Preliminary

Basis for the Use of Forced Convection Heat Transfer Correlations in the AP600 PCS Natural Circulation Driven Air Cooling Path

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Summary

The Westinghouse SSAR calculations of heat and mass transfer from the AP600 containment shell to the PCS riser use a mixed convection correlation. The PCS scaling analysis[1] was performed using only forced convection in the riser, based on the fact that buoyancy effects are second-order and hence, forced convection is adequate for a scaling analysis. This report provides additional information to support the selection of those correlations for the SSAR and scaling analysis.

The approach taken in this analysis is to define the relationship between the Grashof and Reynolds numbers in the riser as the containment shell heats up during a design basis transient. The bulk flow rate in the PCS air flow path was calculated using the control volume equation for mass, momentum, and energy developed in the AP600 PCS Scaling Analysis. Those equations balanced the air flow path buoyancy with the drag forces based on measured form and friction loss coefficients, coupled with heat and mass transfer from the dry and wet containment surface into the riser air.

Following a postulated DECLG transient, the initial containment shell heatup is with a dry exterior containment surface, followed by cooldown with a wet external surface. The calculated Grashof-Reynolds number relationship for the riser air during the transient heatup and cooldown defined the PCS air flow path operating map. Examination of a criterion for the importance of buoyancy proposed by Jackson and Hall[2] shows that heat transfer in the AP600 riser, over the full range of wet and dry operation, is well characterized by forced convection heat transfer, with only minor buoyancy effects. Therefore, forced convection heat transfer correlations are good approximations for the PCS riser, and are fully justified for use in an approximate scaling analysis.

A comparison shows the Westinghouse mixed convection heat transfer correlation (Appendix 1) produces Nusselt numbers approximately 1% less than the Dittus-Boelter correlation for forced convection with AP600 shell-to-environment temperature differences greater than 2 °F. An alternate[3] mixed convection correlation produces Nusselt numbers 3 to 5% less than Dittus-Boelter over the same range of AP600 operation. These comparisons show that the Westinghouse mixed convection correlation is a good approximation to either Dittus-Boelter, or the alternate correlation.

Introduction

The AP600 passive containment cooling system (PCS) maintains the peak containment pressure below the design limit during any design basis accident (DBA). The PCS uses an external air cooling system to remove heat from the containment shell during DBA. The external air cooling system consists of air inlet ports located around the upper circumference of the shield building, an annular downcomer that is bounded by the shield building and the baffle, and a riser bounded by the baffle and containment shell. A chimney extends the riser above the shield building. The design is shown in Figure 1.

A large tank of water at the chimney elevation supplies an external cooling water film to the outside of containment when initiated by a high containment pressure signal. The development of the water film requires a period of time to initiate flow, fill piping, fill weirs, and wet the surface. It is conservatively assumed that this process takes approximately 11 minutes. Prior to 11 minutes the containment surface is assumed to be dry.

As the containment shell heats up, heat transfer to the riser air induces a buoyant, natural circulation air flow through the downcomer and riser. The hotter the shell, the greater the buoyant forces and the higher is the resulting PCS air flow rate. The PCS air flow rate and temperatures were calculated by solving the coupled momentum and energy equations for the PCS air flow path and its boundaries (shell, baffle, and shield building) with shell temperature treated as a parameter. With the air flow rate and temperatures, the riser Reynolds and Grashof numbers were calculated, defining an operating map for PCS operation. The Reynolds and Grashof numbers corresponding to the AP600 operating map were used to characterize the mixed convection flow in the riser.

Quantification Summary

Following a postulated LOCA, heat removal from the dry shell is by radiation to the baffle and convection to the riser air. The dry shell heat fluxes are low, on the order of 500 B/hr-ft^2-°F. In contrast, the condensation heat transfer from the containment gas to the shell inner surface is on the order of 10,000 B/hr-ft^2-°F. The large difference between the condensation heat flux into the shell and the dry heat flux out of the shell is accommodated by the heat capacity of the shell. The shell heat capacity gives it a time constant of approximately 4 minutes, so the shell outer surface heats up rather slowly, i.e., from its initial temperature of 120 °F to its peak temperature of approximately 266 °F with a time constant of 4 minutes. In contrast, the time constant for heating up the air in the PCS air flow path is approximately 40 seconds for a shell temperature of 130 °F, and decreases with increasing shell temperature. With a time constant significantly less than that of the shell, the external PCS can be modeled as quasi-steady. A quasi-steady model of the AP600 external PCS was developed and used in the PCS scaling analysis[1].

<u>AP600 Operating Map</u>

The model of the external PCS developed for the scaling analysis was used to calculate the dry shell heat removal and PCS air flow rates with the external containment temperature treated as a parameter. The resulting riser Grashof and Reynolds numbers (based on the riser hydraulic diameter) are presented in Figure 2 for a parametric shell-to-environment temperature difference range of 0.5 to 200 °F. For a given difference between the shell and environment temperatures, the energy and momentum equation solutions provide a bulk flow rate and temperature from which the corresponding riser Reynolds and Grashof numbers can be obtained. The relationship between the Grashof and Reynolds numbers corresponding to each shell temperature is shown in Figure 3. This relationship defines an operating map for the dry operation of the AP600 PCS up to a maximum external shell surface temperature of approximately 266 °F (or shell-to-environment temperature difference of 146 °F). Although the PCS may start up from a shell-to-environment temperature difference less than 0.5 °F, such low temperatures are of little practical interest, since they imply that internal containment pressures are very low. In practice, the total heat removed from the shell prior to wetting is quite small[1], so the behavior of the PCS prior to wetting has no significant effect on containment pressure.

After the external liquid film develops, the buoyancy induced by the sensible heating of the riser and the additional buoyancy of the light water vapor maintain the high velocity riser air flow. Heat removal from the containment shell is primarily by evaporation of liquid water into the riser air. Smaller quantities of heat are also removed from the containment shell by convection to the cooler riser air and by radiation to the cooler baffle. Operating points at 2000, 4000 and 8000 sec. were taken from the scaling analysis, resulting in the wet operating points shown in Figure 3. Note that wet operation has nearly the same Grashof-Reynolds number relationship as dry operation.

The wet and dry AP600 PCS operating maps define the relationship between the riser Reynolds and Grashof numbers. The Reynolds, Grashof, and Prandtl numbers define the Nusselt numbers from which the convective heat transfer to the riser is determined. Because the mole fraction of steam in the riser is always less than approximately 10%, the Prandtl number can be simply approximated as that of dry air, with a value of 0.72 over the entire PCS operating range. Thus the dimensionless groups that characterize convective heat transfer to the riser are known over the entire PCS operating range.

Jackson and Hall Criterion

Jackson and Hall recommend a criterion for determining when the influence of buoyancy on the heat transfer coefficient is less than 5% of the forced convection value:

$$\frac{\overline{Gr}_b}{Re_b^{2.7}} < 10^{-5} \tag{1}$$

$\overline{Gr}_b$ is approximately equal to Gr/2. Using this relationship, the Jackson and Hall criterion is approximately:

$$\frac{Gr_b}{Re_b^{2.7}} < 2x10^{-5} \tag{2}$$

The values of $Gr/Re^{2.7}$ are plotted in Figure 4 over the full range of AP600 PCS operation. With a peak value of $Gr/Re^{2.7}$ equal to $2x10^{-5}$ at a Grashof number corresponding to 0.5 °F and values of $Gr/Re^{2.7}$ significantly less than $2.0x10^{-5}$ for wet operation and higher temperature dry operation, the Jackson and Hall criterion shows the influence of buoyancy on boundary layer heat transfer correlations is less than 5% over the full range of AP600 operation.

Westinghouse Correlation Comparison to Dittus-Boelter

The Jackson and Hall criterion indicates that heat transfer in the riser is dominated by forced convection. Consequently, the Westinghouse mixed convection correlation (Appendix 1) was compared to the well-known Dittus-Boelter forced convection correlation:

$$Nu_F = 0.023\,Re^{0.8}Pr^{0.4} \tag{3}$$

over the range of AP600 PCS operation. The results are presented in Table 1, and show that the Westinghouse mixed convection correlation Nusselt numbers are within 1% of the Dittus-Boelter Nusselt numbers for shell temperatures greater than 2 °F above the environment. The Westinghouse correlation, although a mixed convection correlation, predicts almost pure forced convection for the AP600 riser.

Alternate Correlation Comparison to Dittus Boelter

An alternate correlation that has been suggested for consideration[4] is the Cotton and Jackson equation for ascending flow in a uniformly heated vertical tube:

$$\frac{Nu}{Nu_F} = \left[1 - \frac{8x10^4 Gr^*}{Re^{3.425} Pr^{0.8}} \left(\frac{Nu}{Nu_F}\right)^{-2}\right]^{0.46} \tag{4}$$

The absolute value of the quantity in brackets is taken before raising to the 0.46 power.

The Grashof number based on wall heat flux, Gr^*, can be shown to be equal to the product GrNu, so Equation 4 becomes:

$$\frac{Nu}{Nu_F} = \left[1 - \frac{8x10^4 Gr Nu}{Re^{3.425} Pr^{0.8}} \left(\frac{Nu_F}{Nu}\right)^{2}\right]^{0.46} \tag{5}$$

The Cotton and Jackson correlation is compared to the Dittus-Boelter correlation in Table 1. The comparison shows the Cotton and Jackson correlation predicts Nusselt numbers within 5% of the Dittus-Boelter correlation for the AP600 riser with a shell temperature greater than 2 °F greater than the environment. The Table 1 comparisons show that in all cases of interest, heat transfer in the AP600 riser is well into the forced convection dominated region, regardless of whether the Cotton and Jackson correlation is used, or the Westinghouse mixed convection correlation is selected.

Conclusions

The Jackson and Hall criterion indicates that the riser operates predominantly in forced convection and consequently, buoyancy induced mixed convection effects are expected to be minor. The use of forced convection in the riser scaling analysis is justified by the dominance of forced convection.

The Cotton and Jackson correlation for upward buoyant mixed convection flow produces a Nusselt number for the AP600 riser that differs only a few percent from the Nusselt number for pure forced convection with the difference smaller at higher Reynolds numbers (higher shell-environment temperature differences).

The Westinghouse mixed convection correlation produces results nearly identical to the Dittus-Boelter correlation for pure forced convection. The Westinghouse mixed convection correlation is appropriate for the operating range of the PCS riser.

References

1. Letter, N. J. Liparulo (Westinghouse) to R. W. Borchardt (U.S. NRC), NTD-NRC-94-4318, October 27, 1994, "AP600 Passive Containment Cooling System Scaling Report", (WCAP-14190, Scaling Analysis for AP600 Passive Containment Cooling System (PCS), October, 1994).

2. J. D. Jackson and W. B. Hall, Influences of Buoyancy on Heat Transfer to Fluids Flowing in Vertical Tubes under Turbulent Conditions, *Turbulent Forced Convection in Channels and Bundles Theory and Application to Heat Exchangers and Nuclear Reactors, 2,* Advanced Study Institute Book (eds. S. Kakac, and D. B. Spalding), 1979, 613-640.

3. M. A. Cotton and J. D. Jackson, "Comparison Between Theory and Experiment for Turbulent Flow of Air in a Vertical Tube with Interaction," *Mixed Convection Heat Transfer -- 1987,* edited by V. Prasad, I. Catton, and P. Cheng, ASME, New York, 1987, HTD-Vol. 84, pp. 43-50.

4. Letter, T. J. Kenyon (US NRC) to N. J. Liparulo (Westinghouse), "Request for Additional Information on the AP600", Docket No. 52-003, April 29, 1994. Enclosure 2, "References Heat Transfer Correlations", from R. Viskanta.

Nomenclature

c_p	Constant pressure specific heat
d	Channel hydraulic diameter
g	Gravitational acceleration
h	Heat transfer coefficient
k	Thermal conductivity
q_w	Wall, or shell, heat flux
V	Bulk channel velocity
β	Coefficient of volume expansion = $1/T_{abs}$ for a perfect gas
ρ	Density
$\bar{\rho}$	The average density across the boundary layer defined:

$$\bar{\rho} = \frac{1}{(T_w - T_b)} \int_{T_w}^{T_b} \rho \, dT \qquad (6)$$

ν	Kinematic viscosity
μ	Dynamic viscosity

Subscripts

abs	The absolute temperature
b	The bulk flow value
F	Forced convection
w	The surface value

Dimensionless Groups

Grashof Number	$Gr = gd^3(\rho_w - \rho_b)/\rho_b \nu^2$
	$Gr^* = g\beta d^4 q/k\nu^2$
	$\bar{Gr} = gd^3(\bar{\rho} - \rho_b)/\rho_b \nu^2$
Nusselt Number	$Nu = hd/k$
Prandtl Number	$\mu c_p/k$
Reynolds Number	$Re = Vd/\nu$
Richardson Number	$Ri = Gr/Re^2 = gd(\rho_w - \rho_b)/V^2\rho$

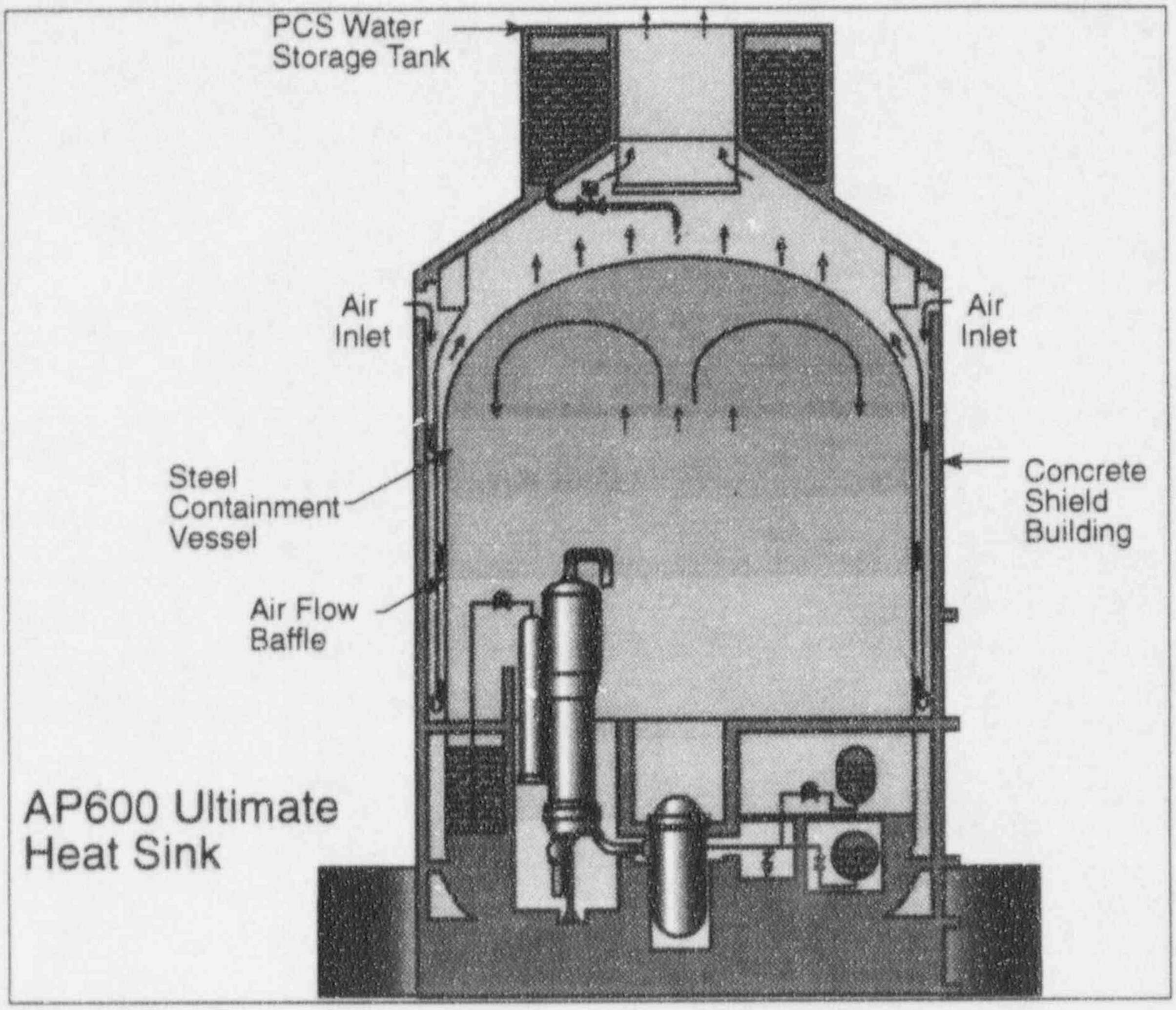

Figure 1 AP600 Passive Containment Cooling System Design Schematic

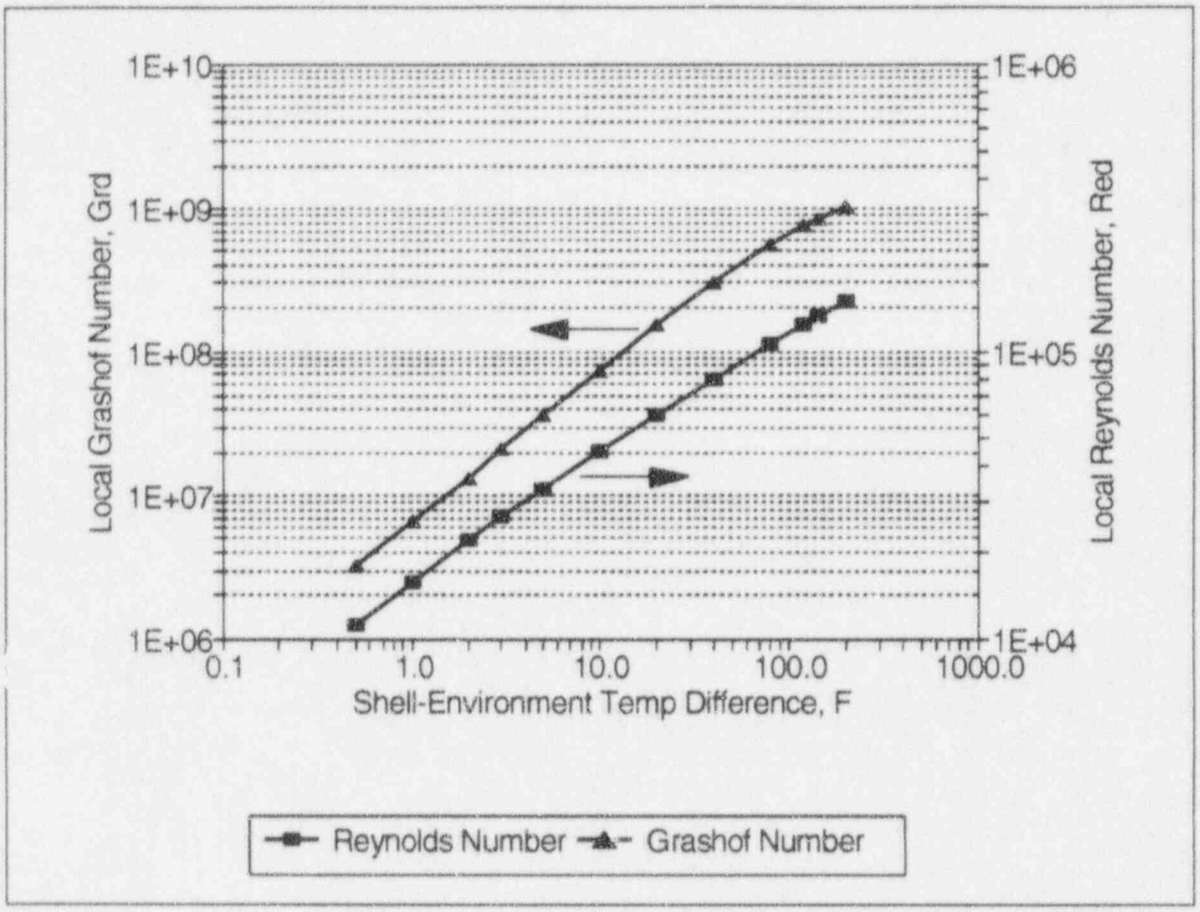

Figure 2 Dry PCS Operation with Shell-to-Environment Temperature Difference as a Parameter

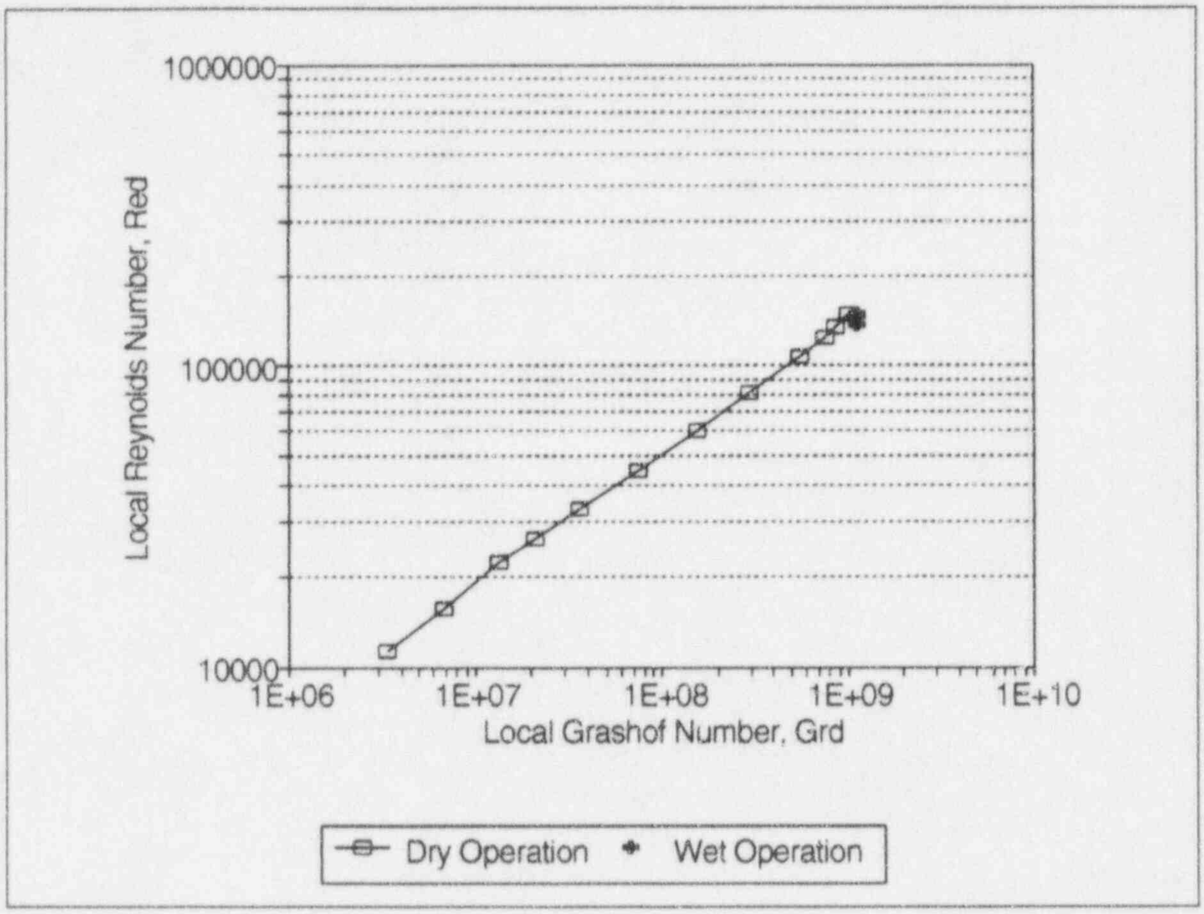

Figure 3 Parametric Relationship between PCS Air Flow Path Reynolds and Grashof Numbers

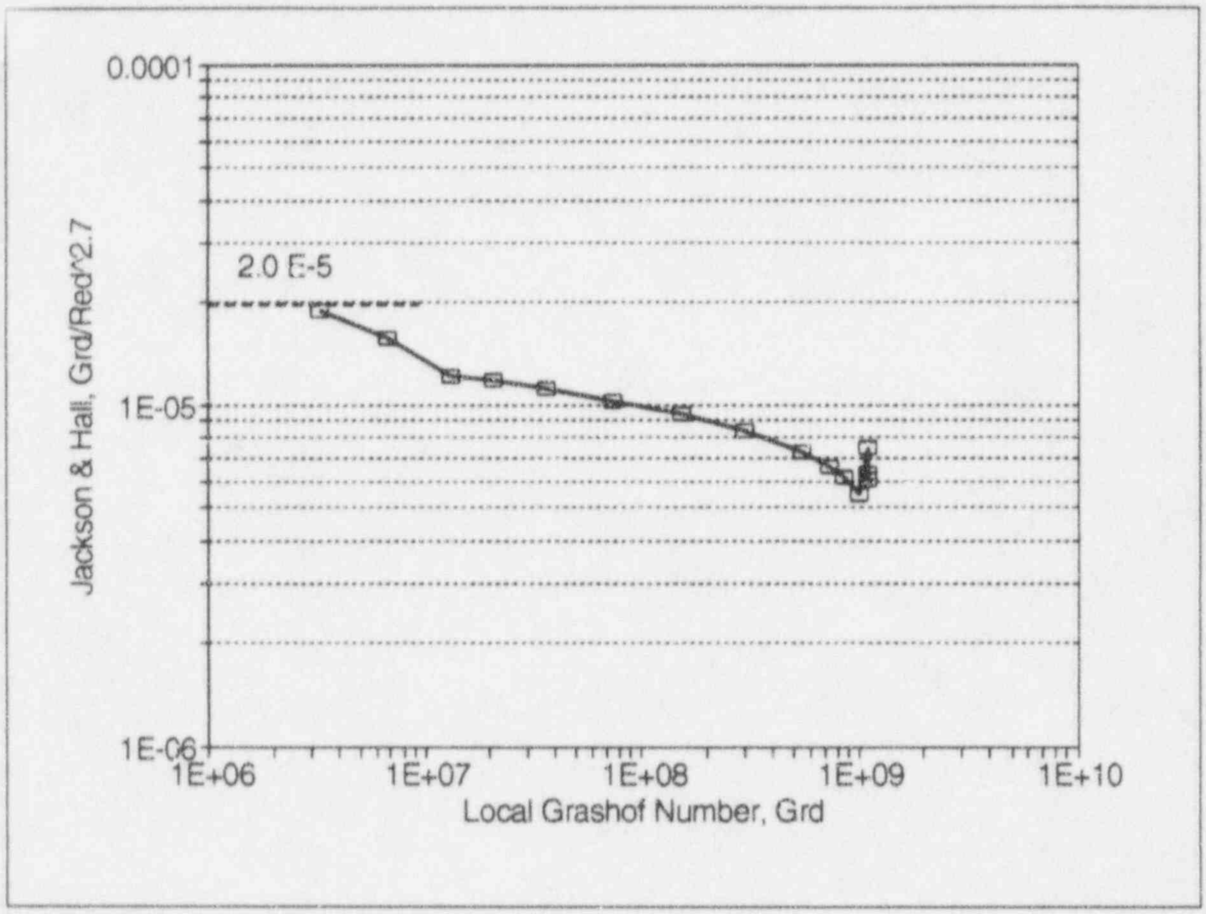

Figure 4 AP600 Riser Richardson Number and Comparison to Jackson and Hall Criterion for Significant Buoyant Effects

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Table 1
Parametric Characterization of Heat Transfer in AP600 PCS Riser

ΔT (°F)	Gr_d	Re_d	Jackson & Hall $Gr/Re^{2.7}$	Nusselt Numbers			Nusselt Ratios	
				Dittus-Boelter	Cotton & Jackson	Westing-house	West/D-B	C&J/D-B
0.5	3.29e+06	11232	1.91e-05	35.08	32.09	34.44	0.982	0.915
1.0	6.69e+06	15683	1.57e-05	45.81	42.59	45.15	0.985	0.930
2.0	1.32e+07	22187	1.22e-05	60.47	57.18	59.91	0.991	0.946
3.0	2.09e+07	26579	1.18e-05	69.87	66.13	69.16	0.990	0.947
5.0	3.60e+07	33156	1.12e-05	83.39	79.11	82.51	0.989	0.949
10.0	7.48e+07	44839	1.03e-05	106.17	101.06	105.02	0.989	0.952
20.0	1.53e+08	60372	9.45e-06	134.69	128.66	133.22	0.989	0.955
40.0	2.98e+08	80817	8.37e-06	170.08	163.24	168.41	0.990	0.960
80.0	5.49e+08	106629	7.30e-06	212.31	204.76	210.58	0.992	0.964
120.0	7.46e+08	124497	6.53e-06	240.32	232.63	238.83	0.994	0.968
146.0	8.50e+08	133768	6.13e-06	254.53	246.87	253.25	0.995	0.970
200.0	1.01e+09	149287	5.41e-06	277.89	270.49	277.15	0.997	0.973
56.5	1.12e+09	137655	7.47e-06	260.43	250.75	257.47	0.989	0.963
66.8	1.12e+09	146119	6.36e-06	273.17	264.56	271.29	0.993	0.968
69.8	1.11e+09	148597	6.02e-06	276.87	268.61	275.35	0.995	0.970

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Appendix 1: Westinghouse Convective Heat Transfer Correlations

1.1 Outside Containment

The McAdams[1] turbulent free convection heat transfer correlation:

$$Nu_{d,free} = 0.13(Gr_d Pr)^{1/3} \tag{1}$$

and the Colburn[2] turbulent forced convection heat transfer correlation:

$$Nu_{d,forc} = 0.023 Re_d^{4/5} Pr^{1/3} \tag{2}$$

are used for the external convective heat transfer to or from the surfaces.

1.2 Inside Containment

The McAdams turbulent free convection heat transfer correlation:

$$Nu_{x,free} = 0.13(Gr_x Pr)^{1/3} \tag{3}$$

and the smooth flat plate[3] turbulent forced convection heat transfer correlation:

$$Nu_{x,forc} = 0.0296 Re_x^{4/5} Pr^{1/3} \tag{4}$$

are used for the internal convective heat transfer to or from the surfaces.

1.3 Combined Free and Forced Convection

The correlations for combined free and forced convection heat transfer from Churchill[4] are, for turbulent opposed free and forced convection:

$$Nu = (Nu_{free}^3 + Nu_{forc}^3)^{1/3} \tag{5}$$

and for assisting free and forced convection, Nu is the larger of the following three expressions:

$$abs(Nu_{free}^3 - N_{forc}^3)^{1/3} \quad ; \quad Nu_{free} \quad ; \quad 0.75 Nu_{forc} \tag{6}$$

The lower limit in the latter equation, that prevents the value of Nu from going to zero when Nu_{free} and Nu_{forc} are equal, comes from Eckert and Diaguila[5].

References

1. W. H. McAdams, *Heat Transmission*, Third Edition, McGraw-Hill, 1954.

2. A. P. Colburn, "A Method of Correlating Forced Convection Heat Transfer Data and a Comparison With Fluid Friction", *Transactions of the AIChE*, Vol. 29 (1933), p. 174.

3. H. Schlichting, *Boundary Layer Theory*, Sixth Edition, McGraw-Hill.

4. S. W. Churchill, "Combined free and Forced Convection Around Immersed Bodies", Section 2.5.9, and "Combined Free and Forced Convection in Channels", Section 2.5.10 in E. U . Schlunder, Ed.-in-Chief, *Heat Exchanger Design Handbook*, Hemisphere Publishing Corp. 1983.

5. E. R. G. Eckert and A. J. Diaguila, "Convective Heat Transfer for Mixed, Free, and Forced Flow Through Tubes", *Transactions of the ASME*, May, 1954, pp 497-504.