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AP600 PASSIVE CONTAINMENT COOLING SYSTEM
INTEGRAL SMALL-SCALE TESTS
FINAL REPORT

August 1994

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SUMMARY

The AP600 reactor design includes a passive containment cooling system (PCS) to remove heat released to the containment following any postulated event and to transfer this heat from the containment to the environment. This system employs natural draft air cooling and the evaporation of a water film from the outside of the steel containment shell to transfer heat from the containment vessel to the environment.

Past AP600 small-scale PCS tests were conducted to demonstrate its performance. These early tests investigated water film behavior, mass transfer (evaporation), and convective heat transfer on the external surface of a steel vessel initially filled with one atmosphere of air and heated on the inside by dry steam, as well as a range of design basis conditions for steam/air internal pressure, external cooling air velocity, air temperature, air humidity, and water film flow rates.

The purpose of the integral small-scale tests is to demonstrate the operation of the AP600 PCS over an increased range of operating conditions, including postulated severe accident conditions, and to evaluate the impact of low environmental temperatures on the containment and air baffle structures. Results from selected integral small-scale tests were reported in PCS-T2R-006, Interim Report (Reference 6.3). The data from these tests will be applied to the development and verification of analytical models used to predict performance of the full-scale PCS.

This report represents results from the AP600 small-scale PCS integral extension tests, which will be used for additional validation of the WGOTHIC computer code used to predict containment performance.

1.0 INTRODUCTION

1.1 Background

The AP600 reactor is a pressurized water reactor designed to utilize the passive containment cooling system (PCS) as the safety-grade means to remove heat released to the containment following any postulated event and transfer this heat to the environment. The PCS utilizes a steel containment vessel by condensation of the steam. Heat is conducted through the steel wall and transferred to the environment from the outside surface of the containment by convection to air and evaporation of a water film.

Following a design basis loss-of-cooling accident (LOCA), the outer surface of the containment above the operating deck initially needs to be wetted by a water film to remove sufficient heat to limit the containment pressure to around 40 psig. The cooling water is supplied by gravity drain from an elevated storage tank. As the core decay heat decreases as a function of time, the energy released to the containment will decrease and the amount of evaporative cooling required will also decrease. Therefore, the water flow rate introduced to the containment surface is passively decreased over time as a result of the decrease in the water head in the tank and a staggered drain pipe arrangement. Sufficient water is stored to provide cooling water film flow for three days, which affords sufficient time for other water sources to be made available. However, even with no additional water after three days, natural convective air cooling alone could remove the remaining decay heat and ensure containment integrity.

Operation of the AP600 PCS was demonstrated over a range of design basis conditions (Reference 6.1) by the original PCS tests. The system was also demonstrated over additional design basis conditions in the PCS integral extension tests. These tests, performed using the integral containment cooling test facility located at the Westinghouse Science & Technology Center in Churchill, PA, utilized a [ ]^{a,c} steel pressure vessel to simulate the containment shell. The pressure vessel was surrounded by a cooling air annulus and equipped with provisions to provide cooling water film flow over the external vessel surface. The test apparatus also included provisions for varying annulus inlet air temperature and humidity, annulus air velocity and containment water film flowrate and temperature. The vessel, initially filled with one atmosphere of air, was pressurized (heated) with steam to provide data over a range of containment pressures and prototypic air and water film flowrates. These tests provided heat transfer data at prototypic conditions for both inside and outside containment heat transfer.

The AP600 small-scale PCS integral containment cooling extension tests were conducted using the same facility originally constructed for demonstrating operation of the AP600 PCS. To permit testing over the broader range of operating conditions specified for the extension tests, the test facility was modified to test in cold weather and to simulate abnormal operating conditions. The test facility was also upgraded to improve or automate particular test measurements, such as condensate flowrate based on experience gained from previous tests.

1.2 Test Objectives

The purpose of the integral small-scale tests is to provide additional test data obtained over a broader range of operating conditions than previously tested and to supplement the results from the previous integral containment experiment. Repeat tests at selected operating conditions were also performed to establish the validity and repeatability of results obtained for earlier tests. Tests were also run at off-nominal conditions using hot and cold film water temperatures and cold cooling air. The purpose of these tests was to avoid ice formation in the annulus region and to determine any effects cold weather had on the passive containment cooling capabilities.

This report addresses the results obtained from the AP600 small-scale PCS integral containment cooling test extension tests. An interim report of selected results from these extension tests can be found in Reference 6.3.

1.3 Test Matrix

A test matrix detailing test conditions and parameters can be found in Table 1.3-1. There were nine categories of tests performed for the small-scale PCS testing. They include base cases with constant steam flow, repeated base cases with prototype injection in the steam inlet, water film levels of cooling, water film distribution limits of cooling, vessel air content impact on inside heat transfer (HT), baffle air flow limits, water film temperature (phases 1 and 2), transient steam flow, and ice formation and melt demonstration.

The test matrix includes a full range of expected design basis external water flowrates, the maximum and minimum expected cooling air velocities, and the cooling air inlet temperature and relative humidity. Provided below is a discussion of each of the key test variable ranges used to establish the test conditions. The effects of these test parameters on heat transfer capability are discussed separately in Section 4.0, test results.

1.3.1 External Baffle Cooling Air Velocity

Based on the AP600 containment transient analyses, three cooling air velocities, 16 ft./sec., 12 ft./sec. and 8 ft./sec. were examined. Sixteen (16) ft./sec. corresponds closely to the maximum calculated air velocity during wetted heat transfer shortly after PCS initiation following a postulated loss-of-coolant accident (LOCA), and to the calculated air velocity with a dry containment surface when containment internal pressure is 40 psig.

The 12 ft./sec. velocity is the calculated air velocity in the baffle when containment pressure is 20 psig.

The 8 ft./sec. velocity is the air velocity calculated to generate by natural circulation when the containment pressure is 10 psig. This pressure and air velocity define the condition where containment cooling would transition from a wetted external surface to a dry surface, if not operator action is taken after three days when the stored PCS water has been used.

1.3.2 External Containment Surface Water Flow

The water flowrates supplied to the top external surface of the test vessel correspond to the prototypic water flow rates onto the AP600 containment which were used in the LOCA response containment transient analysis. These AP600 maximum and minimum PCS water supply flows are 205 gpm and 54 gpm, respectively. Since some of the supplied water is evaporated from the containment dome, the amount of water which reaches the top of the cylindrical portion of the containment will initially be ~150 gpm and reduces to ~36 gpm at 3 days when the PCS water is exhausted. Since the perimeter of the cylindrical portion of the AP600 containment is [ ]^{a,c} the maximum and minimum flow/ft. around the containment top, exterior, vertical surface is 0.398 gpm/ft. and 0.095 gpm/ft., respectively. These flows are matched on the three ft. diameter PCS test vessel and an intermediate flow of 0.25 gpm/ft. was included.

1.3.3 Cooling Air Inlet Temperature/Relative Humidity

For all wetted test conditions the cooling air inlet temperature was maintained at ~130°F. This elevated air inlet temperature approximates the average air temperature that would occur in the full sized AP600 cooling path, based on the maximum environmental air temperature of 115°F currently specified in the EPRI Requirements Document. The inlet air relative humidity was raised to 30 percent to approximate the average specific humidity that will be achieved in the full sized AP600 PCS cooling path, based on the 85°F maximum wet bulb temperature for the inlet air specified in the EPRI Requirements Document. For comparison, identical test conditions with low relative humidity inlet air were also examined.

Table 1.3-1 is not included in the non-proprietary version of this document.

2.0 TEST FACILITY DESCRIPTION

2.1 Introduction

The AP600 small-scale PCS integral containment cooling extension tests were performed using the integral containment cooling test facility located at the Westinghouse Science & Technology Center in Churchill, PA. This facility, originally constructed for demonstrating operation of the AP600 PCS, was modified to permit testing over a broader range of operating conditions.

The integral containment cooling test facility uses a [ ]^{a,c} pressure vessel to simulate the AP600 steel containment shell. The vessel can contain air or nitrogen at one atmosphere when cold and is supplied with steam at pressures up to 80 psig. A transparent acrylic cylinder installed around the vessel forms the air cooling annulus. The test vessel wall is 3/8-in. thick. A water film is formed from water added at the top of the pressure vessel, which flows down covering the vessel external surface. Air flow up the annulus outside the vessel cools the vessel surface resulting in condensation of the steam inside the vessel.

Figure 2.1-1 is a section view of the test apparatus. Saturated steam from a boiler is throttled to a variable, but controlled, pressure and supplied to the bottom of the vessel which initially contained one atmosphere of air. The steam is distributed inside the vessel by one of two steam distributor arrangements. The uniform steam distributor, shown in Figure 2.1-1, provides for slow radial flow, uniform along and around the central supply pipe that runs the full height of the test vessel. The uniform distributor was expected to produce the most limiting steam condensation conditions.

To establish the total heat transfer from the test vessel, measurements are recorded for steam inlet pressure, temperature, and condensate flow and temperature from the vessel. Twenty-four thermocouples located on the outer surface of the vessel's 0.375-in.-thick circumference. The measured thermocouples are weighted by the respective vessel wall areas sensed by the thermocouples and summed to obtain the average vessel outside surface temperature.

An axial fan, which is used to control the cooling air velocity, is located in the chimney region above the test vessel and forms the upper chimney for the cooling air flow path.

Water can be added at the top of the vessel to create a water film on the external surface. External water film flow rates onto the vessel and the flow rate of excess water from the bottom of the vessel that is not evaporated are measured. The difference in water mass is the liquid evaporated, which, multiplied by heat of vaporization change plus the enthalpy increase of the drained fluid above inlet conditions; and the heat transfer to air (i.e., its temperature rise multiplied by its specific heat and its measured flow rate) give a third measurement of the total heat transfer.

The tower that supports all but the pressure vessel itself provides three floors for workers to erect the components, install instrumentation, and operate instrument traverses and rakes. External cooling air temperature, humidity, and velocity can be surveyed by traversing at several elevations in the cooling annulus. Air velocity is also measured at the inlet to the air heating coil. Thermocouple potentials are sequentially sampled and converted by data acquisition equipment and recorded on paper tape or computer memory and disk.

A photograph of the test apparatus, Figure 2.1-2 shows many of the test components, including the transparent cylinder and test vessel. The tower, which supports all but the pressure vessel, provides three floors for workers to assemble components, install instrumentation, and conduct instrument traverses.

2.2 Facility Component Description

2.2.1 Foundation and Tower

The foundation for the pressure vessel and tower is a [ ]^{a,c} square pad of reinforced concrete located next to Building 301 at the Westinghouse Science & Technology Center. The tower was constructed using 6-in. square structural tubing for posts and 6 x 4-in. angles for platform supports. The tower has three [ ]^{a,c} square work platforms with a [ ]^{a,c} square center opening to accommodate the test vessel and annulus baffle. The three work platforms are located at elevations [ ]^{a,c} above the foundation.

The pressure vessel is supported by four 6-in. steel angle legs attached to a 5-ft. diameter steel ring base. The ring and the tower's corner posts are anchored to the concrete foundation. The pressure vessel weighs approximately 5,000 lbs. empty and the tower weight is approximately 8,600 lbs. With the air baffle in place, the assembly can withstand winds in excess of 100 miles per hour.

2.2.2 Pressure Vessel

The pressure vessel is a [ ]^{a,c} vessel with elliptical heads and a 0.375-in. thick steel wall. Overall length of the vessel including the heads is [ ]^{a,c} At the bottom of the vessel, a standard 150-lb. class, 20-in. weld-neck flange is welded into the head on the vessel centerline. The 20-in. diameter opening formed by the weld-neck flange serves as a manway. The manway opening is covered by a 150-lb. class, 20-in. blind flange. A 4-in. diameter hole through the center of the 20-in. blind flange is covered by a 150-lb. class, 4-in. blind flange. A 2-in. pipe nipple is welded into the 4-in. blind flange to permit connection of the external steam supply pipe to the internal steam distributor. A threaded 1-in. pipe nipple is welded into the 20-in. blind flange to provide for the condensate drain.

At the top of the vessel, a standard 150-lb. class, 10-in. weld-neck flange is welded into the head on the vessel centerline. It is also covered by a blind flange. The top vessel blind flange serves as a feedthrough for vapor trap pigtails that connect with reference pressure lines and nitrogen charging lines inside the vessel. The top vessel opening was also utilized for installation and centering of the internal steam distributor.

The pressure vessel is rated for its intended use, 100 psig, although the extra heavy walls would permit a higher rating. The heavier wall thickness was specified to better model wall heat transfer without making fabrication and erection unduly difficult.

The vessel support legs provide 60 in. of clearance between the bottom flange and the foundation to accommodate installation of steam supply and condensate drain piping. The inner and outer surfaces of the vessel were sprayed with a 0.006- to 0.008-in. thick coating [ ]^{a,c} to prevent corrosion. Prior to application [ ]^{a,c} the vessel was prepared by sandblasting the surfaces with G-40 size steel shot.

2.2.3 Steam Supply

Saturated steam was supplied by a 10,000 lbs. per hour gas fired boiler, which was maintained at 100 psig during testing. Full firing was maintained at the boiler to avoid cycling and pressure swings that would result in unsteady operation of controls in the test apparatus. Excess steam was vented through a pressure-limiting relief valve and flow silencer above the boiler. Laboratory demineralized water was used for boiler water makeup; no condensate was returned to the boiler.

The steam was supplied to the test tower through 88 ft. of 4 in., schedule 40 piping insulated with 1.5 in. of glass fiber insulation. Electrical trace heaters were installed over 40 ft. of the steam supply piping to reduce piping heat losses and ensure that superheated steam conditions (after throttling from 100 psig to the lower test pressure) were maintained for all tests.

Since the heaters can operate continuously only to 500°F, thermocouples were installed on the pipe, under the heaters, and at the inlet and outlet ends of the heated section. An automatic temperature controller connected to a 40 ampere, 240 volt, three-phase SCR power controller to which pairs of heaters are connected, automatically cycles the heaters on and off. The set point for the controller can be up to 500°F, but usual use will be for a maximum of 350°F. This installation also permits the pipe to be used as a storage heater to provide initial steam flow with a large, prototypical superheat. The addition of the electrical heaters ensures superheated steam conditions for all test conditions.

At the test tower, steam was delivered from the main 4-in. supply manifold to the test vessel inlet, the air humidifier, and the air heating coil through individually valved 2-in. insulated pipes.

A 2-in. steam supply pipe size is installed downstream of the flow control valves to minimize dynamic pressure effects in the internal steam distributors. The 2-in. steam supply pipe is connected to the 2-in. nipple, which is welded into the 4-in. blind flange at the bottom of the pressure vessel.

2.2.4 Steam Inlet to the Vessel for Containment Simulation

The pressure vessel provides for installation of different types of steam distributors. The uniform and prototype steam distributors were used for all of the AP600 small-scale containment cooling extension tests represented by this report.

The uniform distributor consists of [ ]^{a,c} long sections which are connected by pipe couplings. The uniform distributor extends from the steam inlet nipple up into the neck of the weld-neck flange at the top of the vessel. The weld-neck flange retains the distributor while allowing it to slide up and down inside the neck to allow for differential thermal expansion. The distributor sections were fabricated from 48-in. lengths of threaded schedule 40 stainless steel pipe containing fourteen 0.125-in. diameter metering holes. The metering holes were drilled in pairs, [ ]^{a,c} spaced 6 in. apart, with alternate pairs 90 degrees from the others. In order to prevent jetting of steam into the vessel, each inner distributor section is surrounded by a 3.5-in. outside diameter, 0.065-in. thick wall, stainless steel shield tube. Each shield tube contains [ ]^{a,c} holes. The 0.75-in. holes were drilled in sets [ ]^{a,c} with each set spaced at 6-in. intervals. The shield tubes were designed such that when they are assembled over the inner distributor sections, the 0.75-in. diameter shield tube holes are centered between the distributor metering holes. Disks welded on each end of each shield tube loosely center it over the inner distributor section. The shield tubes slide over the inner distributor sections and rest on the pipe couplings, which join the assembled distributor sections. The inner distributor section and shield tube for one section of the uniform distributor are shown in Figure 2.2-1.

The prototypic distributor directs steam upward from an outlet formed between a 2-in. (2-3/8) in. outside diameter) schedule 40, stainless steel supply pipe and a cylinder that has 8-1/2 in. outside diameter made from 18 gauge stainless steel sheet. The supply pipe has [ ]^{a,c} metering holes with 3/16-in. diameter spaced [ ]^{a,c} at five axial locations spaced 5 in. apart near the bottom of the pipe. A conical section, [ ]^{a,c} welded to the supply pipe 2-1/2 in. from its lower end is seam-welded to the cylindrical section, which is [ ]^{a,c} long. To assure a uniform velocity out of the distributor, two fine mesh copper screens are held 1 in. apart in the lower end of the cylinder annulus using a brass hub with six spokes that is held by screws through the cylinder. The supply pipe is [ ]^{a,c} long. The complete "conical" distributor is shown in Figure 2.2-2. The upward velocity of steam out of this distributor is usually less than 5 ft./sec, and much less when it is elevated and the water level in the vessel is raised to within 5 ft. of the top of the vessel.

The conical steam distributor was installed near the bottom of the pressure vessel. Its discharge is approximately [ ]^{a,c} above the bottom of the vessel. The bottom of the vessel is the weld joint between the elliptical head and the welding neck flange.

2.2.5 Condensate Handling

Condensate that is formed on the inside wall of the pressure vessel flows down and collects in the neck of the 20-in. flange at the bottom of the vessel. The condensate is removed through a 1-in. pipe connected to a liquid drain trap (vapor trap or steam trap) and cooled below 90°F by a condensate cooling heat exchanger. The cooled condensate is collected in a weigh tank consisting of a 55-gallon drum, which rests on an electronic scale. The mass of condensate collected in the weigh tank is measured by the electronic scale and this reading is continuously communicated to the data acquisition system (DAS) over an RS232 interface. A level probe installed in the weigh tank is connected to a solenoid valve installed in the weigh tank drain line and provides for automatic draining when the weigh tank is filled.

2.2.6 External Cooling Annulus and Air Ducting

The AP600 small-scale PCS test cooling air annulus is formed by a transparent acrylic cylinder installed around the pressure vessel. Initially, all PCS tests were performed with a 15-in. annulus width using the same acrylic cylinder used for previous AP600 PCS tests. As a result of a series of natural convection tests performed using the integral test facility, a new acrylic cylinder that formed a 5-in. annulus was designed and constructed. Upon completion of the natural convection tests, a decision was made to retain the 5-in. annulus configuration and conduct additional tests at conditions investigated in the 15-in. annulus configuration, thus, providing supplemental test data.

The 15-in. air cooling annulus was fabricated from 0.25-in. thick acrylic sheets hot formed to a 33-in. inside radius. Aluminum angles were used to reinforce the edges of the acrylic panels; the angles also served as flanges that were used to join adjacent panels. The panels were stiffened using flat aluminum bars. The components were assembled, using screws to fasten the acrylic to the aluminum supports, to form a cylinder 259 in. (21 ft., 7 in.) high and 66-in. inside diameter. The entire cylinder assembly was attached to the tower using aluminum angle supports. The bottom of the acrylic cylinder was located at an elevation 35.75 in. above the bottom of the vessel, even with the top of the inlet air duct.

The cylinder was fabricated from 48 in. x 48 in. or 48 in. by 33 in. acrylic sheets that cover nearly 90 degrees of arc. They are supported on each edge by three in. by 3 in. x 0.25 in. thick high strength, corrosion resistant 6061-T6 aluminum angles that are 259 in. high. The angles complete the 90 degree sector. Screws are used to fasten acrylic sheets to the angles. Four such assemblies are joined to form the cylinder by bolting pairs of aluminum angles together with butted angle sections outside the cylindrical form. A rectangular bar, 1.5 in. x 1.25 in. 6061-T6, is used to reinforce the acrylic panels. Horizontal bars are rolled to support acrylic panels near their edges on the outside, and for the 48-in. panels, in the middle also. Long bars on the inside run vertically at the center of panels to link them. Gaps are taped with clear plastic.

At the top or outlet of the cooling air annulus, a 9.75-in. high conical section provided a transition between the 66-in. diameter annulus wall and the 48-in. diameter axial fan housing.

The air inlet to the cooling annulus was formed by a dished 0.125-in. thick steel pan at the bottom and a dished heavy gauge galvanized steel sheet at the top. The inlet duct has a circular shape with a 66-in. outer radius. The duct was located such that its centerline was offset 12 in. from the vessel centerline toward the inlet side of the duct. The sides of the inlet duct were approximately 31-in. high and covered by galvanized steel sheet. The air inlet duct was joined to a trapezoidal-shaped galvanized sheet metal transition duct which was fastened to the air heating coil. The shape and size of the air duct is to provide uniform, low air velocities around the vessel and accelerate flow up into the containment annulus.

The 5-in. air cooling annulus was fabricated using 3/32-in. thick acrylic sheets supported by aluminum ribs attached to the original 15-in. annulus baffle. The gap between the 15-in. annulus baffle and the 5-in. annulus baffle was sealed at the top and bottom using a washer type ring cut from styrofoam sheet to prevent air from short circuiting the cooling air flow path between the 5-in. baffle and the vessel wall.

2.2.7 Axial Fan

Controlled velocity air flow in the cooling air annulus was provided by a 48-in. diameter, variable speed axial fan inside a 48.5-in. diameter, 36-in. high housing. It is belt driven up to 890 rpm by a four-pole (1750 rpm) totally enclosed fan cooled 10-horsepower, 460 volt induction motor. Motor speed is varied by a 10-horsepower, 460 volt variable frequency power supply. The fan is capable of supplying up to 32,000 cfm of air, which corresponds to 32 ft. per second air velocity in the 15-in. wide cooling air annulus. Operation of the fan is required to simulate natural draft air velocities that can be achieved in the full-scale plant. This is due to the flow resistance of the inlet air heating coil and because the smaller height of the test facility produces a less buoyant head of air than the actual full-size plant.

2.2.8 Air Heating and Humidification

To permit simulation of cooling air conditions expected to occur at upper sections of the actual AP600 PCS flow path, the integral test facility includes provisions for heating and humidifying air supplied to the test facility inlet duct. Air supplied to the test facility inlet duct is heated using a steam/air heat exchanger. The air heater has an 8-ft. wide x 4-ft. high inlet that provides 32 ft^2 of frontal flow area. The air heater, when supplied with 1380 lbm/hour steam flow, will heat 16,000 cfm of air (or air flowing at a velocity of 16 ft./sec. in the containment annulus) from 50°F to 124°F.

After heating, air flow supplied to the annulus may be humidified to levels above ambient to simulate the relatively high moisture content expected near the top of the AP600 containment due to the evaporation of the cooling water film. Steam is injected into the inlet air flow in the air inlet duct just downstream of the air heating coil. Four injection tubes are used to inject dry saturated steam into the air supply stream. The steam immediately condenses as a fog and becomes well mixed with the inlet air before entering the cooling air annulus.

Humidity is controlled using a humidity sensor element mounted inside the inlet air duct farthest from the humidifier that provides the control input to the humidifier control valve. The sensor is able to accurately respond to relative humidity (water vapor partial pressure divided by saturated vapor pressure) for air dry bulb temperatures from -20°F to 175°F.

2.2.9 Water Film Flow

Water for the evaporating film flow is provided using the city water supply. A 30 kW electric water heater is installed in the film water supply system to preheat the film water to simulate expected temperatures for the film part way down the actual containment; the temperature of the water contained in the AP600 film water storage tank can also be simulated.

The film water flow is distributed onto the test vessel head at a radius just outside the 10-in. diameter (11-in. outside diameter) flange. A supply manifold formed by a 15-in. inside diameter ring of tubing with twenty-eight 1/16-in. inside diameter metering tubes spaced 1.75 in. apart provides a uniformly distributed, non-splashing water feed at a radius of approximately 8 in. on the vessel head. In addition to the water distributing supply manifold, additional water distribution is provided using a weir arrangement formed by strips of rubber glued to the vessel head surface.

Water film flow that does not evaporate from the vessel walls is collected in the drain pan at the bottom of the air inlet duct. A drain hose carries the film drain water to an acrylic cylinder that releases entrained air. From there it flows by gravity to a counterflow heat exchanger that cools the film drain water enough to pass through a water meter used to measure film water drain flowrate.

2.3 Instrumentation

This section provides a description of each piece of instrumentation used on the facility. Table 2.3-1 summarizes the instrumentation and accuracy of each measurement based on manufacturer's data or upon operating experience (whichever is greater).

2.3.1 Steam and Condensate Flow, Temperature, and Pressure

Steam flow rates to the vessel were not measured directly; however, steam that condensed on the inside vessel wall was measured by collecting the condensate in a weigh tank. The mass of condensate collected in the weigh tank was measured using an electronic scale. The scale reading was communicated to the data acquisition system (DAS) over RS232 interface and recorded along with the coinciding time at the same sampling rate selected for recording temperature measurements.

During previous tests, a highly accurate water meter, the same as used in commercial and residential fresh water supply lines, was used to register the total condensate passed. Temperatures above 90°F damage the meter's oscillating piston-measuring mechanism so the flow is cooled by counterflow heat exchange cooled by fresh city water. The meters have a nominal 20 gallons per minute capacity and register down to 0.25 gallons per minute with an accuracy between 98 percent and 100.8 percent over the range. At a rate of 0.25 gallons per minute, the meter registers 98 percent of actual flow; at 2 gallons per minute, 100.8 percent; at 13 gallons per minute, 100 percent; and at 20 gallons per minute, 99.8 percent. Two gallons per minute condensate flow is typical for an average test. A flow volume, typically 10 or 20 gallons, was timed using a stop-watch to measure flow rate. Measurements were found to be very repeatable. With the liquid drain trap below the vessel, unmeasured condensate inventory was small and flow to the meter very steady. Even with the raised trap used when the vessel was filled with water to within 5 ft. of the top, trap operation still provided a very steady flow.

In order to simplify and directly record the measurement of the condensate flow, a 330 lb. capacity, splash-proof scale with digital output to the DAS was used to measure condensate flow rates together with a 55-gallon stainless steel condensation collection drum. During testing, the drum drain is closed and the collected condensate weights are recorded along with the time. An automatic level switch is used to cycle the drain valve as the drum is filled. Condensate flow rates are then calculated using the differences in the recorded weights and times. The condensate cooling exchanger is used to suppress loss of condensate from flashing liquid as its pressure is reduced when leaving the vapor trap.

The temperature of steam after the automatic pressure-reducing valve and just before the coupling to the steam distributor in the vessel is registered by a 1/16-in. diameter stainless steel sheathed copper-constantine thermocouple held in the steam flow by a 1/16-in. inner-diameter tube, using a 1/16-in. pipe male connector threaded into the supply pipe wall. The thermocouple is connected through controlled purity, extension wire to a low-level volt meter and analog to digital conversion circuit in the data acquisition equipment. The thermocouple to extension coupling is installed inside water-resistant rubber boots. Condensate temperature is similarly measured as it leaves the vessel.

Steam pressure was measured using a pressure transducer that provided a continuous record of steam pressure in the vessel.

$$P_{internal} = 0.309 * I_{110} - 38.636$$

where $P_{internal}$ = internal vessel pressure (psia)
I_{110} = data output channel (110) (mV)

The enthalpy of the steam entering the vessel and the condensate leaving the vessel were determined using the steam inlet temperature, vessel pressure, and condensate drain temperature. Condensate mass flowrate was calculated by dividing the mass of condensate collected over a given time interval by the corresponding time duration. The heat input to the vessel or the total heat transfer from the vessel was determined by multiplying the difference of the steam and condensate enthalpies by the condensate mass flowrate.

2.3.2 Containment Vessel Wall Temperatures

Twenty-four 0.032-in. diameter stainless steel sheathed copper-constantine thermocouples attached to the outer vessel wall measured vessel surface temperature. Each thermocouple junction end was installed in a 1/32-in. deep, 1/32-in. wide groove approximately 0.75-in. long and peened into place. The grooves were filled with solder and finished to provide a smooth outer surface.

The thermocouples were installed at locations representative of all the vessel heat transfer areas. The temperature measurement locations are shown schematically in Figure 2.3-1. Two sets of three thermocouples were located at each median radius of two equal areas on the vessel head with the thermocouples in each set located 120 degrees apart around the circumference. Three sets of three thermocouples were located on the top vessel side wall; these thermocouples were located in the middle of three 18-in. high areas with three thermocouples spaced 120 degrees apart in each area. The lower three sets were located in the middle of three 72-in. high areas on the lower vessel side wall also with three thermocouples spaced 120 degrees apart in each area.

The heat flux meters were used to find local heat flux density, average heat flux density, and total heat flux from the vessel. Average heat flux density at a certain vessel elevation was found by averaging the data for temperature differences for each pair of three thermocouples at that elevation. The averages were then multiplied by the fractional area on the vessel in which the set was centered. The area fraction assigned to the two sets of three thermocouples on the top vessel head was [ ]^{a,c} the area fraction assigned to the upper three sets of thermocouples on the vertical side wall was [ ]^{a,c} and the lower three sets of thermocouples on the vessel vertical side wall were assigned area fractions of [ ]^{a,c} giving a total of [ ]^{a,c} for the eight sets of three thermocouples. The resulting average temperature difference when multiplied by 835 Btu/hr-ft^2-°F is the average heat flux density in Btu/hr-ft.^{2}. When multiplied by [ ]^{a,c} the heat transferring area of the vessel, the total heat flux is obtained.

Thermocouples are copper and constantine in a 0.032-in. diameter stainless steel sheath that is 5 ft. long to a transition to flexible wire and then a connector to extension wire. The connectors are enclosed in water-resistant sealing rubber boots. Each thermocouple junction end was installed in a 1/32 in. deep, 1/32-in. wide groove approximately 0.75 in. long. Each was peened in place by using a chisel to make indentations on each side of the groove. Junctions of a pair are located opposite each other on either side of the wall and the leads run parallel in opposite directions. The inner thermocouples are led into the vessel through 1/16-in. diameter tubing held by 1/16-in. pipe thread male connectors. Thermocouple grooves and the thermocouple penetration inside the vessel were filled with solder and finished to provide smooth inner and outer surfaces.

2.3.3 Containment Annulus Air Flow, Temperature, and Humidity

The velocity of air entering the annulus inlet air duct is measured using a four in. diameter rotating vane anemometer. The air velocity is measured at the center of 12 equal areas at the inlet face of the air heating coil. The areas are 16-in. high and 24-in. wide. The anemometer dial registers ft. per second, and the time was typically 30 seconds. (Note: When testing in the 5-in. annulus configuration, one half of the heater inlet face area is blocked to maintain the axial fan motor rpm at an efficient operating speed and the number of velocity measurements is reduced from 12 to 9). The volumetric cooling air flowrate entering the inlet duct is determined by multiplying the average measured inlet face velocity by the heating coil inlet face area, 16 ft.^{2} in the 5-in. annulus configuration, 32 ft.^{2} in the 15-in. annulus configuration.

$V_a = V_i * A_i * 60$

where V_a = volumetric air flow rate
V_i = inlet face velocity (anemometer)
A_i = steam heating coil inlet face area

The average air temperature entering the annulus is measured using three thermocouples located in the middle of the inlet to the annulus at three locations 120 degrees apart. The average air temperature leaving the annulus was measured using three sets of three thermocouples centered in equal areas at the outlet of the air annulus before the fan with the three thermocouples in each set located 120 degrees apart. The average annulus air temperature was calculated by averaging the annulus inlet and outlet temperatures.

$T_{annulus} = (0.5 * (T_{inlet} + T_{outlet}))$

where $T_{annulus}$ = overall annulus air temperature
T_{inlet} = annulus inlet temperature
T_{outlet} = annulus outlet temperature

The annulus air velocity is calculated by multiplying the average heating coil inlet face velocity by the ratio of ambient air density to average annulus air density and the coil inlet face to annulus flow area ratio. The annulus flow area is 4.47 ft.^{2} for the 5-in. annulus width resulting in a coil inlet face to annulus flow area ratio equal to 3.579 for the 5-in. annulus configuration. The annulus flow area is 16.69 ft.^{2} for the 15-in. annulus width resulting in a coil inlet interface to annulus flow area ratio equal to 1.917 for the 15-in. configuration.

$V_{annulus} = [(V_{air} * A_{sh})/A_{annulus}]/60$

V_{air} = air velocity at the steam heater (ft/min)
A_{sh} = area of steam heater (32 ft.^{2} or 16 ft.^{2})
$A_{annulus}$ = area of annulus (16.69 or 4.47)

Relative humidity of the annulus air is measured by traversing, at the same locations used for air temperature traverse measurements, the annulus flow path using a portable hand-held digital relative humidity and temperature indicator.

The sensor in the probe is a one-micron thin polymer film that absorbs water vapor and whose electrical capacitance is sensed. It and the instrument have ± 2 percent accuracy for the 0 to 80 percent relative humidity (RH) and ± 3 percent from 80 percent RH to 100 percent RH. Its repeatability is better than 1 percent RH. It may be used between temperatures of -40°F and 199°F, up to 100 percent RH, and for temperatures up to 220°F and 80 percent RH. Thus, its range is more than adequate for all test conditions. A calibrator is used with the instrument to maintain accuracy.

2.3.4 Annulus Wall Temperatures

The outer surface temperature of the 15-in. annulus wall was measured using four 0.032-in. diameter, stainless-steel-sheathed, copper-constantine thermocouples attached to the outer surface of the acrylic cylinder. The wall thermocouples were located at four elevations adjacent to four vessel wall temperature locations, as shown in Figure 2.3-1.

In the 5-in. annulus test configuration, four additional thermocouples were attached to the outside surface of the inner (5-in. annulus) acrylic cylinder at locations adjacent to the thermocouples installed on the outer (15-in. annulus) cylinder.

The inner (5-in.) and outer (15-in.) annulus wall temperature measurements were used to calculate the temperature differences across the annulus walls and evaluate heat losses due to convection and radiation from the annulus walls to ambient air.

2.3.5 Wind Speed and Direction

A weather vane/anemometer was mounted on the roof of the building adjacent to the test tower approximately 12 ft. above ground level. The wind speed and direction indicated by the anemometer were continuously monitored and recorded on the DAS channels 112 and 113, respectively. This data was used to confirm that the steady state test data was not influenced by high or gusting (>5 mph) local wind effects.

$V_{wind} = 100 * I_{112}$
$D_{wind} = I_{113}$

where V_{wind} = velocity of wind (mph)
I_{112}, & I_{113} = data output channels 112 & 113
D_{wind} = direction of wind

2.3.6 Water Film Flowrate and Temperature

The flow rate of the water film added on the outside of the vessel at the top is measured using a water meter. The volumetric flow rate is obtained by recording the elapsed time required to pass a number of gallons of water through the water meter. The temperature of water passing through the meter is observed using a copper-constantine thermocouple installed in the water line, so that water density can be determined to permit calculation of the mass flowrate from the measured volumetric flowrate.

The water film flow that does not evaporate is collected in the drain pan and measured using a water meter after a drain water cooler (heat exchanger). A thermocouple registers water temperature before the meter. Flow rates are again found by measuring the time it takes to pass a given volume of water through the meter. For some test conditions, the film drain flow is so small that a graduated cylinder is used instead of the water meter.

A 1/16-in. diameter, sheathed copper-constantine thermocouple immersed in the inlet line to the film water flow distributor was used to measure the actual temperature of the film water supplied to the test vessel.

2.4 Data Acquisition System (DAS)

Test measurements, such as annulus air velocity and atmospheric pressure, were obtained using portable instruments and manually recorded in a data log along with the time at which the observations were made. Thermocouple temperature measurements and collected condensate weight were processed by a DAS.

Thermocouples were connected to the system by 20 AWG, copper and constantine, special limit (controlled-purity) duplex extension wires with solid PVC insulation. All thermocouple outputs were recorded using an electronic data logger unit. Thermocouple extensions were connected to isothermal terminal blocks that plugged into sets of low-level input cards on the data logger or an extender chassis that connected with the data logger. The voltage signals were converted to digital temperatures as the data logger sequentially sampled the inputs.

The data acquisition was done according to a pre-selected sequence programmed into the data logger in accordance with the channel assignment shown in Table 2.4-1. The data logger acquires a set of data within an 8-second interval, but requires an additional 8 seconds to store the data to an internal floppy disk or approximately 30 seconds to store to the computer. Short term transient data is therefore accumulated on the internal floppy disk, while the remainder of the data is accumulated on the computer. Condensate weigh-tank output is only obtained during computer-controlled data acquisition and the data transformation program inserts a constant value during all times when the condensate data is unavailable.

2.5 Facility Operation

The small-scale tests were essentially performed in the same manner. Specific differences, such as vessel pressure, steam flow rate, degree of water coverage, rate of annulus flow and presence of noncondensables are detailed in the description of the individual tests. The following is a summary of the detailed test procedure used for the small-scale tests.

1. Test instrumentation calibration and setup are verified prior to the start of the test. Three strip charts were setup to receive real time signals throughout the test for key variables such as steam flow rate and temperature, vessel pressure and internal velocity meter outputs. The ambient conditions of pressure and humidity are recorded. Tests are conducted only when the ambient air relative humidity is 100 percent or less at 88°F and lower temperature, 80 percent at 95°F or 68 percent at 100°F.

2. The test vessel is closed up and the data acquisition system is cycled to provide the initial vessel conditions prior to the start of the test.

3. The steam boiler is fired up to produce steam flow and pipe trace heating is activated to prevent condensation in the steam supply lines to the test section. Prior to starting the test, the steam is vented to atmosphere to get to full operating conditions.

4. The fan is turned on to provide the specified air flow.

5. The electrical superheating of the steam line is started with control set to 350°F. The steam flow to the air heating coil is then started with the control set to 130°F, except for tests 132, 133, and 134, when there is no heating supplied.

6. The initial air pressure is established in the vessel.

7. The water heater is vented and control set to 10°F above specified film water delivery temperature. The film water flow is started at the specified rate.

8. The air flow (at face of heating coil), film water flow and film water temperature are measured and adjusted if necessary.

9. The DAS is activated prior to the start of steam flow to the test section. In the majority of tests, the DAS was operated in local mode for the first 10 minutes of each transient before reverting to computer mode with data accumulated approximately every 2 minutes. The data is stored to unique file names for later data handling. The steam system is opened to provide the required steam flow to the test vessel.

10. During performance of the test, the distribution of water coverage at the bottom of the baffle was recorded during the steady state portion of the test. Noncondensible samples were taken at approximately one hour intervals as required by the individual test to determine the concentration of noncondensables.

TABLE 2.3-1
SUMMARY OF TEST INSTRUMENTATION

Sensor Description	Model No.	Channel No.	Calibration Range	Accuracy
Pressure Transmitter	Foxboro Ref. No. 90F40362-18A1 Ref. No. 23925	110	0 to 85 psig	± 0.5%
Steam Control Valve Actuator Pressure Transducers	Foxboro Model No. 843DP-A211NS-M SN 23527	111	0 to 15 psig	± 0.5%
Pressure Gauges	Marshall Town Model No. G28771 Ref. No.'s 23827 & 23828	N/A	30 in. Hg to 60 psig	± 0.5 psig
Water Meters	Rockwell Model No. SRII SNs 41927375, 40551131, 40551132 and 40434331	N/A	0 to 12 gal	± 4.5%
Thermo-Anemometer	Alnor Instrument Co. Model 8500 SN 1891	N/A	10 to 2000 FPM	± 3%
Vane Anemometers	Taylor Instrument Co. Model No. 3132-A-4 SNs E576 and 2256	112, 113	150 to 2000 FPM	± 3%
Humidity/Temperature Meter	Vaisala Co. Model No. HMI-31 SN 241913	N/A	0 to 100% RH -40 to 199°F	± 2% 0 to 80% ± 3% 80 to 100% ± 0.6°F
Fluke Data Logger	Fluke Co. Model No. 2240-B SN 2385069	N/A	32 to 750°F	± 0.8°F
Thermocouples	TT	Numerous	TT: 0 to 350°C	TT: ± 0.5°C
Platform Scale	A.N.D. Co. Model No. FV-150 KA1 SN C1810326	N/A	10 to 200 lbs.	± 0.05 lb

TABLE 2.4-1
DATA LOGGER FUNCTIONS

Channel	Functions
1	TC, Steam to Vessel
2	TC, Condensate From Vessel
3	TC, Condensate After Cooler
4	TC, Water at Film Distributor
5	TC, Water Drain From Pan
6	TC, Drain Water After Cooler
7	TC, Ambient Air Under Cooler
8, 9, 10	TC, Air Inlet to Annulus; Front, Side, Back Sector
11, 12, 13	TC, Air Outlet, Front Sector, Inner, Mid, Outer Rad
14, 15, 16	TC, Air Outlet, Side Sector, Inner, Mid, Outer Rad
17, 18, 19	TC, Air Outlet, Back Sector, Inner, Mid, Outer Rad
20, 21, 22	TC, Air Traverse, Top, Front, Side, Back Sector
23, 24, 25	TC, Air Traverse, 2/3 Height, Front, Side, Back
26, 27, 28	TC, Air Traverse, 1/3 Height, Front, Side, Back
29	TC, Spare
30 - 35	TC, Vessel Wall, Dome, Inner Rad, Even Inside
36 - 41	TC, Vessel Wall, Dome, Outer Rad, Even Inside
42 - 47	TC, Vessel Wall, Upper 1.5 Ft.
48 - 53	TC, Vessel Wall, 2nd 1.5 Ft.
54 - 59	TC, Vessel Wall, 3rd 1.5 Ft.
60 - 65	TC, Vessel Wall, Next 6 Ft.
66 - 71	TC, Vessel Wall, 2nd 6 Ft.
72 - 77	TC, Vessel Wall, Bottom 6 Ft.

Note: 30-77 are even inside wall, odd on outside, front, side and back in order

TABLE 2.4-1 (Cont.)
DATA LOGGER FUNCTIONS

Channel	Functions
78	TC, Steam Inlet to Control Valve
79	TC, City Water Inlet to Water Heater
80	TC, Steam Pipe at Inlet to Superheating Section
81	TC, Steam Pipe at Outlet From Superheating Section (Not recorded, but displayed on controller)
82 - 87	TC, Upper Probe, Tip, Tip, 5, 10, 15, 20 in. From Tip
88 - 93	TC, Lower Probe, Tip, Tip, 5, 10, 15, 20 in. From Tip
94 - 99	TC, Inside Vessel Traverse, Upper, 2nd, 3rd 1.5 Ft., Next, 2nd, Bottom 6 Ft., All Front Sector
100 - 103	TC, Outer (15") Baffle Wall, Top 4.5 Ft., Next, 2nd, Bottom 6 Ft., All Front Sector
104 - 107	TC, Inner (5") Baffle Wall
110	Vessel Pressure Transducer
111	Steam Valve Pneumatic Control Air Pressure Transducer
112	Wind Speed
113	Wind Direction

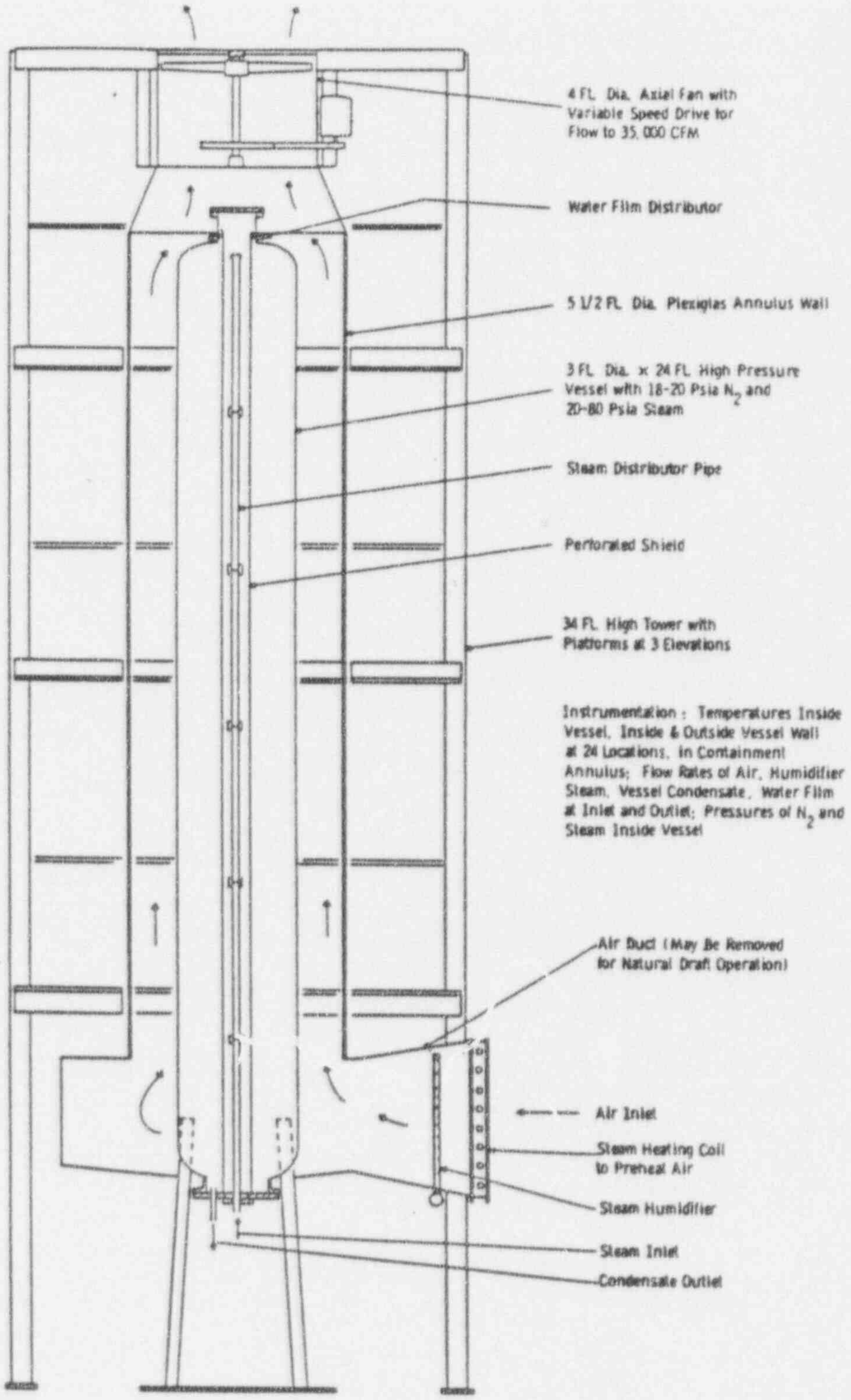


Figure 2.1-1 Section View of AP600 Integral Small-Scale Test

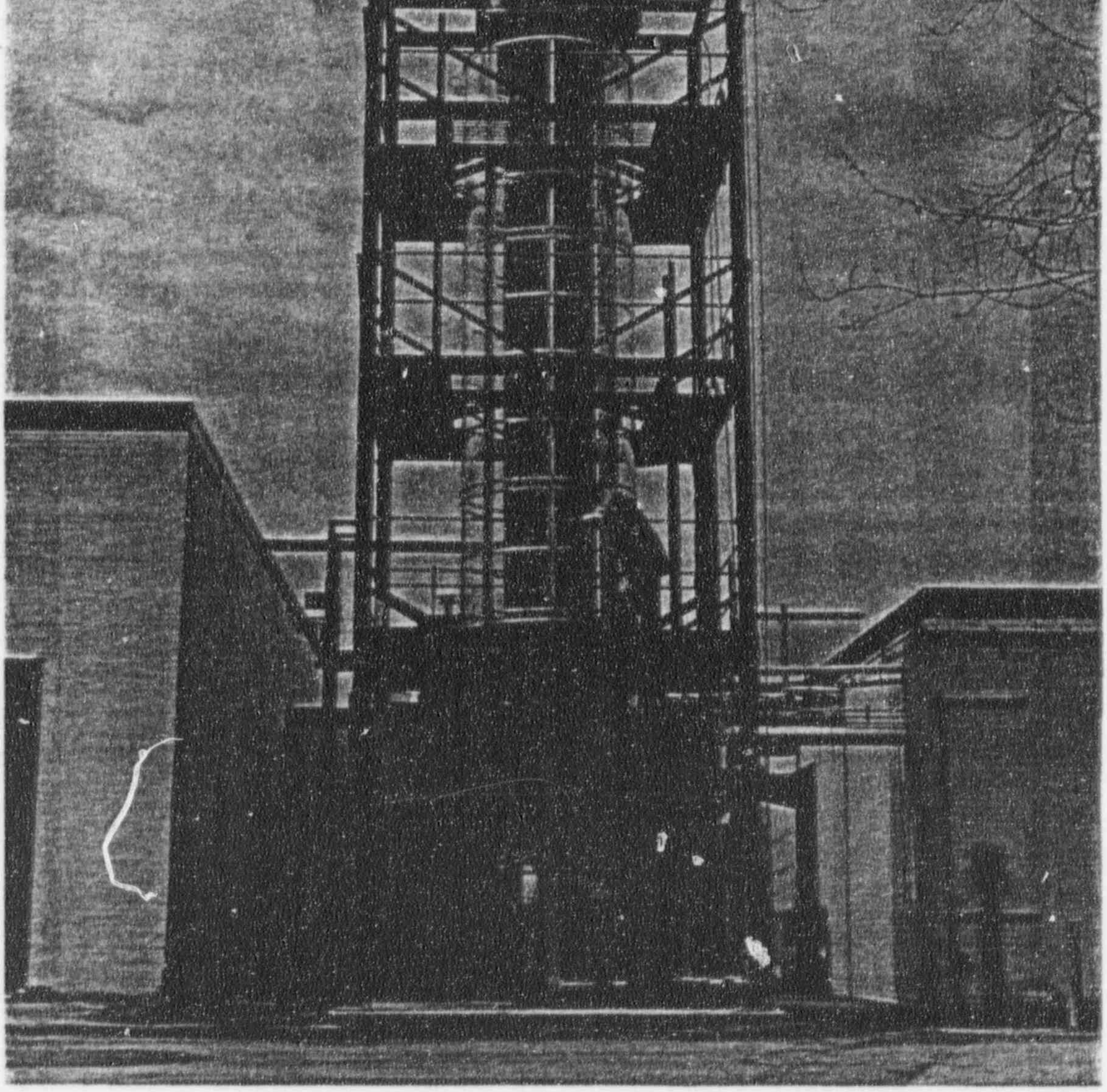


Figure 2.1-2 Passive Containment Cooling System Test Apparatus

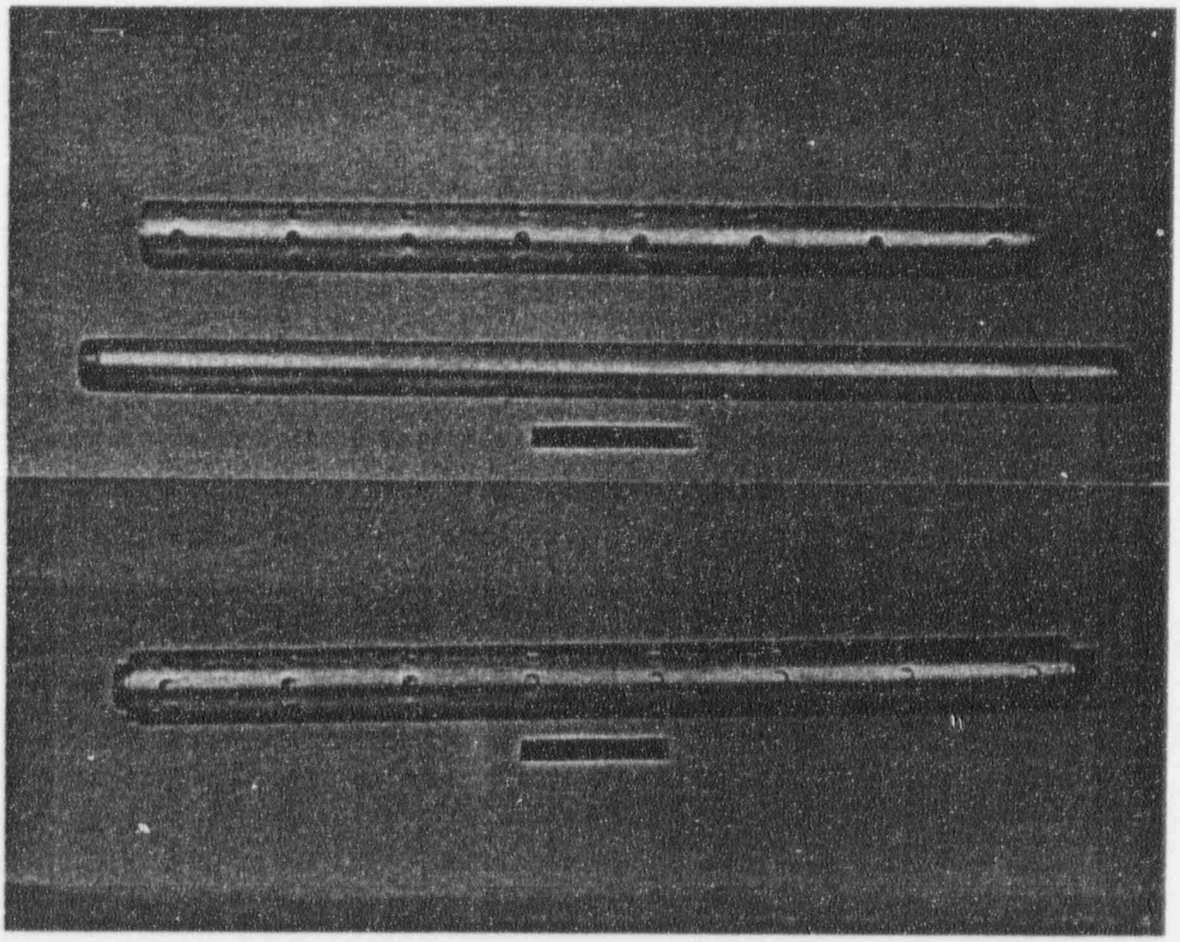


Figure 2.2-1 **One Section of the Uniform Steam Distributor.** (The upper photo shows the outer shield tube above the supply pipe with metering holes. The lower photo shows the two assembled.)

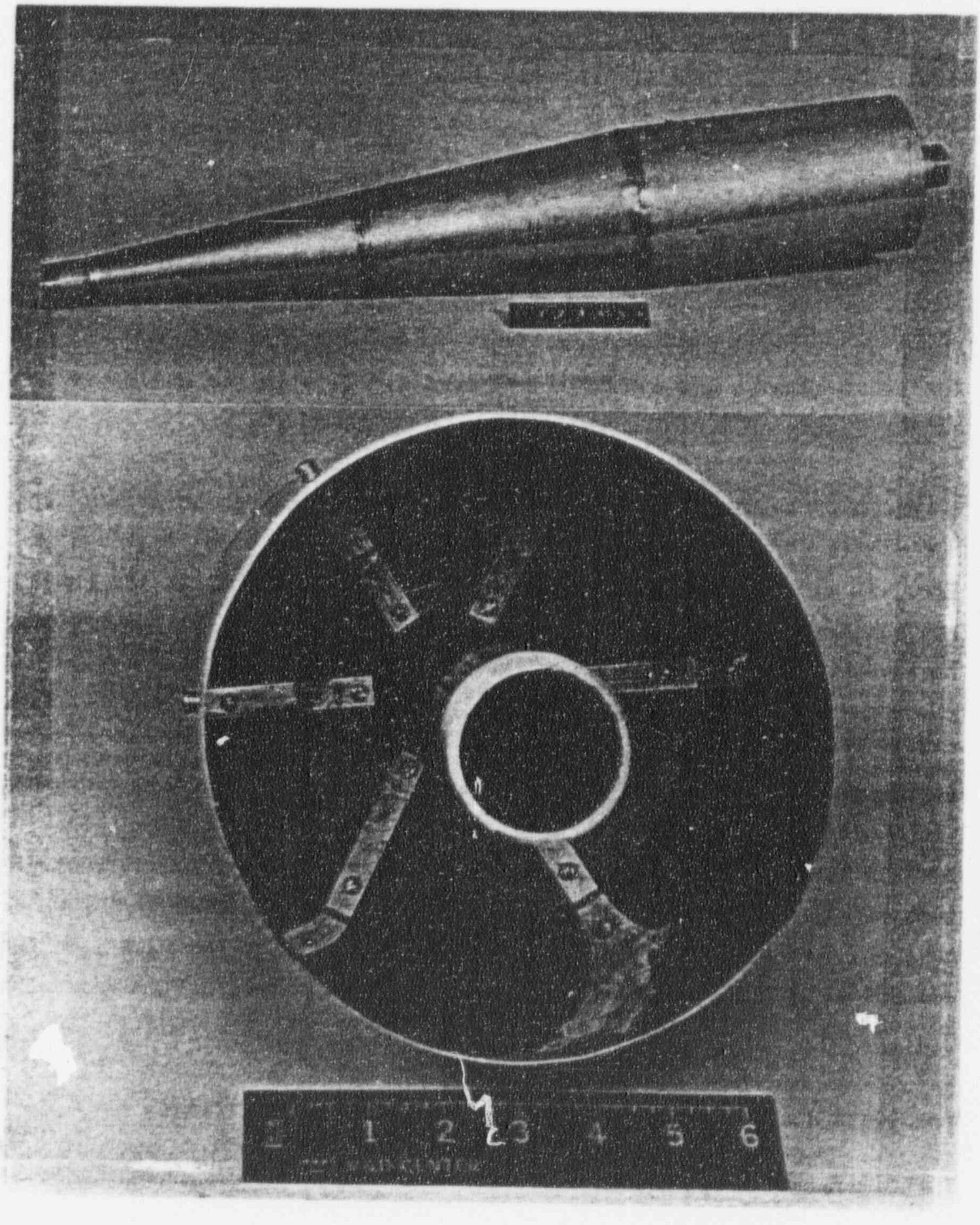


Figure 2.2-2 Side and Top View of Prototype Steam Distributor

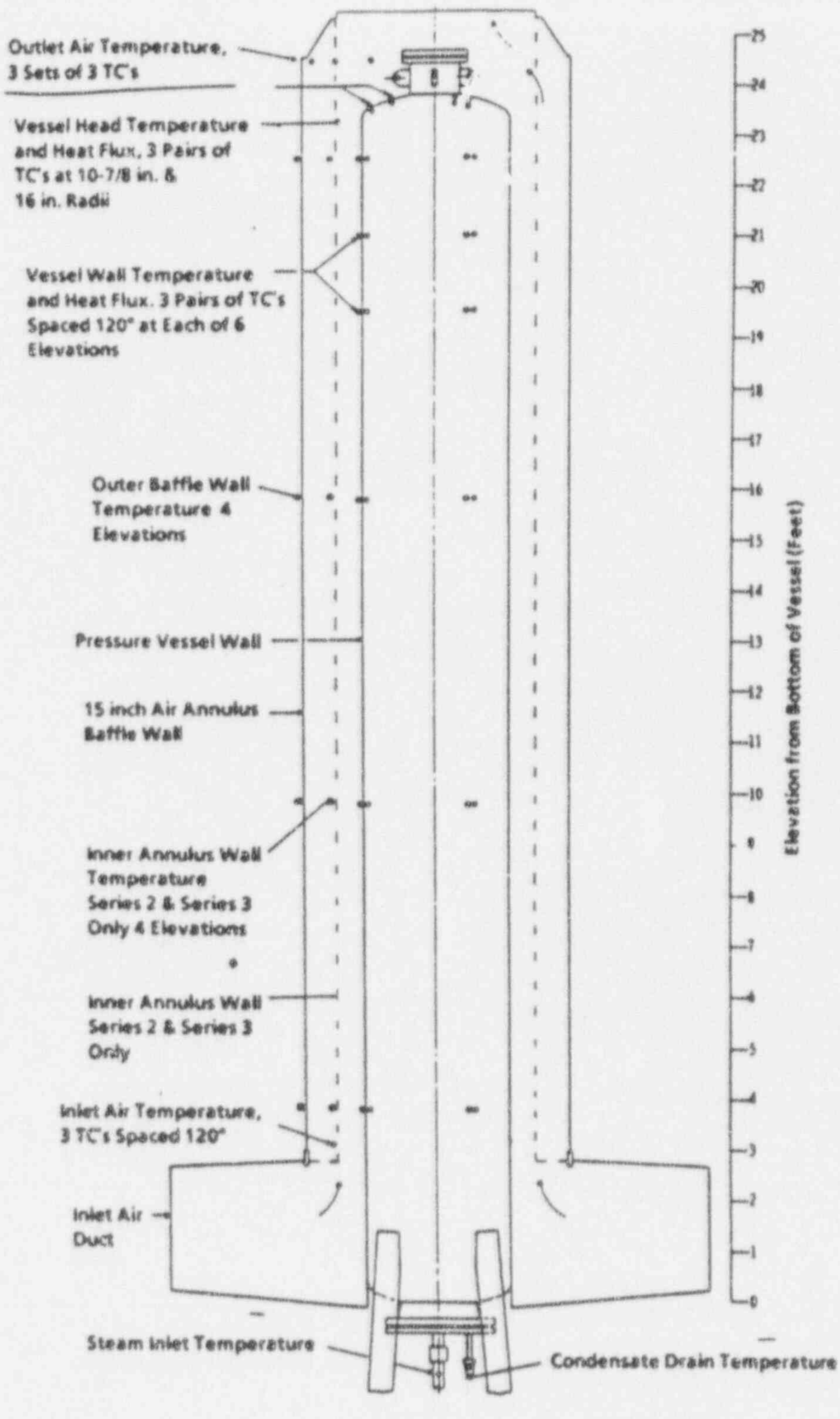


Figure 2.3-1 **Temperature Measurement Locations.** (Locations for vessel wall temperature measurements, baffle wall temperature measurements, containment annulus air temperature measurements, and inlet steam and condensate temperature thermocouples.)

3.0 DATA REDUCTION

3.1 Introduction

As discussed in Section 2.4 of this report, test measurements such as thermocouple temperature and condensate weight are processed by a data acquisition system. The thermocouple outputs are recorded using an electronic data logger unit. The output from the data logger is transferred to a computer for display and storage. The data is then transferred to a floppy disk, entered into a data log book, and then all data is analyzed by a following data reduction program.

The data is accumulated during the tests in the following forms:

1. The test record book, which provides documentation of the conduct of the test, includes any anomalies that may be experienced during the conduct of the test, and contains a record of the history of the test facility.

2. The Fluke data acquisition system (DAS) output, which is stored directly to disk (note that no data reduction is performed during these operations). The thermocouples are directly recorded in degrees Fahrenheit, and all others in actual millivolt or volt signals.

3. Strip chart recorders, which provide a qualitative description of the test trends for selected channels; such as temperatures, vessel pressure, etc.

4. Data recorded by hand on data sheets and on the individual test procedures. This data includes gas sampling data, helium concentration data, atmospheric pressure and weather conditions.

The primary source of test data is that recorded on the Fluke DAS. The Fluke data is in the form of an ASCII file containing the values of the 113 channels, listed in Table 2.4-1. The Fluke data file is manually edited to include the following two types of hand input data: 1) test identification and prerequisite data, which must be included in each hand input data set and 2) recorded data, which is data recorded by the test engineer and may or may not exist for a given test.

A data reduction program, written in FORTRAN, is used to reduce the data stored on the edited data files.

The ASCII files created by the data reduction program are then manipulated by the FoxPro database program and the Quattro Pro plotting program to create select data spreadsheets. Figure 3.1-1 illustrates the data reduction process.

3.2 Test Validation

All data generated by the small-scale tests were reduced by the FORTRAN code PCCS.FOR.

3.2.1 Test Acceptance

The following test acceptance criteria were established for the tests reported herein:

1. Data on forcing functions are available, i.e., steam flow rate, fan speed, water flow rates, inlet temperatures of steam, water and air. Strict adherence to the specific absolute pressures and flow rates is not necessary but values should be nearly constant as defined in the test matrix.
2. Data on response variables are available, i.e., condensate flow rates, excess water flow rates, air, water and steam outlet temperatures, vessel pressure, 80 percent of the vessel and fluid temperatures, and vessel water coverage measurements were taken.
3. Unplanned excursions must be evaluated on a case by case basis. Failures that may result in faulty data outputs are not acceptable.
4. The vessel pressure is maintained within the specified pressure limits during the constant pressure portions of these tests.

3.3 Test Analysis

3.3.1 Heat Balance

Heat balances were performed to determine, in a rough manner, the acceptability of the test data and instrument performance. Heat loads were calculated for the reported tests from the various methods listed below:

- Condensate mass flow rate
- External heat loss (water and air)
- Heat flux across the wall

Figure 3.3-1 illustrates the external heat loss and wall meter heat balance values relative to the heat losses calculated form the condensate measurements.

3.3.1.1 Internal Steam Condensation

The measured rate of steam condensation inside the test vessel represents the total heat removed from the vessel. The condensate heat load is determined by multiplying the difference of the steam and condensate enthalpies by the condensate mass flowrate. Enthalpies of the steam entering the vessel and the condensate leaving the vessel are determined using the steam inlet temperature, vessel pressure, and condensate drain temperature. Condensate mass flowrate is calculated by dividing the mass of condensate collected, in a weigh tank, over a given time interval by the corresponding elapsed time.

3.3.1.2 External Water Film Evaporation and Air Cooling

The rate of external water film evaporation is determined by measuring the difference in the mass flowrate of water supplied to the vessel surface and the mass of water collected in the drain pan at the bottom of the vessel. The mass flowrate of water film evaporated multiplied by the change in enthalpy between the water supplied to the vessel surface and the vapor leaving the film is the heat removed from the external vessel surface by evaporation. The average temperature of the evaporated vapor is calculated based on the average vessel outer surface temperature and the local heat flux measurements.

The heat added to the excess water film is determined by the drain mass flowrate multiplied by the change in enthalpy of the water supplied to the water film distributor and the collected drain water.

Heat transfer to the air flowing through the annulus between the vessel wall and the baffle results due to convection from the vessel water film (or vessel surface at dry areas) and baffle wall surfaces and cooling of the water vapor evaporated from the surface of the vessel. The heat transferred to the air from the vessel water film and baffle surfaces is determined by the product of the air mass flowrate, the difference in temperatures measured at the annulus inlet and outlet, and the specific heat of air evaluated at the average annulus air temperature. Heat transfer due to the cooling of water vapor is determined by the product of the mass flowrate of vapor evaporated from the vessel water film, the difference between the average temperature of the vapor leaving the water film and the annulus outlet temperature, and the vapor specific heat.

3.3.1.3 Vessel Wall Heat Flux Meters

The inner and outer vessel wall thermocouples (heat flux meters) are used to determine the local vessel heat flux, the average vessel heat flux, and the total heat transferred from the test vessel. Local heat flux is determined by multiplying the measured temperature difference for a given thermocouple pair by the vessel area fraction associated with the thermocouple pair and the vessel thermal conductivity (835 Btu/hr-ft^2-°F based on a thermal conductivity of 24 Btu/hr-ft^2-°F for carbon steel) and dividing by the distance between the thermocouples (11/32 in. thickness of steel between paired thermocouple junctions). Average heat flux for a given vessel elevation (total of 8) is similarly determined based on the average temperature difference for each of the three thermocouple pairs located at that specific elevation. The overall area weighted average vessel heat flux is determined by multiplying the average heat flux at each vessel elevation by the respective fractional surface area assigned to the three thermocouple pairs located at the given elevation. An area fraction [ ]^{a,c} is assigned to each of the two sets of three thermocouples located on the upper vessel head. An area fraction [ ]^{a,c} is assigned to each of the upper three sets of three thermocouples located on the vessel side wall. The lower three sets of three thermocouples located on the vessel side wall are each assigned an area fraction [ ]^{a,c} resulting in a total [^{a,c} for the eight sets of thermocouple pairs. The total vessel heat transfer determined by the heat flux meters is obtained by multiplying the calculated average area weighted heat flux by the total vessel heat transfer surface area [ ]^{a,c} exclusive of the lower vessel head and the region within the perimeter of the water film distributor ring).

3.4 Matrix Tests

The test matrix which identifies all tests that encompass the PCS small scale tests is shown in Table 1.3-1 of this document. Following is a list of the categories for these tests:

- Base Cases with Constant Steam Flow
- Base Cases repeated with Prototypic Steam Injection
- Water Film Limits of Cooling
- Water Film Distribution Limits of Cooling
- Vessel Air Content Impact on Inside Heat Transfer
- Baffle Air Flow Limits
- Water Film Temperature (Phase I)
- Water Film Temperature (Phase II)
- Transient Steam Flow
- Ice Formation and Melt Demonstration

These categories are summarized below. Results from selected tests in each category, except ice formation and melt demonstration, are reported in Section 4 of this document.

3.4.1 Base Cases with Constant Steam Flow

Seventeen base case tests with constant steam flow were run to establish internal vessel pressures under nominal cooling conditions. The steam flow was 0.25 lb/sec. All other conditions were identical except for the size of the annulus. Two tests were performed using the 5 in. annulus, the other 15 used the 15 in. annulus. Two of the tests, one using the 15 in. annulus (Run 38, Test 106-15U), the other using the 5 in. annulus (Run 70, Test 106-5U) gave invalid data. Test 106-15U was aborted because of rain and test 106-5U was aborted because the annulus air velocity was incorrect.

3.4.2 Base Cases Repeated with Prototypic Steam Injection

Four tests were run in this category. They were all performed using the 5 in. annulus width. These tests were performed using prototypic steam injection through a 6-in. diameter pipe at 5-6 ft elevation, offset from the center of the test pressure vessel. All these tests generated valid data.

3.4.3 Water Film Limits of Cooling

Three tests were run in this category. These tests determined the effects of film flowrate on coolability. The flowrates used were 0.5, 1.0, and 2.5 gpm. The flowrate was changed to degrade cooling until the limits of coolability were reached. One of these tests was not completed due to a computer failure.

3.4.4 Water Film Distribution Limits of Cooling

Nine tests were run in this category. These tests determined the effect of water film distribution on cooling. The water film distribution was varied from 100 to 66 to 33 percent. This variable was changed to degrade cooling until the limits of coolability were reached. All of these generated valid data.

3.4.5 Vessel Air Content Impact on Heat Transfer

Three tests were run to determine the impact of vessel air content on heat transfer. The initial air content was varied from 1.0 atm to 2.0 atm or 14.7 psia to 22 psia. The 5-in. annulus was used with prototypic injection. All of these tests generated valid data.

3.4.6 Baffle Air Flow Limits

These six tests were run to determine the limits of air flow in the baffle. The air flow was varied from 12 ft./sec to natural circulation. All data generated from these tests were valid.

3.4.7 Water Film Temperature (Phases I and II)

There were two tests run in Phase I to determine the effects of the shell water film temperature. In both cases, the temperature was raised to 120°F, increased from the normal temperature of 80°F. These tests were performed with two steam flows; 0.1 lb/sec and 0.25 lb/sec. Both of these tests produced valid data. Phase II tests varied both the steam flow and the water film temperatures. In the Phase II tests, the temperatures were 40 and 80°F. These tests also produced valid data.

3.4.8 Transient Steam Flow

These tests varied the steam flowrate. Rather than a steady state value for steam flow, a transient rate from 1.5 lb/sec to 0.1 lb/sec over 30 seconds was used for both tests in the category. The data generated was valid.

3.4.9 Ice Formation and Melt Demonstration

These tests were run to determine the onset of ice formation in the annulus region and any other effects of cold weather of the passive containment cooling capabilities of the system. Both tests produced valid data.

3.5 Test Summary

Table 3.5-1 provides a summary of the channels that are considered failed during small-scale testing. Table 3.5-2 provides a summary of all the test runs performed and how they vary from the base cases. Test runs identified with an asterisk identify the qualified matrix tests reported herein.

TABLE 3.5-1
SUMMARY OF REPORTED TESTS AND FAILED CHANNELS

Test Matrix No.	Run No.	Channel Numbers										
		17	52	65	77	81	89	90	93	95	97	99
105-15U	009	X	X	X	X	X						X
106-15U	047		X	X		X		X				
106-15U	041		X	X		X						
106-15U	039		X	X		X						
107A-15U	003	X	X	X	X	X						
107C-15U	007	X	X	X	X	X						
106-5U	074		X	X		X		X	X			
106-5U	071		X	X		X						
107A-5U	072		X	X		X		X		X		
109B-15U	009	X	X	X	X	X						X
113A-15U	014	X	X	X	X	X						
113B-15U	015	X	X	X	X	X						
114A-15U	007	X	X	X	X	X						
114B-15U	007	X	X	X	X	X						
107A-5P	076		X	X		X	X	X		X	X	
111-5P	084		X	X		X	X	X	X			
117C-15U	012	X	X	X	X	X						
120A-15U	012	X	X	X	X	X						
121-15U	018	X	X	X	X	X						
131A-15U	006	X	X	X	X	X						
132A-15U	019	X	X	X	X	X						
132B-15U	020	X	X	X	X	X						

TABLE 3.5-2
SUMMARY OF TEST RUNS

Test Run #	Test Matrix #	Comment
3*	107A-15U	Low steam flow rate
4	133A-15U	Cold baffle temps
5	133B-15U	Cold baffle temps
6A*	130-15U	Base Case, water film temperature
7B*	107C-15U	High steam flow rate
7C*	114A-15U	100% water distribution
7D*	114B-15U	66% distribution, low film flow
8	108B-15U	Low steam supply and water film flow
9A*	105-15U	High baffle flow
9B*	109B-15U	Low water film flow rate
9C	109A-15U	Water film limits of cooling
9D	117B-15U	Water film limits of cooling
10A	109C-15U	Water film limits of cooling, test incomplete
10B	115A-15U	Low steam supply flow
10C	115B-15U	66% water distribution, low film flow
11	115C-15U	33% water distribution, low film flow
12A*	120A-15U	Water film temp. increased
12B	120B-15U	Water film temp. increased
12C*	117C-15U	Low baffle flow rate
13A	117E-15U	No baffle air flow
13B	113B-15U	Film flow decreased, distribution 66%
14*	113A-15U	High film flow, low steam flow
15A*	113B-15U	Low film flow rate, 66% distribution
15B	113C-15U	Film flow decreased, distribution 33%

TABLE 3.5-2 (Cont.)
SUMMARY OF TEST RUNS

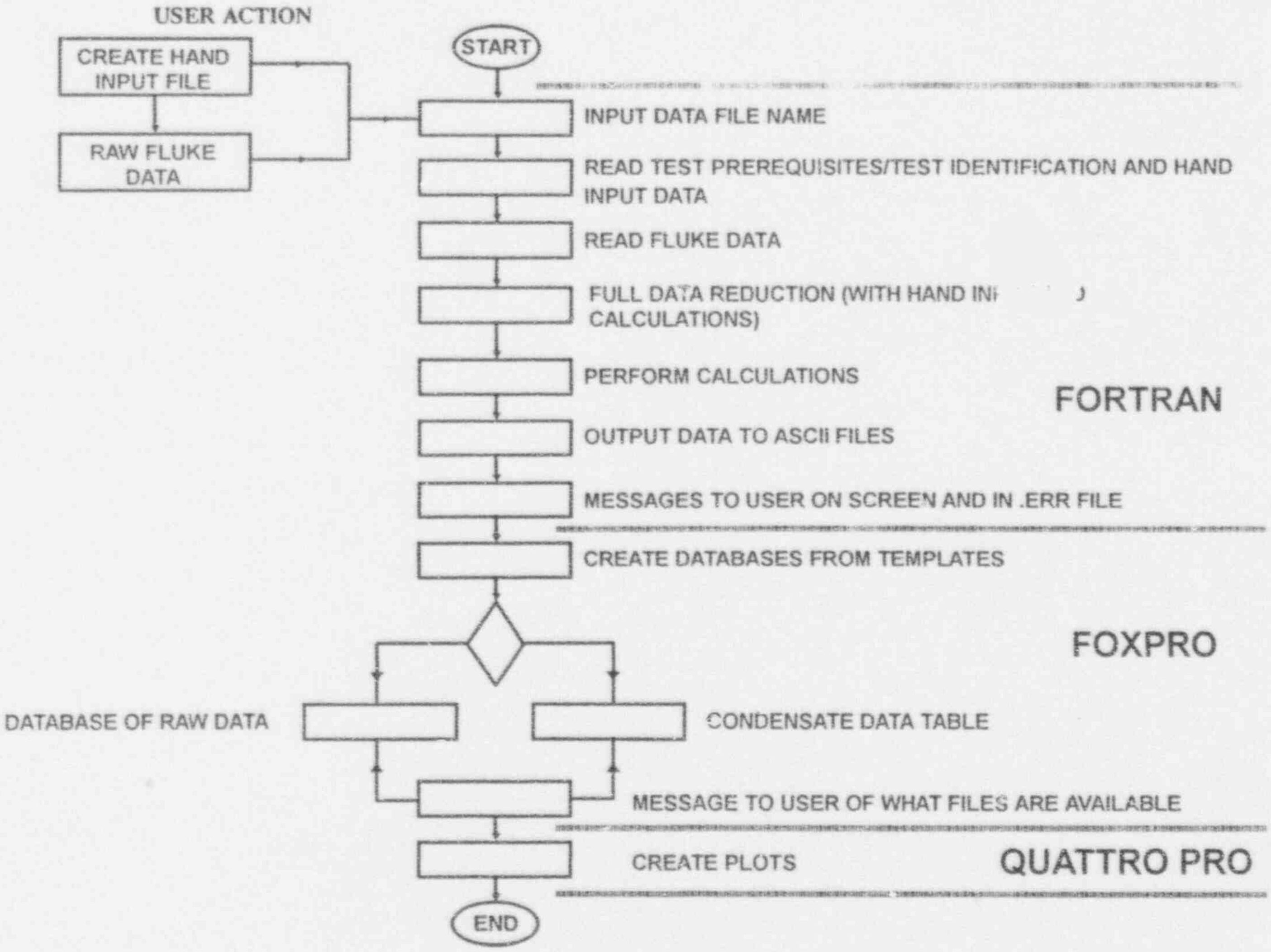
Test Run #	Test Matrix #	Comment
16A	118A-15U	No baffle air flow, low steam supply flow
16B	118B-15U	No baffle air flow
17A	119A-15U	No air flow, no water film flow
17B	119B-15U	No air flow, low steam supply flow
18A*	121-15U	Transient steam flow
18B	122-15U	Transient steam flow
19*	132A-15U	Low annulus air temperature
20A*	132B-15U	Water film temp decreased, low steam flow
20B	106-15U	Base Case
21	106-15U	Base Case
22	106-15U	Base Case
23	106-15U	Base Case
24	106-15U	Base Case
25	106-15U	Base Case
38	106-15U	Base Case
39*	106-15U	Base Case
40	106-15U	Base Case
41*	106-15U	Base Case, without water film
42	106-15U	Base Case
44	106-15U	Base Case
45	106-15U	Base Case
46	106-15U	Base Case
47*	106-15U	Base Case
70	106-5U	Smaller baffle width
71*	106-5U	Smaller baffle width
72*	107A-5U	Smaller baffle width

TABLE 3.5-2 (Cont.)
SUMMARY OF TEST RUNS

Test Run #	Test Matrix #	Comment
74*	106-5U	Smaller baffle width
76*	107A-5P	Base Case with prototype steam distributor
78	106-5P	Base Case with prototype steam distributor
79	106-5P	Base Case with prototype steam distributor
80	106-5P	Base Case with prototype steam distributor
81	111-5P	Base Case with smaller baffle, prototype steam distributor
82	112-5P	Base Case with smaller baffle, prototype steam distributor
84*	111-5P	Base Case with smaller baffle, prototype steam distributor

Note:

* included in Section 4.0, Test Results



I:\fw\work\dataproc.pre

Figure 3.1-1 Data Handling Process

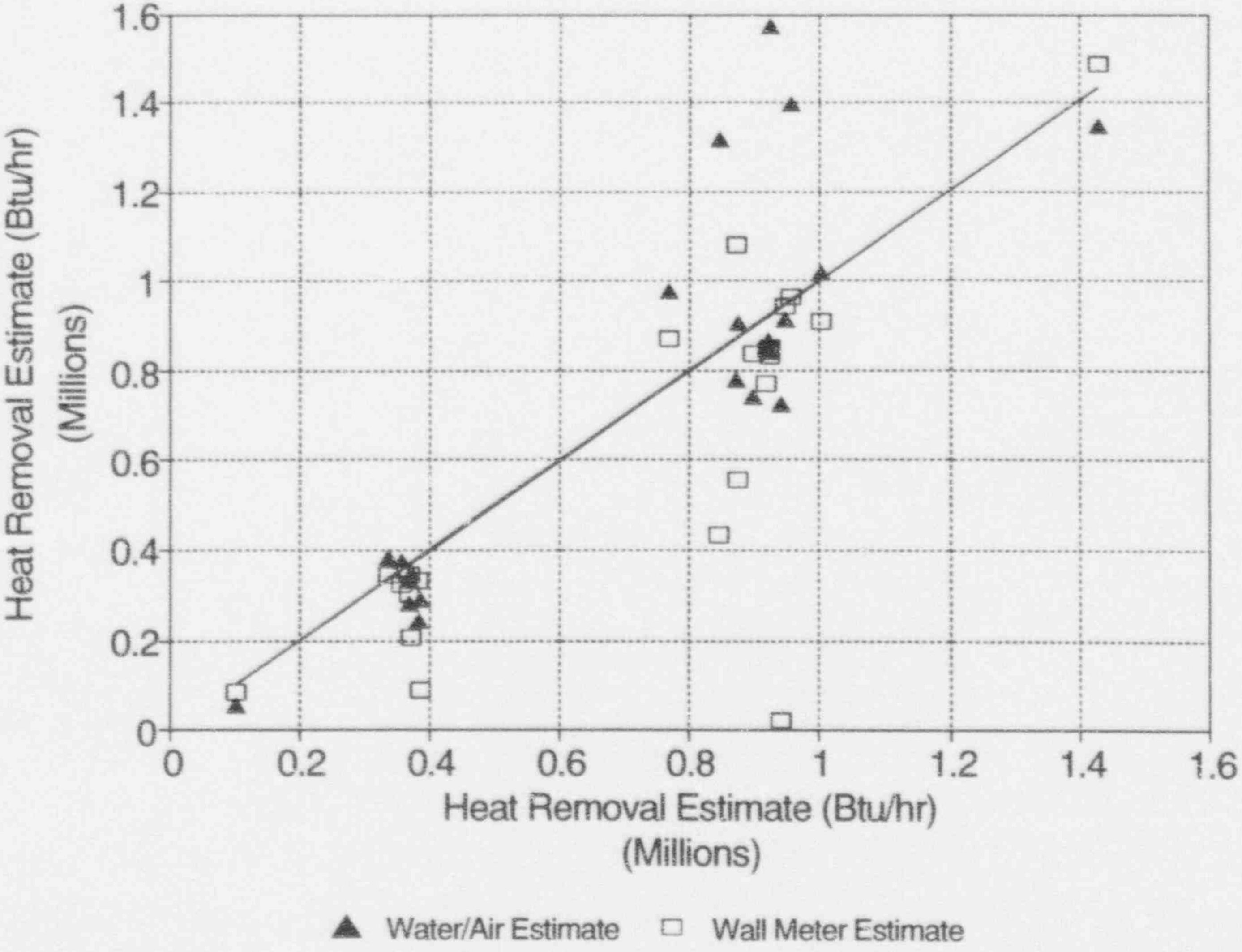


Figure 3.3-1 Comparison of Small-Scale Heat Removal Rates

4.0 TEST RESULTS

4.1 Matrix Tests Description

A summary of test conditions is listed in Table 4.1-1 for each of the reported tests. The table also categorizes each test according to its variation from the baseline tests. The following descriptions briefly summarize the purpose of each test and describe problems that may have occurred during each test. Tables 4.1-2 through 4.1-23 present a comparison (for each test) of the inside and outside wall temperatures recorded during a steady-state period from which each test's results were obtained. If a water film survey was conducted for a specific test, the table also describes the water film condition of the vessel and more specifically the condition in the area of each thermocouple. Figures 4.1-1 through 4.1-22 show the vessel pressure and condensate flow history versus time for each test.

4.1.1 Test 105-15U, Run 9A

This test is a baseline, constant steam flow test designed to measure the effects of a 0.25 lbm/sec steam flow rate on containment cooling. A higher-than-normal air baffle flow rate of 16 ft./sec was used to determine high baffle flow effects. The test was performed with the intention of using full vessel film distribution for cooling; however 100 percent film distribution was not obtained.

4.1.2 Test 106-15U, Run 47

This test is a baseline, constant steam flow test designed to measure the effects of a 0.25 lbm/sec steam flow rate on cooling. A normal baffle flow rate of 12 ft./sec was used, and an air sample test was conducted. Full vessel film distribution was intended; however a vessel survey was not conducted to determine actual film coverage.

4.1.3 Test 106-15U, Run 41

This test is a constant steam flow test that was performed without water film to determine the difference between film and air cooling and air cooling only. As in Run 47 the steam flow rate was 0.25 lbm/sec, the baffle flow rate was 12 ft/sec, air sampling was performed, and full vessel film distribution was intended but a survey was not taken. Table 4.1-4 indicates the higher inside and outside wall temperatures due to the dry outside vessel wall.

4.1.4 Test 106-15U, Run 39

This test is a repeat of Run 47. The superheat controller failed during conduct of the test and manual heater control was used. Note from Table 4.1-5 that the wall temperatures were slightly higher than those for Run 47.

4.1.5 Test 107A-15U, Run 3

This test is a baseline, constant steam flow test designed to measure the effects of a low steam flow rate (0.10 lbm/sec) on containment cooling. The baffle flow rate was again 12 ft./sec; however air samples were not taken.

4.1.6 Test 107C-15U, Run 7B

This test is a baseline, constant steam flow test designed to measure the effects of a high steam flow rate (0.50 lbm/sec) on containment cooling. Due to the limitations of the steam trap, the maximum steam flow rate achieved was 0.435 lbm/sec.

4.1.7 Test 106-5U, Run 74

This test used a 5-in. air baffle width to determine the effects of a smaller baffle on containment cooling. The remaining test conditions were intended to be the same as those for Test 106-15U. A water film survey was not taken; however air sampling was performed.

4.1.8 Test 106-5U, Run 71

This test is a repeat of Run 74. Just as in Run 74 a water film survey was not taken; however 6 to 8 film distributor tubes on the north side of the vessel were reportedly blocked causing dry streaks in that area of the vessel. Note from Table 4.1-9 the higher inside and outside wall temperatures.

4.1.9 Test 107A-5U, Run 72

This test used a 5-in. air baffle width to determine the effects of a smaller baffle on containment cooling. The remaining test conditions were intended to be the same as those for Test 107A-15U.

4.1.10 Test 109B-15U, Run 9B

This test is designed to measure the effects of a low water film flow rate (1.5 gpm) on containment cooling. The remaining test conditions were intended to be the same as those for Test 106-15U.

4.1.11 Test 113A-15U, Run 14

This test is intended to measure the effects of a high water film flow rate (2.5 gpm) and a low steam flow rate (0.10 lbm/sec) on containment cooling. The remaining test conditions were intended to be the same as those for Test 106-15U.

4.1.12 Test 113B-15U, Run 15A

This test is a repeat of Test 113A-15U with one exception; a 1.7 gpm film flow rate was used and a 66 percent film distribution on the outside of the vessel was achieved.

4.1.13 Test 114A-15U, Run 7C

This test is designed to measure the effects of a 2.5 gpm film flow rate and a 0.25 lbm/sec steam flow rate (100 percent film distribution). The remaining test conditions were identical to Test 113A-15U. Table 4.1-14 presents the inside and outside wall temperatures and the extent of water film coverage.

4.1.14 Test 114B-15U, Run 7D

Test conditions were identical to Test 114A-15U with one exception; a 1.7 gpm film flow rate was used and a 66 percent film distribution on the outside of the vessel was achieved. Table 4.1-15 presents the wall temperatures and the extent of water film coverage.

4.1.15 Test 107A-5P, Run 76

This test is designed to determine the effects of the prototype steam distributor (with a 0.10 lbm/sec steam flow rate) on containment cooling. The remaining test conditions were intended to be the same as those for Test 107A-5U. Three film distributor tubes were reportedly blocked during conduct of the test.

4.1.16 Test 111-5P, Run 84

This test is designed to determine the effects of the prototype steam distributor (with a 0.25 lbm/sec steam flow rate) on containment cooling. The remaining test conditions were intended to be the same as those for Test 106-5U.

4.1.17 Test 117C-15U, Run 12C

This test is designed to determine the effects of a low air baffle flow rate (8 ft./sec). The remaining test conditions were intended to be the same as those for Test 106-15U.

4.1.18 Test 120A-15U, Run 12A

This test is designed to determine the effects of a 120°F water film temperature on containment cooling. Remaining test conditions were intended to be the same as those for Test 107A-15U. Reported problems included the boiler shutting down, dirty water and the regulator for the steam valves not supplying enough pressure; however all problems were resolved.

4.1.19 Test 121-15U, Run 18A

This test is designed to determine the effects of a transient steam flow (1.5 lbm/sec down to 0.15 lbm/sec) on cooling. With the exception of steam flow the remaining test conditions were intended to be the same as those for Test 106-15U. An informal measurement of the water film was taken and indicated that significant dry-out of the vessel occurred during high vessel pressures. Full wetting of the vessel occurred when the pressure reached approximately 20 - 30 psi.

4.1.20 Test 131A-15U, Run 6A

This test is designed to determine the effects of a cold water film temperature (40°F, i.e, winter weather conditions) on cooling. The remaining test conditions were intended to be the same as those for Test 106-15U. A formal measurement of the water film was not taken; however the vessel was reportedly 87 percent wetted in lieu of the intended 100 percent film coverage.

4.1.21 Test 132A-15U, Run 19

Test conditions in this test were the same as Test 131A-15U with one exception; the annulus air temperature was below 32°F. A new method of firing the boiler was attempted to save water. The method did not work however so testing was continued using the old method.

4.1.22 Test 132B-15U, Run 20A

This test is test conditions same as Test 132B-15U with one exception; the steam flow rate was 0.10 lbm/sec.

4.2 Comparison of Results

Tables 4.2-1 and 4.2-2 give a summary of results for selected AP600 integral extension tests. The results listed in Table 4.2-1 were determined by the LOTUS data reduction program and the results given in Tables 4.2-2 were determined by the PCS data reduction program. Any comparisons of heat transfer/ flux results made in Section 4.2 refer to the heat transfer/flux obtained from the condensation of steam inside the vessel, since this type of heat transfer measurement was judged to be the most accurate. Figures 4.2-1 through 4.2-22 describe the vessel pressure and heat transfer (due to steam condensation) histories versus time.

Tables 4.1-1 through 4.1-23 and Tables 4.2-1 and 4.2-2 are not included in the non-proprietary version of this document.

Figures 4.1-1 through 4.1-22 and Figures 4.2-1 through 4.2-22 are not included in the non-proprietary version of this document.

5.0 CONCLUSIONS

The tests reported herein were considered acceptable since the test objectives and the criteria of Section 3.2.1 were met. Assessment of the heat balances of the reported tests shown in Figure 3.3-1 indicate that the majority of the tests performed in a consistent fashion and were in good agreement with respect to the three methods of heat balancing. The success and results of the tests should be based on the heat transfer/flux values determined form steam and condensate since these values were calculated using the condensate mass flow measured by a weigh tank (and directly recorded by the Fluke DAS) and did not involve human error or obvious irregularities in measuring techniques. Based on those values, the tests performed as expected and their overall behavior was repeatable.

6.0 REFERENCES

6.1 Tests of Heat Transfer and Water Film Evaporation from a Simulated Containment to Demonstrate the AP600 Passive Containment Cooling System, WCAP 12667, January 1990.

6.2 AP600 Integral Extension Test Specification, SEE-TE-0080, June 1990.

6.3 AP600 Small-Scale PCS Integral Extension Test Interim Test Report, PCS-T2R-006, March 1992.

6.4 AP600 Small-Scale PCS Test Preliminary Data Transmittal, February 28, 1992.

6.5 "Tests of Heat Transfer and Water Film Evaporation on a Heated Plate Simulating Cooling of the AP600 Reactor Containment," WCAP-12665, April 24, 1992.

6.6 AP600 Passive Containment Cooling System Data Reduction Program Manual, SEE-TE-0096, August 30, 1990.

6.7 "Integral Containment Cooling Test Extension—Test Specification," WCAP-13315, PCS-TIP-003.

APPENDIX A
DRAWINGS

This appendix is not included in the non-proprietary version of this document.

APPENDIX B
DATA FILES

This appendix is not included in the non-proprietary version of this document.