**U. S. Department of Energy
Nuclear Engineering Education Research
Award Number: DE-FG07-02ID14341**

**HORIZONTAL HEAT EXCHANGER DESIGN AND ANALYSIS
FOR PASSIVE CONTAINMENT HEAT REMOVAL SYSTEMS**

**Final Technical Report
June 1, 2002 through May 31, 2005**

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August 29, 2005

Executive Summary

This project has investigated the major aspects of horizontal heat exchanger performance in “passive” containment heat removal from a light water reactor following a design basis accident LOCA (Loss of Coolant Accident). Passive systems are those that do not require any external input such as AC power sources or operator action to function. Passive systems for containment heat removal following a design basis accident LOCA are one of the strategies for achieving simplification and improving safety and reliability of future nuclear reactors.

The heat exchanger design in this work is intended for advanced and innovative reactors. For passive containment heat removal, advantages of horizontal heat exchangers over the current vertical heat exchanger designs include potentially better heat transfer efficiency, economic benefits from the reduced containment size, less tube fouling for reduced maintenance and greater seismic resistance.

This project consisted of experimental investigations and development of empirical and physics-based analysis models for incorporation into industry safety computer programs such as TRAC and RELAP. Two graduate students (one of which was supported by NEER funding) and seven undergraduate students from Purdue University’s School of Nuclear Engineering participated in the project. The students acquired experimental experience and received instruction in the development of physics models and the use of reactor safety codes. Of the seven undergraduate students, one is now a Master’s student in the PI’s laboratory. Another three are anticipated to further their nuclear engineering education through at least a master’s degree and have expressed interest in graduate research related to the topics of this project.

Since the application to passive containment heat removal is new, the following aspects of heat exchanger behavior were investigated:

1. the condensation heat transfer characteristics when the incoming fluid contains noncondensable gases
2. the effectiveness of condensate draining in the horizontal orientation
3. the conditions that may lead to unstable condenser operation or highly degraded performance
4. multi-tube behaviors with the associated primary-side and secondary-side effects

A single-tube horizontal test facility and a six-tube horizontal tube bundle facility were constructed and operated to perform these investigations. The goal of the single-tube facility was to provide basic heat transfer data for better understanding of steam condensation from steam/ noncondensable gas mixtures and to obtain a data base for validation of a physics-based model for condensation heat transfer. New methods were developed for measuring the local heat flux on condenser tubes and a data base of heat fluxes at the top and bottom of the condenser tube were obtained.

These single-tube data demonstrated that the heat transfer rates are symmetric about the condenser tube very close to the tube inlet. Downstream, the heat transfer rates are much higher from the top side of the tube than the bottom, most likely due to additional thermal resistance from a thicker condensate film along the bottom of the tube. ***These data are unique and important because they provide multi-dimensional information (circumferential angle and distance from inlet) for condensation heat transfer rates without disturbance of the condensation phenomena by intrusive instrumentation or thermal distortion of the condenser tube geometry.***

The tube-bundle facility was constructed to examine system behavior of a horizontal heat exchanger. Steady-state tests were performed to observe tube-to-tube effects and transient tests were run to simulate LOCA scenarios at actual pressure with prototype condenser tube dimensions. The transient tests in the tube bundle demonstrated that horizontal heat exchangers can self-regulate their heat removal rates to match the incoming heat load. Retention of noncondensable gases in the downstream portion of the tubes is essential to this important function. By confirmation of the self-regulation capabilities and positive results regarding the four aspects listed above, ***horizontal heat exchangers have been shown to possess the capabilities of vertical heat exchangers for PCCS heat removal.***

Empirical correlations for the heat removal rates at the top and bottom of the tubes were derived from the single-tube facility data. These were incorporated into the RELAP5 code and the tube bundle conditions at steady state was successfully reproduced. A physics-based model was developed independent of the test data and verified against the single-tube test data.

This report documents the research completed during the entire project period from June 1, 2002 through May 31, 2005.

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Acronyms

DBA	Design Basis Accident
LOCA	Loss of Coolant Accident
PCCS	Passive Containment Cooling System
SBWR	Simplified Boiling Water Reactor

Nomenclature

Symbol	Explanation	unit
A	area	m^2
C_c	coefficient in calculating Sherwood number	
C_s	coefficient in calculating Nusselt number	
D_{AB}	mass diffusivity	m^2/s
D_{in}	Inner diameter of condenser tube	m
D_{out}	Outer diameter of condenser tube	m
D_{co}	Hydraulic diameter of coolant annulus	m
$E1$	parameter in calculating the void fraction	
$E2$	parameter in calculating the void fraction	
F	Ratio of shear force and gravity force	
f_v	Friction factor defined by (8)	
g	gravitational acceleration	m/s^2
h_c	condensation heat transfer coefficient	$W/(m^2K)$
h_{co}	coolant heat transfer coefficient	$W/(m^2K)$
h_f	film heat transfer coefficient	$W/(m^2K)$
$h_{f,top}$	film heat transfer coefficient at the top	$W/(m^2K)$
h_{fg}	latent heat of vaporization	J/kg
h_s	sensible heat transfer coefficient	$W/(m^2K)$
h_w	combination of film, wall and coolant heat transfer coefficient	$W/(m^2K)$
k_c	condensation thermal conductivity	$W/(mK)$
k_{co}	coolant thermal conductivity	$W/(mK)$
k_l	condensate thermal conductivity	$W/(mK)$
k_s	mixture thermal conductivity	$W/(mK)$
k_{wall}	wall thermal conductivity	$W/(mK)$
M	constant used in flow regime identification	= 5.13
M_s	mole mass of steam	kg
M_{air}	mole mass of air	kg
$\dot{m}_{air}$	Air mass flow rate	kg/s
$\dot{m}_{co}$	Coolant mass flow rate	kg/s
$\dot{m}_l$	Condensate mass flow rate	kg/s
$\dot{m}_s$	Steam mass flow rate at the inlet of a section	kg/s
$\dot{m}_{s,in}$	Inlet steam mass flow rate	kg/s
Nu	Nusselt number	
Nu_{co}	coolant Nusselt number	

Nu_f	film Reynolds number	
P	pressure	Pa
Pr	Prandtl number	
q''_{bot}	wall heat flux at the bottom	W/m ²
$q''_{top,in}$	inner wall heat flux at the top of the tube	W/m ²
Q	heat transfer rate for a section	W
R	gas law constant =8.31451	J/(mol K)
R_{wall}	wall resistance	m ² K/W
Re_{co}	Coolant Reynolds number	
Re_l	Condensate Reynolds number	
Re_{mix}	Mixture Reynolds number	
S	Slip ratio	
Sc	Schmidt number	
Sh	Sherwood number (for mass transfer)	
T_{cl}	centerline temperature	K
T_{co}	coolant temperature	K
T_i	interface temperature	K
T_{mix}	temperature to obtain the material properties in calculating the local heat transfer	K
T_{wi}	wall inner surface temperature	K
T_{wo}	wall outer surface temperature	K
x	Local quality defined	
x_{gb}	bulk gas mole fraction	
x_{gi}	interfacial gas mole fraction	
Y	parameter in calculating the void fraction	
α	void fraction	
δ^+	nondimensional film thickness	
δ_f	film thickness	m
τ_g^*	Nondimensional interfacial shear	
μ_l	condensate viscosity	Pa*s
μ_{mix}	mixture viscosity	Pa*s
σ	surface tension	N/m
ϕ	log-mean gas concentration	
ρ_l	condensate density	kg/m ³
ρ_{mix}	mixture density	kg/m ³
θ	Stratification angle	
X	Martinelli parameter	
Δx	length of tube section	m

1 Introduction

1.1 Significance of the Project

This project investigated the major factors of horizontal heat exchanger performance in passive containment heat removal from a light water reactor following a design basis accident (DBA) LOCA (Loss of Coolant Accident). Passive Containment Cooling Systems (PCCS) with horizontal heat exchangers are a possible contributor towards reactor safety, however there is a lack of mechanistic understanding of the heat transfer phenomena occurring in horizontal heat exchanger tubes and the need for confirmation of overall heat exchanger performance.

The significance of the research into heat exchanger performance lies in the contributions towards improved safety and reliability of essential safety functions of future LWR's. There is a strong move towards passive safety systems because the equipment is driven by failsafe forces or mechanisms such as gravity and natural circulation. Horizontal heat exchangers bring benefits over the current vertical heat exchanger designs such as lower maintenance requirements, reduced capital costs due to reduced containment building height and volume, improved security due to reduced building height and better seismic resistance characteristics. The heat exchangers investigated in this work may be used in both advanced and innovative reactors. The modeling methods that result from this study will provide a tool for evaluating system performance and will be necessary for the reactor design and licensing processes.

The importance of the condensation heat transfer investigation lies in clarification of a basic phenomenon that occurs in heat exchangers of several nuclear and non-nuclear applications.

This NEER grant has provided a significant opportunity to develop nuclear engineering students. Two graduate students (one of which was supported by the NEER funding) from Purdue University's School of Nuclear Engineering participated in the experimentation phase throughout the entire program and the analysis phase in the third year. A total of seven undergraduate students provided assistance with the experimental research for various shorter terms. Notably, the program attracted two undergraduates on a 2005 Summer Undergraduate Research Fellowship and one student on scholarship from the School of Nuclear Engineering's "Undergraduate Research Experience" program. The students acquired experimental experience and received instruction in the analysis of data, development of physics models and use of reactor safety codes. Of the seven undergraduate students, one is now a Master's student in the PI's laboratory. Another three are anticipated to further their nuclear engineering education through at least a master's degree and have expressed interest in graduate research related to the topics of this project.

1.2 Passive Containment Heat Removal

Passive systems for containment heat removal following a design basis accident LOCA are one of the strategies for achieving simplification and improving safety and reliability of future nuclear reactors. Passive systems are those that do not require any external input such as AC power sources or operator action to function. Compared with active systems, the passive designs are much simpler because they do not depend on the availability of large power supplies and they do not rely on safety-grade containment cooling systems, both of which add cost and complexity. Yadigaroglu [1999] provides a review of the various passive designs.

The driving forces for these systems are relatively small forces such as natural circulation for cooling and gravity for condensate return. In particular, the heat transfer processes are driven by small pressure and temperature differences. Thus, to achieve the needed cooling rates, heat transfer with phase change is necessary. In one of the first passive concepts, General Electric designed the Simplified Boiling Water Reactor (SBWR) PCCS with vertical heat exchangers that condense containment steam and transfer the heat to a pool outside the containment [Vierow, 1991, 1992]. This design is the basis for passive systems of several current plant designs in the US, Europe and Japan.

1.3 Condensation Heat Transfer in Horizontal Tubes

In contrast to the vertical tube situation, in horizontal heat exchanger tubes, steam condenses along the inner surface of the tube and runs along the periphery to the bottom. At the top portion of the tube, the condensate layer is thin and provides a relatively small thermal resistance. The primary heat transfer mode through the condensate layer is conduction. When the flow is stratified, the primary heat transfer modes at the bottom are conduction and forced convection; however, the thicker condensate layer at the bottom provides more heat transfer resistance, rendering the upper section of the tube a more effective heat transfer surface. The situation is further complicated by the droplet entrainment and mist formation by gas shear.

Heat transfer rates at both the top and bottom sections of the tube must be known in order to evaluate the PCCS heat transfer capabilities. This requires detailed knowledge of the local temperatures and condensate film characteristics at the condenser tube top and bottom. Ideally, data would be taken at several angles around the tube; however if the approximate location of the stratified condensate level is known, a reasonable two-region approximation for the heat transfer rates can be obtained.

An additional factor in the PCCS system must be taken into account. The intake gas mixture from the containment will contain a significant concentration of noncondensable gases. These gases accumulate along the condensate film to form a boundary layer that inhibits steam from reaching the film surface. With a lower vapor partial pressure at the condensation surface, the heat transfer rate is decreased and the heat exchanger performance is degraded.

Figure 1 illustrates a possible flow development in horizontal PCCS tubes. The thickness of the condensate layer is a function not only of the distance from the

condenser tube inlet, but also of the circumferential angle. The condensate film represents a heat flow resistance and the heat transfer rate is an inverse function of the condensate film thickness. It is noted that the film thickness is generally on the order of millimeters or less and is exaggerated in the figure.

Figure 1 Profiles in Horizontal Tubes of Condensing Steam-Air Mixtures

In the PCCS situation, annular flow and stratified flow are the dominant regimes. Annular flow occurs at the test section inlet where there is minimal condensate present and the steam has a high velocity. The steam may have a high enough velocity along the condenser tube to entrain condensate droplets and form annular mist flow. Stratified flow is expected downstream. As steam condenses, the steam velocity will decrease and the condensate will run down along the sides of the tube to accumulate at the tube bottom. Stratified flow may also be observed close to the inlet for low steam flow rates.

Two main determinants of the flow regime in horizontal tubes with condensing steam are gravity and interfacial shear. The gravitational force causes the phenomena to be asymmetric at any cross section, resulting in a stratified condensate layer at the bottom of the tube. Shear determines the flow-through of the condensate as it is pulled along by the steam as well as the surface condition of the condensate layer, i.e. the heat transfer surface and the droplet entrainment. The development of the flow regime is quite complicated, yet its effect on heat transfer rates is significant and must be understood to develop mechanistic models of the phenomena.

1.4 Advantages of Horizontal Heat Exchangers

Horizontal condensation heat exchangers have traditionally found many industrial applications, including in the process industry and the air conditioning and refrigeration industry [Kakac, 1998]. Within the nuclear industry, the steam generators of Russian-design VVER reactors are horizontal heat exchangers. The horizontal heat exchanger proposed in this project is applicable to advanced reactors and to some of the Generation

IV reactors that use passive cooling. The modeling is applicable to the heat exchangers of the German SWR 1000 [Twilley, 2002].

Extensive past experience has shown that horizontal exchangers offer several advantages over vertical exchangers including less tube fouling and higher structural earthquake resistance. Heat transfer rates are expected to be very high. For the passive heat removal application investigated herein, there is an economic benefit because the shorter coolant pool allows for reduction in the containment building height and volume. Further, the reduced containment building height may reduce the visibility of the plant from the outside and address the security concern.

2 Project Objectives and Tasks

2.1 Objectives

The project's overall objectives are to:

- a. design a horizontal heat exchanger for passive containment heat removal from a light water reactor following a design basis accident LOCA
- b. experimentally investigate the major aspects of the heat exchanger's behavior
- c. develop analytical tools for incorporation into reactor safety codes
- d. investigate condensation heat transfer mechanisms
- e. develop nuclear engineering students and a junior faculty.

2.2 Tasks

The specific tasks to accomplish these objectives are to:

- a. construct an experimental facility with a single-tube test section
- b. obtain fundamental data for local condensation heat transfer coefficients and condensate draining in a horizontal tube
- c. construct an experimental facility with a tube-bundle test section
- d. obtain data on the heat removal performance of the tube-bundle test section
- e. investigate conditions leading to highly degraded performance or unstable condenser operation of the heat exchanger
- f. develop a heat transfer coefficient correlation for implementation into a reactor safety code
- g. incorporate the heat transfer model into a reactor safety code and verify the model against experimental data

3 Project Activities

3.1 Single-tube Experiments

3.1.1 Scaling Analysis

The single-tube experimental facility was designed to simulate a single PCCS tube. The goals of the single-tube experiments are to obtain fundamental data for local condensation heat transfer coefficients and condensate draining in a horizontal tube under DBA LOCA conditions.

To ensure proper test section design and experimental conditions, a scaling analysis was performed. The scaling analysis was based on the General Electric's 1200 MWe ESBWR [Challberg et al., 1998]. This design has four PCCS units that condense steam from the containment drywell and transfer heat to outside water pools at a rate of 13.5 MW each unit.

The scaling of the heat transfer rate through the condenser is given by:

$$Q = N_{tubes} N_{units} U A_i \Delta T_{total} \quad (1)$$

where Q is total heat removal rate, N_{tubes} is the number of PCCS condenser tubes per unit, N_{units} is the number of units, U is the overall heat transfer coefficient, A_i is the inner surface area of a condenser tube, and ΔT_{total} is the temperature difference from the steam/noncondensable gas bulk to the PCCS pool, which is the driven potential of the heat transfer.

From this calculation, the steam mass flow rate per heat exchanger tube for an ESBWR PCCS is about 3.4 – 9.4 g/s/tube. This determines the range of steam mass flow rates for the experimental conditions. However, in preliminary experiments, it is found that at 6.0 g/s steam flow rate, complete condensation occurs in the single tube facility even for the highest air mass fraction (0.20). Thus it is not necessary to perform experiments with lower steam mass flow rates. For the higher limit of the steam flow rate, 9.4 g/s corresponds to only 21 kW heater power on the steam generator, which is far below the capacity of the steam generator. Since a purpose of the single-tube experiments is also to provide a wide data base for condensation heat transfer, the inlet steam velocities were extended to allow for annular and stratified flow and the condensate Reynolds numbers cover the range of both laminar and turbulent flow. The upper limit of the inlet steam mass flow rate was raised to 46.0 g/s, which corresponds to about 100 kW heater power.

The condenser tube is a straight tube and the length of the tube along which condensation can occur is 120 inches (3.04 m). Little data is available on the length necessary for complete condensation in a horizontal tube. In the 10 MW units of the vertical SBWR condensers, the 2.4 m tube length is longer than needed for long term heat

removal in 2-inch diameter tubes. A length of 3 m was chosen based on calculations using the Chato correlation [1962] for condensation heat transfer in horizontal tubes with stratified flow, because PCCS steam/air flow is expected to be stratified except near the condenser tube entrance.

A 1-1/4 inch tube diameter was chosen with a 0.083 in. tube wall thickness. This wall is thick enough to install thermocouples for measuring the tube wall inner and outer surface temperatures using a new technique developed in this project. From these temperatures, the local heat flux through the tube wall can be accurately determined. The condenser tube and coolant annulus specifications are listed in Table 1.

Table 1 Condenser Tube and Coolant Annulus Specifications

Condenser tube inner diameter	1.08 in. (27.50 mm)
Condenser tube outer diameter	1.25 in. (31.75 mm)
Condenser tube length	120 in. (3.0 m)
Annulus outer diameter	2.5 in. (63.5 cm)
Annulus width	0.63 in. (15.9 mm)

3.1.2 Facility Description

3.1.2.1 Integral Test Loop

As shown in Figure 2 Facility Layout, the major components of the test facility are: the steam supply, the noncondensable gas supply, the coolant water supply, the test section, the condensate collection system, the associated piping and water storage tanks, the instrumentation and the data acquisition system. All heated components except the test section secondary side are thermally insulated with Microlok fiberglass insulation. Each of the components and the instrumentation are described in the following sections.

The test section is a single-tube, concentric-pipe heat exchanger. The secondary side consists of an annulus through which cooling water flows. The coolant annulus design is necessary for these fundamental experiments because it allows for determination of the heat flux axial profile along the condenser tube. The actual PCC units will be located in pools of water that become a saturated, two-phase mixture after some time of operation. Since local secondary-side conditions will be difficult to attain in the tube bundle tests, most data will be integral data (e.g. steam venting from the secondary side, inlet and outlet condenser tube temperatures).

Figure 2 Facility Layout

3.1.2.2 Steam Supply

The steam supply consists of a pressure vessel, immersion heaters and a control panel. The pressure vessel was designed by the contractor and manufactured by Kennedy Tank and Manufacturing Co., Inc.

The steam supply, uninsulated and insulated, is shown in Figure 3 and Figure 4 respectively. The pressure vessel shell is Schedule 10 stainless steel 304 pipe, 60 inches in height and 24 inches in diameter, along with two 24-inch, Schedule 10 stainless steel end caps that were welded to the top and bottom of the body. A drain line from the bottom center, two blowdown lines from the top, weld necks for the heaters and several other penetrations were added. The vessel was built for 150 psi and hydrotested at 180 psi for 12 hours.

Figure 3 Steam Supply Vessel (before insulating)

The two pressure relief valves are $\frac{1}{2}$ in. bronze valves manufactured by Kunkle and are factory-set to open at 130 psig. A vacuum breaker line with a check valve inboard of the vacuum breaker is also attached. During cooldown, the pressure falls below atmospheric pressure, causing the vacuum breaker valve to open. This allows air to flow into the steam supply and avoids placing the vessel under a vacuum.

Figure 4 Steam Supply Vessel (after insulating)

Three 8-inch, flanged immersion heaters manufactured by Watlow Process Systems were purchased for the current project. The heaters have Inconel sheaths. Each heater has a total output of 50 kW. Two of the heaters have two 25 kW circuits and one heater has eight 6.25 kW circuits. The possible power levels are up to 150 kW in increments of 6.25 kW. Type K thermocouples are used to measure the heater sheath temperature. Watlow Series 146 Temperature Regulators receive the thermocouple signal and break the circuit if an overtemperature condition is detected. The temperature regulators are set to activate when the sheath temperature exceeds 600 F.

Power is supplied to the steam supply from a 190 amp, 157 kW source via the control panel. The control panel was custom designed and manufactured by Watlow. Systems engineering support for the steam supply, heaters and control panel is supplied by Thermtech Systems, Inc.

Orion Instruments supplied the magnetic liquid level indicator. This device shows the water level inside the steam supply without exposing glass to high pressure, as is the usual design of sight glasses. The Atlas model indicator comes with a Reed switch device which the PI hooked up to shut off all heater power when the liquid level falls below a prescribed level.

A separator removes water droplets from steam exiting the steam supply. Because the separator is also a pressure vessel, an ASME-certified unit was purchased from Clark Reliance Corporation. A one-inch stainless steel steam line is used to transport steam

from the separator to the test section steam inlet. A vortex flow meter for measuring the steam flow rate and the air supply line are on this steam line.

Heat loss tests were conducted on the steam supply after it had been insulated. The closed system that was tested included the pressure vessel, the water level indicator, the separator and associated piping up to the isolation valve on the steam line. The pressure vessel is expected to be the largest source of the integral system heat loss because, although the heater penetrations become warm (60°C), they must be uninsulated to avoid electrical terminal connection damage.

For the heat loss tests, the system was heated to saturation temperature at 100 kPa and let to sit with a 2 kW heater on. Because the system temperatures and pressure slowly increased, the heat loss was known to be less than 2 kW. A second test was performed at 400 kPa. The saturation temperature is 143°C, and thus the temperature difference driving heat losses is 1.5 times that at 100 kPa. Again, with a 2 kW heater on, the system temperatures and pressure gradually rose, indicating a heat loss of just less than 2 kW.

Heat loss tests for the remainder of the system can not be conducted because the test section heat loss would not be prototypical.

3.1.2.3 Noncondensable Gas Supply

The noncondensable gas is air. The University's air compressor is rated at 135 psia and a pressure regulator prevents a surge of high pressure air from entering the steam line. Two mass controllers were purchased to enable good regulation of air inflow over the entire test matrix.

3.1.2.4 Coolant Water Supply

City water is used for the secondary-side cooling water supply. A 1-1/2 horsepower centrifugal pump from American Machine & Tool Co. is used to circulate the cooling water through the secondary side of the test section. To ensure a uniform supply of coolant through the coolant annulus cross section, coolant enters and exits through four ports, located 90° apart. The coolant outlet with 4 exit ports is shown in Figure 5. The flow rate during testing is maintained at the same rate into each port. The secondary side is pressurized to about 200 kPa to promote uniform flow in the coolant annulus.

3.1.2.5 Test Section

The test section consists of a stainless steel condenser tube and optical grade, polycarbonate plastic blocks with a 2-1/2 inch hole axially through the center. Specifications are given in Table 1.

Steam and noncondensable gas are mixed and directed into the condenser tube. Steam condenses in the tube, while uncondensed steam, condensate and air flow through. The exiting condensate water is collected in a tank. Cooling water flows through the secondary-side annular cooling jacket counter current to the primary side flow. Stainless

steel spacers are provided along the tube to ensure that the tube is centered in the cooling jacket.

Figure 5 Coolant Exit from Test Section of Single-tube Facility

The secondary side is made of 6 pieces of polycarbonate plastic, three on the top and three on the bottom. The upper and lower sections of plastic are bolted together and Neoprene rubber strips are placed in grooves between the upper and lower plastic sections to prevent coolant leakage. Figure 6 shows the lower 3 sections of plastic with the rubber strips in place. Acrylic flanges are attached near the end of each section and the flanges are bolted to hold axially-adjacent plastic sections together. O-rings prevent coolant leakage from between the axially-adjacent sections.

The secondary side is not insulated because the heat loss through the polycarbonate plastic is negligible. Polycarbonate plastic has a low thermal conductivity similar to that of thermal insulation material and the difference between the secondary side and ambient temperatures is very small. The mean temperature of the secondary side is less than 45°C and ambient temperature is about 22-27°C.

This secondary side design was chosen for the following reasons:

- a. The transparent plastic allows for visual observation of the secondary side. Air purging can be confirmed and the thermocouple attachments can be seen, should any of them become faulty.
- b. Instrumentation can easily be brought out of the plastic blocks because they have sufficient thickness to render options such as drilling pipe threads for thermocouple fittings possible.
- c. The plastic blocks have a minimum contact width of 0.75 in. (19 mm) of contact surface with which to make a seal.

Figure 6 Lower Half of Coolant Channel

The new secondary side design allows easy access to instrumentation on the condenser tube, should repair or addition be necessary. If the plastic were not cut in half axially, access would be extremely difficult. Further, all of the 98 thermocouple wires and sheaths from the test section would have to be run through the coolant inlet or outlet of the test section. These wires would cause severe distortion of the coolant patterns around the thermocouples, which would affect thermocouple measurements and the phenomena themselves. They would also prevent coolant from flowing through the annulus.

The centerline, tube wall and coolant thermocouples are placed in sets at fourteen axial locations along the test section. These locations are closely spaced near the condenser inlet and spaced further apart with distance from the inlet. The thermocouple locations at each of 14 cross sections are shown in Figure 7 and Figure 8.

Figure 7 Axial Locations of Thermocouples in Single-tube Facility

Figure 8 Cross Sectional View of Thermocouple Locations in Single-tube Facility

For condenser tube inner surface temperatures, a hole is drilled nearly through the tube wall and a thermocouple with a plug design is tapped into the hole. The plug is made of stainless steel. Type T thermocouple wires (copper and constantan) are threaded through holes in the plug and soldered together to form the measurement junction. This is an innovative inhouse thermocouple design, described further in the heat flux measurement section.

Following the basic ideas of Kuhn and Peterson's method [1995, 1997] for the tube outer wall temperatures, a groove is made on the condenser tube outer surface and a sheathed thermocouple is anchored in the groove. The coolant axial temperature profiles are measured along the top and bottom of the tube. Coolant thermocouples are inserted into the middle of the annulus. Figure 9 shows the first meter of the test section after it has been fully assembled.

Figure 9 Assembled Test Section (first meter near the steam inlet)

3.1.2.6 Condensate Collection System

As the test proceeds, condensate flows out the test section and collects in the condensate collection tank. By watching the water level in a sight glass on the tank and controlling the valve opening on the drain line, the operator can maintain a steady condensate water level in the condensate collection tank. As steady state data is being recorded, condensate water is not drained and the change in condensate level with time is recorded. From this, the condensate flow rate may be calculated. Measurements are taken over several-minute time durations, ensuring accurate measurement. The change in the water level did not affect the steady state because venting maintained a constant pressure.

3.1.2.7 Instrumentation

A total of 119 temperatures, four flow rates, three absolute pressures, two gauge pressures and one differential pressure are recorded. The bulk of the temperature measurements are made along the test section where there are seven measurements at each of fourteen specified distances from the starting point of condensation, for a total of 98 measurements.

The data is recorded by a data acquisition system assembled from National Instruments components. The LABVIEW software is used to display, scan and save data.

All thermocouples are T-type (copper-constantan). The sheathed coolant thermocouples are mounted through CONAX fittings sealed with Teflon disks. All other thermocouples along the condenser tube wall are brought out of the coolant annulus between plastic sections.

Flow rates are recorded for the inlet steam, the inlet noncondensable gas, the condensate outflow and the coolant water. For the inlet steam flow rate, a 1-in. (25.4 mm) Foxboro vortex flow meter, model 83F-A, is used.

For absolute pressure measurements on the steam line, Honeywell absolute pressure transmitters are used. One pressure tap is located near the vortex flow meter and the other is just upstream of the steam inlet to the condenser tube. These are model STA940 devices, with a range of 0-500 kPa. For pressure differential across the test section, a Honeywell differential pressure transmitter, model STD924, with a range of 0 to 400 in. (0 to 99.6 kPa) H₂O is used. Although the expected pressure drop falls in only in the lower 1/20 of this range, technical sales representatives state that accuracy standards are maintained at the low end of the instrument's pressure range.

3.1.2.8 Data Acquisition System

A total of 109 temperatures, four flow rates, two absolute pressures, three gauge pressures and one differential pressure are recorded. The bulk of the temperature measurements are made along the test section where there are seven measurements at each of fourteen specified distances from the starting point of heat transfer (total of 98 measurements).

The configuration of the data acquisition system is shown in Figure 10. A National Instruments SCXI signal conditioning system which can handle 128 channels is used to preprocess the signals, the function of the signal condition system is to filter, amplify the signals. National Instruments PCI-6034E 16-bit data acquisition board is used convert the analog signals to digital signal. LabVIEW software is used to manage the system and record data. The thermocouple signals are analog inputs of between 0 and 10 mV, these signals are amplified in by the signal conditioning modules. The pressure transducer and flow meter signals are 4 to 20 mA that are converted to 1-5 V DC signals by an electrical box made in-house. The mass controllers return 0 to 5 V signals corresponding to the real mass flow rates.

Figure 10 Configuration of the Data Acquisition System

3.1.3 Heat Flux Measurement Methods

3.1.3.1 Inner surface thermocouples

Pairs of wall inner surface-to-wall outer surface thermocouples are used to measure the temperature gradient across the tube wall and then calculate the local heat flux. For condenser tube inner surface temperatures, a novel thermocouple with a plug design is used. T-type thermocouple wires of 0.25 mm diameter were threaded through holes in the plug and soldered together to form the measurement junction. The details of the plug thermocouples are shown in Figure 11, with photographs in Figure 12.

Figure 11 Details of Tube Wall Surface Temperature Thermocouples

Thermocouple Far View

3.2 mm
Thermocouple Closer View
(before sealing with epoxy)

Figure 12 Condenser Tube Inner Wall Thermocouple Pictures

A hole was drilled nearly through the tube wall and the plug thermocouple was tapped into the hole. A fabrication procedure was developed to drill the hole without deforming the tube inner surface. If the inner surface were deformed, then the film flow inside the tube would be disturbed and the phenomena would be changed. The process includes starting from a small diameter hole and then increasing the diameter of the drill bit gradually. Finally the bottom of the hole was milled to be flat. The plug is made of the same material as the condenser tube, stainless steel 304. Small grooves were machined on the bottom of the plug to accommodate the thermocouple wires and junctions; this also ensures that the plug sits flat in the hole.

The 5.0 mm distance between the plug thermocouple and the outer surface thermocouple is to minimize the effect of plug thermocouple on the coolant flow. Any effects are negligible because the outer surface thermocouple is at least 23 wire diameters downstream of the plug. It is noted that the temperatures measured by the wall inner surface and the outer surface thermocouples are not the actual values at the physical inner and outer surfaces of the tube wall. These temperatures must be corrected by a heat conduction calculation to obtain the temperature values at the physical boundaries of the wall.

3.1.3.2 Calibration

The basic consideration of calibration is to measure the temperature difference between the inner surface thermocouple and the outer surface thermocouple under a series of known heat fluxes, then plot the temperature difference as a function of heat flux. In order to provide a uniform known heat flux, boiling heat transfer was chosen as the heat transfer mechanism on the secondary side of the condenser tube. To achieve a high heat flux comparable to the heat flux values expected during experiment, forced convection and condensation were chosen as the heat transfer mechanisms on the primary side of the condenser tube. Further, to eliminate the asymmetrical effect, the condenser tube was set to vertical for the calibration. The calibration device and setup are shown in Figure 13.

Figure 13 Calibration Device and Setup

The main parts of the calibration setup are the boiling section and the condenser. The boiling section is made of a 0.254 m long, 0.108 m I.D stainless steel pipe with flanges attached on both ends. The boiling section is thermally insulated during the calibration tests using 25 mm thick fiberglass insulation material to minimize the heat loss. By moving the boiling section along the condenser tube, all of the thermocouple pairs could be calibrated, two pairs at a time. The coiled tube condenser has a capacity that is high enough to condense all of the steam generated from the boiling section.

Before each calibration test, the boiling section was filled with water up to a level close to the top of the boiling section. The thermocouples to be calibrated were set approximately at the middle of the boiling section. During the run, while steam of a given pressure and flow rate traveled through the condenser tube, the water in the annulus was heated to the saturation temperature. The water that was boiled off in the boiling section was sent through the condenser and then collected in a graduated cylinder. The time-averaged boiling rate was calculated by simply dividing the collected water mass by the collection time. From the boiling rate, the time-averaged heat transfer rate was calculated. The time-averaged and length-averaged heat flux was obtained by dividing the heat transfer rate by the heat transfer area. By changing the primary steam pressure and flow rate, different heat fluxes could be obtained.

Figure 14 Tube Wall Inner and Outer Surface Temperatures during Calibration Process
(Cross section at 0.013 m from inlet, heat flux of 68.8 kW/m^2)

The time-averaged inner surface-to-outer surface temperature differences were compared with the heat fluxes. A linear fit between the heat flux and the temperature difference provided the calibration curves for the inner surface-to-outer surface thermocouple pairs. The heat fluxes during the condensation experiments could then be calculated from these equations. The calibration results for a typical temperature measurement cross section are shown in Figure 15. Error bars of $\pm 0.7^\circ\text{C}$ were added to the temperature differences to account for the $\pm 0.5^\circ\text{C}$ uncertainty of the sheathed thermocouples and $\pm 0.2^\circ\text{C}$ of the plug thermocouples.

Figure 15 Thermocouple Pair Calibration Results for a Typical Cross Section (0.419 m from inlet) (“top” and “bottom” indicate the locations when the tube is horizontal)

The figure shows that the wall heat flux is approximately a linear function of the temperature difference across the tube wall. Conduction calculation also shows that the wall heat flux is a linear function of the temperature difference across the wall if the thermal conductivity of the wall material is considered to be constant and the effect of the small amount of epoxy is neglected. Based on the design value of the thermocouple embedding depth, the wall heat flux could be expressed as:

$$q''_{wall,in,*}(z) = \frac{k_w (T_{wi,*}(z) - T_{wo,*}(z))}{r_{wall,in} \ln \left(\frac{r_{wo}}{r_{wi}} \right)} = 10.34 \left(\frac{\text{kW}}{\text{m}^2 \cdot ^\circ \text{C}} \right) \Delta T_w \quad (2)$$

in which the wall thermal conductivity is taken as 15.1 W/m/K. The slopes of the calibration curves are generally larger than the value in equation(2), which means that the two thermocouple junctions are closer together than the designed distance.

The calibration curves for all thermocouple pairs along the test section are listed in Table 2. The ranges of the constants in the calibration curves reflect the differences in the thermocouple pair installation. The axial locations of the measurement cross sections are also shown in the table.

Table 2 Calibration Constants

Distance from inlet (m)	Thermocouple pair at top of tube (kW/m ² /°C)	Thermocouple pair at bottom of tube (kW/m ² /°C)
0.013	N/A	13.84
0.064	N/A	12.07
0.114	23.24	20.26
0.216	16.91	17.46
0.318	17.50	N/A
0.419	17.19	10.82
0.521	10.68	N/A
0.724	14.34	13.04
1.029	N/A	16.81
1.334	13.14	N/A
1.689	N/A	N/A
2.070	10.24	17.72
2.451	20.57	16.64
2.934	25.07	14.57

Table 2 shows that the values of calibration constants vary by a maximum factor of 2.5. Ideally, they would all have the same values. Since the calibration data were linear and repeatable for most pairs of thermocouples, thermal resistances are known to be constant over the heat flux range for each pair of thermocouples. Therefore the heat flux measurements are accurate even though calibration constants vary among the thermocouple pairs. It is noted that several pairs of thermocouples showed nonlinearity in the calibration curve. These pairs of thermocouples were disqualified from the data set (marked as N/A in Table 2) and their data are not used in the horizontal tube experiments.

The variation in the calibration constants is most likely due to the contact resistance between the tube inner surface thermocouple and the bottom of the hole in which the plug thermocouple is inserted. The plug is pushed flush against the bottom of the hole into which it is inserted and high conductivity epoxy is used to reduce the thermal resistance. Contact resistance, however, cannot be eliminated. A higher slope in the calibration shows that there is a larger contact resistance between the inner surface thermocouple junction and the bottom of the hole.

3.1.3.3 Error Analysis

For the fourteen measurement cross sections, there are 28 inner surface-to-outer surface thermocouple pairs on the test section to measure the heat flux. Calibration equations were obtained for 20 pairs of the 28 pairs. For the remaining 8 pairs,

calibration curves were not obtained because the calibration data showed nonlinear pattern or some of the thermocouples were damaged. For the 20 pairs for which calibration curves were obtained, a statistical analysis performed for the slope of the calibration curves showed that the slopes were between 10.27 and 25.07 kW/m²/°C, with an average value of 16.24 kW/m²/°C and standard deviation of 3.93 kW/m²/°C. This indicates the consistency of the fabrication process and provides suggestions for improving the process.

The maximum relative uncertainty in the averaged wall heat flux is ±9.5%. It occurs when the imposed wall heat flux is at the minimum value, which is about 12.8 kW/m². For the highest heat flux, 145.8 kW/m², the relative uncertainty is ±2.3%. The major sources of uncertainty are the boiling section heat loss, the uncertainty in the condensate rate measurement and the uncertainty in the water level measurement in the boiling section.

3.1.4 Experimental Procedures

3.1.4.1 Pretest Procedures

In preparing the equipment for a run, all valves are checked to ensure that they are all shut. This leaves the steam generator and test section isolated from each other. The steam generator pressure vessel is filled with chemically-treated water to a water level high enough to keep the heaters covered throughout the testing. The condensate collection tank, condensate drain lines, and pressure transducer lines are all primed with water.

Power supplies are turned on and all data acquisition channels are checked for proper functioning.

Atmospheric pressure from the absolute pressure transducer on the test section and the room temperature are recorded for purposes of heat loss calculations.

Heaters are turned on to a predetermined power level to heat the water in the pressure vessel. Once atmospheric pressure is surpassed in the steam generator by a few kPa, the valve to the blow-down tank is opened, steam and air mixture is discharged to the blow-down vessel long enough to vent at least 10 times the volume of the steam generator gas space. This process purges the steam generator of air and ensures a pure steam delivery to the test section. The steam generator is pressurized with steam until the desired pressure is achieved.

During this time, air flow is started and the air pre-heater is turned on to bring the air flow into the test section to the proper temperature. Coolant flow through the test section is also begun. Air is purged from the coolant annulus via the vent valve on the secondary side and by pressurization procedures. The coolant flow is adjusted to provide for a coolant temperature rise of about 15°C at steady state, except for at the highest steam flow rate cases where the temperature rise is about 30°C.

Once the steam generator pressure is at a prescribed level, the steam line is opened to the test section to start the approach to steady state.

3.1.4.2 Steady Operation Procedures

To reach steady state, the inlet steam flow rate and air flow rate must be steady. Steady steam mass flow rate is achieved by critical flow. The heater power is set such that the rate of steam generation is slightly higher than the desired steam flow rate into the test section. The pressure of the water is kept at a constant value which is more than two times the test section pressure. The 1-inch globe valve upstream of the vortex flow meter controls the steam flow rate by critical flow. The ½-inch globe valve on top of the separator vents steam to the environment to keep the pressure of the steam generator constant. By this way, the steam mass flow rate into the test section can be kept constant regardless of the pressure fluctuation in the test section during the approach to steady state.

Steady air mass flow rate is achieved by the automatic adjustment of the air mass flow controllers corresponding to the preset flow rate.

With the steam/air mixture flowing into the test section, the condensate exits to the condensate collection tank. The condensate collection tank water level is maintained at a steady level, towards the lower end of the sight glass. Any uncondensed steam is discharged to the outside atmosphere from the test section outlet. The rate of discharge is controlled by the globe valve at the test section outlet.

For runs at pressure above atmospheric pressure, the condensate sight glass level is maintained constant by adjusting a ball valve on the drain line. With experience, the operator can find a valve position that maintains the level nearly constant, to within a few centimeters, over a period of several minutes. The air temperature and coolant flow rates are checked periodically to ensure that they are at the correct values.

Key to obtaining a steady state is the position of the globe valve at the test section outlet that controls the discharge rate of steam/air mixture. The position must be set so that the flow rate out balances the air flow rate into the test section. Otherwise, the test section depressurizes if the valve is open too much and pressurizes if the valve is open too little.

Control parameters are plotted as a function of time on the data acquisition PC. When the temperatures in the steam generator and the pressure and temperature in the test section have been constant for at least 5 minutes, the system is deemed to be at steady state. Data is then recorded for each channel for at least a two-minute period. Measurement of the condensate collection rate is taken over a 2 to 5-minute time duration, ensuring accurate measurement.

Regarding measurement of the condensate collection rate, while steady state data is being recorded, the condensate collection tank drain valve is closed and the change in the

water level is recorded. The steady state was found to be insensitive to the water level change in the condensate collection tank.

Consecutive tests can be run by changing the steam generator power level and/or the air mass flow rate and the cooling water flow rate.

3.1.4.3 Shutdown Procedures

System shutdown commences by shutting the steam line valve to isolate the steam generator from the test section. The steam generator power is then shut off.

The air pre-heater is turned off and once the pre-heater has been cooled to below 80°C, the air flow is terminated. After the test section has cooled down, the coolant water pump is turned off and water is drained out of the secondary side coolant annulus. The state of the steam generator is monitored until it is in a safe enough state to stop monitoring.

3.1.5 Experimental Test Conditions

The tests were performed at post-LOCA, low-pressure (up to 400 kPa) conditions. The target ranges of conditions are shown in Table 3. The steam flow rates correspond to the actual steam flow rates and velocities expected into a single PCCS tube.

Table 3 Targeted Single-Tube Experimental Conditions

Parameter	Range
Primary side pressure (kPa)	100-400
Steam inlet flow rate (g/s)	6.0 – 46.0
Steam/air mixture inlet velocity (m/s)	8.0-37.0
Noncondensable gas inlet concentration (Mass %)	0 – 20
Secondary side pressure (kPa)	200
Secondary side water flow rate (kg/s)	1.48
Secondary side coolant inlet temperature (°C)	45.0

Originally the coolant flow rate was set to provide about a 15.0°C coolant temperature rise assuming complete condensation. Later CFD simulation of the coolant flow with simplified boundary conditions revealed that there is significant thermal stratification in the coolant with such a low velocity. Thus the coolant inlet flow rate was increased to 1.48 kg/s, which corresponds to a velocity of 0.6 m/s. Under this velocity, thermal stratification within the coolant annulus can be neglected. Also, to improve the accuracy of the heat flux calculation, the inlet coolant temperature was increased to 45.0°C.

Table 4 provides a detailed list of test conditions.

Table 4 Test Matrix for the Single Tube Experiments (targeted conditions)

Test No.	Inlet Pressure [kPa]	Inlet Steam Mass Flow [g/s]	Air Mass Flow Rate [g/s]	Inlet Air Mass Fraction [-]	Coolant Inlet Temperature [°C]	Coolant Inlet Mass Flow [kg/s]
7	100.00	6.00	0.06	0.01	45.00	1.48
6	100.00	6.00	0.12	0.02	45.00	1.48
5	100.00	6.00	0.32	0.05	45.00	1.48
4	100.00	6.00	0.67	0.10	45.00	1.48
12	100.00	6.00	1.06	0.15	45.00	1.48
11	100.00	6.00	1.50	0.20	45.00	1.48
3	100.00	11.50	0.12	0.01	45.00	1.48
2	100.00	11.50	0.23	0.02	45.00	1.48
1	100.00	11.50	0.61	0.05	45.00	1.48
10	100.00	11.50	1.27	0.10	45.00	1.48
9	100.00	11.50	2.03	0.15	45.00	1.48
8	100.00	11.50	2.87	0.20	45.00	1.48
20	200.00	6.00	0.06	0.01	45.00	1.48
21	200.00	6.00	0.12	0.02	45.00	1.48
22	200.00	6.00	0.32	0.05	45.00	1.48
23	200.00	6.00	0.67	0.10	45.00	1.48
50	200.00	6.00	1.06	0.15	45.00	1.48
16	200.00	6.00	1.50	0.20	45.00	1.48
101	200.00	11.50	0.12	0.01	45.00	1.48
100	200.00	11.50	0.23	0.02	45.00	1.48
99	200.00	11.50	0.61	0.05	45.00	1.48
15	200.00	11.50	1.27	0.10	45.00	1.48
14	200.00	11.50	2.03	0.15	45.00	1.48
13	200.00	11.50	2.87	0.20	45.00	1.48
59	200.00	23.00	0.00	0.00	45.00	1.48
18	200.00	23.00	0.23	0.01	45.00	1.48
19	200.00	23.00	0.47	0.02	45.00	1.48
32	200.00	23.00	1.19	0.05	45.00	1.48
33	200.00	23.00	2.55	0.10	45.00	1.48
34	200.00	23.00	4.06	0.15	45.00	1.48
35	200.00	23.00	5.74	0.20	45.00	1.48
24	400.00	11.50	0.12	0.01	45.00	1.48
25	400.00	11.50	0.23	0.02	45.00	1.48
26	400.00	11.50	0.61	0.05	45.00	1.48
51	400.00	11.50	1.27	0.10	45.00	1.48
52	400.00	11.50	2.03	0.15	45.00	1.48
53	400.00	11.50	2.87	0.20	45.00	1.48
27	400.00	23.00	0.23	0.01	45.00	1.48
28	400.00	23.00	0.47	0.02	45.00	1.48

Test No.	Inlet Pressure [kPa]	Inlet Steam Mass Flow [g/s]	Air Mass Flow Rate [g/s]	Inlet Air Mass Fraction [-]	Coolant Inlet Temperature [°C]	Coolant Inlet Mass Flow [kg/s]
36	400.00	23.00	1.19	0.05	45.00	1.48
37	400.00	23.00	2.55	0.10	45.00	1.48
38	400.00	23.00	4.06	0.15	45.00	1.48
54	400.00	23.00	5.74	0.20	45.00	1.48
46	400.00	34.50	0.35	0.01	45.00	1.48
47	400.00	34.50	0.70	0.02	45.00	1.48
39	400.00	34.50	1.81	0.05	45.00	1.48
40	400.00	34.50	3.82	0.10	45.00	1.48
41	400.00	34.50	6.06	0.15	45.00	1.48
42	400.00	34.50	8.62	0.20	45.00	1.48
48	400.00	46.00	0.46	0.01	45.00	1.48
49	400.00	46.00	0.92	0.02	45.00	1.48
43	400.00	46.00	2.37	0.05	45.00	1.48
44	400.00	46.00	5.11	0.10	45.00	1.48
45	400.00	46.00	8.11	0.15	45.00	1.48

3.1.6 Experimental Results and Discussion

3.1.6.1 Heat Transfer Coefficient Experiments

Axial Temperature Profiles

Figure 16 shows the temperature profiles along the test section for one of the tests. This test had an inlet steam mass flow rate of 11.5 g/s, an inlet air mass fraction of 0.05 and an inlet pressure of 200 kPa. The vertical lines in the figure show the axial locations of the local heat transfer measurement cross sections. As expected the temperature profiles reveal the highest temperatures along the centerline. The condenser tube inner surface temperatures are well below the centerline temperatures, with temperatures at the top of the tube being higher than those at the bottom of the tube. Finally, the temperature rise in the coolant running countercurrent to the primary side is evident.

Figure 16 Typical Temperature Profiles for Single-tube Tests (Test 99)
(vertical lines represent the locations of temperature measurement cross sections)

To obtain reliable data, the coolant flow must be reasonably uniform around the test section. The heat transfer rates would be greatly affected if the heat removal from the bottom of the tube resulted in coolant thermal stratification. Prior to the experiment, a CFD calculation for simplified boundary conditions (constant condenser tube wall temperature) confirmed that the thermal stratification in the coolant flow was negligible (less than 1.0°C). Thus in the current test, the much higher coolant temperature rise on the top of the test section indicates that the heat transfer was significantly greater than on the bottom half.

The large difference in the “ T_{wi} ” and the “ T_{wo} ” profiles demonstrates the success of the new embedded thermocouple design. For most tests, the temperature drop across the condenser tube wall is as large as 15.0°C . This temperature difference is large compared to the error associated with the thermocouples ($\pm 0.3^{\circ}\text{C}$). Some other experimental programs employed a much thicker wall in order to obtain an accurate local heat flux measurement, but the heat flux is reduced due to the wall thickness.

The temperature dip in the condenser tube wall thermocouple readings in the first 0.3 m from the inlet is an unexpected result and it is attributed to entrance effects. Complications near the inlet are common in condensation heat transfer facilities [Peterson et al., 1997]. In the current experimental series, this effect is under high pressure and high mass flow rate test conditions. Since the entrance complications are not yet well understood and are beyond the scope of the current study, no further investigations are discussed herein.

Since the wall thermocouple junctions are embedded in the wall, the measured condenser tube inner and outer surface temperatures " T_{wi} " and " T_{wo} " are not the temperatures at the physical wall boundaries. These temperatures were corrected to the physical boundaries by a heat conduction calculation and labeled as " $T_{wall,in}$ " and " $T_{wall,out}$ " correspondingly.

Global Effects of Noncondensable Gas

In many heat exchangers, the inlet flow of the condenser tube may contain a significant concentration of noncondensable gas. The noncondensable gas accumulates along the condensate film to form a boundary layer that inhibits steam from reaching the film surface. With a lower vapor partial pressure at the condensation surface, the heat transfer rate is decreased and the heat exchanger performance is degraded. For forced flow condensation, the effect of the noncondensable gas should be much less significant than in the case of a stagnant vapor because the mixing induced by the flow promotes steam contact with the heat transfer surface [Sparrow et al., 1967].

The effect of the noncondensable gas can be examined by comparing the centerline temperature profiles. The centerline temperature corresponds to the local steam saturation temperature. In these experiments, since the pressure drop along the heat transfer length is less than 1.0 kPa for most of the test conditions, the assumption that the centerline temperature reflects the local steam partial pressure is valid. At the inlet, the steam partial pressure is close to the total pressure. The difference with the total pressure at higher inlet air mass fractions is due to the air partial pressure. Therefore, the inlet temperature for the conditions with higher air mass fraction is relatively low due to the lower steam partial pressure.

Figure 17 Centerline Temperature Profiles for Various Inlet Air Mass Fractions
(Inlet steam flow rate: 6.0 g/s, Inlet pressure: 100 kPa)

Figure 18 Centerline Temperature Profiles for Various Inlet Air Mass Fractions
(Inlet steam flow rate: 11.5 g/s, Inlet pressure: 200 kPa)

With the mixture flowing through the condenser tube, the mixture temperature drops sharply after a major part of the steam has been condensed. From Figure 17 it can be seen that the condensation length is shorter for lower air mass fraction cases because this source of thermal resistance is small. Also the condensate rate measurement by condensate collection confirmed that for the 0.10, 0.15 and 0.20 inlet air mass fraction cases, steam had been vented out. For the 0.01, 0.02 and 0.05 inlet air mass fraction cases, the condensate collection rate is close to the inlet steam mass flow rate. For the steam flow rate shown in Figure 18, a pure steam test was performed. The inlet pressure could not be maintained due to complete condensation of the steam within the tube, the inlet pressure was about 70.0 kPa, which corresponds to a saturation temperature of 90°C, thus the inlet steam is superheated and the steam near the outlet is saturated. Other temperature profiles in Figure 18 show the same trend, in which the 1% and 2% inlet air mass fraction cases exhibit complete condensation.

Local Flow Regimes

In horizontal heat exchanger tubes, steam condenses along the inner surface of the tube and runs along the periphery to the bottom. The liquid film on the tube wall represents a resistance to the heat transfer. The major flow regimes are annular flow and stratified flow in the present study. For annular flow, most of the current heat transfer models [Shah, 1979; Dobson & Chato, 1998] treat the film thickness as being uniform. For stratified flow, the primary heat transfer modes at the tube bottom are conduction and forced convection; however, the thicker condensate layer at the bottom provides more heat transfer resistance, rendering the upper section of the tube a more effective heat transfer surface.

In a horizontal condenser tube, the local flow regimes strongly affect the heat and mass transfer processes. The most widely used flow regime map for horizontal two-phase flow was developed by Mandhane [1974] for adiabatic two-phase flow, in which the flow regimes were defined on a plot of the liquid superficial velocity vs. gas superficial velocity. For horizontal condensation flow, Jaster and Kosky [1976] defined the major flow regimes as annular, transition and stratified flow based on their experimental observations of steam condensation in a 12.5 mm I.D. tube. Flow regime transition criteria were developed based on a force balance between the gravitational force and the wall shear stress. Soliman [1982, 1983, 1986] investigated the condensation phenomena of refrigerants and classified the flow regimes into mist, annular and wavy flows. The modified Weber number was used to predict the mist-to-annular transition and the Froude number was used to predict the annular-to-wavy transition. Rifert [1998] proposed a chart for horizontal condensing flow of steam and classified the flow regime as annular, asymmetric or gravitational flow based on the nature of local heat transfer coefficients distribution over the tube perimeter. These maps are of reference in determining the flow patterns in the current study. The Mandhane flow regime map is used because it is well established for adiabatic conditions and the Jaster and Kosky criteria is used because it is for steam condensation.

Since the local heat transfer characteristics are closely related to the flow regime, the flow regime at each local cross section must be identified to analyze the heat transfer. To assess the local flow conditions, the local void fraction was estimated. The local coolant top and bottom temperatures were averaged to obtain the local average coolant temperature. Noting that the top and bottom temperatures are generally less than 5°C apart, this is a reasonable simplification. The total heat transfer rate from the condenser tube inlet to the local cross section could then be calculated by knowing the coolant temperature. The local condensate flow rate was calculated assuming the condensate at saturation state. The flow condition at a cross section could then be approximately determined.

To identify the local flow regimes, both the Mandhane flow regime map [1974] and the Jaster and Kosky flow regime identification criteria [1976] were used and their results were compared. For most of the test conditions, the flow regime was either wavy flow or stratified flow regimes on the Mandhane flow regime map and transition or stratified flow using the criteria by Jaster and Kosky:

$$F = \frac{\text{Axial shear force}}{\text{Gravitational body force}} = \frac{\tau_w}{\rho_l g \delta} \quad (3)$$

For $F > 29$, the flow regime is annular, for $29 \geq F \geq 5$, the flow regime is transition flow and for $F < 5$, the flow regime is stratified flow. Figure 19 shows the flow regime transition along the test section for all the inlet steam mass flow rates. The inlet air mass fraction was 0.05 for all the test conditions shown. When the inlet pressure and steam mass flow rate are kept constant, generally the local flow regime does not vary with inlet air mass fraction except for the pure steam tests. From Figure 19, it can be seen that F decreases along the condenser tube. For the low mass flow cases, F comes to asymptotic value, signifying the finish of the condensation.

Figure 19 Flow Regime Transition along Condensation Region
(0.05 inlet air mass fraction)

The flow regime analysis by the Mandhane flow regime map for the test conditions corresponding to Figure 16 shows that the first three cross sections ($z = 0.013, 0.064, 0.114$ m) were in the wavy flow regime. For the cross sections downstream, the flow regime was stratified flow. The results using the Jaster and Kosky criteria show that only the first cross section was in the transition flow regime and the rest of the cross sections were in the stratified flow regime. This discrepancy could be caused by the definition of the flow regimes since both of authors defined the flow regimes by flow visualization.

Local Heat Transfer Coefficients

From the temperature profiles, the local heat fluxes at the top and bottom of the tube were calculated with their corresponding calibration equations. The local heat transfer coefficients are defined as:

$$h_*(z) = \frac{q''_{wall,in,*}(z)}{T_{cl}(z) - T_{wall,in,*}(z)} \quad (4)$$

This heat transfer coefficient includes the condensation heat transfer coefficient and the condensate film heat transfer coefficient because no data on film characteristics are

available to separate the two components. Figure 20 shows the local heat transfer coefficient profile corresponding to the test conditions of Figure 16.

Figure 20 Typical Local Heat Transfer Coefficient Profiles

The local heat transfer coefficient profile shows the highest value at the inlet of the condenser tube and a decrease along the condenser tube. This is due to the decrease of the steam/air mixture velocity, the increase of the condensate film thickness along the condenser tube and the increase of the local air concentration.

It can be clearly seen that near the inlet of the condenser tube, the heat transfer coefficients at the top of the tube are much higher than those at the bottom of the tube. This is mainly due to the asymmetrical film thickness profile. Near the tube inlet, the condensate film is the major heat transfer resistance. Towards the outlet of the condenser tube, most of the steam has been condensed and the predominant heat transfer mechanism changes from condensation heat transfer to single phase gas cooling by forced convection at the top of the tube. At the bottom of the tube, the heat transfer mechanism is forced convection through the condensate. Not much difference is shown between the top and bottom of the tube at the outlet. The source of error is from the calibration process. Error bars of $\pm 10\%$ were added to the plot based on error analysis of the calibration.

3.1.6.2 Stable Operation Conditions

Stable operation condition is defined as: both the local temperature profile and the global parameters do not vary with time for a given test condition.

For the single tube test, all the test conditions presented in Table 4 were stable. From the scaling analysis, the lowest steam inlet flow rate corresponds to 3.4 g/s, but

single tube test was not performed for this low steam flow rate. There were two reasons. First, the steam flow rate was measured with a 1-inch vortex flow meter, the flow meter does not give stable output for flow rate under 5.0 g/s. Second is that from the experience of running the experiment, it is known that complete condensation occurs for all the 6.0 g/s test conditions, which suggests that any flow rate under this flow rate will be complete condensation.

As stated in the proposal, there was a concern that the tube could possibly be plugged by condensate water at low steam mass flow rates, which causes unstable conditions. In the experiment, it can be seen that for test condition no. 20 ($\dot{m}_s=6.0$ g/s, $P = 200$ kPa, inlet air mass fraction = 0.01), which has the lowest steam mass flow rate and air mass fraction, although condensation completed at about 1.5 m from the mixture inlet, stable operation was still maintained. This is possibly because that air accumulated in the tube section after the condensation completed and the condensate is drained by both the shear from the air flow and gravity.

The only unstable operation condition occurred was for inlet steam flow rate 6.0 g/s. No stable inlet pressure was measured due to the complete condensation. The condensation drew the condenser tube to a vacuum condition and steam was sucked in from the inlet. Although the venting valve was closed, small amount was able to leak in to the tube and accumulate.

For pure steam test with high steam flow rate, when complete condensation does not occur, stable operation condition was able to be maintained due to the venting steam. One example was the test condition no. 59 ($\dot{m}_s = 23.0$ g/s, $P = 200$ kPa).

3.1.7 Conclusions

Condensation of steam in a horizontal heat exchanger with a noncondensable gas present has been experimentally studied. . The condenser tube inner surface, outer surface and annular coolant channel temperature profiles were measured along the top and the bottom of the test section along with the condenser tube centerline. For the condenser tube inner surface temperature, an innovative thermocouple design was developed that allowed for nonintrusive measurements. From these and other data, local heat fluxes were obtained for a variety of pressures, steam inlet velocities and noncondensable gas mass fractions.

The local heat fluxes are essential for development of analytical models of horizontal condensation heat exchangers but have not been seen reported elsewhere. In particular, it is important to distinguish the difference in performance between the top and bottom of the condenser tube to evaluate the overall performance. The experimental results also identified that the major flow regimes in the horizontal condenser tubes are generally wavy and stratified flow, with annular flow occurring only for a steam/air mixture velocity greater than ~ 46.0 m/s.

The global effect of noncondensable gas on the heat transfer rate was investigated by comparing the centerline temperature profile and the overall heat transfer rate. For the

low inlet air mass fraction conditions studied in this paper, the effect is not as significant as expected.

The effect of the local air mass fraction, local air concentration distribution, local liquid film characteristics and the local turbulent mixing on the local heat transfer coefficient were qualitatively analyzed. The top and bottom show different dependencies on these factors due to the local flow conditions.

3.2 Tube Bundle Experiments

3.2.1 Relation of Tube Bundle Tests to Single-tube Tests

Design of the horizontal heat exchange for a new PCCS system requires detailed knowledge of the heat transfer phenomena inside the condenser tubes and understanding of the system response to various parameters. Detailed heat transfer information is difficult to attain from a tube bundle facility because the amount of instrumentation could become unfeasible and the secondary side conditions can not be controlled well. This NEER program addressed these concerns by including well-instrumented single-tube experiments which produced fundamental heat transfer data for condensation inside a horizontal tube and including tube bundle experiments which investigated the system behavior. The tube bundle experiments were performed to obtain the following key data:

- a. Tube-to-tube effects, as evidenced by differences in condensate drain rates among the tubes and secondary side observations
- b. System response to the redistribution of noncondensable gases
- c. Transient tests in which the heat input was decreased with time to simulate decay heat

Similar to the previous test series, the conditions were extended beyond post-LOCA conditions to identify the range of conditions associated with highly degraded performance or unstable condenser operation. The pool temperature rise with time and void development around the condenser tubes after secondary-side water has come to saturation temperature introduced transient effects not present in the single-tube experiments.

If the single-tube data can be shown applicable to the tube bundle facility, then it may be assumed that there is a good understanding of the heat transfer in the tube bundle and the analysis methods developed from the single tube experiments may be applied to analyze heat transfer on the primary side of the tube bundle.

The applicability of the single-tube test data to the tube bundle facility was confirmed using the following techniques:

- a. Comparison of tube centerline temperature profiles for both facilities when the tube bundle facility uses only one heat exchanger tube and similar test conditions
- b. Application of the correlations from the single-tube experiments to the tube bundle experiments

The results of the centerline temperature profile are shown in Section 3.2.7 and the results of the second method are discussed in Section 3.3.2.

3.2.2 Scaling Analysis

For proper design of the horizontal tube-bundle test facility, the development of a well-balanced and justifiable scaling approach based on available information on PCCS and horizontal heat exchanger characteristics is essential. A scaling analysis was performed to design a PCCS condenser test facility with reference to the functional requirements for General Electric's 1200 MWe ESBWR [Challberg et al., 1998]. This design has four PCCS units that condense steam from the containment drywell and transfer heat to outside water pools at a rate of 13.5 MW.

The scaling of the heat transfer rate through the condenser is given by:

$$Q = N_{tubes} N_{units} U A_i (T_b - T_{pool}) \quad (5)$$

where Q is total heat removal rate, N_{tubes} is the number of PCCS condenser tubes per unit, N_{units} is the number of units, U is the overall heat transfer coefficient, A_i is the inner surface area of a condenser tube, and T_b and T_{pool} are the steam/noncondensable gas bulk temperature and PCCS pool temperatures, respectively.

For the current test facility, the maximum heat removal rate per heat exchanger tube is equal to that in an ESBWR PCCS tube (maximum 9.4 g/s/tube) as inferred from Challberg [1998]. The number of tubes determines Q for the test facility.

The "height effect" for horizontal heat exchangers, in which the upper tubes' heat removal efficiency may be impaired by vapor blanketing on the tube secondary side, is more important than having "inner" and "outer" tubes. The minimum number of tubes that could represent this multi-tube behavior (N_{tubes}) was determined to be six. The available steam supply allows for operation at up to 11 g/s/tube ($Q = 145$ kW).

The test facility has one PCCS heat exchanger (N_{units}).

U is calculated from heat transfer correlations and A_i follows from a determination of the condenser tube length and diameter. Since the secondary side is generally experiencing pool boiling, T_{pool} is the boiling temperature of water.

The condenser tube diameter was chosen to preserve the expected range of inlet steam/gas Reynolds numbers. Complete condensation inside the condenser tubes was the criteria for determining the condenser tube length. Several correlations, including those by Chato [1962], Shah [1979], and Cavallini [1974], were used to estimate the condensation heat transfer coefficient for pure steam. The lengths required for complete condensation were compared against Wu's test data [2005], obtained as part of this NEER project. The Chato estimation of the pure steam condensation length matched Wu's data well. Arai's formulation [2003] for the noncondensable gas effect in combination with the Chato correlation was found to best predict the steam/air mixture condensation lengths.

The characteristics of a tube bundle are different from that of an isolated single tube due to the liquid circulation and potential for vapor blanketing of top tubes. The staggered configuration of the condenser tubes allows for investigation of the height effect and potentially better secondary side circulation than a rectangular configuration.

A pitch-to-diameter ratio of 1.65 was selected because this value falls within the range of standard horizontal heat exchangers and it is similar to the ESBWR ratio inferred from [Challberg, 1998].

The scaling analysis results are summarized in Table 5.

Table 5 Results of Scaling Analysis for Tube Bundle Facility

Parameter	ESBWR Value	Scaled Value
Maximum inlet flow rate of steam/tube	9.4 g/s ¹	2-11 g/s
Number of condenser tubes	2688 ¹	6
Outer diameter	PUMA value of 50.8 mm for SBWR assumed ²	38.1 mm (1.5 in. pipe size)
Length of condenser tube	No reference value	4 m
Tube configuration	Not stated	Staggered
Pitch-to-diameter ratio	“Row pitch” of 40 mm ¹	1.65

¹Challberg et al., 1998

²Ishii et al., 1996

3.2.3 Facility Description

3.2.3.1 Integral Test Loop

The test loop other than the test section is described in Section 3.1.2. The only parts that differ are the test section and the steam supply line. For the tube bundle tests, the steam flow rate can be as high as 66 g/s. This higher steam flow rate requires a larger vortex flow meter than used for the single-tube experiment. A 1-1/2 in. (38.1 mm) Foxboro vortex flow meter, model 83F-A, is used to measure the inlet steam flow rate.

A schematic of the integral test loop for the tube bundle tests is illustrated in Figure 21.

Figure 21 Tube Bundle Test Facility

3.2.3.2 Test Section

The test section consists of six stainless steel condenser tubes and a secondary-side pool in which coolant water removes heat by boiling heat transfer. Tube specifications are given in Table 6.

Table 6 Tube Bundle Test Section Specifications

Condenser tube outer diameter	38.1 mm (1.5 in.)
Condenser tube wall thickness	3.05 mm (0.12 in.)
Condenser tube heat transfer length	3.96 m (156 in.)
Number of condenser tubes	6
Pool length	3.96 m (156 in.)
Pool width	0.13 m (5.25 in.)
Pool height	0.30-0.41 m (12-16 in.)

The scaling study determined the tube bundle configuration and dimensions. A secondary-side end plate showing the tube bundle configuration is shown in Figure 22.

Figure 22 Test Section End Plate with Tube Bundle Configuration

Steam condenses in the tubes, while uncondensed steam, condensate and air flow through. The exiting condensate water is collected in a separate condensate collection tank for each tube.

Heat is removed by a coolant pool that surrounds the six condenser tubes. The pool is heated to saturation and eventually boils. The cover of the secondary side pool is

inclined upward towards the center. This allows for venting of steam from the pool through the vent above the test section. Four sight glass windows were installed in the pool structure to allow for visual observation of the secondary side boiling and void distributions around the condenser tubes. Two sight glasses are located at each end of the pool structure, on opposite sides. One of these sight glasses is shown in Figure 23, along with the condenser tubes inside.

Figure 23 Sight Glass on Secondary Side Pool Structure

Along the active heat transfer sections of each of the tubes, 5 equidistant thermocouples were installed to measure bulk centerline temperature profiles. In contrast to the single-tube experiments, the secondary side boundary condition is pool boiling after an initial warmup period. The tube outer surface temperature at the tube top and the coolant temperature are measured at three axial locations. These correspond to the inlet, outlet and middle cross sections. Following the basic ideas of Kuhn and Peterson's method [1995, 1997] for the tube outer wall temperatures, a groove is made on the condenser tube outer surface and a thermocouple is anchored in the groove.

All temperatures are measured with Type T (copper-constantan), 1.0 mm, ungrounded sheathed thermocouples. Figure 24 shows the locations of the thermocouples at a cross section where there are tube outer surface and coolant thermocouples.

Figure 24 Cross Sectional View of Thermocouple Locations for Tube Bundle Facility

A rubber silicon gasket between the pool and the pool cover prevents leakage of pool water and steam. Extensive leak testing was performed to achieve a leak-free test section.

3.2.3.3 Coolant Water Supply

The coolant water on the secondary side of the test section is de-ionized water. A secondary-side refill system, consisting of a stainless steel tank for water storage and associated piping, allows for continuous supply of coolant water to the test section secondary side. As a pre-test procedure, heated water may be supplied to the tank from the steam generator. A 5-kW heater on the storage tank maintains the water at the desired refill temperature. A high-temperature pump supplies water from the storage tank to the secondary side and a rotameter measures the flow rate.

3.2.3.4 Condensate Collection System

As steam condenses in the tubes, the condensate flows out each tube into its respective condensate collection tank, driven by the momentum of condensate water, any uncondensed steam and the noncondensable gas. The tanks are made of Schedule 10, 6-inch SS304 pipes and are vertically mounted as shown in Figure 25. The water levels in the tanks are controlled by valve positions on the drain lines and each tank has a sight glass with a measuring tape for measurement of the rate of change of the water level.

As the test proceeds, condensate flows out the test section and collects in the condensate collection tanks. By watching the water levels and controlling the valve positions on the drain line, the operators could maintain steady condensate water levels in the condensate collection tanks. Since there are six tanks, however, maintaining steady water levels is not feasible. Instead, once the tanks become about half full, they are drained until the water level is at about the bottom of the sight glass. As steady state data is being recorded, condensate water is not drained and the change in condensate level with time is recorded. From this, the condensate flow rate may be calculated.

Measurements are taken over several-minute time durations, ensuring accurate measurement.

Figure 25 Condensate Collection Tanks for Tube Bundle Facility

3.2.3.5 Instrumentation

A total of 77 temperatures, two flow rates, one absolute pressure and five differential pressures are recorded. The data is recorded by a data acquisition system assembled from National Instruments components. The LABVIEW software is used to display, scan and save data.

Flow rates are recorded for the inlet steam and the inlet noncondensable gas. One of two Foxboro vortex flow meters, model 83F-A, measures the inlet steam flow rate. A 1-in. (25.4 mm) flow meter is used at lower flow rates and a 1-1/2 in. flow meter is used at higher flow rates.

Two absolute pressure measurements are taken on the steam line. A Honeywell absolute pressure transducer is installed near the vortex flow meter and an OMEGA gauge pressure transducer, model PX303 with a range of 0-680 kPa, is installed just upstream of the steam inlet to the condenser tube. For pressure differential across the test section, a Honeywell differential pressure transmitter, model STD924, with a range of 0 to 400 in. (0 to 99.6 kPa) H₂O is used. Although the expected pressure drop falls in only in the lower 1/20 of this range, technical sales representatives state that accuracy standards are maintained at the low end of the instrument's pressure range. A Dwyer pressure transducer is placed on the air line.

3.2.4 Development of Automated Venting System

In the actual PCCS, a noncondensable gas vent line extends from the outlet plenum of the heat exchanger to a prescribed submergence depth in the suppression chamber

water pool. Should the pressure of the PCCS heat exchanger exceed the suppression chamber pressure by a large enough pressure difference to overcome this submergence, noncondensable gases will be vented to the suppression chamber. Other experimental programs have been performed with manual venting of the heat exchanger tubes by the operator. Manual control does not let the system respond to noncondensable gas redistribution in its natural manner.

An automated venting system was added to the current facility such that the system will vent as dictated by the noncondensable gas distribution. Figure 26 shows the venting system schematic. The pressure controller obtains a signal from the pressure transducer, which is sensing the gauge pressure of the PCCS vent tank. If the pressure of the test section is higher than the set pressure of the pressure controller, the pressure controller sends a signal to the solenoid valve to open, venting noncondensable gas and possibly steam from the test section.

The pressure controller is from Dwyer Inc., model 16A2133. The solenoid valve is a 1/4-in. valve.

Figure 26 Schematic Diagram of Pressure Control

This system simulates the behavior of gas venting through a submerged pipe. A desired steady state pressure is determined prior to the test and the upper pressure limit on the pressure controller is set to the steady state pressure plus the head due to vent line submergence. As shown in the experimental results section, the automated venting was successfully implemented into the tube bundle facility.

3.2.5 Experimental Procedures

3.2.5.1 Pretest Procedures

In preparing the equipment for a run, the steam supply pressure vessel is filled with water to a water level high enough to keep the heaters covered throughout the testing. The pressure transducer lines are primed with water and the water levels in the six condensate collection tanks, the PCCS vent tank and the coolant vent tank are adjusted to be near the bottom of the respective sight glasses. Power supplies are turned on and all data acquisition channels are checked for proper functioning.

Heaters are turned on to a predetermined power level to heat the water in the pressure vessel. Once the pressure is surpassed in the steam supply reaches about 130 kPa, the valve to the blow-down tank is opened and steam is discharged to the blow-down vessel long enough to vent at least 10 times the volume of the steam supply gas space. This process purges the steam supply of air and ensures a pure steam delivery. The steam supply is pressurized with steam until the desired pressure is achieved.

As the steam supply is pressurizing, air flow is started and the air pre-heater is turned on to bring the air flow into the test section to the proper temperature. The secondary side pool is filled with de-ionized water.

Once the steam supply pressure is at a prescribed level, the steam line is opened to the test section to start the approach to steady state. Heat transfer through the condenser tubes brings the secondary side pool water up to saturation and eventually boiling begins. Evaporated cooling water exits via the coolant vent tank. The condensate exits to the condensate collection tanks. Noncondensable gas and any uncondensed steam are discharged to the outside atmosphere from the test section outlet at a rate determined by the automated venting system.

While the system is progressing towards a steady state, the water levels in the six condensate collection tanks are continuously monitored to maintain the water levels in the lower 5 cm of the sight glass level.

3.2.5.2 Steady-state Test Operation Procedures

At steady state, the length of the secondary side pool that is boiling is constant. The extent of boiling at the first and last meter of the test section may be observed through the sight windows on the pool. The temperatures throughout the system and the venting frequency of the automated venting system also do not change with time. No adjustments are needed to the positions of the valve on the steam line, the condensate drain valve positions or the test section steam/gas outlet valve, as were needed in the first year of tube bundle testing. This is evidence that the system is successful in venting in response to system behavior and that the flow rate out nearly balances the air flow rate into the test section. The balance is not exact because the water levels in the condensate collection tanks are slowly rising. The decrease in gas space should be causing a slightly higher air flow rate out of the test section than into it, however the water level rise is slow enough that any perturbations on the steady state are undetectable in the recorded data.

The only necessary adjustment during a test is to the air temperature if the pre-heater does not maintain the air at the inlet steam temperature.

Control data are plotted on the LabVIEW display and monitored. After the temperatures in the steam supply and the test section and the venting frequency have been constant for at least 5 minutes, the system is deemed to be at steady state. Data is then stored for each channel for at least a two-minute period and the changes in the water levels of the six condensate collection tanks are manually recorded for at least 8 data points. The steady state was found to be insensitive to the water level change. The six condensate collection tanks are assigned to two or three students so that each person is recording the water level change for two or three tanks.

Consecutive tests can be run by changing the steam mass flow rate and/or the air mass flow rate, refilling the secondary side if necessary.

3.2.5.3 Transient Test Operation Procedures

The startup procedures are the same for steady state and transient tests. The heater power and the air mass controller setting are changed according to a predetermined function of time. Data is recorded by the Data Acquisition System and for condensation collection tank water levels from the time steam is first introduced into the test section. This complete set of data makes possible the assessment of the effect of secondary-side pool heat up and decreasing decay heater power.

In some tests, secondary-side water is replenished during the test from the secondary side refill tank. Water refill allows the test to proceed longer without condenser tube uncover. Water refill with cold water also makes possible the investigation of thermal stratification in the pool and simulation of pool refill in a real PCCS.

3.2.5.4 Shutdown Procedures

System shutdown commences by closing the steam line valve to isolate the steam supply from the test section. The steam supply power is then shut off.

The air pre-heater is turned off and once the pre-heater has cooled to below 40°C, the air flow is terminated. The state of the steam supply is monitored until the pressure has decreased.

After the steam supply has depressurized to atmospheric pressure, a vacuum breaker valve on the pressure vessel opens. The vacuum breaker lets air into the steam supply as it cools down and pressure decreases below atmospheric pressure, to prevent a large vacuum in the steam supply tank.

3.2.6 Experimental Test Ranges

An initial set of tests were performed to confirm that the detailed data from the single-tube test facility was applicable to the tube bundle test facility. For these tests, the

tube bundle facility was configured to use only one of the six condenser tubes. The test matrix is listed in Table 7.

Table 7 Test Matrix for the Tube Bundle Experiments Using One Tube

Test Name	Inlet Pressure [kPa]	Inlet Steam Mass Flow [g/s]	Air Mass Flow Rate [g/s]	Inlet Air Mass Fraction [%]
1.dat	200	11.5	0.12	1
2.dat	200	11.5	0.12	1
3.dat	200	11.5	0.23	2
4.dat	200	6.0	0.06	1
5.dat	200	12.0	0.12	1
6.dat	200	12.0	0.48	4
7.dat	200	12.0	0.60	5
8.dat	200	12.0	0.84	7
9.dat	200	12.0	0.60	5
10.dat	400	12.0	0.60	5
11.dat	400	12.0	0.96	6
12.dat	200	18.0	0.18	1
13.dat	200	18.0	0.54	3
14.dat	200	18.0	0.90	5
15.dat	200	24.0	0.24	1
16.dat	200	24.0	0.48	2
17.dat	200	24.0	0.96	4
18.dat	200	12.0	1.2	10
19.dat	200	12.0	2.4	20
20.dat	200	6.0	1.2	20
21.dat	400	12.0	1.2	10
22.dat	400	12.0	2.4	20
23.dat	400	12.0	2.4	20
24.dat	200	6.0	0.6	10
25.dat	400	6.0	0.6	10

The tube bundle tests were performed at post-LOCA, low-pressure (up to 400 kPa) conditions. The target ranges of conditions are shown in Table 8 and a listing of each test is included in Table 9. The steam flow rates correspond to the actual steam flow rates and velocities expected into a single PCCS tube.

Table 8 Targeted Tube Bundle Experimental Conditions

Parameter	Range
Primary side pressure (kPa)	200-400
Steam flow rate (g/s)	12 – 48

Steam velocity (m/s)	1-7
Noncondensable gas inlet mass fraction	0.0-0.20
Secondary side pressure (kPa)	100

Table 9 Test Matrix for the Tube Bundle Experiments (targeted conditions)

Test No.	Inlet Pressure [kPa]	Inlet Steam Mass Flow [g/s]	Air Mass Flow Rate [g/s]	Inlet Air Mass Fraction [%]	Solenoid Valve Operational?
1	200	12	0.6	5	No
2	200	24	1.2	5	No
3	200	18	1.8	10	Yes
4	200	24	4.8	20	Yes
5	200	30	3.0	10	Yes
6	200	36	1.8	5	Yes
7	200	36	3.6	10	Yes
8	200	36	7.2	20	Yes
9	400	18	0.9	5	Yes
10	400	18	1.8	10	Yes
11	400	18	3.6	20	Yes
12	400	36	3.6	10	Yes
13	400	36	7.2	20	Yes
14	200	36	3.6	10	No
15	200	36	3.6	10	No
16	200	36	3.6	10	No
17	200	36	3.6	10	Yes
18	200	36	7.2	20	Yes
19	200	24	2.40	10	Yes
20	200	24	2.40	10	Yes
21	200	36	0.36	1	Yes
22	200	36	0.72	2	Yes
23	200	36	1.80	5	Yes
24	200	36	3.60	10	Yes
25	200	36	3.60	10	Yes
26	400	36	0.36	1	Yes
27	400	36	0.73	2	Yes
28	400	36	1.90	5	Yes
29	400	36	4.00	10	Yes
30	400	36	9.00	20	Yes
31	250	48 - 0	0 - 2	0-5	Yes
32	400	36	0.73	2	Yes
33	400	36 - 0	0 - 2	0 - 5	Yes
34	400	36	0.73	2	Yes

3.2.7 Experimental Results and Discussion

3.2.7.1 Comparisons between Single-Tube and Multi-Tube Facility Data

The condenser tube in the single-tube tests was extensively fitted with thermocouples to measure axial temperature profiles of the tube centerline, condenser tube wall and coolant. The local heat fluxes and heat transfer coefficients were obtained from this fundamental data as a function of several parameters, as described in Section 3.1 and Section 3.3.1. To compare data from the tube bundle facility against that of the single-tube facility, the steam inlet was reconfigured to deliver steam to the lowest tube of the tube bundle. City water was circulated through the secondary-side pool so that the secondary side conditions were closer to those of the single-tube facility. The primary side conditions in the tube bundle facility replicated those of selected tests run with the single-tube facility.

While the conditions in the two facilities were matched as closely as possible, the discrepancies in the geometries and secondary-side conditions should be noted. Firstly, the diameter and length of the condenser tubes are different. These differences can affect the flow patterns (stratified vs. annular), the condensate behavior, the condensation length, and the local heat transfer coefficient. The inlet steam/air Re was matched in these tests to promote similar hydrodynamic behaviors.

Detailed geometries are shown in below Table 10.

Table 10 Test Section Geometries

	O.D. [mm]	I.D. [mm]	A_i cross [m ²]	Wall Thickness [mm]	Tube Length [m]
Single-tube condenser tube	31.75	27.53	0.00060	2.11	3.048
Tube-bundle condenser tube	38.10	31.75	0.00079	3.18	3.962

The secondary-side conditions were different in terms of flow geometry, coolant flow rate and coolant temperatures. Both test sections had countercurrent flow of the primary and secondary side. The flow geometry in the tube bundle facility was that of coolant entering the bottom of the pool tank at 25 cm (10 inches) from the condenser tube outlet, flowing along the rectangular open channel with the water level at the middle of the second-lowest tube, and exiting the bottom of the pool tank at 25 cm (10 inches) from the condenser tube inlet. The maximum flow rate possible in the tube bundle facility was used and the coolant temperature rise was larger than in the single-tube facility.

Data which may be compared are the centerline temperature profiles, the length required for complete condensation, the condensation collection rates and the trends

when a boundary condition is varied. Figure 27 through Figure 29 show that the centerline temperature profiles and condensation lengths are very similar between the single tube test and the test on the tube bundle facility using only the lowest condenser tube. Essentially all of the steam condensed in these tests, which indicates that the condensation rate is the same. The good agreement is shown for two pressures and two inlet air mass fractions.

For the tests shown in Figure 27, the inlet steam/air Re was 41,000 for the single-tube test facility and 41,200 for the tube bundle test facility. The inlet steam flow rates were 11.5 g/s for the single-tube and 12 g/s for the tube-bundle facilities respectively.

Figure 27 Centerline Temperature Profile at 200 kPa, 0.01 Air Mass Fraction

For the tests shown in Figure 28, the inlet steam/air Re and air mass fraction were 41,000 and 5% for the single-tube test facility and same parameters were 40,500 and 4% for the tube bundle test facility. The steam flow rates were the same as for the cases in Figure 27.

Figure 28 Centerline Temperature Profile at 200 kPa, 0.04-0.05 Air Mass Fraction

For the tests shown in Figure 29, the inlet steam/air Re was 38,620 for the single-tube test facility and 33,350 for the tube bundle test facility. The steam flow rates were the same as for the cases in Figure 27.

Figure 29 Centerline Temperature Profile at 200 kPa, 12g/s Steam 0.05 Air Mass Fraction

The differences in the geometries and secondary side conditions do not affect the results to a noticeable extent. The coolant flow rate was about 1480 g/s in the single-tube test facility and 630 g/s in the tube bundle facility. The inlet and outlet coolant temperatures were about 45°C and 48°C respectively for the single-tube facility. For the tube bundle facility tests, the inlet temperature was between 14°C to 15°C and the outlet temperature was between 21°C and 27°C. The heat transfer coefficient on the secondary side is smaller in the tube-bundle facility than in the single-tube facility, in part because of the lower coolant flow rate. The comparison shows that this is compensated for by a larger heat transfer area and a larger temperature difference between the tube wall and the coolant.

Inlet Air Mass Fraction Effect

The tube-bundle tests with a single tube clearly exhibited the expected trend that an increasing inlet air mass fraction increases the condensation length and maintains centerline temperatures at higher values. Figure 30 through Figure 33 show this trend for three inlet steam flow rates at 200 kPa and one steam flow rate at 400 kPa. The 0.05 air mass fraction tests in Figure 30 demonstrate good repeatability.

Figure 30 Centerline temperature profile at 200kPa, 12 g/s steam with various air mass fractions

Figure 31 Centerline temperature profile at 200kPa, 18 g/s steam with various air mass fractions

Figure 32 Centerline temperature profile at 200kPa, 24 g/s steam with various air mass fractions

Figure 33 Centerline temperature profile at 400kPa, 12g/s steam with various air mass fractions

Inlet Steam Flow Rate Effect

Figure 34 through Figure 37 show the effect of the steam inlet mass flow rate on the centerline temperatures and the condensation length. As the inlet flow rate increases, the length required for complete condensation also increases.

Figure 34 Centerline temperature profile at 200kPa, 1% air mass fraction with various inlet steam flow rates

Figure 35 Centerline temperature profile at 200kPa, 4% air mass fraction with various inlet steam flow rates

Figure 36 Centerline temperature profile at 200kPa, 10% air mass fraction with various inlet steam flow rates

Figure 37 Centerline temperature profile at 400kPa, 10% air mass fraction with various inlet steam flow rates

3.2.7.2 Steady State Multi-tube Experiments

The steady state tests consist of condensation in six tubes with boiling on the secondary side and steam release to a coolant vent tank positioned above the test section. A key feature of these tests is the automated gas venting system. This system eliminates the need for manual venting and allows the system to vent when enough noncondensable gas has accumulated at the test section outlet to raise the pressure above a setpoint corresponding to vent line submergence in a PCCS.

The venting system is described in Section 3.2.4. The following Figure 38 through Figure 40 demonstrate the proper function of the venting system at steady state. During this time, the air flow rate into and out of the test section was constant, minus a small correction for the gradually changing water levels in the condensate collection tanks.

The venting system opened at regular intervals with a pressure oscillation magnitude proportional to the primary side pressure. For a 200 kPa test and 0.05 inlet air mass fraction, the oscillation was about 6 kPa, corresponding to almost 1 psi or 60 cm of water head. For a 400 kPa test and the same air inlet mass fraction, the oscillation was on average about 11 kPa, or 1.1 m of water head. No effects of the venting pressure fluctuations are observable in the condensate rate data at steady state.

A future issue is to investigate the amount of vented gas upon each opening. Noting that the oscillations were larger in the higher pressure tests, it is possible that too much gas was vented through the ½-inch solenoid valve and a ¼ inch may be preferable. A proportional solenoid valve, in which the valve opening is proportional to the difference between the setpoint pressure and the upstream pressure, may be preferable.

Figure 38 Test Section Inlet and Outlet Pressures for Test 23
(200 kPa, 36 g/s steam, 5% air)

Figure 39 Test Section Inlet and Outlet Pressures for Test 21
(200 kPa, 36 g/s steam, 1% air)

Figure 40 Test Section Inlet and Outlet Pressures for Test 28
(400 kPa, 36 g/s steam, 5% air)

Axial Temperature Profiles

Figure 41 shows the temperature profiles for each of the six tubes separately during a typical test. Centerline temperatures were measured at five axial locations because these were considered to be critical data. Surface temperatures were measured at three locations because there was concern about the amount of uncertainty in these data. Pool temperatures were measured at three axial locations because this data was anticipated to be a general monitor of secondary side conditions and pool stratification. The pool temperatures were not expected to be particularly useful in quantitatively evaluating heat transfer rates.

The conditions in these tests were a primary side pressure of 394 kPa, steam flow rate of 37 g/s and 0.0097 inlet air mass (Test No. 26).

Figure 41a Tube 1 Temperatures, Test 26

Figure 41b Tube 2 Temperatures, Test 26

Figure 41c Tube 3 Temperatures, Test 26

Figure 41d Tube 4 Temperatures, Test 26

Figure 41e Tube 5 Temperatures, Test 26

Figure 41f Tube 6 Temperatures, Test 26

Centerline temperatures decrease with a shallow slope until most of the steam has condensed. The temperature decrease is steeper as the local partial pressure of steam falls rapidly. Condensation heat transfer is minimal downstream of this steep slope. The pool temperatures are at the local saturation temperature. There is a small water head on the tubes which elevates the temperature above 100°C by a few degrees. The surface temperatures are closer to the pool temperatures than the centerline temperatures. These thermocouples are soldered into grooves on the condenser tube outer surface and they measure the wall temperature but are heavily influenced by the surrounding coolant conditions. The exact location of the thermocouple junction, the location and amount of solder and imperfections in the tube manufacture all contribute to uncertainty in the surface temperature data.

To investigate the relative tube heat removal rates, the centerline, surface and pool temperatures are plotted in separate figures (Figure 42 through Figure 44). The data show that the centerline temperatures are similar for all six tubes, with tube 5 showing slightly lower temperatures than the other tubes. Only at the 2 m location do the temperatures differ significantly and this is due to the end of the condensation region

oscillating around the 2 m mark. This is a positive indication that each tube carries approximately the same heat load and all heat transfer surfaces are fully active. The pool temperatures are within the uncertainty of the thermocouples throughout the pool. A larger spread is exhibited in the surface temperatures, again showing the difficulty in obtaining a uniform and well-known measurement. The conditions in these tests were a primary side pressure of 394 kPa, steam flow rate of 37 g/s and 0.0097 inlet air mass (Test No. 26).

Figure 42 Centerline Temperature Profiles for Tube Bundle, Test 26

Figure 43 Tube Surface Temperature Profiles for Tube Bundle, Test 26

Figure 44 Pool Temperature Profiles for Tube Bundle, Test 26

The lower temperatures in tube 5 are believed to be due to an unintended inclination in the tube. This could be due to warping in the tube or faulty installation. This tube consistently showed lower temperatures and lower condensation heat removal rates in all tests. The inclination is believed to be due to spacers installed inside the pool that may not have completely prevented the tubes from sagging. The elevations of the condenser tubes are well-known at the inlet and outlet of the secondary side pool because they are fixed by the end plates (Figure 22).

The condensation collection rates for each condensation collection tank of Test No. 26 are shown in Figure 45. Other than tube 5 which shows a lower condensate collection rate, the collection rates in the other five tubes are within a few percent. Note that some tubes are measured in inches and others in centimeters. The measurement tape on condensate collection tank 6 was installed upside down.

Figure 45a Condensation Collection Rate for Tube 1, Test 26

Figure 45b Condensation Collection Rate for Tube 2, Test 26

Figure 45c Condensation Collection Rate for Tube 3, Test 26

Figure 45d Condensation Collection Rate for Tube 4, Test 26

Figure 45e Condensation Collection Rate for Tube 5, Test 26

Figure 45f Condensation Collection Rate for Tube 6, Test 26

As verification of the degree of steady state obtained in the system, the centerline temperatures for Test 26 are plotted as a function of time in Figure 46. Tube 5 temperatures are deleted since this tube was showing anomalous behavior as described above. The temperatures vary in response to the noncondensable gas venting. Considering that the manufacturer's stated uncertainty is $\pm 0.5^\circ\text{C}$, the temperatures are considered to be at a steady state when the heat exchanger is venting. The temperatures at the third measurement location vary because the end of the high condensation region is oscillating back and forth about this point (see Figure 42).

Figure 46a Time-dependence of Steady State Centerline Temperatures for Tube Bundle at First Measurement Location, Test 26

Figure 46b Time-dependence of Steady State Centerline Temperatures for Tube Bundle at Second Measurement Location, Test 26

Figure 46c Time-dependence of Steady State Centerline Temperatures for Tube Bundle at Third Measurement Location, Test 26

Figure 46d Time-dependence of Steady State Centerline Temperatures for Tube Bundle at Fourth Measurement Location, Test 26

Figure 46e Time-dependence of Steady State Centerline Temperatures for Tube Bundle at Fifth Measurement Location, Test 26

The centerline temperature figures also show that the top tubes carry the steam/air mixture at a temperature 3 to 5°C higher than the lower tubes. This small degree of thermal stratification that was present in all of the tests, despite a minimum thickness of 2 in. fiberglass insulation around the steam inlet plenum and the connection to the condenser tubes. The effect is to have a slightly larger temperature difference driving the heat transfer in the upper tubes, although the tube overall condensation rates and condensation lengths were not affected.

Figure 47 shows that the surface temperatures are constant. There is no relative trend in the order of the tube temperatures due to the uncertainty in the exact measurement details, as discussed above. Similar trends are seen at the other measurement locations.

Figure 47 Time-dependence of Surface Temperatures for Tube Bundle at First Measurement Location, Test 26

The pool temperatures in Figure 48 are at saturation throughout the pool. Similar trends are seen at the other locations.

Figure 48 Time-dependence of Pool Temperatures for Tube Bundle at First Measurement Location, Test 26

Effects of Noncondensable Gas

The effect of the NC gas is investigated by comparing the centerline temperature profiles. The centerline temperature corresponds to the local steam saturation temperature. Because Figure 38 through Figure 40 show that the pressure drop from test section inlet to vent outlet is less than 5.0 kPa and the oscillations due to venting are small, the pressure can be assumed constant in the condenser tubes. The centerline temperature can then be assumed to reflect the local steam partial pressure. At the inlet, the steam partial pressure is close to the total pressure. The difference with the total pressure at higher inlet air mass fractions is due to the air partial pressure. Therefore, the inlet temperature for the conditions with higher air mass fraction is relatively low due to the lower steam partial pressure.

From Figure 49 it can be seen that the condensation length is shorter for lower inlet air mass fractions because there is less condensation degradation by the noncondensable gas. The condensate rate measurement by condensate collection confirmed that for the 0.10 and 0.20 inlet air mass fraction cases, steam had been vented out. For the 0.01, 0.02 and 0.05 inlet air mass fraction cases, the condensate collection rate is close to the inlet steam mass flow rate and complete condensation may be assumed.

Figure 49a Tube 1 Centerline temperature profiles for various inlet air mass fractions
(Inlet steam flow rate: 36 g/s, Inlet pressure: 400 kPa)

Figure 49b Tube 2 Centerline temperature profiles for various inlet air mass fractions
(Inlet steam flow rate: 36 g/s, Inlet pressure: 400 kPa)

Figure 49c Tube 3 Centerline temperature profiles for various inlet air mass fractions
(Inlet steam flow rate: 36 g/s, Inlet pressure: 400 kPa)

Figure 49d Tube4 Centerline temperature profiles for various inlet air mass fractions
(Inlet steam flow rate: 36 g/s, Inlet pressure: 400 kPa)

Figure 49e Tube 5 Centerline temperature profiles for various inlet air mass fractions
(Inlet steam flow rate: 36 g/s, Inlet pressure: 400 kPa)

Figure 49f Tube 6 Centerline temperature profiles for various inlet air mass fractions
(Inlet steam flow rate: 36 g/s, Inlet pressure: 400 kPa)

Similar trends for the same flow rate at 200 kPa are seen in Figure 50. These tests did not experience condensation to the extent that the higher pressure tests did. The pressure effect is discussed below.

Figure 50a Tube 1 Centerline temperature profiles for various inlet air mass fractions
(Inlet steam flow rate: 36 g/s, Inlet pressure: 200 kPa)

Figure 50b Tube 2 Centerline temperature profiles for various inlet air mass fractions
(Inlet steam flow rate: 36 g/s, Inlet pressure: 200 kPa)

Figure 50c Tube 3 Centerline temperature profiles for various inlet air mass fractions
(Inlet steam flow rate: 36 g/s, Inlet pressure: 200 kPa)

Figure 50d Tube 4 Centerline temperature profiles for various inlet air mass fractions
(Inlet steam flow rate: 36 g/s, Inlet pressure: 200 kPa)

Figure 50e Tube 5 Centerline temperature profiles for various inlet air mass fractions
(Inlet steam flow rate: 36 g/s, Inlet pressure: 200 kPa)

Figure 50f Tube 6 Centerline temperature profiles for various inlet air mass fractions
(Inlet steam flow rate: 36 g/s, Inlet pressure: 200 kPa)

Pressure Effect

The effect of pressure is to increase the temperature difference that drives heat transfer and shorten the condensation length. Figure 51 shows the pressure effect for an inlet steam flow rate of 36 g/s and inlet mass fraction of 0.01. Figure 52 shows the pressure effect for the same inlet flow rate and an inlet air mass fraction of 0.05. Figure 53 shows the pressure effect for the same inlet flow rate and an inlet air mass fraction of

0.10. Only Tube 1 is shown since the centerline temperature profiles are nearly the same in the other five tubes.

The centerline temperature decreases sharply at the end of significant condensation and will approach the pool temperature when primary-to-secondary side heat transfer has terminated. The 400 kPa cases show the length required for nearly complete condensation increases with increasing inlet air mass fraction. Among the 200 kPa cases, only the 0.01 inlet air mass fraction case approached complete condensation.

This pressure affect is important for the PCCS during transient operation. A goal of the PCCS is to suppress containment pressure. These results indicate a higher heat removal rate at higher pressure. The system can then be expected to find a steady balance between a state of higher pressure and higher heat removal, versus lower pressure and less heat removal.

Figure 51 Tube 6 Centerline temperature profiles for various pressures (Inlet steam flow rate: 36 g/s, Inlet air mass fraction 0.01)

Figure 52 Tube 6 Centerline temperature profiles for various pressures (Inlet steam flow rate: 36 g/s, Inlet air mass fraction 0.05)

Figure 53 Tube 6 Centerline temperature profiles for various pressures
(Inlet steam flow rate: 36 g/s, Inlet air mass fraction 0.10)

Tube-to-Tube Effects

The effects that tubes have on each other can be important in a tube bundle heat exchanger. A prime concern in horizontal heat exchangers is that the upper tubes could become blanketed by the vapor leaving the lower tubes and the heat transfer would be reduced to forced convection to a vapor coolant.

Tests with vigorous boiling on the secondary side were run to investigate this effect. The void fraction was observed through the sight glasses to be higher around the upper tubes, however the heat removal rates were not reduced. In fact, the higher tubes were consistently performing at average or higher-than-average heat removal rates.

An alternative method to investigate the tube-to-tube effect is to continuously refill the secondary-side pool with coolant water from the bottom, such as would be done in an actual PCCS system, and observe the effect of a stratified coolant pool. In this case, cooler water was fed in to the bottom of the pool at two locations, at a flow rate that balanced the coolant vaporization rate.

Figure 54 shows the temperature distribution in the coolant pool during steady state with continuous refill (Test 34). The pool water surround the lower two tubes was cooler than the upper water by 2°C to 4°C. No differences in the performance of the individual condenser tubes were observed.

Figure 54 Secondary-side Pool temperatures with Continuous Pool Refill

Unexpected Observations

An unexpected observation was that the tests with vigorous coolant boiling experienced strong vibrations throughout the test section. This vibration probably affected the condensate distribution within the condenser tubes, although no direct measurement of this was possible. The vibration could be reduced to levels not noticeable to the students by maintaining the secondary side water level below the sight glass in the coolant vent tank or by reducing the steam flow rate. Although undetectable to nearby students, the vibrations may still have been present and affecting condensation phenomena.

Conditions for Stable and Satisfactory Operation

The vibrations mentioned under the Unexpected Observations title and the inability to condense all of the steam are the two operational factors that could affect adversely the heat exchanger performance. The vibrations could pose a threat to the structural integrity of the device if the boiling is extremely violent. On the other hand, the vibrations may enhance heat transfer. Any movement of the tubes should cause a thinner condensate layer on the condenser tube inner surface and the vapor bubbles on the outer surface were clearly shaken from the tubes. Vibrations should not be taken into account when sizing the heat exchanger because they are not enhancing heat transfer at low heat removal rates and gentler boiling conditions.

The inability to condense all of the steam at high steam flow rates and/or high air mass fractions suggests that the heat exchanger is undersized for these conditions. The pressure also played an important role in determining the amount of condensation. Most of the tests for which a large amount of steam passed through the heat exchanger were at 200 kPa, which is below the normal DBA operating range. In an actual LOCA scenario, the higher pressure tests which vented steam corresponded to conditions which would exist for a short time during the transient and could be tolerated by the PCCS and containment.

3.2.7.3 Transient Experiments

Test Overview

Transient tests were run to investigate the horizontal heat exchanger's behavior during the long-term cooling phase. The steam flow rate was decreased as a function of time to simulate the decrease of decay heat. The air flow rate was controlled as a function of time. Initially, the air mass fraction was low because the initial blowdown was over and most of the noncondensable gas would have been blown into the wetwell. The air mass fraction increased because, after the initial blowdown and due to the decreasing RPV steam release rate, noncondensable gases could rise from the lower drywell and other pockets of low circulation. The PCCS would consequently be ingesting large amounts of noncondensable gas from the drywell. Later, the air mass fraction was low because most of the noncondensable gases would have been purged to the wetwell. Initial conditions for these tests were obtained by looking at past SBWR analysis and estimating appropriate parameters.

Figure 55 and Figure 56 show the heater power and air mass flow rate curves for a test that began at 650 kPa in the steam generator and 100 kW of heater power. A second transient was run that began at a lower heater power. Note that these tests are not intended to predict the long-term pressure. These tests are to examine the heat exchanger component performance. The primary side was set to vent automatically when the pressure exceeded 250 kPa, which is near to the expected long-term containment pressure. Since the trends are similar to the first transient test, only the first test is detailed below.

Figure 55 Heater Power for Transient Test (Test 31)

Figure 56 Air Flow Rate for Transient Test (Test 31)

The test section including the secondary-side pool was initially at about 40°C and the condenser tubes were filled with stagnant air at the same temperature. The steam generator was heated to the prescribed initial pressure of 650 kPa. The condensate collection tanks were not drained so that information on the heat removal rates could be obtained. Data was recorded for the entire transient starting from opening of the steam line and cold pool startup until the steam flow rate decreased to zero.

Transient Behavior of Tube Bundle Heat Exchanger

The pressure difference between the steam generator and test section and the heater power determined the inlet steam flow rate. Figure 57 and Figure 58 show the pressures and the inlet steam flow rate respectively. The decrease in test section pressure and increase in inlet steam flow rate at about 1,600 seconds was caused by draining of the condensate tanks.

Figure 57 Steam Generator and Test Section Pressures (Test 31)

Figure 58 Inlet Steam Flow Rate (Test 31)

The rate of venting is shown in Figure 59. The highest frequency of vent opening occurred prior to 400 seconds, corresponding to the highest inlet air mass fraction in the test section. Once the inlet air mass fraction had been decreased to its long-term constant value, the venting frequency also decreased. From 400 sec. to the end of the test, although the air mass fraction was constant, the venting frequency further decreased. The heat exchanger was better able to condense all of the steam while venting noncondensable gas at a slower rate because the inlet steam flow rate was also decreasing. ***These venting trends are typical of the behavior expected in the PCCS and partially verify that the horizontal heat exchanger would be an appropriate PCCS component.*** The automated venting system also represents a significant improvement in PCCS studies over other those that had manual venting because the system is allowed to take on its natural behavior.

Figure 59 Test Section Inlet Pressure (Test 31)

The heatup and eventual cooldown of the primary side steam/gas mixture in the heat exchanger tubes is shown in Figure 60 for the five axial measurement locations. In the legend, “C” stands for centerline temperature, the first digit provides the location number and the second digit provides the tube number with Tube 1 being the top tube. The first location heats up to saturation temperature at the primary side pressure within 30 seconds, with the second, third and fourth locations taking progressively longer. Temperatures at

the fifth location rise to the pool temperatures, indicating that the condensation region does not extend for the full length of the tubes. The fourth location is showing evidence of condensation only for a short time. For the transient simulated here, the steam could be condensed upstream of the fourth measurement section and the condenser is oversized. By the end of the test, only the first measurement section remained at saturation temperature.

Figure 60a Transient Centerline Temperatures at First Cross Section (Test 31)

Figure 60b Transient Centerline Temperatures at Second Cross Section (Test 31)

Figure 60c Transient Centerline Temperatures at Third Cross Section (Test 31)

Figure 60d Transient Centerline Temperatures at Fourth Cross Section (Test 31)

Figure 60e Transient Centerline Temperatures at Fifth Cross Section (Test 31)

The secondary-side pool heatup is shown in Figure 61. In the legend, “P” stands for pool temperature, the first digit provides the location number and the second digit provides the tube number with Tube 1 being the top tube. Pool temperatures were not measured at the second or fourth measurement location. All three measurement cross sections exhibited a fairly linear heatup rate to saturation temperature, with the top of the pool heating earlier and faster than the bottom of the pool. The first, third and fifth measurement locations begin to heat up at about 43 sec., 84 sec. and 120 seconds, respectively.

Figure 61a Transient Pool Temperatures at First Cross Section (Test 31)

Figure 61b Transient Pool Temperatures at Third Cross Section (Test 31)

Figure 61c Transient Pool Temperatures at Fifth Cross Section (Test 31)

The details of the heatup of each tube are shown in Figure 62. As the front of the hot steam/air gas mixture traveled further down the tubes, the temperatures began to rise. The outlet of each tube did not increase to the primary side saturation temperature because the condensation region did not extend to the downstream sections of the tubes. Instead, these sections leveled out within a few degrees of the pool-side temperatures.

The oscillations in the decreasing temperatures at measurement location three (C3-*n*) between 1000 and 2000 seconds were synchronized with the noncondensable gas venting. The end of the condensation region was shifting back and forth across this measurement region due to the venting.

Figure 62a Transient Centerline Temperatures for Tube 1 (Test 31)

Figure 62b Transient Centerline Temperatures for Tube 2 (Test 31)

Figure 62c Transient Centerline Temperatures for Tube 3 (Test 31)

Figure 62d Transient Centerline Temperatures for Tube 4 (Test 31)

Figure 62e Transient Centerline Temperatures for Tube 5 (Test 31)

Figure 62f Transient Centerline Temperatures for Tube 6 (Test 31)

The condensate heat removal rate decreased in response to a decreasing inlet steam flow rate. Figure 63 and Figure 64 show the rate of water level change prior to condensate draining in condensate collection tanks 1 and 6, for the top and bottom tube respectively.

Figure 63 Transient Condensate Collection Rate for Tube 1 (Test 31)

Figure 64 Transient Condensate Collection Rate for Tube 6 (Test 31)

The change in the condensation length, or portion of the tubes which are experiencing condensation heat transfer, provides insight for the heat exchanger sizing. Figure 65 and Figure 66 illustrate how the active condensation region initially increases to match the heat load and then decreases as the steam supply rate diminishes. The heat exchanger is shown capable of following the heat load.

Figure 65a Tube 1 Early Term Transient Condensation Length (Test 31)

Figure 65b Early Term Tube 2 Transient Condensation Length (Test 31)

Figure 65c Early Term Tube 3 Transient Condensation Length (Test 31)

Figure 65d Early Term Tube 4 Transient Condensation Length (Test 31)

Figure 65e Early Term Tube 5 Transient Condensation Length (Test 31)

Figure 65f Early Term Tube 6 Transient Condensation Length (Test 31)

Figure 66a Long Term Tube 1 Transient Condensation Length (Test 31)

Figure 66b Long Term Tube 2 Transient Condensation Length (Test 31)

Figure 66c Long Term Tube 3 Transient Condensation Length (Test 31)

Figure 66d Long Term Tube 4 Transient Condensation Length (Test 31)

Figure 66e Long Term Tube 5 Transient Condensation Length (Test 31)

Figure 66f Long Term Tube 6 Transient Condensation Length (Test 31)

The horizontal heat exchanger was seen able to perform well and regulate the heat transfer rate according to the heat load. This self-regulation is in large part due to the noncondensable gases, which play an important role in system operation. As the heat

load decreased, the noncondensables occupied an increasing fraction of the tube volumes, as indicated by Figure 66. The automated venting system let the heat exchanger purge noncondensables when the pressure increased above a setpoint, which corresponds to a water head on a submerged pipe in a real plant. This self-regulation of the heat transfer rate is one of the attractive features of the vertical tube bundles. A major conclusion of these tests is the confirmation of the horizontal tube bundles to self-regulate their heat removal rate.

3.2.8 Conclusions Regarding Tube Bundle Tests

The transient tests provided valuable data regarding horizontal tube bundle performance under Design Basis Accident conditions.

The tube bundle experiments were performed to obtain the following key data:

1. Tube-to-tube effects, as evidenced by differences in condensate drain rates among the tubes and secondary side observations
2. System response to the redistribution of noncondensable gases
3. Transient tests in which the heat input was decreased with time to simulate decay heat

As described in this section, all of these data were obtained and positive conclusions about the horizontal heat exchanger's capabilities were reached.

3.3 Analysis Model Development

The analysis model development was carried out as a two-step process. First, a nonlinear fit was performed for the single-tube experimental data to obtain correlations for the condensation heat transfer coefficients for the top and bottom of the tube separately, and then these correlations were incorporated into the reactor safety analysis code RELAP5 to model a single tube of the bundle experimental facility and compare with the experimental data. In the second step, a mechanistic model independent of the experimental data was developed based on the knowledge of the horizontal condensation phenomena gained from experiment and the evaluation of some current models. Then this mechanistic model was implemented into a stand-alone program to simulate the single tube and compare with the experimental data. In such a way, the mechanistic model can be verified since it was developed independent of the experimental data.

3.3.1 Condensation Heat Transfer Coefficient Empirical Model

This empirical model was developed in a manner convenient for incorporating into a reactor safety analysis code. Since the codes usually use non-iterative methods to calculate the heat transfer coefficient and only give control volume-averaged outputs, the consideration used in developing the empirical correlation is that it should use the current available parameters in the code to calculate the heat transfer coefficient and the method should be simple to save computational time.

From the experimental data, the heat transfer coefficients at the top and the bottom of the tube were calculated along the tube. Then the local condensation Nusselt number could be calculated. Nonlinear fitting of the experimental data gave two correlations for the Nusselt number as follows:

$$Nu_{top} = 73.94 Re_{mix}^{0.204} Pr_l^{-1.927} M_a^{-0.103} \quad (6)$$

$$Nu_{bot} = 1758.20 Re_{mix}^{-0.171} Pr_l^{-2.436} M_a^{-0.106} \quad (7)$$

From the correlations, the functional dependencies can be seen. For the top of the tube, the Nusselt number increases as the mixture Reynolds number increases and decreases as the air mass fraction increases. Conversely, for the bottom of the tube, it can be seen that the Nusselt number decreases as the mixture Reynolds number increases. This trend is purely from the experimental data. A positive exponent was expected for the mixture Reynolds number in both correlations. It is believed that mixture Reynolds number exponent for the bottom is negative because a factor incorporating the condensate film effect was not included, while the single-tube data is well represented by this correlation, its applicability to other experiment is questionable. For the Prandtl number dependency, the condensate temperature decreases due to heat transfer to the coolant, therefore liquid Prandtl number increases along the tube. The Nusselt number also decreases along the tube, resulting in a negative exponent for the Prandtl number.

From the experimental data, when the cross section averaged heat flux is taken as the average of the top and bottom local heat fluxes, integration along the tube results in an overall heat removal rate very close to the overall heat transfer rate calculated from the measured coolant temperature rise across the test section. Thus it is reasonable to take the cross section-averaged Nusselt number as the average of Nu_{top} and Nu_{bot} . The condensation heat transfer coefficient can be calculated as:

$$h_c = \frac{(Nu_{top} + Nu_{bot})}{2} \frac{k_l}{D_{in}} \quad (8)$$

where the condensation heat transfer coefficient accounts for heat transfer from the bulk steam/air mixture to the tube inner surface.

3.3.2 Model Implementation and Verification

This correlation was incorporated into the RELAP5 code to simulate the single tube facility and the tube bundle facility.

For the RELAP5 calculation, an input deck corresponding to the single tube facility was setup as shown in Figure 67.

Figure 67 RELAP5 Nodalization of the Horizontal Single-tube Experimental Facility

The heat transfer length is divided into 30 control volumes; the length of each control volume is 0.1 m. The inlet steam flow rate, inlet pressure, inlet air mass fraction, coolant flow rate, and the coolant inlet temperature are specified at the same value as in the experiment.

As shown in

Figure 68, the experiment results are compared with RELAP5 predictions. The data compared are the local heat fluxes. RELAP5 calculates the control volume-averaged heat fluxes on the condenser tube wall, whereas the experimental results provide the heat fluxes on the top and bottom of the condenser tube.

Figure 68 Heat Flux Comparison of Experiment Results and RELAP5 Predictions

From

Figure 68 it can be seen that the code underpredicts the local heat fluxes. Also, the change in condensation heat transfer models does not make a significant difference in the local heat flux. This is probably due to the fact that the coolant side is in forced convection. Since the heat transfer coefficient for the forced convection is generally one order in magnitude less than the condensation heat transfer coefficient, the major resistance from the tube centerline is dominated by the coolant side resistance. Thus the change in the condensation model does not give a significant change in the overall local heat transfer coefficient. The forced convection heat transfer on the coolant side should be improved.

For the bundle test, since no pool side tube-to-tube effect model has been developed, only one of the six condenser tubes was modeled. The steam flow and air flow rates are given as one sixth of the total flow rates. The nodalization is setup as shown in Figure 69.

Figure 69 RELAP5 Nodalization of a Single Tube of the Tube-bundle Facility

In the tube bundle facility, the coolant boils on the outer surface of the tube, thus the coolant side can be modeled as a pipe of one control volume. The environment is modeled by time-dependent volume 320. The steam that is boiled off from the coolant side vents through the single junction 310 into the environment. The primary side is modeled the same as the single tube facility, except that the condenser tube is divided into forty control volumes of 0.1 m length since the heat transfer length of the condenser tubes in the bundle test facility is about 3.96 m.

In the calculation, the inlet pressure is given to be the same as that in the experiment. The inlet steam flow rate and air mass flow rate is given as one-sixth of those values in the experiment. The coolant pool initial temperature is given as very close to the boiling temperature at one atmosphere pressure so that the calculation can reach a steady state quickly.

As for the comparisons with the experimental data, since the tube bundle facility is only instrumented to measure the centerline, the tube outer surface and the coolant temperatures in a few locations along the tube, no detailed local heat fluxes can be calculated on the tube. The local variables that can be directly compared with the experimental data are the centerline temperatures and the tube surface temperatures. Also as stated in previous sections, the measured tube outer surface temperatures has large uncertainty, this data is not compared with the RELAP5 simulation results.

Figure 70 shows a comparison of the centerline temperatures of the original RELAP5 and the modified RELAP5 simulation results as well as the measured values from the experiment. The result clearly shows that the modified RELAP5 gives better predictions. The same trend was seen in the results for 0.02, 0.05, 0.10 and 0.20 inlet air mass fractions.

Figure 70 Comparison of RELAP5 Results with Experimental Data of Tube-bundle Facility Centerline Temperatures

For the tube bundle facility, the pool side heat transfer mode is saturated nucleate boiling, thus the heat transfer coefficient is much higher compared to the forced convection heat transfer coefficient. Under this condition, the effect of the condensation heat transfer coefficient is significant, as shown in Figure 71. The calculation results for local heat fluxes in the condensation region also show much higher values for the modified RELAP5.

Figure 72 shows a comparison of condensate collection rates for one of the test conditions. A schematic of the relative tube location within the pool is also shown in the figure. From the calculation results, for all 5 inlet air mass fractions, complete condensation occurred. The difference in experimental values of the condensate collection rate for each individual tube may be a result of the tube-to-tube effect in the bundle as discussed in section 3.2.

Figure 71 Comparison of Wall Heat Flux Calculations between the Original and Modified RELAP5

Figure 72 Comparison of Measured and Predicted Condensate Collection Rates

3.3.3 Mechanistic Model Development

It is well known that the empirical correlations directly derived from the experimental data have the limitation of strong dependencies on the experimental facility and parameters. This mechanistic model is developed independent of the current experimental data. Instead it is based on the phenomena and theoretical models that reflect the physics of the phenomena, thus it has a wider range of applicability compared to the empirical correlations.

3.3.3.1 Description of the Theories

The flow regimes inside the condenser tube are mainly wavy and stratified flow. The characteristics of these two flow regimes are that the film on the top part of the tube flows down the wall to the bottom of the tube and forms a liquid pool at the bottom of the tube. The pool flows in the axial direction due to the shear of the steam/air mixture. Thus the tube is peripherally divided into two sections, the top and the bottom sections. For the top section, the phenomenon is film condensation and for the bottom section, the phenomenon is forced convection condensation.

For the steam/air mixture that flows in the core, diffusion layer theory developed by Peterson [1993] is applied. The theory is based on the analogy between heat and mass transfer. The detailed description of the theory will be stated later.

It is obvious that the heat transfer mode through the tube wall is heat conduction and the heat transfer from the wall to the coolant water is single-phase forced convection for a concentric tube heat exchanger.

From the above description, the definitions of different temperatures associated with the phenomena are shown in Figure 73. The steam and air mixture is assumed to be in thermal equilibrium and the temperature is T_b^s . The gas mixture/liquid interface temperature has different values from the top to the bottom and the temperatures are represented as $T_{i,top}$ and $T_{i,bot}$. The tube inner and outer surfaces are defined as $T_{wi,top}$, $T_{wo,top}$, $T_{wi,bot}$ and $T_{wo,bot}$. The coolant temperature is assumed to be uniform along the periphery as T_{co} . It should be noted that the liquid film thickness on the tube wall is exaggerated.

From the definitions of temperatures and the illustration of the phenomena, an equivalent resistance network can be drawn as shown in Figure 74. The form of the resistance network is symmetrical from the mixture bulk outward, though the values are different.

Each of the resistances is described here from the bulk mixture to the coolant. The heat transfer from the bulk mixture to the interface is separated into condensation and sensible heat transfer. The heat transfer coefficients are represented as h_c and h_s . The heat transfer from the interface to the wall inner surface is defined as the film heat

transfer coefficient h_f . The resistance of the wall is represented as R_{wall} . The convective heat transfer coefficient from the wall outer surface to the coolant is represented as h_{co} .

Figure 73 Temperature Definitions for Horizontal Condensation Phenomena with Forced Convection Coolant Side Heat Transfer

Figure 74 Equivalent Heat Transfer Resistance Network

After these resistances or heat transfer coefficients are defined, appropriate methods to calculate these heat transfer coefficients are developed. This includes the evaluation of existing models and development of new models as necessary.

The first and one of the most important models related to the phenomena is the heat transfer from the bulk mixture to the interface. For this heat transfer, the diffusion layer model developed by Peterson [1993] is applied. The diffusion layer model separates the heat transfer into condensation and sensible components. Due to the condensation on the

interface, the steam concentration near the interface is lower than that in the bulk. Thus the interface temperature is lower than the mixture bulk temperature. The sensible component considers the part of the heat transfer due to the temperature difference. Due to the steam concentration gradient from the bulk mixture to the interface, the movement of steam from the bulk to the interface is considered as a diffusion process and the heat transfer associated with the diffusion mass transfer process is defined as the condensation heat transfer component. The diffusion layer theory also includes an equivalent condensation thermal conductivity which is similar to the mixture thermal conductivity for sensible heat transfer, thus the condensation heat transfer coefficient can be calculated easily. Detailed model equations are stated later in this section.

Since the heat transfer mode on the top part of the tube is considered as film condensation, the film heat transfer coefficient for this part is calculated from the Nusselt theory. The heat transfer through the liquid pool at the bottom is considered as forced convection and is calculated using a formulation similar to the Dittus-Bolter equation.

The wall resistance is calculated using the conduction equation in cylindrical coordinates. The coolant convective heat transfer coefficient is calculated by a single-phase forced convection heat transfer model.

Detailed model equations of the heat transfer models stated above as well as flow regime models will be presented in the next section.

3.3.3.2 Model Implementation

These models and correlations were integrated together into a stand alone computer program. The program divides the test section into small cells along the axial direction and takes the inlet steam mass flow rate, inlet air mass flow rate, condenser tube inlet pressure, and coolant inlet mass flow rate and inlet temperature as input variables. Then the heat transfer and flow conditions within each small section are calculated based on the inlet condition of that section.

The major steps of the program are listed as follow:

1. Estimate the coolant outlet temperature. This is because that the heat exchanger modeled is a counter-current double-pipe type. The coolant outlet corresponds to the steam/air mixture inlet.
2. Calculate the heat transfer rate of a tube section based on the exit conditions of the previous tube section. (For the first tube section, assume an energy loss rate upstream).
3. Evaluate the flow condition and fluid properties at the exit of the current tube section.
4. Identify the flow regime at the exit of the current tube section.
5. Calculate the void fraction at the exit of the current tube section.
6. Repeat steps 2 to 5 for all the tube sections.

7. Compare the calculated coolant inlet temperature with the given coolant inlet temperature. If the difference exceeds a threshold (0.01 °C), then modify the coolant outlet temperature and repeat from 2.

A flow diagram of the program is shown in Figure 75. The capital letters represents the major modules of the program. The function of each module and detailed model equations used in each individual module are given in this section.

T_CO_OUT:

This module predicts the coolant outlet temperature from the input variables. It assumes that all the steam is condensed.

$$T_{co,out} = T_{co,in} + \frac{\dot{m}_s h_{fg}}{\dot{m}_{co} C_p} \quad (9)$$

HEAT_TRANSFER:

This module calculates the total heat transfer rate within a cell based on the local flow regime at the inlet of the cell. The flow chart of this module is shown in Figure 76.

The subroutine Q_TOP calculates the local wall heat flux when the flow regime is in stratified or wavy flow. The subroutine Q_BOTTOM calculates the heat transfer when the heat transfer through the film is forced convection. This is generally true for the liquid pool at the bottom of the tube when the flow regime is stratified or wavy flow and applies to the whole tube periphery when the flow regime is annular flow. For the bottom liquid pool of the stratified and wavy flow and annular flow, the heat transfer is shear dominated; for the top part of the tube when the flow regime is stratified or wavy flow, the heat transfer is gravity dominated.

After the local wall heat flux is calculated, the program calculates the total heat transfer rate of the cell based on the stratification angle θ . The definition of stratification angle is shown in Figure 77. The stratification angle is calculated as 2π for annular flow:

$$Q^n = \left(q''_{bot} \frac{\theta}{2} + q''_{top} \left(\pi - \frac{\theta}{2} \right) \right) D_{in} \Delta x \quad (10)$$

Figure 75 Flow Chart of the Mechanistic Model Program

Figure 76 Flow Chart of the Heat Transfer Module

Figure 77 Definition of Stratification Angle for Stratified and Wavy Flow

FLOW:

This module calculates the mass flow rates of each phase at the exit of the current cell based on the heat transfer rate calculated by the module HEAT_TRANSFER. The coolant temperature at the exit of current cell is also calculated.

$$\dot{m}_s^{n+1} = \dot{m}_s^n - \frac{Q^n}{h_{fg}} \quad (11)$$

$$T_{co}^{n+1} = T_{co}^n - \frac{Q^n}{\dot{m}_{co} C_p} \quad (12)$$

FLOW_REGIME

This module calculates the flow regime at the outlet of the current cell. The method proposed by Jaster and Kosky [1976] is used. This flow regime transition criterion is for pure steam condensation in a horizontal tube.

Calculate the interface friction factor [Wallis, 1969]:

$$f_v = \frac{0.079}{\text{Re}_{mix}^{0.25}} \quad (13)$$

The mixture Reynolds number is calculated as:

$$\text{Re}_{mix} = \frac{4(\dot{m}_s + \dot{m}_{air})}{\pi D_{in} \mu_{mix}} \quad (14)$$

Local mixture quality:

$$x = \frac{\dot{m}_s + \dot{m}_{air}}{\dot{m}_{s,in} + \dot{m}_{air}} \quad (15)$$

Martinelli parameter [Wallis, 1969]:

$$X^2 = \left(\frac{\mu_l}{\mu_{mix}} \right)^{0.25} \left(\frac{\rho_{mix}}{\rho_l} \right) \left(\frac{1-x}{x} \right)^{1.75} \quad (16)$$

Liquid phase Reynolds number:

$$\text{Re}_l = \frac{4m_l}{\pi D_{in} \mu_l} \quad (17)$$

Non-dimensional film thickness [Kosky, 1971]:

$$\delta^+ = \begin{cases} \sqrt{\frac{\text{Re}_l}{2}} & \text{if } \text{Re}_l \leq 1250 \\ 0.0504 \text{Re}_l^{0.875} & \text{if } \text{Re}_l > 1250 \end{cases} \quad (18)$$

Flow regime identification criteria (derived by Jaster and Kosky [1976]):

$$F = \frac{(1 + X^{\frac{2}{M}})^{\frac{3M}{2}} \rho_l \left(\frac{\dot{m}_s + \dot{m}_{air}}{0.25 \pi D_{in}^2} \right)^3 f_v^{\frac{3}{2}}}{2\sqrt{2} (\rho_{mix} \rho_l)^{\frac{3}{2}} \mu_l \delta^+ g}, \quad M = 5.13 \quad (19)$$

$$Flow\ Regime = \begin{cases} Annular & \text{if } F > 29 \\ Transitional & \text{if } 29 \geq F \geq 5 \\ Stratified & \text{if } F < 5 \end{cases} \quad (20)$$

VOID_FRACTION

This module calculates the local void fraction based on the flow condition and flow regime.

For annular flow, the CISE model is used [Premoli et al., 1971]. The CISE model is a slip ratio model that is formulated as follows.

$$Y = \frac{\rho_l}{\rho_{mix}} \frac{x}{1-x} \quad (21)$$

$$E1 = 1.578 \left(\frac{(\dot{m}_s + \dot{m}_{air}) D_{in}}{\mu_l} \right)^{-0.19} \left(\frac{\rho_l}{\rho_{mix}} \right)^{0.22} \quad (22)$$

$$E2 = 0.0273 \left(\frac{(\dot{m}_s + \dot{m}_{air})^2 D_{in}}{\sigma \rho_l} \right) \left(\frac{(\dot{m}_s + \dot{m}_{air}) D_{in}}{\mu_l} \right)^{-0.51} \left(\frac{\rho_l}{\rho_{mix}} \right)^{-0.08} \quad (23)$$

Slip ratio:

$$S = 1 + E1 \sqrt{\frac{Y}{1 + Y \cdot E2}} - Y \cdot E2 \quad (24)$$

Void fraction:

$$\alpha = \frac{1}{1 + \frac{1-x}{x} \frac{\rho_{mix}}{\rho_l} S} \quad (25)$$

For stratified and wavy flow, the Taitel and Dukler model [1976] is used to calculate the nondimensional liquid level and thus the void fraction, the stratification angle is also calculated. It is noticed that for the current test conditions and flow regime transition criterion, the void fraction profile along the tube is a smooth curve. No significant discontinuities between models were seen.

Q_BOTTOM

This heat transfer subroutine calculates the wall heat flux when the heat transfer model through the condensate layer is forced convection, which applies to annular flow and the bottom part of the stratified flow. In the process, the heat transfer coefficients in Figure 3.7 are calculated using their corresponding models; an iterative scheme is applied

to calculate the interfacial temperature and gas mole fraction. The structure of this subroutine is as follows:

Step 1: Calculate the bulk air mole fraction in the mixture at the cell inlet.

$$x_{gb} = \frac{\dot{m}_{air}/M_{air}}{\dot{m}_s/M_s + \dot{m}_{air}/M_{air}} \quad (26)$$

Step 2: Assume local interface gas mole fraction and obtain interface temperature.

$$x_{gi} = x_{gb} + 0.001 \quad (27)$$

$$T_i = Tsat(P^*(1 - x_{gi})) \quad (28)$$

Step 3: Evaluate mixture properties at T_{mix} .

$$T_{mix} = \frac{1}{2}(T_b^s + T_i) \quad (29)$$

$$\rho_{mix} = \frac{(\dot{m}_s + \dot{m}_{air})}{\left(\frac{\dot{m}_s}{\rho_s} + \frac{\dot{m}_{air}}{\rho_{air}}\right)} \quad (30)$$

Mixture viscosity [Peterson et al., 1993]:

$$\phi_v = \frac{\left[1 + (\mu_{air}/\mu_s)^{0.5} (M_s/M_{air})^{0.25}\right]^2}{\left[8(1 + M_{air}/M_s)\right]^{0.5}} \quad (31)$$

$$\phi_{air} = \frac{\left[1 + (\mu_s/\mu_{air})^{0.5} (M_{air}/M_s)^{0.25}\right]^2}{\left[8(1 + M_s/M_{air})\right]^{0.5}} \quad (32)$$

$$\mu_{mix} = \frac{x_{air}\mu_{air}}{x_{air} + (1 - x_{air})\phi_v} + \frac{(1 - x_{air})\mu_s}{x_{air}\phi_{air} + (1 - x_{air})} \quad (33)$$

Mixture thermal conductivity (thermal conductivity for sensible heat transfer):

$$k_s = k_{mix} = \frac{x_{air}k_{air}}{x_{air} + (1 - x_{air})\phi_v} + \frac{(1 - x_{air})k_s}{x_{air}\phi_{air} + (1 - x_{air})} \quad (34)$$

Mixture heat capacity:

$$c_{p,mix} = x_{air}c_{p,air} + (1 - x_{air})c_{p,s} \quad (35)$$

The binary mass diffusivity is calculated using the correlation [Perry et al., 1985]:

$$D_{AB} = \frac{10^{-7} T_{mix}^{1.75} [(M_{air} + M_s)/(M_{air}M_s)]^{0.5}}{P \left(v_{air}^{\frac{1}{3}} + v_s^{\frac{1}{3}} \right)^2} \quad (36)$$

Step 4: Calculate the wall resistance.

$$R_{wall} = \frac{D_{in} \ln \left(\frac{D_{out}}{D_{in}} \right)}{2k_{wall}} \quad (37)$$

Step 5: Calculate the heat transfer coefficient on the coolant side.

Nusselt number:

$$Nu_{co} = 0.023 Re_{co}^{0.8} Pr^{0.4} \quad (38)$$

Heat transfer coefficient:

$$h_{co} = \frac{Nu_{co} k_{co}}{D_{co}} \quad (39)$$

Step 6: Calculate the sensible and condensation heat transfer coefficients.

Single-phase friction factor [Moody, 1944]:

$$f_m = 1.375 \times 10^{-3} \left(1 + \frac{2154.4}{Re_{mix}} \right) \quad (40)$$

Interfacial shear coefficient [Whalley and Hewitt, 1968]:

$$f_i = f_m \left(1 + 24 \left(\frac{\rho_l}{\rho_{mix}} \right)^{\frac{1}{3}} \frac{\delta}{D_{in}} \right) \quad (41)$$

Sherwood number for condensation heat transfer [Gnielinski, 1976]:

$$Sh = \max \left(\frac{\frac{f_i}{2} (\text{Re}_{mix} - 1000) Sc}{1 + 12.7 \left(\frac{f_i}{2} \right)^{\frac{1}{2}} (Sc^{\frac{2}{3}} - 1)} ; 10 \right) \quad (42)$$

Nusselt number for sensible heat transfer [Gnielinski, 1976]:

$$Nu = \max \left(\frac{\frac{f_i}{2} (\text{Re}_{mix} - 1000) \text{Pr}}{1 + 12.7 \left(\frac{f_i}{2} \right)^{\frac{1}{2}} (\text{Pr}^{\frac{2}{3}} - 1)} ; 10 \right) \quad (43)$$

Calculate the equivalent thermal conductivity of condensation:

$$\phi = - \frac{\ln \left(\frac{1 - x_{gb}}{1 - x_{gi}} \right)}{\ln (x_{gb} / x_{gi})} \quad (44)$$

$$k_c = \frac{1}{\phi T_{mix}} \left(\frac{h_{fg}^2 P M_s^2 D_{AB}}{R^2 T_{mix}^2} \right) \quad (45)$$

Sensible heat transfer coefficient:

$$h_s = Nu \frac{k_s}{D_{in}} \quad (46)$$

Condensation heat transfer coefficient:

$$h_c = Sh \frac{k_c}{D_{in}} \quad (47)$$

Step 7: Calculate the film heat transfer coefficient.

Film thickness:

$$\delta_f = \frac{D_{in}}{2} \left(1 - \sqrt{1 - \frac{2\pi(1-\alpha)}{2\pi - \theta}} \right) \quad (48)$$

Phase Velocity:

$$u_l = \frac{j_l}{1-\alpha}, \quad u_{mix} = \frac{j_{mix}}{\alpha} \quad (49)$$

Interface roughness factor [Thome et al., 2003]:

$$f = 1 + \left(\frac{u_{mix}}{u_l} \right)^{0.5} \left(\frac{(\rho_l - \rho_{mix}) g \delta^2}{\sigma} \right)^{\frac{1}{4}} \left(\frac{G}{G_{strat}} \right) \quad (50)$$

$$G = \frac{\dot{m}_{s,in} + \dot{m}_{air}}{\frac{\pi D_{in}^2}{4}} \quad (51)$$

$$G_{strat} = \left\{ \frac{(226.3)^2 A_{Ld} A_{Vd}^2 \rho_{mix} (\rho_l - \rho_{mix}) \mu_l g}{x^2 (1-x)} \right\}^{\frac{1}{3}} + 20x \quad (52)$$

$$x = \frac{\dot{m}_s + \dot{m}_{air}}{\dot{m}_{s,in} + \dot{m}_{air}} \quad (53)$$

$$Re_l = \frac{4 \dot{m}_l \delta}{(1-\alpha) \mu_l} \left(\frac{4}{\pi D_{in}^2} \right) \quad (54)$$

Film heat transfer coefficient [Thome et al., 2003]:

$$h_f = 0.003 Re_L^{0.74} Pr_L^{0.5} \frac{k_l}{\delta} f \quad (55)$$

Step 8: Calculate the local heat flux.

Total heat transfer coefficient:

$$h_{total} = \frac{1}{\frac{1}{h_f} + \frac{D_{in}}{D_{out}} \frac{1}{h_{co}} + R_{wall} + \frac{1}{h_c + h_s}} \quad (56)$$

Local heat flux:

$$q''_{bot} = h_{total} (T_b^s - T_{co}) \quad (57)$$

Step 9: Calculate the interface temperature and air mole fraction.

Interface temperature:

$$T_i = T_b^s - \frac{q''_{bot}}{(h_c + h_s)} \quad (58)$$

Interface air mole fraction:

$$x_{gi} = 1 - P_{sat}(T_i)/P \quad (59)$$

Step 10: Compare the interface air mole fraction with the assumed value in Equation (27). If the difference is greater than 1.0e-3, then update the values and calculate from Step 3 again until convergence is achieved.

Q_TOP

This heat transfer subroutine calculates the wall heat flux when the heat transfer model through the condensate layer is conduction, which applies to the top part of the tube when the local flow regime is stratified or wavy flow. The difference between Q_TOP and Q_BOTTOM is only in the way the film heat transfer coefficient is calculated. Q_TOP calculates the film heat transfer coefficient using an iterative method. From Step 1 to Step 6, the equations used are same.

Step 7: Calculate the local heat flux and the film heat transfer coefficient.

- a. Calculate a wall heat flux assuming no film exists.

$$q''_{top,1} = \frac{T_b^s - T_{co}}{\frac{1}{h_c + h_s} + R_{wall} + \frac{D_{in}}{D_{out}} \frac{1}{h_{co}}} \quad (60)$$

- b. Calculate the film heat transfer coefficient [Thome et al., 2003].

$$h_f = 0.655 \left[\frac{\rho_l (\rho_l - \rho_{mix}) g h_{fg} k_l^3}{\mu_l D_{in} q''_{top,1}} \right]^{\frac{1}{3}} \quad (61)$$

- c. Recalculate the local heat flux considering the film resistance.

$$q''_{top} = \frac{T_b^s - T_{co}}{\frac{1}{h_c + h_s} + R_{wall} + \frac{D_{in}}{D_{out}} \frac{1}{h_{co}} + \frac{1}{h_f}} \quad (62)$$

- d. Compare $q''_{top,1}$ and q''_{top} , if the difference less than 1 W/m², then stop iteration, otherwise, set $q''_{top,1} = q''_{top}$, repeat from b.

Step 8: Calculate the interface temperature and air mole fraction.

Interface temperature:

$$T_i = T_b^s - \frac{q_{bot}''}{(h_c + h_s)} \quad (63)$$

Interface air mole fraction:

$$x_{gi} = 1 - P_{sat}(T_i)/P \quad (64)$$

Step 9: Compare the interface air mole fraction with the assumed value in Equation (27). If the difference is greater than 1.0e-3, then update the values and calculate from Step 3 again until convergence is achieved.

3.3.3.3 Model Verification

The mechanistic model is verified against the single tube experimental data. In the program, the 3.0 m heat transfer length is divided into 300 cells of 0.01 m in length. The program is assessed on both global and local bases. For local phenomena, the local wall heat fluxes, heat transfer coefficients and temperature profiles were compared with the experimental data. All of the test conditions listed in Table 4 were calculated.

For the global assessment, the overall heat transfer rate of the condenser is compared with the experimental data. The results are listed in Table 11.

From the comparison, the overall performance of the condenser was predicted reasonably accurately. Figure 78 shows a comparison of the calculated and experimental heat transfer rate of the test section. All the predicted heat transfer rates fall within $\pm 20\%$ of the experimental heat transfer rates.

Table 11 Comparison of the Heat Exchanger Overall Performance

Test No.	Inlet Pressure (kPa)	Steam Mass Flow Rate (kPa)	Air Mass Flow Rate (g/s)	Experimental Heat Transfer Rate (kW)	Calculated Heat Transfer Rate (Kw)	Relative Error %
7	104.42	6.01	0.06	13.64	13.55	0.63
6	104.63	5.99	0.12	13.75	13.51	1.76
5	104.84	6.06	0.32	13.63	13.66	-0.22
4	105.23	5.90	0.67	12.95	13.31	-2.75
12	108.17	5.92	1.08	12.74	13.27	-4.19
11	109.47	5.93	1.51	12.51	13.23	-5.72
3	104.88	11.49	0.11	25.58	25.92	-1.30
2	105.06	11.49	0.24	25.03	25.92	-3.55
1	105.81	11.44	0.61	24.05	25.65	-6.69
10	112.40	11.54	1.34	23.05	25.51	-10.69
9	115.92	11.55	2.08	22.53	25.20	-11.84
8	121.17	11.49	2.93	22.25	24.92	-12.03
20	201.57	5.85	0.06	14.23	12.88	9.52
21	197.54	5.84	0.13	14.33	12.87	10.20
22	198.71	6.05	0.32	14.49	13.32	8.09
23	199.85	5.85	0.67	14.41	12.88	10.65
50	200.30	5.99	1.05	13.98	13.19	5.69
16	201.12	6.16	1.49	14.08	13.57	3.65
101	200.50	11.46	0.12	28.18	25.24	10.44
100	200.59	11.48	0.24	28.07	25.27	9.98
99	202.31	11.33	0.61	27.27	24.94	8.53
15	201.05	11.41	1.26	25.43	25.12	1.24
14	200.84	11.60	2.08	25.27	25.53	-1.03
13	199.67	11.60	2.83	25.08	25.53	-1.83
18	201.20	23.63	0.24	45.34	38.93	14.13
19	198.57	23.73	0.47	44.95	38.50	14.34
32	200.73	23.23	1.18	42.74	37.86	11.40
33	199.86	23.16	2.52	40.82	36.94	9.50
34	200.53	23.08	4.17	38.46	36.09	6.15
35	201.06	23.15	5.91	37.38	35.36	5.39
24	401.26	11.44	0.11	28.05	24.41	12.97
25	400.02	11.30	0.23	27.51	24.11	12.34
26	400.54	11.53	0.61	27.47	24.61	10.43
51	400.34	11.57	1.29	27.83	24.69	11.30
52	399.44	11.75	2.08	27.97	25.08	10.33
53	398.83	11.45	2.89	27.30	24.45	10.44
27	401.26	23.19	0.23	53.44	47.81	10.53
28	401.77	22.89	0.47	51.77	47.12	8.97
36	402.18	22.97	1.08	51.59	46.32	10.21
37	401.28	23.04	2.58	49.51	45.12	8.86
38	400.63	23.01	4.16	47.65	44.24	7.17
54	400.51	23.02	5.73	45.20	43.57	3.61
46	403.31	35.14	0.35	63.68	50.88	20.10
47	400.22	34.61	0.71	62.94	50.45	19.85
39	403.91	34.73	1.70	61.48	49.91	18.81
40	398.66	34.82	3.53	59.05	48.77	17.41
41	398.82	34.58	6.03	56.13	47.70	15.01
42	401.62	34.48	8.54	53.92	46.79	13.22
48	401.80	45.92	0.47	61.84	51.68	16.42
49	401.29	45.92	0.95	61.14	51.40	15.94
43	400.63	45.61	2.29	59.97	50.68	15.49
44	401.12	45.38	5.19	58.42	49.57	15.14
45	401.07	45.16	8.20	56.20	48.39	13.88

Figure 78 Comparison of Experimental and Calculated Wall Heat Fluxes with the Mechanistic Model

For the local assessment, the calculated local variables were compared with the experimental results. First, the calculated temperature profiles along the tube were compared with the experimental data. Figure 79 shows the local temperature profiles of test no. 9. The calculated results are shown as the solid lines and experimental data are shown as symbols. The colors of the symbols match the lines for the same periphery position. The temperatures are defined in Figure 73.

Figure 79 Comparisons of Temperature Profiles Corresponding to Single-tube Test No. 9

From the comparison, it can be noticed that the wall temperatures appear to be generally overpredicted. This is a consequence of the measured inner wall temperatures not being at the physical inner surface. Further, from the discussion in the experimental section, it is known that the wall temperature measurements have uncertainty due to difficulty in fabrication process. Since the experimental wall heat fluxes were obtained from calibrated temperature differences and not from absolute temperature measurements, a comparison of the local wall heat fluxes would be a more reasonable way of verification.

The wall heat flux comparison for test no. 9 is shown in Figure 80. From the comparison, the heat fluxes on the top near the inlet are underpredicted. It should be noted that the measured heat fluxes are at the top of the tube, thus the experimental value is the maximum value of heat flux in the film condensation region. In contrast, the calculated heat fluxes at the top are average values for the film condensation region.

Figure 80 Comparison of Local Heat Flux Profiles Corresponding to Single-tube Test No. 9

From the description of the phenomena, it is expected that due to the higher condensation heat flux at the top of the tube, the gradient of steam concentration at the top of the tube is higher than that at the bottom of the tube. Thus it is expected that the interfacial air concentration is higher at the top of the tube. The difference between the top interface air concentration and the bottom interface air concentration becomes smaller along the tube.

Figure 81 shows the local interface air mole fraction at the top and bottom of the tube, no experimental values are shown because this is not a measured value. From the comparison, it can be seen that the trend is correct.

Figure 81 Comparison of the Local Interface Air Concentration for Single-tube Test No. 9

4 Problems Encountered and Departures from Planned Methodologies

The problems encountered in this project and departures from the planned methodologies are summarized in Table 12, along with an assessment of their impact on the project results. Details follow the table. *No problems were encountered that prevented completion of project goals.*

Table 12 Problems Encountered and Methodology Departures

Problem/Departure	Project Aspect	Solution/Impact
Problem: accurate measurement of local heat fluxes was difficult	Single-tube	Inability to use commercial equipment provided motivation for new instrumentation development. Resulted in a journal paper
Departure: coolant jacket was not designed for unidirectional flow	Single-tube	CFD calculations were performed to obtain flow conditions for a nearly uniform flow path
Problem: measurement of heat flux as various angles around the tube requires extensive instrumentation	Single-tube	Data was restricted to the tube top and bottom
Departure: number of tubes was originally estimated as 5	Tube-bundle	Numbers of tubes was increased and tube bundle configuration was modified
Problem: straightness of condenser tubes was difficult to quantify	Tube-bundle	One tube may have an undesirable angle. Effect was quantified by comparing with other tubes
Problem: tube wall temperatures have a high uncertainty	Tube-bundle	Data are difficult to interpret, but other thermocouples provide data
Problem: confirmation of applicability of single-tube data to tube-bundle data	Tube-bundle	Although some boundary conditions do not match, there appears to be no major discrepancy in data
Departure: vibrations of secondary side	Tube-bundle	Possible effect on condensation and boiling heat transfer rates, although the same should occur in a real plant
Problem: optimization of venting system	Tube-bundle	Potentially venting too much air on each opening for higher pressure tests

Problem/Departure	Project Aspect	Solution/Impact
Problem: lack of detailed information on condensate behavior inside tubes	Analysis	Some parameters in the model are oversimplified
Problem: fouling on tube outer surface	Single tube and tube-bundle	No measurable effect

4.1 Single-tube Experiments

Measurement of local heat fluxes was the most challenging problem in the single-tube testing. In response, a graduate student and the PI developed a new measurement methodology that has led to one archival journal paper to date. As detailed in Section 3.1.3, a new thermocouple design enabled the measurement of the condenser tube wall temperature very close to the tube wall inner surface. Pairs of thermocouples were installed at several locations along the tube to measure the temperature difference between the inner and outer surfaces, from which the local heat flux can be inferred. Essential to these measurements was a careful calibration of each thermocouple pair. The graduate student developed calibration procedures that allow for heat flux determination with an uncertainty of less than 10%.

In departure from the planned methodologies, the coolant jacket was not designed to minimize the flow rate in the circumferential direction. The initial intention was to have a one-dimensional flow on the secondary-side through multiple parallel channels. The coolant temperature profile at any circumferential angle would then reflect the heat removal along the section of the condenser tube about that angle. Designs for one-dimensional flow proved impractical because they interfered with the instrumentation, they introduced additional complexities in the heat transfer paths and appropriate channel angles were difficult to determine. These problems were bypassed by the use of coolant flow conditions that brought about the same effect. Computational Fluid Dynamics (CFD) simulations of the coolant jacket provided flow conditions that resulted in a nearly uniform flow with minimal circumferential mixing.

To capture the circumferential variations due to horizontal stratification, temperatures at given distances from the condenser tube inlet were to be measured at four circumferential locations. The temperatures were measured only along the top and bottom surfaces of the condenser tube. Excessive instrumentation would have been required to measure heat transfer rates at other peripheral locations. The PI intends to have data taken at other angles by rotating the condenser tube.

In summary, the problems and departures spurred new instrumentation development, prevented data acquisition at various circumferential angles, but did not detract from the data quality.

4.2 Tube-bundle Experiments

The conceptual design of the tube bundle had five condenser tubes with one central and four surrounding tubes. A configuration with a “central tube” that could be surrounded by four other tubes and vapor on the secondary side was thought to be the

most limiting tube configuration. Upon further study, a configuration with a “height effect”, in which higher tubes become vapor blanketed by steam traveling up from the lower tubes was determined to be more of a challenge to the heat transfer efficiency. The design was changed to a six-tube, alternating tube configuration.

The degree to which the tubes are straight and horizontal is difficult to confirm. Tubes with an upward inclination from inlet to outlet may have difficulty draining. Spacers to support the tubes straight within the test section were inserted however they do not appear to be strong enough to support the tubes adequately. The impact on the results is one of six condenser tubes shows heat removal rates inconsistent with data from the other five tubes.

The tube wall temperatures were difficult to interpret. A high uncertainty was expected and mentioned in the proposal. The causes of the difficulty are insufficient knowledge of the exact measurement location of the thermocouple junction, the location and amount of solder and imperfections in the tube manufacture. The effect is an inability to obtain detailed heat transfer data from each tube. To obtain such essential data, the single-tube testing was included in this test program and demonstrated to be applicable to the tube-bundle condenser tubes.

To demonstrate the applicability of the single-tube data to the tube-bundle condenser tubes, several steps were taken, as described in Section 3.2.1. While the conditions in the two facilities were matched as closely as possible, the discrepancies in the geometries and secondary-side conditions should be noted. The diameter and length of the condenser tubes are not identical, and the secondary-side conditions were different in terms of flow geometry, coolant flow rate and coolant temperatures. Several sets of data including centerline temperatures, condensation lengths and the effects of inlet air mass fraction, pressure and steam flow rate were compared. The comparisons showed very good correlation between the single-tube and tube bundle data.

An unexpected observation was the amount of vibration from the secondary side of the tube-bundle test facility. While this is expected to also occur in the actual plant, the extent to which it affected condensation and boiling is not known.

Finally, there is some optimization that remains to be done on the noncondensable gas venting system. While the frequency of venting is realistic, the size of the solenoid valve may be affecting the amount of gas venting per opening. In future studies, it is advisable to install a proportional valve that has a flow area opening proportional to the pressure drop across it.

4.3 Analysis Model Development

With regard to empirical model development, two pieces of information that would be extremely valuable are the film characteristics, including thickness at various circumferential angles, wave formation and frequency.

For the mechanistic modeling, heat transfer coefficients at the top and bottom of the tube were derived. These must be averaged in a manner to reflect the location of the water film inside the tube. An assumed averaging method was adopted.

4.4 Miscellaneous Problem

Fouling was observed on the tube outer surfaces in both facilities after several tests with city water as coolant were performed. Tests were repeated before and after tube cleaning and no measurable effects were found.

5 Applicability of Horizontal Heat Exchangers for PCCS Heat Removal

To investigate the applicability of horizontal tube-bundle condensers for passive containment heat removal, the following aspects of heat exchanger behavior were investigated:

1. the condensation heat transfer characteristics when the incoming fluid contains noncondensable gases
2. the effectiveness of condensate draining in the horizontal orientation
3. the conditions that may lead to unstable condenser operation or highly degraded performance
4. multi-tube behaviors with the associated primary-side and secondary-side effects

Regarding the heat transfer characteristics, local heat fluxes were measured by novel techniques developed herein and shown to be on the order of heat fluxes seen in vertical heat exchangers. These measurements were obtained from the single-tube facility and demonstrated to be valid for the tube-bundle facility. Of particular importance, noncondensable gases did not degrade the heat transfer to an extent of concern. Increased pressure was shown to increase the heat removal capacity.

Condensate draining was successful in the horizontal orientation. Momentum transfer from the incoming steam and shear from the steam/noncondensable gas mixture to the condensate surface promoted condensate flow to the heat exchanger outlet. A small degree of downward inclination is recommended because difficulties may arise in the manufacturing process to ensure that the tubes are all horizontal and without any deformation that could inhibit draining.

Tests were conducted outside of the expected DBA LOCA range of conditions to identify unstable or highly degraded heat exchanger performance. Pure steam tests at low flow rates were found to be unstable unless noncondensable gases could be ingested from the tube outlet. Steam condensation reduced the pressure to an extent that the heat exchanger pressure was less than atmospheric pressure and unstable behavior followed. Highly degraded performance occurred at low pressure (1 or 2 atmospheres) and high inlet air mass fractions corresponding to the initial vessel blowdown phase in a DBA LOCA or severe accident conditions. During vessel blowdown, most of the heat would be absorbed by the suppression pool in a BWR and the horizontal heat exchanger would be activated for the subsequent period.

Multi-tube behaviors with the associated primary-side and secondary-side effects were investigated in the tube-bundle tests. The condenser tubes behaved similarly with respect to condensation heat removal. One tube removes less heat than the other tubes and it appears to be either inclined at an unfavorable angle or sagging slightly within the pool. This suggests that better methods are needed in the assembly process. On the secondary side, visual observations showed that although the upper tubes were surrounded by a higher void region than the lower tubes, the heat removal rates were not measurably affected.

The transient tests in the tube bundle demonstrated that horizontal heat exchangers can self-regulate their heat removal rates to match the incoming heat load. Retention of noncondensable gases in the downstream portion of the tubes is essential to this important function. By confirmation of these two points, *horizontal heat exchangers have been shown to possess the capabilities of vertical heat exchangers for PCCS heat removal.*

The heat exchanger examined in this work is intended for advanced and innovative reactors in which passive heat removal systems are adopted to improve safety and reliability. The following tasks remain before a final design can be proposed to reactor designers:

1. optimization of the tube configuration
2. transient tests in the tube-bundle facility with higher inlet steam flow rates (up to the design limit of 10 g/s)
3. tests with a lighter gas to simulate hydrogen
4. development of analysis methods for tube bundle behavior in a pool
5. analysis of tube bundles with prototypic numbers of tubes
6. recommendations for actual heat exchanger dimensions and configurations
7. recommendations for manufacturing and assembly processes

6 Accomplishments

All of the goals and objectives of the project were successfully completed as described in Section 3, Project Activities, and Section 5. The primary technical goals of the proposed work were:

1. to investigate the major factors of horizontal heat exchanger performance in passive containment heat removal from a light water reactor following a design basis accident LOCA
2. to develop modeling methods for use in nuclear thermal-hydraulic codes
3. to advance the understanding of condensation heat transfer mechanisms in horizontal tubes when noncondensable gases are present

Goals related to the NEER program were:

1. to support basic research in nuclear engineering
2. to assist in developing nuclear engineering students
3. to contribute to strengthening the academic community's nuclear engineering infrastructure.

The following accomplishments beyond those promised in the proposal were achieved.

1. development of novel measurement methods for the local heat flux
2. automated venting method for releasing noncondensable gases from the test section as they would be in an actual PCCS.

One journal submission has been accepted for publication and a second is in review. A Purdue University publication was produced that documents the research. Several ANS conference papers have also been presented.

Table 13 shows a complete list of project milestones, anticipated completion dates as in the proposal and actual completion dates.

Table 13 Project Milestones

ID Number	Task / Milestone Description	Planned Completion	Actual Completion	Comments
1	Experimental Studies			
1.1	Single-tube Experiments			
1.1.1	Construction of Facility	12/31/02	05/01/03	Steam supply delay and late arrival of grad student
1.1.2	Shakedown Testing	03/31/03	05/15/03	
1.1.3	Data Reduction Software Dev.	06/31/03	05/31/03	
1.1.4	Heat Transfer Coef. Expt.	12/31/03	01/31/05	Testing extended due to development of new measurement techniques
1.1.5	Stable Operation Experiments	03/31/04	01/31/05	
1.2	Tube Bundle Experiments			
1.2.1	Construction of Facility	06/30/04	05/31/04	
1.2.2	Shakedown Testing	09/30/04	05/31/04	
1.2.3	Tube Bundle Experiments	03/31/05	05/15/05	
2	Analytical Model Development			
2.1	Heat Transfer Coef. Correlation	03/31/04	02/28/05	
2.2	Mechanistic Model Development	05/31/05	05/15/05	
2.3	Model Verification and Analysis	05/31/05	05/15/05	
3	Reports			
3.1	Yearly Progress Report	08/29/03	08/12/03	
3.2	Yearly Progress Report	08/29/04	08/20/04	
3.3	Final Report	08/29/05	08/29/05	

7 Conclusions

This project has investigated the major aspects of horizontal heat exchanger performance in passive containment heat removal from a light water reactor following a design basis accident LOCA. For passive containment heat removal, advantages of horizontal heat exchangers over vertical heat exchangers include potentially better heat transfer efficiency, economic benefits from the reduced containment size, less tube fouling for reduced maintenance and greater seismic resistance.

The heat exchanger designed in this work is intended for advanced and innovative reactors, in which passive heat removal systems are adopted to improve safety and reliability. Since the application to passive containment heat removal is new, the following aspects of heat exchanger behavior were investigated:

1. the condensation heat transfer characteristics when the incoming fluid contains noncondensable gases
2. the effectiveness of condensate draining in the horizontal orientation
3. the conditions that may lead to unstable condenser operation or highly degraded performance
4. multi-tube behaviors with the associated primary-side and secondary-side effects

A single-tube horizontal test facility and a six-tube horizontal tube bundle facility were constructed and operated to perform these investigations. The goal of the single-tube facility was to provide basic heat transfer data for better understanding of steam condensation from steam/ noncondensable gas mixtures and to obtain a data base for validation of a mechanistic model for condensation heat transfer. New methods were developed for measuring the local heat flux on condenser tubes. A data base of local heat fluxes at the top and bottom of the condenser tube were obtained. These data are unique and important in that they provide multi-dimensional information for condensation heat transfer rates without disturbance of the condensation phenomena by intrusive instrumentation or thermal distortion of the condenser tube geometry.

The tube-bundle facility was constructed to examine system behavior of a horizontal heat exchanger. Steady state tests were performed to observe tube-to-tube effects and transient tests were run to simulate LOCA scenarios at actual pressure with prototype condenser tube dimensions. The transient tests in the tube bundle demonstrated that horizontal heat exchangers can self-regulate their heat removal rates to match the incoming heat load. Retention of noncondensable gases in the downstream portion of the tubes is essential to this important function. By confirmation of these two points and positive results regarding the four aspects listed above, ***horizontal heat exchangers have been shown to possess the capabilities of vertical heat exchangers for PCCS heat removal.***

Empirical correlations for the heat removal rates at the top and bottom of the tubes were derived from the single-tube facility data. These were incorporated into the RELAP5 code and the tube bundle behavior at steady state was successfully reproduced.

The data obtained in both facilities provided quantitative information on the effects of pressure, noncondensable gas mass fraction and steam/gas Reynolds number on condensation heat transfer. These led to a better understanding of condensation heat transfer mechanisms in horizontal tubes and valuable insight for model development. A mechanistic model was developed independent of the test data and verified against the single-tube test data.

The PI intends to continue the modeling work by developing analysis methods that can predict the secondary-side pool boiling around the tube bundle and proposing design details for a PCCS horizontal heat exchanger. The models developed as part of this project will serve as the starting point.

In addition to successful completion of the technical goals, this project was also successful in supporting a Ph.D. graduate student (a second Ph.D. student with other support also participated), upgrading the experimental laboratory of a junior faculty and providing research experience to seven undergraduate students in Purdue University's School of Nuclear Engineering. Of the seven undergraduate students, one is now a Master's student in the PI's laboratory. Another three are anticipated to further their nuclear engineering education through at least a master's degree and have expressed interest in graduate research related to the topics of this project.

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9 Publications Arising from this Award

Archival Journal Publications

T. Wu, K. Vierow, “A Local Heat Flux Measurement Technique for Inclined Heat Exchanger Tubes”, accepted for publication in *Experimental Heat Transfer*, Jul. 2005.

Submitted Journal Publications

T. Wu, K. Vierow, “Experimental Study of Steam Horizontal In-Tube in the Condensation in the Presence of a Noncondensable Gas”, submitted to *International Journal of Heat and Mass Transfer*, Mar. 2005.

Conference Papers

T. Wu, K. Vierow, “Fundamental Condensation Heat Transfer Studies for Horizontal PCCS Heat Exchanger”, Proc. of 2005 ANS Annual Meeting, San Diego, CA, Jun. 2005.

T. Wu, K. Vierow, “Horizontal In-Tube Condensation in the Presence of a Noncondensable Gas”, accepted for publication in 11th International Topical Meeting on Nuclear Reactor Thermal Hydraulics (NURETH-11), Avignon, France, (14 pp.) Oct. 2005.

Y. J. Song, K. Vierow, “Horizontal Heat Exchanger Scaling for Passive Containment Heat Removal System Experiments”, accepted for publication in Proc. of the ANS 2005 Winter Meeting, Washington, DC, Nov. 2005.