(i.e., the efficiency of the algorithm), is found to depend on this choice. The effects of adaptively modifying ε_1 and ε_{2a} are shown in Table 2. Note C_3 provides a basis for selectively testing the elements to split. Without C_3 the computation either stops before meeting the convergence criterion, or inefficiently splits every element in order to continue. Figure 3 displays a typical refinement history, showing clearly the adaptive integration of two surfaces meeting at 60 deg.

Conclusions

An effective adaptive algorithm for implementing mesh refinement or increasing quadrature in the computation of radiation configuration factors is demonstrated. While the adaptive technique functions well in simple geometries—avoiding oversplitting the domain and undersampling the integral—the results indicate the need for developing improved adaptive algorithms for finding more efficient refinement paths.

Combining h- and p-refinement is strongly recommended for complex, curved geometries, especially since h-refinement effectively captures the singularity in the integrand, while p-refinement models the trigonometric terms well.

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Combined Radiation-Convection Heat Transfer in a Pipe

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Introduction

DURING the refill and reflood periods of a postulated loss of coolant accident (LOCA), U.S. federal regulations require that for reflood rates of less than 1 in./s, clad-to-coolant heat transfer should be based on single-phase steam.

For typical reflood conditions where the fuel cladding temperature may reach 1400 K with coolant pressure in the range of 1.7–5.1 atm, the radiant heat transfer from fuel cladding to steam may become important. In a reactor safety analysis, the effect of radiation to steam is traditionally treated by the superposition model which assumes that the total Nusselt number equals the sum of the radiation and the pure convective Nusselt numbers.

The radiative and convective heat transfer are interdependent since both contain temperature-dependent material properties in such a way that a change in the heat flux due to one component causes a change in the heat flux due to the other. Therefore, the radiant energy transport should be coupled to the convection energy transports and solved simultaneously when the radiant energy becomes important. The combined radiation and convection problem is very difficult to solve due to the highly nonlinear nature of radiation transfer. In the literature, various simplifying assumptions were made depending upon the application.[1-3] Recently, Kim and Viskanta[4] studied the interaction of convection and radiation heat transfer in high pressure and temperature steam. They found that the convective Nusselt number was significantly reduced when compared with the Nusselt number of pure convection.

This study considers the effect of radiation to steam under the reflood steam cooling condition. The calculation results compared very well with the test data obtained from the steam cooling experiments performed by Larsen and Lord[5] where the pressure, power, flow rates, and the hydraulic diameter of test section were chosen to simulate the pressurized water reactor (PWR) LOCA conditions.

Analysis

The energy equations of an axisymmetric pipe flow can be written as

$$\rho C_p\left(u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial r}\right) = \frac{1}{r}\frac{\partial}{\partial r}\left[r(k + k_t)\frac{\partial T}{\partial r}\right] + u\frac{\mathrm{d}P}{\mathrm{d}x} + (\mu + \mu_t)\left(\frac{\partial u}{\partial r}\right)^2 - \operatorname{div} q''_R \tag{1}$$

where q''_R is the radiative heat flux to the fluid element.

Before Eq. (1) can be solved, the turbulent viscosity μ_t, turbulent conductivity k_t and q''_R must be expressed in terms of the existing variables. Van Driest's mixing length hypothesis[6] is used for μ_t, and Cebeci's model[7] is used to express the turbulent conductivity.

Landram et al.[2] proposed

$$-\operatorname{div} q''_R = 4K_p(T_w)\frac{T_w}{T}E_b(T_w) - 4K_p(T)E_b(T) \tag{2}$$

The Planck mean absorption coefficient K_p for steam shown in Eq. (2) is obtained from two sources. First, the exponential wide-band model developed by Edward[8] is used. Edward expressed the mean steam absorptivity by

$$\alpha_g = \frac{1}{\alpha T^4}\sum_i^m B(\nu_i, T_w)A_i \tag{3}$$

where B is the blackbody radiosity, ν is the wave number, A_i is the total absorptance of ith band, and m is the total band band number with $m = 5$ for steam. The Edward's absorption coefficient and absorptivity is referred to Beer's Law as $\alpha_g = 1 - \exp(-K_pD)$. The mean absorption coefficient can also be obtained from

$$K_p = P\left[5.6\left(\frac{1000}{T}\right)^2 - 0.3\left(\frac{1000}{T}\right)^4\right] \tag{4}$$

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Equation (4) was derived from the curve presented by Abu-Romia and Tien (A-T).[9]

In the experimental setup there is a fairly long unheated entry length of about 45 diam. Measurements from the first thermocouple, which is located within the heated section, were used to define the initial conditions. The initial condition for fluid entering the circular pipe is therefore assumed to be a uniform temperature T_i and a developed velocity profile; the boundary conditions are obtained from the measured wall temperature.

In predicting the hydrodynamic and heat characteristics of pipe flows the Patankar-Spalding (P-S) computer code[10] has demonstrated its accuracy on several occasions such as the results shown in Ref. 10. In this study the P-S code[10] was used as a base code and then modified to include the above-mentioned turbulence conductivity and the radiation model. During the forward-marching calculation, the previous step-calculated fluid temperatures are used to evaluate the current step's radiating heat transfer. This avoids the complication due to the nonlinear radiative heat transfer. No further iteration for radiative heat transfer by the current step's fluid temperatures is necessary. This is because there is no heat source in the fluid, the extreme temperature exists at the wall boundary, and there is no abrupt temperature change on the wall boundary.

Results and Discussion

The present calculations simulate several chosen test cases which were reported to have overall good flow stability and heat balance in Larsen and Lord's experiment.[5] Before the simulations were made for steam-cooled tests, the prediction for a nitrogen test case was performed. Since nitrogen is transparent to radiation (i.e., it does not absorb or emit the radiant energy), the heat transfer between the pipe wall and the fluid is purely convective. The predicted gas temperatures agree closely with the reported gas temperatures. This simulation confirms that the P-S code[10] is accurate in the absence of radiation.

Three steam-cooled test cases with varying system pressure (1.7, 3.4, and 5.1 atm, respectively) were then simulated. In each case three different calculations were performed. The radiation model was bypassed in the first calculation to obtain the pure convection results. The combined radiation-convection problems with the absorption coefficients of Edward[8] and A-T[9] were respectively solved in the second and third calculations.

The predicted bulk steam temperatures are compared with data. The calculations with A-T absorption coefficient[9] agree well with data. With Edward's absorption coefficient[8] the steam temperatures are consistently underpredicted. Fig. 1 shows

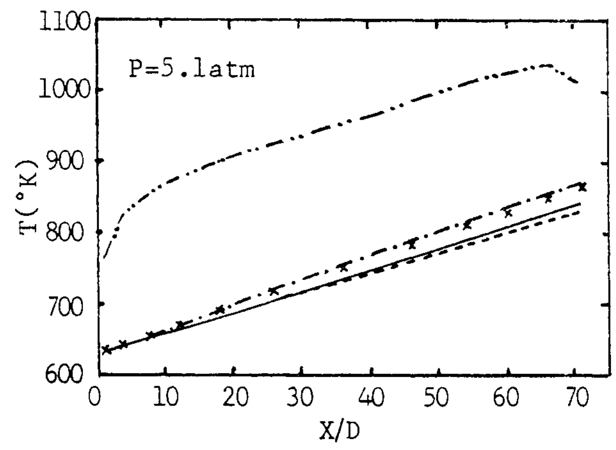

Fig. 1 Comparison of steam bulk temperature: x data, ----- pure convection (no radiation model), —— radiation model with Edward's absorption coefficient,[8] -·-·-· radiation model with A-T absorption coefficient,[9] ·-·-·-·· measured wall temperature.

Table 1 Nusselt number ratio

	X/D	1	10	20	40	60
(1)[a]	A	3.107	1.140	1.114	1.130	1.147
	B	3.011	1.046	1.018	1.029	1.042
	C	3.010	1.044	1.014	1.022	1.033
(2)[a]	A	3.069	1.087	1.106	1.157	1.206
	B	2.949	0.950	0.958	0.993	1.027
	C	2.901	0.928	0.939	0.979	1.019
(3)[a]	A	3.087	1.210	1.198	1.235	1.272
	B	2.870	0.988	0.961	0.986	1.006
	C	2.876	0.988	0.969	0.987	1.008

[a](1) p = 1.7 atm (2) p = 3.4 atm (3) p = 5.1 atm.

the comparison for the case of test 58 (P = 5.1 atm). Therefore, the absorption coefficient from A-T is a better choice for our interesting pressure range.

Table 1 depicts the ratios of the calculated Nusselt numbers to the well-known relation

$$Nu_{ST} = 0.023 Re^{0.8} P_r^{1/3} \left(\frac{\mu_b}{\mu_w}\right)^{0.14} \tag{5}$$

Equation (5) was proposed by Sieder and Tate[11] for a fully developed turbulent flow with the consideration of variable viscosity due to T_w.

In Table 1, row C represents Nu_{NR}/Nu_{ST}, where Nu_{NR} is the calculated Nusselt number when the radiation model is turned off and Nu_{ST} is defined in Eq. (5). When the radiation model is turned on and A-T absorption coefficient is used in the radiation model, the calculated total and convective Nusselt numbers are presented at row A and row B, row A is Nu_T/Nu_{ST} and row B stands for Nu_c/Nu_{ST}, where $Nu_T = q''_{\text{total}} \cdot D/[k \cdot (T_w - T_b)]$, and $Nu_c = (q''_{\text{total}} - q''_R) \cdot D/[k \cdot (T_w - T_b)]$.

The radiant energy in the combined problem will raise the steam temperature and reduce the temperature difference between wall and steam. The higher steam temperature in turn will impede the convection heat flux from the boundary. Since the Nusselt number is proportional to the heat flux and inversely proportional to the temperature difference, the convective Nusselt number may be either larger or smaller due to the effect of radiative heat transfer to the steam. In the combined problem calculation the convective Nusselt numbers only (row B of Table 1), compare closely with the values obtained from the pure convection problem (row C of Table 1). This means the decrease in the wall-steam temperature difference due to radiation approximately compensates for the decrease in convective heat flux. Assuming the total Nusselt number equals the sum of the pure convective Nusselt number plus the radiative Nusselt number indicates that the traditional superposition model is applicable to steam cooling under reflood conditions (steam pressure from 1.7 to 5.1 atm). This is different from what was found by Kim and Viskanta.[4] A possible reason for this is that the system pressure used in their analysis cases was 68 atm, which is much higher than the present cases.

Conclusions

A one-dimensional radiative heat flux model is included in the energy transport equation, which is solved simultaneously with other conservative equations. The results show 1) the predictions of the present combined problem with A-T absorption coefficient[9] agree fairly well with data where the bulk steam temperature is increased from 6.5 to 18.3%, and radiation heat transfer contributes up to 27% of the heat transfer; and 2) the convective Nusselt number in the present combined problem agrees well to the Nusselt number obtained in the pure convection calculation. This result suggests that the traditional superposition model is applicable for reflood analysis.

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Effects of Baffle Length on the Performance of Pipe Insulation

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Nomenclature

D = gap width, $r_0 - r_i$
g = acceleration due to gravity
H = length of a baffle
h = average heat transfer coefficient
K = permeability
k = effective thermal conductivity of porous medium
k_b = thermal conductivity of baffle
Nu = average Nusselt number, hD/k
R = dimensionless radial distance, r/D
Ra = Rayleigh number, $Kg\beta(T_i - T_0)D/\alpha\nu$
r = radial distance
T = temperature
T_m = mean temperature, $(T_i + T_0)/2$
V_r = dimensionless velocity in the r direction, $V_r = 1/R(\partial\psi/\partial\theta)$
V_θ = dimensionless velocity in the θ-direction, $V_\theta = -\partial\psi/\partial R$
w = dimensionless thickness of baffle
α = thermal diffusivity of porous medium
β = coefficient of thermal expansion
γ = orientation of the referenced baffle with respect to the vertical diameter
θ = angular coordinate
ν = kinematic viscosity
τ = dimensionless time
ψ = stream function
$\odot$ = dimensionless temperature, $(T - T_m)/(T_i - T_0)$

Introduction

It is well known that natural convection within insulation accounts for the major loss of useful energy. To suppress this nonbeneficial convective transport has become one of the most important considerations in process design. Recently, it has been shown that using internal partitions can effectively reduce convective heat losses.[1] The authors have successfully demonstrated the feasibility of using radial baffles to conserve energy[2] for pipes. In addition, they have shown that partial baffles are generally more effective than full baffles. However, the dependence of heat transfer on the baffle length has not been examined. Therefore, it is the purpose of the present study to investigate if there exists an optimal baffle length such that it would reduce the heat loss to a minimum.

Formulation and Numerical Method

For most pipe insulation available today it is adequate to model them as a porous annulus (Fig. 1). For a typical application, the inner cylinder is heated at a constant temperature T_i while the outer cylinder is maintained at the ambient temperature T_0 ($T_i > T_0$). Having invoked the Boussinesq approximation, the dimensionless governing equations based on Darcy's law are given by

$$\frac{\partial V_\theta}{\partial R} + \frac{V_\theta}{R} - \frac{1}{R}\frac{\partial V_r}{\partial \theta} = Ra\left(\cos\theta\frac{\partial \odot}{\partial R} - \frac{\sin\theta}{R}\frac{\partial \odot}{\partial \theta}\right) \quad (1)$$

$$V_r\frac{\partial \odot}{\partial R} + \frac{V_\theta}{R}\frac{\partial \odot}{\partial \theta} = \frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial \odot}{\partial R}\right) + \frac{1}{R^2}\frac{\partial^2 \odot}{\partial \theta^2} \quad (2)$$

with the boundary conditions given by

$$\odot = \tfrac{1}{2},\ V_r = 0, \quad \text{on the inner wall} \quad (3a)$$

$$\odot = -\tfrac{1}{2},\ V_r = 0, \quad \text{on the outer wall} \quad (3b)$$

As for the baffles, they are assumed to be very thin such that the angular temperature gradient is negligible. There-

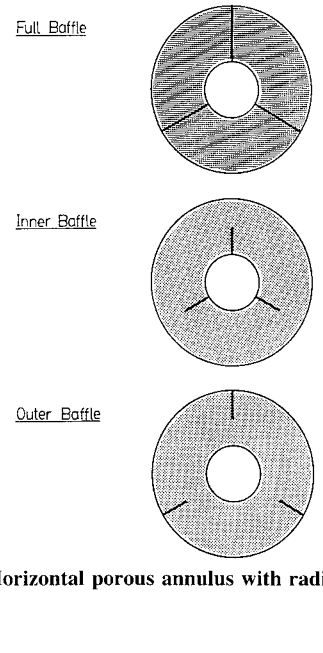

Fig. 1 Horizontal porous annulus with radial baffles.

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