1 of 1

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Page 1

**FLOW AND HEAT TRANSFER IN
VERTICAL ANNULI**

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INTRODUCTION

In shell-side boiling heat exchangers narrow crevices that are formed between the tubes and the tube support structure provide areas for local conditions which differ significantly from the bulk fluid conditions. A quasi-two-dimensional model which was developed to describe the local phenomena in a vertical, cylindrical crevice was described previously¹. The present work compares experimentally obtained flow and tube temperature distributions to those predicted by the model. The results confirm the characteristic "W" shape² of the tube temperature profile centered at the line contact between a heated tube and tube support.

ANALYTICAL MODEL

The flow field in an eccentric annular crevice can be approximated by a circumferential summation of flows in discrete vertical strips. In these strips the flow is characterized by a succession of isolated bubble, "coalesced" bubble, churn-annular flow regimes and dryout from inlet at the bottom to exit at the top. The model neglects the two-dimensional effects which may be important in cases where the axial velocities are relatively low, but were not significant for the tests evaluated in this report.

Each flow regime and transition from one regime to another has been described analytically. The most significant aspect of the analysis is a Lagrangian formulation of the "coalesced" bubble motion. In this regime the heat transfer from the tube surface to the liquid is obtained using the empirical correlation recommended by Ishibashi and Nishikawa³.

$$h_{CB} = \frac{k_l}{\delta} 200 \left(\frac{\bar{D}}{F\delta^2} \right)^{-2/3} P_r^{-2/3} \left(\frac{\rho_l}{\rho_g} \right)^{-1/2} \quad (1)$$

and

$$F = 1.365 \times 10^{-9} q''_{CB} \delta^{-3/2} P_r^{1.627} \left(\frac{\rho_l}{\rho_g} \right)^{1.085} \quad (2)$$

where	h_{CB}	=	boiling heat transfer coefficient
	k_l	=	liquid thermal conductivity
	δ	=	crevice radial gap size
	$\bar{D}$	=	thermal diffusion constant in water
	P_r	=	liquid Prandtl number
	ρ_l	=	liquid density
	ρ_g	=	vapor density

In Equation (2), the heat flux q''_{CB} must be in kcal/hr-m₂.

A detailed description of the analytical model for the four flow regimes was given in Reference (1). The model consists of a set of first order differential equations along with the relevant constitutive relationships. The resulting equations can be integrated with the appropriate boundary conditions. In this case, the pressure drop and the inlet void fraction have been assumed to be given.

The flow rate in each strip is calculated with an assumed initial inlet velocity which is iterated until the calculated pressure drop agrees with the given value.

COMPARISON TO DATA

An experimental program was undertaken to measure pressure drops, water flow rates and tube wall temperatures in crevices with 1.27 cm (0.5 inch) axial length. A total of 262 tests were performed with diametral gap sizes from 0.127 mm (0.005 inch) to 0.762 mm (0.030 inch), pressures from 2068 kPa (300 psia) to 4137 kPa (600 psia), water flow rates from 1.4 kg/hr (3.0 lbm/hr) to 36.3 kg/hr (80.0 lbm/hr), heat fluxes from 78,800 W/m² (25,000 Btu/hr-ft²) to 299,400 W/m² (95,000 Btu/hr-ft²) and inlet conditions from subcooled to 95% void fraction. Of these, 85 were in concentric geometries with the remainder having the tube in line contact with a quartz support. Figure 1 shows predicted versus measured flow rates. The best fit line (not shown) has a slope of 0.99. The corresponding correlation coefficient is 0.92. The largest errors occurred when low flow rates were combined with low or zero heat flux in the larger gap size (0.762 mm). In these cases the crevice pressure drops were as low as 0.2 kPa (0.03 psi) which were reduced from the pressure readings 1.9 cm (0.75 inch) below the crevice inlet and 7.0 cm (2.75 inch) above the crevice exit. Therefore, the sources of some of the errors in these cases are likely to be experimental.

The tube wall temperatures were measured with 12 thermocouples imbedded in 0.56 mm (0.022 inch) grooves on the outer tube surface, 6 of which were at the mid-plane. These thermocouples were "bucked" to a reference thermocouple placed 2.54 cm (1.0 inch) above the crevice exit. To further reduce the experimental errors, "in-place-calibrations" were performed at various heat fluxes without the support plate.

Figure 2 is an example of comparisons between the predicted and measured temperatures at the mid-plane. It shows the effects of heat flux. Note the characteristic "W" shape² of the temperature profile. Temperatures above the horizontal line indicate degraded heat transfer to mostly vapor flow. The data reflects the unaccounted effects of heat conduction to the

support plate. Temperatures below the line indicate enhanced heat transfer due to "coalesced" bubbles.

The predicted temperatures closely follow the trends observed in the tests.

CONCLUSIONS

A quasi-two-dimensional model of vertical crevice thermal-hydraulics has been developed and compared with test data. Some differences between the predictions and the experimental data do exist. These differences are in part due to experimental technique, the instrumentation, and data reduction. In general, the heat transfer correlation of Ishibashi and Nishikawa for the "coalesced" bubble regime represent the experimental data quite well.

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Measured Flow Rate, Kg/Hr

FIGURE 1 - Crevice Flow Rates

Diam. Clear. = 0.4 mm, Pres. = 2760 KPa, Flow Rate = 13.6 Kg/Hr

FIGURE 2 - Wall Temp. Variation with Heat Flux

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