6</th>
<th>Nominal + f = 3f_{iso}</th>
<th>Nominal + q''=6kW/m²</th>
<th>Nominal + 0.5h</th>
<th>Nominal + 0.5h + f=3f_{iso}</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vessel Temp. (K)</td>
<td>679.4</td>
<td>701.6</td>
<td>705.8</td>
<td>713.4</td>
<td>692.5</td>
<td>731.2</td>
</tr>
<tr>
<td>Riser Front Wall Temp. (K)</td>
<td>455.3</td>
<td>454.9</td>
<td>525.3</td>
<td>477.5</td>
<td>493.4</td>
<td>580.2</td>
</tr>
<tr>
<td>Riser Air Exit Temp. (K)</td>
<td>468.0</td>
<td>468.0</td>
<td>568.8</td>
<td>468.0</td>
<td>468.0</td>
<td>568.8</td>
</tr>
<tr>
<td>Riser Mass Flow Rate (kg/sec)</td>
<td>0.0633</td>
<td>0.0633</td>
<td>0.0378</td>
<td>0.0633</td>
<td>0.0633</td>
<td>0.0378</td>
</tr>
</tbody>
</table>

250
above temperatures and air flow rates as various parameters are changed are shown in Table 9.3. The nominal case results are also shown in the table for comparison. As can be seen from the table, the decrease of heat transfer coefficient and vessel emissivity and increase of vessel heat flux all have considerable effects on the vessel wall temperature. However, the increase in the riser friction factor is of more interest because its increase will decrease the Re number, and may cause the flow to move deeper into mixed convection. The decrease of the Re number will by itself reduce the Nu value calculated from the Dittus-Boelter correlation. If the flow moves into the mixed convection regime by Re number reduction, the already-reduced Nu number may be further decreased by a maximum of 50%, as already shown in the riser mockup experiment. The results of tripling the riser friction factor and reducing the heat transfer coefficient by a factor of two, as shown in Table 9.3, is the worst scenario. The vessel wall temperature increases by more than 50 K. Although the final vessel wall temperature, 731 K, does not exceed the limiting temperature, 811 K, the safety margin has been greatly reduced.

Figures 9.16 to 9.19 present the configurational study results. Figure 9.16 shows the results of vessel wall temperature variations as vessel-riser distance changes at fixed riser geometry. Figure 9.17 shows the vessel temperature vs the riser width to length ratio, W/L at fixed riser flow area. Figure 9.18 is similar to Figure 9.17 except it is for fixed riser perimeter (total heat transfer area). Figure 9.19 shows the vessel wall temperature variation when riser length increases, (hence flow area and heat transfer area increase) at fixed vessel-riser distance and riser width. In general, system performance is not sensitive to moderate changes in configuration.

9.11 Impact of Present Analysis on MHTGR RCCS Design

Riser Heat Transfer Coefficient

The riser heat transfer coefficient can be described using Eq.(9.21) in the range 5,000 < Re < 30,000. Equation (9.21) shows that the largest Nu reduction is by about half
Figure 9.16  Vessel Temperature vs. Riser Distance from the Vessel
Figure 9.17  Vessel Temperature vs. Riser Width at Fixed Riser Flow Area
Figure 9.18 Vessel Temperature vs. Riser Width at Fixed Riser Perimeter (Heat Transfer Area)
Figure 9.19  Vessel Temperature as a Function of Riser Length at Fixed Width
of the Nu number calculated from the Dittus-Boelter correlation, which occurs at the lower end of the Reynolds number range. Since the nominal operating condition of the RCCS system is at the middle and upper part of the Reynolds number range (Re > 10,000), the use of a forced convection Nusselt number correlation, such as the Dittus-Boelter correlation, will not cause a significant overestimate of the heat transfer coefficient. Even when the Reynolds number decreases to the lower end of the range (how it can be decreased will be discussed later), the 50% or so decrease in the heat transfer coefficient will only affect the vessel wall temperature moderately, as already shown in Table 9.3. Therefore, the potential change in the riser heat transfer capability will affect RCCS performance, but poses no threat to reactor safety.

Flow Resistance

Flow channel blockage is a potential cause of large increases in channel flow resistance, and hence can shift the RCCS operating point towards mixed convection. Consequently, the RCCS may operate at a smaller flow rate and lower heat transfer coefficient, and therefore much higher vessel wall temperature. In the current RCCS design, which has several parallel chimneys and cold flow paths, and more than 200 riser tubes, the flow resistance of the riser tubes is dominant. However, blockage would most likely occur in the cold flow paths or chimneys. The calculation results in Chapter 6 show that even when the flow resistance of the cold flow paths and chimneys increases by a factor of 5, it has almost no effect on the vessel wall temperature (less than 1 K). Therefore, unless all parallel flow paths are blocked, blockage outside the riser section will have little impact on RCCS performance. Conversely, an increase in riser flow resistance can have a significant impact on RCCS performance. Since the riser tubes are located inside the cavity, in the middle of the flow path and consist of more than 200 parallel channels, severe blockage is not considered to be credible. Even with blockage sufficient
to triple the riser flow resistance, the calculation results in Table 9.3 show that the vessel wall temperature is still well below the limiting temperature.

**Riser Geometry and RCCS Configuration**

A change in riser geometry or its location will affect both convective heat transfer inside the riser and the radiation heat transfer within the cavity. Any design change must consider the effects on both sides of the riser wall carefully. However, as indicated by the calculation results, when extreme conditions are avoided, such as a very narrow riser or a very small flow area, the vessel wall temperature is not sensitive to geometry changes in the region of real design interest (vessel-to-riser front wall distance larger than 12 inches). Consequently, there is no single parameter dominating RCCS performance. Nevertheless, by increasing the riser total heat transfer area and flow area, a moderate vessel wall temperature decrease can be achieved.

9.12 Conclusions and Recommendations

1) The measured Nusselt numbers are in qualitative agreement with the Jackson correlation which is considered the best among published correlations. The present experiments provide an opportunity to examine the local Nusselt number behavior along the flow direction under mixed convection conditions, since air properties change substantially. The Jackson correlation is also in good agreement with the experimental results regarding the extent of the heat transfer impairment, both showing a maximum decrease in heat transfer of about 50%, but it does not replicate well the variation of Nusselt number with Bo number. This bound is, however, very useful in RCCS integral heat removal calculations since the limiting "worst case" is of most concern from a reactor safety point of view.

2) Therefore, a better Nusselt number correlation or even a better dimensionless parameter group, instead of Bo, should be developed in the future. A Grashof number
based on the total heat input instead of local heat flux is recommended for future riser mixed convection analysis, because it can accommodate the heat flux difference caused by geometry differences, and non-uniform heat flux effects. For present purposes the most useful approach would appear to be to present the data on a modified Dittus-Boelter Nu versus Re plot, as in Figure 9.11, where systematic differences from the Dittus-Boelter correlation are readily evident and have been empirically correlated. The recommended heat transfer correlation represented in this figure is Equation (9.21), where all parameters are based on local properties:

$$\text{Nu} = \frac{\text{Nu}_{DB}}{\left[1 + \left(\frac{4200}{\text{Re}}\right)^3\right]}$$  \hspace{1cm} (9.21)

for \(\text{Re} \geq 5000; 1 \times 10^7 \leq \text{Gr}_q \times 10^8\)

3) The friction factor behavior under the present experiment's conditions depends on which properties are used in evaluating the friction factor, because the air temperature and its properties change substantially along the channel. For the condition where the inlet temperature is used as the reference temperature for the evaluation of experimented data, a maximum friction factor increase of 50% over isothermal correlation values is inferred. For the condition where the channel average bulk temperature is used, an initial increase (to \(\text{Bo} = 0.1\)), followed by a decrease in the friction factor, is evident from the experiments. The reason for this difference is the much larger effect of density variations on the friction factor evaluated from the friction pressure drop measurements, compared with the effect of viscosity variation on the friction factor obtained from empirical correlations. The results do not support the recommendation of the Petukhov correlation, Equation (9.8). For design purposes, a value of \(f/f_0 = 1.1\) for \(\text{Bo} < 0.1\) is recommended, where density is computed at the axially-averaged bulk air temperature.

3) The parametric and configurational studies show that no single parameter dominates RCCS performance. The worst scenario is to considerably increase the riser flow resistance, which appears to be impossible in the present design. On the other hand,
the studies also show that by carefully choosing the riser geometry, the vessel wall temperature can be moderately reduced from the present level.
REFERENCES


Appendix A
The Structure of the RECENT Code and Sample Input and Output

The RECENT code has a main program and eight subroutines. These subroutines are INPUT, VIEW, RADIOSITY, FLOW, CONDU, SLINE, CHECK and OUTPUT. The functions of these subroutines and main program are as follows.

INPUT — read the input data

VIEW, CHECK and SLINE — evaluate the view factors between each pair of walls

RADIOSITY — calculate the radiosity, hence the heat transfer between each pair of walls

FLOW — calculate the riser/chimney convective heat transfer

CONDU — calculate lateral conduction within the riser walls

OUTPUT — print out the calculated results

MAIN program — execute the iteration and control the flow of the calculation process

A flow chart has been shown in Figure 7.3. In addition to the above subroutines, there is an input data file, READIN.DAT. It contains the input data and will be discussed next.

The code nominal input and output are shown in Figures A.1 and A.2. In the input file, READIN.DAT, the first card is the total surface number, N, of the outer enclosure.

The second card is the control variables. The first variable, INRA, controls inside riser radiation (=0, without inside radiation calculation; >0, with inside riser radiation). The second variable, IFLOW, controls the riser/chimney convection heat transfer calculation (=0, without convection heat transfer calculation; >0, with convection heat transfer calculation). The third variable, ICONV, controls cavity convection calculation
(=0, without cavity convection; >0 with cavity convection). The fourth variable, ICOND, controls the conduction calculation within the riser wall (=0, without conduction; >0, with conduction). The fifth variable, IHDB, selects the different heat transfer coefficient inside the riser (=0, Eq.(5.4) is used; >0, Dittus-Boelter correlation is used). The last variable, IPRT, is the print indicator. When IPRT > 0, detailed iterating results will be printed out.

The third card contains three variables. The first one, LAMIDA, is a coefficient used in the iteration process, and usually has a value of 0.15. The second variable is the power peaking factor and the last variable is the ratio of the riser height to the vessel height.

The four card contains six variables. The first one is the number of risers. The second and third variables are the multiplication factors in the friction factor and Nusselt number correlations, respectively. The fourth variable is riser hydraulic diameter. The fifth and sixth variables are the riser flow area and total heat transfer area, respectively.

The following N card are each wall's specified conditions: the first column is wall dimension; the second column is the wall emissivity; the third column is a mode number specifying the boundary condition of the wall — 1 for specifying wall heat flux and 0 for specifying wall temperature; the fourth one is the wall heat flux and the last one is the wall temperature. For the walls with specified heat flux, the wall temperature can be assigned an arbitrary number. For the walls with specified temperature, the heat flux can be assigned an arbitrary number. Values of these parameters are calculated by the code. However, a good estimate for these values will reduce the running time.

Finally, the last card is the air inlet temperature, heat transfer coefficient (used for no riser/chimney convection case), initial guessed riser air exit temperature, cavity convection heat transfer coefficient and the riser conductivity.

Most items in the output file are self-explanatory. The first part of the output is the image of the input data. It is followed by the view factors for both enclosures. The results for each wall are temperature T(I), emissivity E(I), area AR(I) and heat flux QT(I). For temperature and heat flux, both SI unit and English unit are given.
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Figure A.1   Nominal Case Input Data
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CAVITY CONVECTION INDICATOR ICONV = 1
RISER WALL CONDUCTION INDICATOR ICOND = 1
ITERATION PRINT INDICATOR IPRT = 0
ITERATION CONVERGING PARAMETER LAMIDA = 0.10
POWER PEAKING FACTOR (FPEAK) = 1.00
HEAT FACTOR DUE TO AREA DIFFERENCE (FAREA) = 1.00
RISER HYDRAULIC DIAMETER = 0.07060
RISER FLOW AREA = 0.0109070
RISER HEAT TRANSFER AREA = 10.9742002

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INSIDE RISER AIR TEMPERATURE = 316.00 K
INSIDE RISER HEAT TRANSFER COEFFICIENT = 21.76 W/K M2
GUESSING OUT - 500.00 K
INITIAL AIR AVERAGE TEMPERATURE = 408.00 K
RISER WALL THERMAL CONDUCTIVITY = 45.00W/M K

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Figure A.2 Nominal Case Output
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**INSIDE RISER VIEW FACTOR MATRIX**

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**FINAL ITERATION DATA**

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**HEAT FLUX AT VESSEL WALL** 4346.429 W/M² 0.1378E+12 Btu/hr-ft²

**VESSEL WALL TEMPERATURE** 678.385 DEG K 761.423 DEG F

**RISER WALL AVERAGE TEMP** 424.205 DEG K 303.899 DEG F

**Hr BETWEEN VESSEL AND RISER** 17.100 W/M²-K

Figure A.2 (continued) Nominal Case Output

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APPENDIX B

TEMPERATURE LIMIT ON EXPERIMENTAL RANGE
AND SUBDIVISION OF EXPERIMENTAL RANGE USING BO NUMBER

B.1 Wall Temperature Limit

As mentioned in Chapter 4, the key limitation on the experimental range is the wall temperature, due to the thermocouple insulation employed in the experiments. To estimate the effect of this limitation on the experimental range, a calculation is undertaken as follows.

At each elevation the heat transfer from the wall to the flowing air is given as

$$ q'' = h (T_w - T_a) \quad (B.1) $$

This will lead to a wall temperature

$$ T_w = T_a + q''/h \quad (B.2) $$

where $T_w$ is the wall temperature, $T_a$ is the air temperature, $q''$ is the heat flux at the wall and $h$ is the heat transfer coefficient from the wall to the flowing air. All above quantities are functions of elevation. The air temperature can be expressed as

$$ T_a = T_{in} + \frac{q''\pi D z}{mc_p} \quad (B.3) $$

where it is assumed that the heat flux is uniform axially. Several factors affect the wall temperature: air temperature, heat transfer coefficient and heat flux. For a rough estimate,
using the channel exit wall temperature (since the air temperature is highest there) and the Dittus-Boelter correlation for the heat transfer coefficient

\[ \text{Nu} = 0.023 \left( \text{Re} \right)^{0.8} \left( \text{Pr} \right)^{0.4} \]  
(B.4)

This approximation was made because the exact heat transfer coefficient was not known before the experiment and was, as a matter of fact, the major quantity sought in the experiment. In addition, for simplicity, heat losses will not taken into account in the calculation. The over-estimation of the heat input will be offset to some extent by over-estimation of the heat transfer coefficient, because the turbulent mixed convection heat transfer coefficient will be lower than predicted by the Dittus-Boelter correlation.

Substituting Eqs.(B.3), (B.4) and \( z = L \) into Eq.(B.2), one has

\[ T_w - T_{in} = \frac{q'' \pi D z}{\dot{m} c_p} + \frac{q''}{0.023 \frac{k}{D} \text{Re}^{0.8} \text{Pr}^{0.4}} \]  
(B.5)

since

\[ \dot{m} = \frac{\text{Re} \alpha v}{D} \]  
(B.6)

and

\[ \text{Gr} = \frac{g \beta p^2 q'' D^4}{k \mu^2} \]  
(B.7)

Eq.(B.5) becomes

\[ \text{Gr} = \frac{g \beta p^2 D^4 (T_w - T_{in})}{k \mu^2 \left[ \frac{\pi D^2 \lambda}{A \mu c_p \text{Re}^2} + \frac{1}{0.023 \frac{k}{D} \text{Re}^{0.8} \text{Pr}^{0.4}} \right]} \]  
(B.8)
Where \( \dot{m} \) is the mass flow rate, \( A_f \) is the pipe cross area, \( L \) is the length of the heated section, \( D \) is the pipe diameter, \( c_p \) is the air heat capacity, \( \mu \) is the air viscosity, and Re and Pr are the Reynolds number and Prandtl number, respectively.

The limiting wall temperature is 400 °C. Assume the air inlet temperature, \( T_{in} \), is 30 °C, and all properties are evaluated at \( T_a \) equal to 300 °C. The above equation gives the \( T_w \) limit line in Figure 4.2

B.2 Subdivision of the experimental range

To better understand the MHTGR RCCS experiment, and permit judicious placement of the the experimental points over the full range of the experiment, a rough estimation of when the experiment is in the mixed convection condition, and when it is in the forced convection condition, is needed. A Bo (buoyancy) number is used to make this evaluation, which is defined as [C-3]

\[
Bo = 8 \times 10^4 \frac{Gr_a^{1/2}}{Re^{3.425} Pr^{0.8}}
\]

It is predicted that when Bo is less than 0.01 the flow is in the forced convection; and when Bo is larger than 0.1 the buoyancy effect will have an obvious effect on the Nusselt number and the friction factor. Substituting different Bo values into the above equation will give a family of lines indicating the progressive extent of the buoyancy effect.
APPENDIX C

BUOYANCY EFFECTS ON NUSSELT NUMBER DUE TO MIXED CONVECTION

The Jackson correlation is as follows [C-3]:

\[
\frac{\text{Nu}}{\text{Nu}_0} = \left[ 1 - \frac{\text{Bo}}{\left( \frac{\text{Nu}}{\text{Nu}_0} \right)^2} \right]^{0.46}
\] (C.1)

However, only a few percent error is introduced if a power of 0.50 instead of 0.46 is used. This will gives

\[
\left( \frac{\text{Nu}}{\text{Nu}_0} \right)^4 - \left( \frac{\text{Nu}}{\text{Nu}_0} \right)^2 + \text{Bo} = 0
\] (C.2)

which has the solution

\[
\left( \frac{\text{Nu}}{\text{Nu}_0} \right) = \frac{1 \pm \sqrt{1 - 4\text{Bo}}}{2}
\] (C.3)

To first order Eq.(C.3) is

\[
\left( \frac{\text{Nu}}{\text{Nu}_0} \right)^2 = 1 - \text{Bo}
\] (C.4)

and once again to first order:

\[
\left( \frac{\text{Nu}}{\text{Nu}_0} \right)^2 = 1 - \frac{1}{2}\text{Bo}
\] (C.5)
Bo = \frac{8 \times 10^4 G_r}{Re^{3.425} Pr^{0.8}} \quad (2.29)

By balance pressure gains and losses in a vertical flow channel, the \( G_r \) as a function of \( Re \) and \( Nu \) can be expressed as

\[ G_r \sim Re^2 Nu \quad (C.6) \]

Then, \( Bo \) number can be written as

\[ Bo = \frac{Nu}{Re^{1.425} Pr^{0.8}} \quad (C.7) \]

The Nusselt number is a function of \( Re \) and \( Pr \) numbers. Using Dittus-Boelter correlation, one has

\[ Bo \approx \frac{1}{Re^{0.625} Pr^{0.4}} \quad (C.8) \]

From Eq.(C.5)

\[ \frac{Nu}{Nu_o} \approx 1 - \frac{1}{Re^{0.625} Pr^{0.4}} \quad (C.9) \]

Equation (C.9) suggested that mixed convection Nusselt number has a form of

\[ Nu = \frac{Nu_o}{1 + (C/Re)^n} \quad (C.10) \]

where \( C \) and \( n \) are coefficients and \( Pr \) has been taken as a constant.
APPENDIX D

ANALYSIS OF EXPERIMENTAL ERROR

The output of an experiment is measured parameters. The determination of these parameters, either directly measured or derived from the direct measurement, are subject to certain error, or uncertainty. The error stems from the experimental equipment capability, which can only have certain precision and may also have certain systematic bias. The error is also due to variations in the quantities being measured. When experimental results are the direct measured parameters, the error analysis is relatively easier to perform than when the experiment output is an indirect measured parameter. The errors associated with the direct measured parameters are the systematic error plus the random error caused by the limited precision of the measuring device. Unfortunately, most experiment results fall into the latter category. In the MHTGR RCCS riser experiment, the final parameters sought are the friction factors and Nusselt numbers. They are derived from the measured pipe wall temperatures, air inlet temperature, pressure drop across the flow rate measurement device (an orifice in this case), and the pressure drop between the two pressure taps on the pipe wall. Therefore, Nu and f are the indirect measured parameters.

To estimate the "uncertainty" associated with a derived output quantity from the uncertainties of direct measured parameters, a general result can be used for the expected error in the derived quantity [B-1]. For any function

\[ y = f(x_1, x_2, \cdots, x_i, \cdots, x_n) \] (D.1)

the uncertainty in \( y \), \( E_y \), can be expressed as
\[ E_y^2 = \sum_{i}^{n} \left( \frac{\partial f}{\partial x_i} \right)^2 E_{x_i}^2 \]  

(D.2)

where \( x_i, i=1, 2, \ldots, n \), are the direct measured parameters, \( y \) is the derived quantity from the directly measured parameters, and \( E \) represents the uncertainty.

In the experiment, \( y \) represents the cold run friction factor, \( f_{\text{iso}} \), the hot run friction factor, \( f \), and the Nusselt number, \( \text{Nu} \). Expressions for these quantities are given in the Chapter 4, repeated here as follows.

**Cold run friction factor**

\[ f_{\text{iso}} = \frac{\Delta p_{\text{meas}}}{L \frac{G^2}{D} 2\rho} \]  

(4.24)

**Hot run friction factor**

\[ \bar{f}_1 = \frac{2D(\Delta p_t + \Delta p_b - \Delta p_a)}{G^2 \int_0^L \frac{dz}{\rho}} \]  

(4.7)

and

\[ \bar{f}_2 = \frac{2\rho^* D(\Delta p_t + \Delta p_b - \Delta p_a)}{L G^2} \]  

(4.13)

**Nusselt Number**

\[ \text{Nu} = \frac{(q''_{\text{nom}} - q''_{\text{loss}}) D}{k (T_w - T_a)} \]  

(4.15)

Before applying Eq.(D.2) to the above three equations to obtain the uncertainties in the
friction factor and Nusselt number, the errors in the air properties calculated from the air temperatures, the uncertainty in the flow rate measurement, the errors in the measurements of the heat flux Δ(\(q''_{\text{nom}} - q''_{\text{loss}}\)) and the total heat input Δ(\(Q_{\text{nom}} - Q_{\text{loss}}\)) to the flowing air, and the errors in the air stream temperature calculation Δ(\(T_2\)) must be evaluated.

1. Air Properties

In obtaining the friction factor and Nusselt number, the air density, heat capacity and thermal conductivity were used. For air density, the perfect gas law is assumed which gives

\[
\rho = \frac{353.12}{T(K)}, \quad \text{kg/m}^3
\]  

(4.9)

The heat capacity is taken as constant, 1018 J/kg. The viscosity and thermal conductivity are calculated from the linear correlations

\[
k = 6.9625 \times 10^{-5}T_a + 0.0053525
\]  

(D.3)

where \(T_a\) is the air temperature, K. All of the above correlations are derived based on air properties published by the National Bureau of Standards [H-3]. The maximum relative error in the values calculated from the above expressions is less than 1.5%. This value of 1.5% will be taken as the systematic error in the following air property error analyses.

1). Air density

Applying Eq. (D.2) to Eq.(4.9) gives

\[
E^2 = \left(\frac{353.12}{T_a^2}\right)^2(\Delta T_a)^2
\]  

(D.4)

then,

\[
\Delta \rho = (\rho \Delta T_a)/T_a + \text{systematic error}
\]  

(D.5)
where $T_a$ represents the air temperature. As can be seen, $\Delta p$ is heavily dependent on the uncertainty of the air temperature.

2). Heat Capacity

Since the heat capacity is chosen as independent of the air temperature in the range of this experiment, there is no uncertainty introduced by the air temperature. Only the systematic error applies, which is taken as the maximum relative error, 1.5%, i.e.,

$$\Delta c_p/c_p = 1.5\% \quad (D.6)$$

3). Thermal Conductivity

Since the thermal conductivity is a linear function of the air temperatures given in Eq.(D.3), using Eq.(D.2) yields

$$(E_k)^2 = (6.9625 \times 10^{-5} \Delta T_a)^2 \quad (D.7)$$

Then, the uncertainty is the sum of the temperature induced error and the systematic error,

$$\Delta k = 6.9625 \times 10^{-5} \Delta T_a + \text{systematic error} \quad (D.8)$$

The uncertainty on thermal conductivity also depends heavily on the uncertainty in the air temperature.

2. The Uncertainty in the Flowrate Measurements

To determine the error associated with the air mass flow velocity, $(\Delta G/G)$, Eq. (D.2) is also applicable. The correlation connecting mass flow rate, $G$, and measured parameters is [L-1]
\[ \frac{\Delta G}{G} = \left( \frac{2\Delta D_0}{D_0} \right)^2 + \left( \frac{\Delta K}{K} \right)^2 + \left( \frac{\Delta Y}{Y} \right)^2 + \left( \frac{\Delta p_1}{2p_1} \right)^2 + \left( \frac{\Delta T_1}{2T_1} \right)^2 + \left( \frac{\Delta p}{\Delta p} \right)^2 \]  \quad (D.10)

where \( C \) is a conversion factor depending on the units used. In the present experiment, \( C = 16.4 \) for the mass flow velocity \( G \) as \((\text{kg/m}^2\text{s})\), when the units for the other parameters are as follows,

\( D_0 = \) orifice diameter, \( 1\frac{3}{64} \) inches

\( K = \) flow coefficient, dimensionless

\( Y = \) expansion factor, dimensionless

\( p_1 = \) absolute pressure before the orifice, \((\text{mmHg})\)

\( T_1 = \) absolute temperature before the orifice \((^\circ\text{R})\)

\( \gamma = \) the specific gravity of gas (air = 1.00), (introduce no errors)

\( y = \) the supercompressibility factor, dimensionless, which is constant for the riser experiment conditions and introduces no error.

\( \Delta p = \) the pressure drop across the orifice, inches water column.

The flow coefficient \( K \) and expansion factor, \( Y \), in Eq. (D.9) for computing \( G \) both have an uncertainty less than \( \pm 0.5\% \) when \( \Delta p_1/p_1 < 0.2 \), which applies here. \( \Delta p_1/p_1 \) is usually very small and can be neglected. The inlet air temperature uncertainty is taken as 2 \(^\circ\text{R}\), the uncertainties in \( D_0 \) and \( \Delta p \) are taken as half of the smallest scale interval of the instrument, where \( \Delta D_0 \) is 1/128 inches and \( \Delta p \) is about 1 mm water column. Substituting these parameters and uncertainties into Eq. (D.10), one has
\[
\left( \frac{\Delta G}{G} \right)^2 = 2.7277 \times 10^{-4} + \left( \frac{\Delta T_1}{T_1} \right)^2 + \left( \frac{\delta \Delta p}{\Delta p} \right)^2
\]  
(D.11)

Calculations using experimental data shows that for the hot run the maximum mass flow rate error at inlet Re numbers larger than 10,000 is less than 4%; at an inlet Re number equal to 7,000 it is less than 7%. More detailed results can be found in Table D.1.

3. Heat Flux and Total Heat Input to the Flowing Air

The heat flux input to the air at any point is the difference between the nominal heat flux minus the heat flux loss through the insulation, \( (q''_{\text{nom}} - q''_{\text{loss}}) \). The uncertainty in this quantity is the square root of the sum of the square of the uncertainties of \( q''_{\text{nom}} \) and \( q''_{\text{loss}} \), i.e.,

\[
\Delta(q''_{\text{nom}} - q''_{\text{loss}}) = [ (\Delta q''_{\text{nom}})^2 + (\Delta q''_{\text{loss}})^2 ]^{0.5}
\]  
(D.12)

Since the nominal heat input is uniformly distributed and \( Q_{\text{nom}} = IV \), then

\[
\Delta q''_{\text{nom}} = \frac{\Delta Q_{\text{nom}}}{\pi DL} = \frac{1}{\pi DL} \sqrt{ (\Delta I \cdot V)^2 + (\Delta V \cdot I)^2 }
\]  
(D.13)

where \( I \) is the current through the heaters, \( A \); and \( V \) is the voltage across the heaters. \( \Delta I \) and \( \Delta V \) are the errors in the reading of the ammeter and voltmeter, respectively. The uncertainty in the heat flux loss through the insulation is from two sources. The first is similar to the nominal heat measurement. Since

\[
q''_{\text{loss}} = \frac{Q_{\text{loss}}}{\pi DL}
\]  
(D.14)
then,
\[
\frac{\Delta Q_{\text{loss}}}{\pi DL} = \frac{1}{\pi DL} \sqrt{(\Delta I \cdot V)_{\text{loss}}^2 + (\Delta V \cdot I)_{\text{loss}}^2}
\]  
(D.15)

where the subscript "loss" indicates that the total heat input \( Q \), the current, \( I \), and the voltage, \( V \), are taken to be the values in the heat loss measurements. The second contribution to the uncertainty in the heat flux loss through the insulation is from the use of a linear correlation to represent the actual measured results, which we denote as \( \Delta q''_{\text{corr}} \). For simplicity, let \( \Delta q''_{\text{corr}} \) equal to the mean value of the errors at each measurement point. This gives about 11.1 W/m², which is not important in most of the data reductions. Then the uncertainty in the heat flux loss term is
\[
\Delta q''_{\text{loss}} = \sqrt{\left(\Delta q''_{\text{corr}}\right)^2 + \frac{1}{(\pi DL)^2}\left[(\Delta I \cdot V)_{\text{loss}}^2 + (\Delta V \cdot I)_{\text{loss}}^2\right]}
\]  
(D.16)

And finally
\[
[\Delta(q_{\text{nom}} - q''_{\text{loss}})]^2 = \left[\frac{(\Delta I \cdot V)^2 + (\Delta V \cdot I)^2}{(\pi DL)^2}\right] + \left(\frac{(\Delta I \cdot V)_{\text{loss}}^2 + (\Delta V \cdot I)_{\text{loss}}^2}{(\pi DL)^2}\right)
\]  
(D.17)

The error in the total heat input to the flowing air is readily obtained after we have derived the error in the heat flux loss term.
\[
\Delta(Q_{\text{nom}} - Q_{\text{loss}}) = \Delta\left(Q_{\text{nom}} - (\pi D)\sum_i (q''_{\text{loss}} \Delta z)_i\right)
\]  
(D.18)

where \( i \) indicates the \( i \)-th axial node in the calculation, \( N \) is the total number of axial nodes, \( \Delta z_i \) is the axial width of the \( i \)-th node, which is a constant since equal length nodes are used
in the calculation, and \((q''_{\text{loss}})_i\) is the heat flux lost through the insulation at node \(i\). Eq. (D.18) can also be written as

\[
\Delta(Q_{\text{nom}} - Q_{\text{loss}}) = \left( (\Delta Q_{\text{nom}})^2 + (\pi D \Delta z)^2 \sum_{i}^{N} (\Delta q''_{\text{loss}})_i \right)
\]  

(D.19)

\(\Delta Q_{\text{nom}}\) can be calculated as follows by using Eq. (D.2)

\[
(\Delta Q_{\text{nom}})^2 = (\Delta I \cdot V)^2 + (\Delta V \cdot I)^2
\]  

(D.20)

The second term on the right hand side of Eq. (D.19) can be expressed as

\[
(\pi D \Delta z)^2 \sum_{i}^{N} (\Delta q''_{\text{loss}})_i^2 = \frac{(\pi DL \cdot \Delta q''_{\text{corr}})^2}{N} + \frac{(\Delta Q_{\text{loss}})^2}{N}
\]  

(D.21)

since \(N \Delta z = L\). It should be noted that the larger the number of axial node the less the error. Finally,

\[
\Delta(Q_{\text{nom}} - Q_{\text{loss}}) = \left( (\Delta Q_{\text{nom}})^2 + \frac{(\pi DL \cdot \Delta q''_{\text{corr}})^2}{N} + \frac{(\Delta Q_{\text{loss}})^2}{N} \right)^{0.5}
\]  

(D.22)

where \((\Delta Q_{\text{nom}})\) is from Eq. (D.20), \((\Delta Q_{\text{loss}})\) is from Eq. (D.15) and \(\Delta q''_{\text{corr}}\) is taken as constant, 11.1 W/m². \(\Delta I\) and \(\Delta V\) are 0.25 A and 2.5 V, respectively, which is half of the minimum scale of the meters used.

4. Uncertainties in Air Stream Temperature Calculations

The air bulk temperature used in the friction factor and Nusselt number calculations is a function of axial location and calculated from the energy balance
\[
(T_a)_n = T_{in} + \frac{(\pi D) \sum_i^n [(q_{nom}'' - q_{loss}'') \Delta z]_i}{\bar{m} c_p}
\]

where subscripts \(i\) and \(n\) indicate the \(i\)-th and \(n\)-th axial nodes, \(T_{in}\) is the inlet air temperature, and \(m\) is the air flow rate and \(c_p\) is its heat capacity. Applying Eq. \((D.2)\) to Eq. \((D.23)\) yields

\[
\Delta(T_a)_n^2 = \left( (\Delta T_{in})^2 + \left( \frac{\Delta \bar{m}}{\bar{m}} \right)^2 + \left( \frac{\Delta c_p}{c_p} \right)^2 + \frac{\sum_i^n \left[ (\Delta q_{nom}'')^2 + (\Delta q_{loss}'')^2 \right]_i}{\sum_i^n (q_{nom}'' - q_{loss}'')^2} \right)
\]

\[(D.24)\]

To estimate the last term on the right hand side of above equation, one has

\[
\left[ \frac{\Delta((T_a)_n - T_{in})}{(T_a)_n - T_{in}} \right]^2 = \left( \frac{\Delta \bar{m}}{\bar{m}} \right)^2 + \left( \frac{\Delta c_p}{c_p} \right)^2 + \frac{\sum_i^n \left[ (\Delta q_{nom}'')^2 + (\Delta q_{loss}'')^2 \right]_i}{\sum_i^n (q_{nom}'' - q_{loss}'')^2}
\]

\[(D.25)\]

This equation can also be written as

\[
\left[ \frac{\Delta((T_a)_n - T_{in})}{(T_a)_n - T_{in}} \right]^2 = \left( \frac{\Delta \bar{m}}{\bar{m}} \right)^2 + \left( \frac{\Delta c_p}{c_p} \right)^2 + \frac{\sum_i^n \left[ (\Delta Q_i)^2 + (\pi D \Delta z \Delta q_{corr}'')^2 + \left( \frac{\Delta z \Delta Q_{loss}}{L} \right)_i^2 \right]}{\sum_i^n (Q_{nom} - Q_{loss})_i^2}
\]

\[(D.26)\]

On the right hand side of Eq. \((D.26)\), \(\frac{\Delta \bar{m}}{\bar{m}} = \frac{\Delta G}{G}\), which can be found from Eq. \((D.11)\).

The second term, the error in the air heat capacity value, can be taken as its maximum error,
1.5%. In the numerator of the third term on the right hand side of Eq.(D.26), the first summation can be found as \( \frac{n}{N}(\Delta Q)^2 \), the second summation is straightforward and equals \( \frac{(\pi DL)^2}{N}(\Delta q''_{corr})^2 \). The last summation is not so easy to estimate because the heat losses, \( \Delta Q_{loss} \), at different axial nodes are different. However,

\[
\sum_{i}^{n} \left( \frac{\Delta z \Delta Q_{loss}}{L} \right)_{i}^2 = \sum_{i}^{n} \frac{\Delta Q_{loss}}{N} \leq \frac{\Delta Q_{loss}}{N} \tag{D.27}
\]

Using \( \frac{\Delta Q_{loss}}{N} \) in the last summation will make the evaluation more conservative. Then the uncertainty in the air stream temperature calculation can be expressed as

\[
[(\Delta T_{a})_{u}]^2 = [(\Delta T_{in})^2 + [(T_{a})_{n}-T_{in}]^2] + \left( \frac{\Delta G}{G} \right)^2 + \left( \frac{\Delta c_{p}}{c_{p}} \right)^2 + \left[ n(\Delta Q_{nom})^2 + (\pi DL \Delta q''_{corr})^2 + (\Delta Q_{loss})^2 \right]
\]

\[
\frac{N}{\sum_{i}^{n} (Q_{nom}-Q_{loss})_{i}^2}
\]

\[
\tag{D.28}
\]

where \( \Delta Q_{nom} \), \( \Delta Q_{loss} \) and \( (\Delta G/G) \) are calculated from Eqs. (D.13), (D.15) and (D.11). At the channel exit, \( n=N \), Eq.(D.28) gives the uncertainty in the air exit temperature as

\[
(\Delta T_{e})^2 = [(\Delta T_{in})^2 + (T_{e}-T_{in})^2] + \left( \frac{\Delta G}{G} \right)^2 + \left( \frac{\Delta c_{p}}{c_{p}} \right)^2 + \left[ n(\Delta Q_{nom})^2 + (\pi DL \Delta q''_{corr})^2 + (\Delta Q_{loss})^2 \right]
\]

\[
\frac{N}{(Q_{nom}-Q_{loss})^2}
\]

\[
\tag{D.29}
\]

Calculations show that the air temperature uncertainties at each point, for every run, are below 5%.

After estimating the uncertainties in the air properties, mass flow velocity, heat input and heat flux input, and the air stream temperature, the errors in the experimental
results, cold friction factor, hot run friction factor and Nusselt number, can be evaluated as follows.

A. Cold Run Friction Factor

The cold run friction factor is calculated from measured air flow rate and pressure drop as follows

\[
f_{iso} = \frac{\Delta p_{meas}}{L \frac{G^2}{D \cdot 2\rho}}
\]  

(4.24)

Applying Eq. (D.2) to Eq. (4.24) yields

\[
\left( \frac{\Delta f_{iso}}{f_{iso}} \right)^2 = \left( \frac{\delta \Delta p_{meas}}{\Delta p_{meas}} \right)^2 + \left( \frac{2\Delta G}{G} \right)^2 + \left( \frac{\Delta T_{in}}{T_{in}} \right)^2
\]  

(D.30)

where \( \Delta p_{meas} \) and \( \delta \Delta p_{meas} \) are the measured pressure drop and the uncertainty associated with that measurement. The calculated results in Table D.1 show that the uncertainties in cold run friction factor are less than 7% when \( Re \geq 10,000 \). When \( Re \) equals to 5600, this uncertainty increases to 25%. The major uncertainties come from the uncertainties in the flow rate measurement. The second power dependence on flow rate of the friction factor makes the uncertainty in the friction factor larger than that in the flow rate.

B. Hot Run Friction Factor

Compared with the cold run friction factor the hot run friction factor is more complicated because of the uncertainties associated with the heat input, air stream densities and air temperatures. The hot run friction factor is calculated from the measured parameters as follows
\[
\bar{f}_1 = \frac{2D(\Delta p_t + \Delta p_b - \Delta p_a)}{G^2 \int_a^L \frac{dz}{\rho}} \tag{4.7}
\]

and

\[
\bar{f}_2 = \frac{2\rho^* D(\Delta p_t + \Delta p_b - \Delta p_a)}{LG^2} \tag{4.13}
\]

Only uncertainty in \(\bar{f}_1\) will be estimated here because it depends upon one more variable, \(T_a\). Eq. (4.7) can be written approximately as

or

\[
\bar{f}_1 = \frac{2D(\Delta p_t + \Delta p_b - \Delta p_a)}{G^2 \sum_i^N \frac{1}{\rho_i} \Delta z_i} \tag{D.31}
\]

Applying Eq. (D.2) to Eq. (D.31) gives

\[
\left( \frac{\Delta \bar{f}_1}{\bar{f}_1} \right)^2 = \left( \frac{\delta \Delta p_t}{\Delta p_t + \Delta p_b - \Delta p_a} \right)^2 + \left( \frac{\delta \Delta p_b}{\Delta p_t + \Delta p_b - \Delta p_a} \right)^2 + \left( \frac{\delta \Delta p_a}{\Delta p_t + \Delta p_b - \Delta p_a} \right)^2 + \frac{\sum_i^N (\Delta T_{a,h})^2}{\left( \sum_i^N 1 \right)^2} \tag{D.32}
\]

where the flow rate uncertainty is calculated from Eq. (D.11). The air temperature uncertainties are from Eq.(D.28). The error in the total pressure drop measurement, \(\delta \Delta p_t\), is from the square root of the standard deviation of the reading. The other two terms, \(\delta \Delta p_b\) and \(\delta \Delta p_a\), need to be further estimated. For \(\delta \Delta p_b\), since
\[ \Delta p_b = \sum_{i}^{N} g(\rho_i - \rho_{in})\Delta z \quad (D.33) \]

then,

\[ (\delta \Delta p_b)^2 = (\Delta z \cdot g)^2 \sum_{i}^{N} \left[ (\Delta \rho_i)^2 + (\Delta \rho_{in})^2 \right] \quad (D.34) \]

where subscript \( i \) indicates the \( i \)-th axial node. Since \( N\Delta z = L \) and \( \rho = 353.12/T_a \), the above expression can be written as

\[ (\delta \Delta p_b)^2 = (\Delta z \cdot g)^2 \sum_{i}^{N} \left( \rho \frac{\Delta T_a}{T_a} \right)_{i}^2 \frac{\left( L \Delta \rho_{in} g \right)^2}{N} \quad (D.35) \]

For \( \delta \Delta p_a \), since

\[ \Delta p_a = G^2 \left[ \frac{1}{\rho_e} - \frac{1}{\rho_{in}} \right] = \frac{G^2}{353.12} \left( T_e - T_{in} \right) \quad (D.36) \]

where Eq. (4.9) has been used, one has

\[ (\delta \Delta p_a)^2 = (\Delta p_a)^2 \left[ \frac{2\Delta G}{G} \right]^2 + \left( \frac{\Delta T_e}{T_e} \right)^2 + \left( \frac{\Delta T_{in}}{T_{in}} \right)^2 \quad (D.37) \]

Eq. (D.32) along with Eq. (D.28), (D.35) and (D.37) can then be used to obtain the uncertainty of the hot run friction factor. The calculated results are given in Table D.2.

C. Nusselt Number

The Nusselt number is calculated from the local heat flux, wall temperature and air
temperature as follows,

\[ \text{Nu} = \frac{(q''_{\text{nom}} - q''_{\text{loss}}) D}{k (T_w - T_a)} \]  

(4.15)

Then, the uncertainty in the Nusselt number is

\[ \left( \frac{\Delta \text{Nu}}{\text{Nu}} \right)^2 = \left( \frac{\Delta k}{k} \right)^2 + \left( \frac{\Delta q''_{\text{nom}}}{q''_{\text{nom}} - q''_{\text{loss}}} \right)^2 + \left( \frac{\Delta q''_{\text{loss}}}{q''_{\text{nom}} - q''_{\text{loss}}} \right)^2 + \left( \frac{\Delta T_w}{T_w - T_a} \right)^2 + \left( \frac{\Delta T_a}{T_w - T_a} \right)^2 \]  

(D.39)

This equation, combined with Eqs. (D.8), (D.13), (D.16) and (D.28) will give the uncertainty in the Nusselt number. The results are given in Table D.3.

5. Results and Discussion

The above equations have been programmed, and the results of the calculations are shown in Table D.1 through Table D.3. In the calculations the systematic errors in the air properties were all taken as 1.5%. The error in the inlet air temperature, \( \Delta T_{\text{in}} \), is 1 °C except in the flow rate calculation, where it is taken as 2 °R (1.11 °C). The error in the wall temperature, \( \Delta T_w \), is taken as 3 °C, which is about the largest undulation in the wall temperature ripple measurements. The errors in current and voltage measurement are due to the minimum scales of the ammeter and voltmeter. They are taken as \( \Delta I = 0.25 \text{ A} \) and \( \Delta V = 2.5 \text{ V} \).

As can be seen in Table D.1 and Table D.2, the higher the inlet Re number, the smaller the flow measurement error. The errors in the flow rate measurements are small in both the cold run and hot run cases when the inlet Re number is larger than 10,000. When inlet Re number decreases to below 10,000, the error increases sharply. At an inlet Re number equal to 5,000, \( \Delta G/G \) has an uncertainty of more than 10%.

In the cold run friction factors, the major error comes from the flow measurement
for most of the cases except for the low pressure drop case at inlet Re numbers lower than 10,000, where the uncertainty in the pressure drop measurement also contributes. For the hot run friction factors, the uncertainties are larger than that of the cold runs. In Eq.(D.32), the last term on the right hand side has a relatively smaller contribution than that of the other two terms. At a Reynolds number larger than 15,000 the second term is dominant, and the uncertainties of the friction factor reflect the uncertainty in the accuracy of the flow rate measurement. The uncertainties on the friction factors are lower than 6%. When Reynolds numbers are about 10,000, the first two terms have approximately the same contributions. The resulting uncertainties are close to 10%. Upon further decrease of the Reynolds number, the uncertainty in the flow rate increases rapidly. So does the uncertainty in the pressure drop measurement, because its denominator becomes much smaller than for the high Reynolds number cases. The resulting uncertainty in the friction factor increases to about 15% at Reynolds numbers equal to 7,000.

For the Nusselt number, calculation shows that there is no major term dominant in the determination of the uncertainty. From Table D.3, the minimum relative error occurs at the channel inlet, and the maximum occurs at the channel exit, for a given run. The maximum error in the entire measurement campaign is less than 15% at the channel exit (Run R920). The increase in the Nusselt number uncertainty along the channel is due to the accumulated uncertainty in the air temperature calculation.
Table D.1  Estimated Uncertainties of Cold Run Mass Flow Velocity and Friction Factor

<table>
<thead>
<tr>
<th>Run Number</th>
<th>Inlet Re Number</th>
<th>$\Delta G/G$ (%)</th>
<th>$\Delta f_c/f_c$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>R1</td>
<td>5660</td>
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<td>25.30</td>
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<tr>
<td>R2</td>
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<td>5.04</td>
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<td>2.30</td>
<td>5.10</td>
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<td>16430</td>
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<td>4.53</td>
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Table D.2  Estimated Uncertainties of Hot Run Mass Flow Velocity and Friction Factor

<table>
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<th>$\Delta f/f$ (%)</th>
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APPENDIX E

EXPERIMENT RESULTS
### Table E.1 Experiment Results — R913

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<th>Gr</th>
<th>Nu</th>
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$m=9.399E-3$ kg/sec $\Delta p_l=3.8508$ Pa $Q_{nom}=VI=144*11=1584$ W

$Q_{loss}=364$ W $T_{pve}=25.71$ °C

### Table E.2 Experiment Results — R914

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<th>Gr</th>
<th>Nu</th>
<th>Bo</th>
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$m=9.399E-3$ kg/sec $\Delta p_l=3.8508$ Pa $Q_{nom}=VI=175*13.5=2363$ W

$Q_{loss}=587$ W $T_{pve}=25.71$ °C
### Table E.3 Experiment Results — R916

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$m=9.449E-03$ kg/sec $\Delta p_l=2.3527$ Pa $Q_{nom}=VI=198*15=2970$

$Q_{loss}=786.13$ W $T_{pve}=23.88$ °C

### Table E.4 Experiment Results — R920

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$m=9.510E-03$ kg/sec $\Delta p_l=0.6527$ Pa $Q_{nom}=VI=202*15.5=3131$ W

$Q_{loss}=735.09$ W $T_{pve}=23.38$ °C
Table E.5 Experiment Results — R923

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$m=1.408E-02$ kg/sec $\Delta p_l=30.8229$ Pa $Q_{nom}=VI=124*9.5=1178$ W

$Q_{loss}=160.1$ W $T_{pve}=14.50$ $^\circ$C

Table E.6 Experiment Results — R924

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$m=1.4115E-02$ kg/sec $\Delta p_l=30.1718$ Pa $Q_{nom}=VI=171*13.2=2257.2$ W

$Q_{loss}=358.55$ W $T_{pve}=16.84$ $^\circ$C

295
Table E.7 Experiment Results — R925

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<th>Gr</th>
<th>Nu</th>
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$m=1.3982E-02$ kg/sec  $\Delta p_t=32.3128$ Pa  $Q_{nom}=VI=230*17.6=4048$ W  $Q_{loss}=679.69$ W  $T_{pve}=25.65$ °C

Table E.8 Experiment Results — R929

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<th>$T_{air} (^\circ C)$</th>
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<th>Nu</th>
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$m=1.9701E-02$ kg/sec  $\Delta p_t=63.6311$ Pa  $Q_{nom}=VI=123*9=1107$ W  $Q_{loss}=84.45$ W  $T_{pve}=19.99$ °C

296
Table E.9  Experiment Results — R930

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<th>$T_w$ (°C)</th>
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$\dot{m}=1.9750 \times 10^{-2}$ kg/sec  $\Delta p_t=69.0402$ Pa  $Q_{nom}=VI=176*13.5=2376$ W

$Q_{loss}=230.52$ W  $T_{pve}=19.97$ °C

Table E.10  Experiment Results — R101

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$\dot{m}=1.9494 \times 10^{-2}$ kg/sec  $\Delta p_t=72.8984$ Pa  $Q_{nom}=VI=233*17.75=4135.75$ W

$Q_{loss}=538.35$ W  $T_{pve}=20.27$ °C
Table E.11  Experiment Results — R102

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<td>5966</td>
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<tr>
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<td>133.07</td>
<td>5583</td>
<td>7.91E+07</td>
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$m=6.3337E-03$ kg/sec $\Delta p_l=-10.2661$ Pa $Q_{nom}=VI=130*9.5=1235$ W $Q_{loss}=488.9$ W $T_{pve}=20.20$ °C

Table E.12  Experiment Results — R103

<table>
<thead>
<tr>
<th>Location z (ft)</th>
<th>$T_w$ (°C)</th>
<th>$T_{air}$ (°C)</th>
<th>Re</th>
<th>Gr</th>
<th>Nu</th>
<th>Bo</th>
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<td>0.872</td>
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<td>6861</td>
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<td>322.31</td>
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<td>5326</td>
<td>7.92E+07</td>
<td>13.96</td>
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<td>5.20E+07</td>
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</table>

$m=6.3831E-03$ kg/sec $\Delta p_l=-16.0756$ Pa $Q_{nom}=VI=169*12.5=2112.5$ W $Q_{loss}=787.67$ W $T_{pve}=18.07$ °C
APPENDIX F

TEMPERATURE RIPPLES DUE TO HEATER PITCH

Due to the relative large pitch of the rope heater wound on the outside of the pipe, the question arises whether the inside wall temperatures vary smoothly in the axial direction. To determine how severe this ripple is, a series of measurements were made. The measurements were made on an arbitrary point (defined to be z=0) then moving either upward or downward in about one inch steps until a full pitch, about 2.75 inches, was fully covered. Typical results are shown in Table F.1. They show that the temperature ripple is insignificant. The absence of ripple is attributed to the relative thick pipe wall, coupled with the reasonably high value of the thermal conductivity of the carbon steel compared to the relative low heat transfer coefficient from the wall to the air stream.

Table F.1 Pipe Wall Temperature Variation Over One Heater Pitch

<table>
<thead>
<tr>
<th>Location</th>
<th>Wall Temperature (Deg C)</th>
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</thead>
<tbody>
<tr>
<td></td>
<td>Measurement No. 1</td>
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<tr>
<td>z = 0 in.</td>
<td>162.7</td>
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<tr>
<td>z = 1 in.</td>
<td>164.8</td>
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<tr>
<td>z = 2 in.</td>
<td>166.0</td>
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<tr>
<td>z = 3 in.</td>
<td>167.2</td>
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