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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, $(T - T_m)/(T_i - T_0)$

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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

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

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

with the boundary conditions given by

$$\odot = \tfrac{1}{2},\ V_r = 0, \qquad \text{on the inner wall} \tag{3a}$$

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

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

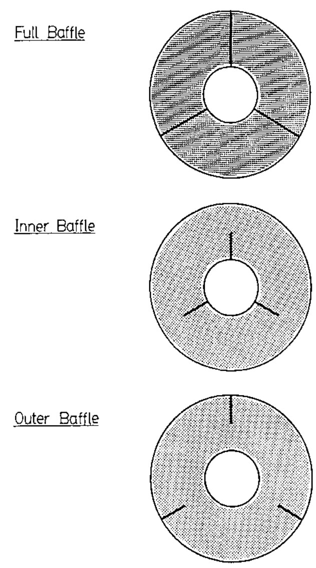

Fig. 1 Horizontal porous annulus with radial baffles.

fore, the appropriate boundary conditions for the baffles are given by

$$V_n = 0 \tag{3c}$$

$$k_b w \frac{\partial^2 \Theta}{\partial R^2} = \frac{k}{R}\left(\frac{\partial \Theta}{\partial \theta}\bigg|_1 - \frac{\partial \Theta}{\partial \theta}\bigg|_2\right) \tag{3d}$$

where subscripts 1 and 2 refer to the two sides of the baffle. Equation (3d) has been derived by performing an energy balance for the baffle. This boundary condition has also been employed by Kwon et al.[3] in a similar study.

To solve the simultaneous equations defined above, the following coordinate transformation is introduced such that the computational domain is mapped onto a rectangular geometry to facilitate calculations:

$$x = \frac{\ln R - \ln R_i}{\ln R_o - \ln R_i} - \frac{1}{2}, \qquad y = \frac{\theta - \theta_i}{\theta_o - \theta_i} \tag{4}$$

Thus, the governing equations are transformed to

$$x_0^2 \frac{\partial^2 \psi}{\partial x^2} + y_0^2 \frac{\partial^2 \psi}{\partial y^2} = -Ra(R_i R_o)^{1/2}\left(\frac{R_o}{R_i}\right)^x \cdot \left[x_0 \frac{\partial \Theta}{\partial x}\cos\left(\frac{y}{y_0} + \theta_m\right) - y_0 \frac{\partial \Theta}{\partial y}\sin\left(\frac{y}{y_0} + \theta_m\right)\right] \tag{5}$$

$$x_0 y_0\left(\frac{\partial \psi}{\partial y}\frac{\partial \Theta}{\partial x} - \frac{\partial \psi}{\partial x}\frac{\partial \Theta}{\partial y}\right) = x_0^2 \frac{\partial^2 \Theta}{\partial x^2} + y_0^2 \frac{\partial^2 \Theta}{\partial y^2} \tag{6}$$

where $x_0 = 1/\ln(R_o/R_i)$, $y_0 = 1/(2\pi)$, $\theta_o = \pi$, $\theta_i = -\pi$, and $\theta_m = (\theta_o + \theta_i)/2$.

The transformed governing equations and boundary conditions are solved numerically by employing a finite-difference method which has been successfully used by the authors. The details of the numerical scheme are omitted here for brevity and may be found in Refs. 4–6. Uniform grids, 41 × 181 in the transformed domain, are used for the present study. It should be noted that further grid refinement did not produce any significant improvement in the calculated Nusselt numbers. As an additional check on the accuracy of the computational results an overall energy balance has been performed after each calculation. For the present study the energy balance is satisfied within 3%. To validate the numerical results thus obtained the solutions have been compared with those reported in the literature for the case of an annulus with no baffles. The agreement is very good as reported in Ref. 2.

Since the emphasis of the present study is placed on the effects of baffle length on the heat transfer results, computations are restricted to the case of $r_i = 1$ and $D = 2$. For simplicity, it is further assumed that the thermal conductivity of baffles is the same as that of the pipe insulation. To effectively reduce the convective heat loss it is clear that the baffle must have a thermal conductivity smaller than that of the pipe insulation. Otherwise, the heat loss by conduction along the baffles may outweigh the convection loss. Therefore, under this assumption the results thus obtained form an upper bound of the problem.

Three different sizes of baffle have been considered, i.e., $H/D = \frac{1}{4}, \frac{1}{2}$, and $\frac{3}{4}$. Numerical results have covered a range of Rayleigh numbers of practical interest (i.e., $50 \leq Ra \leq 500$). For a given Rayleigh number, calculations have been performed to obtain results for various baffle orientations ($\gamma \leq 120$ deg).

Results and Discussion

Due to the page limit the flow and temperature fields from the computations are not documented here but are available in Ref. 7. When the baffles are short (i.e., $H/D = \frac{1}{4}$), the flow and temperature fields differ from those with no baffles only at the regions near the baffles where the streamlines and isotherms are slightly perturbed. As the baffle length increases more convective flow is blocked. As a result, the flow and temperature fields are dramatically changed. For the case of $H/D = \frac{3}{4}$, the flow and temperature fields resemble those with full baffles.

For an annulus with outer baffles the flow patterns are very different from those of inner baffles. The dissimilarity is a result of the difference in basic functions between inner baffles and outer baffles. As the inner baffles are used to block the heated fluid which is on the rise to the top, the outer baffles are used to barricade the cooled fluid which is on the return to the bottom. Because of this fundamental difference the heat transfer results also vary.

When the Rayleigh number is large (e.g., $Ra = 500$), it has been reported[2] that the flow and temperature fields at some orientation angles begin to oscillate. This unstable phenomenon has also been observed here for annuli with outer baffles. A transient analysis has been performed for these special cases to investigate the nature of oscillations. The transient analysis employed here is essentially the same as that reported in Ref. 6, with the exception that a smaller time step ($\Delta\tau = 10^{-5}$) is required for the present case due to the inherently more unstable nature. The variation of flowfield with time is shown in Fig. 2 for an annulus with three outer baffles at $\gamma = 30$ deg. As observed, the flow instability is primarily localized on the region directly above the inner cylinder. In comparison to the case of full baffles[2] it is interesting to note that the flowfields for these two cases have a similar profile, especially for the evolution of secondary flows. However, the flow transition for the present case falls behind those of full baffles.

It is well-known that natural convection at a high Rayleigh number may either lead to a steady multicellular flow or undergo a transition from steady-state to time-dependent flow.[8,9] It is also recognized that the mechanism which triggers the flow instability is the heat dissipated rate.[10,11] If the heat generated could not be dissipated fast enough, the disturbances resulted from the excess heat would initiate a periodic flow. For an annulus with no baffles, the flowfield can reach a steady state at $Ra = 500$ since the heat generated by the inner cylinder can be effectively removed from the outer cylinder. When inserted with inner baffles, the flowfield can still reach a steady state at $Ra = 500$ because the cooled fluid is not totally blocked by the baffles and heat can be dissipated through much of the surface of the outer cylinder. On the other hand, for an annulus with outer baffles (especially for the case of $H/D = \frac{3}{4}$), the cooled fluid is almost entirely blocked by the baffles and heat can only be dissipated through a limited

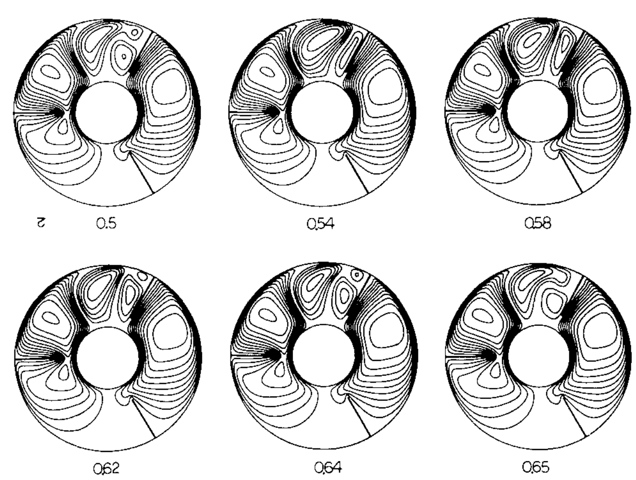

Fig. 2 Oscillatory flowfields for an annulus with three outer baffles of $H = 3D/4$ ($\Delta\psi = 2$).

surface directly above the inner cylinder. As a result, an oscillatory flow is initiated as shown in Fig. 2.

The result of practical interest in many applications is the heat transfer coefficient. The heat transfer coefficient in terms of the Nusselt number is given by

$$Nu = \frac{hD}{k} = -\frac{1}{R_i \ln (R_o/R_i)} \int_{-1/2}^{1/2} \frac{\partial \Theta}{\partial x}\bigg|_{x=-1/2} \mathrm{d}y \tag{7}$$

However, the heat transfer results are most informative if the Nusselt number thus obtained is normalized by the conduction value. It is easy to show that the conduction Nusselt number can be evaluated as

$$Nu_{\mathrm{cond}} = \frac{1}{R_i \ln (R_o/R_i)} \tag{8}$$

The normalized Nusselt numbers are plotted in Figs. 3 and 4 for inner and outer baffles, repectively. From these figures, it is confirmed once again that radial baffles are effective in reducing heat losses from a pipe insulation.

When the Rayleigh number is small (e.g., $Ra = 50$), the convective flow is weak. No secondary flow is formed in the annulus such that the normalized Nusselt number is a smooth continuous function of the orientation angle. As the Rayleigh number increases, the buoyancy-induced flow becomes strong and the flowfield is further complicated by the baffles. As a result, secondary flows appear in the annulus and the flowfield is forced to undergo a transition. It has been reported[2] that a transition in the flow structure will also result in a change in the heat transfer characteristics. This is exactly what is observed in Figs. 3 and 4. For the oscillatory flows, the averaged Nusselt number along with its extreme values are plotted in Figs. 3d and 4d. It is interesting to observe that the

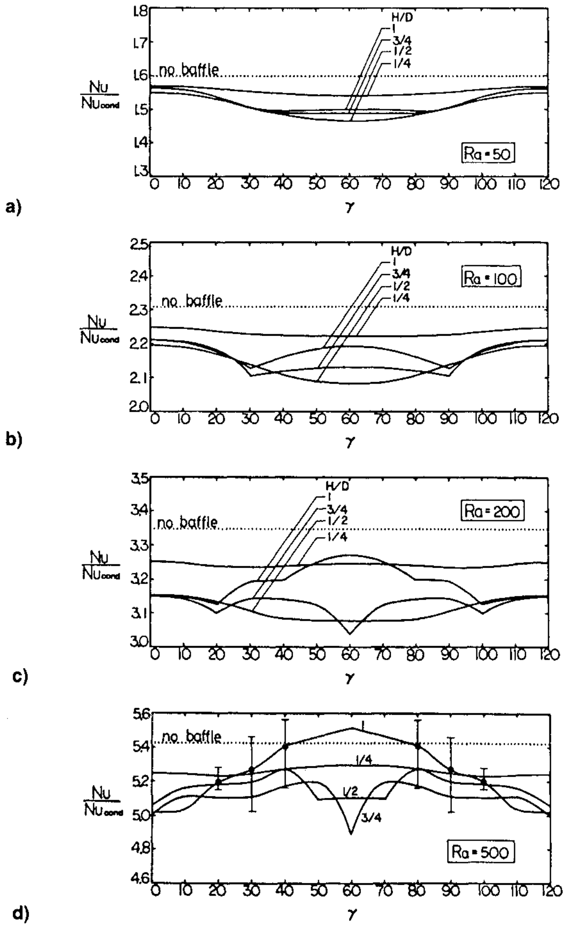

Fig. 3 Effects of baffle length on the heat transfer result for an annulus with three inner baffles.

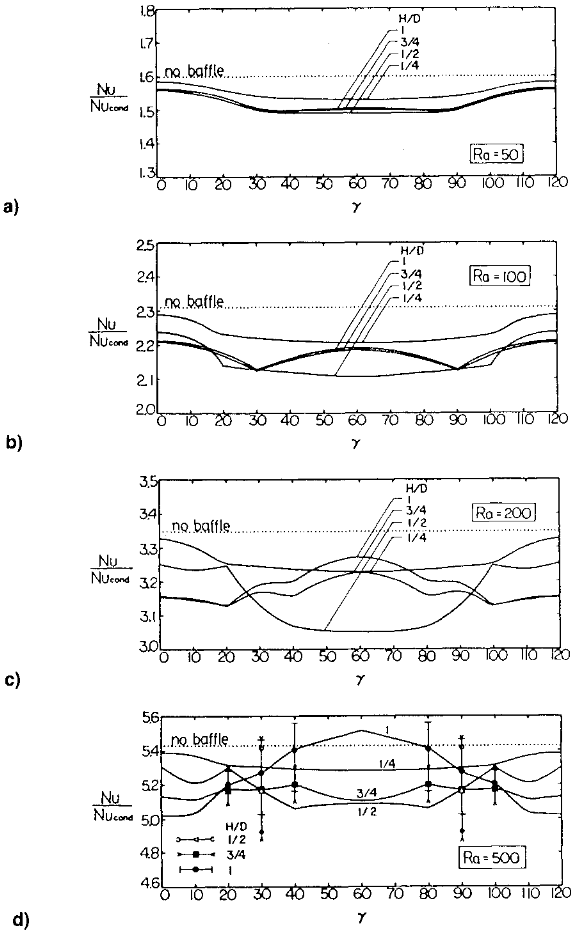

Fig. 4 Effects of baffle length on the heat transfer result for an annulus with three outer baffles.

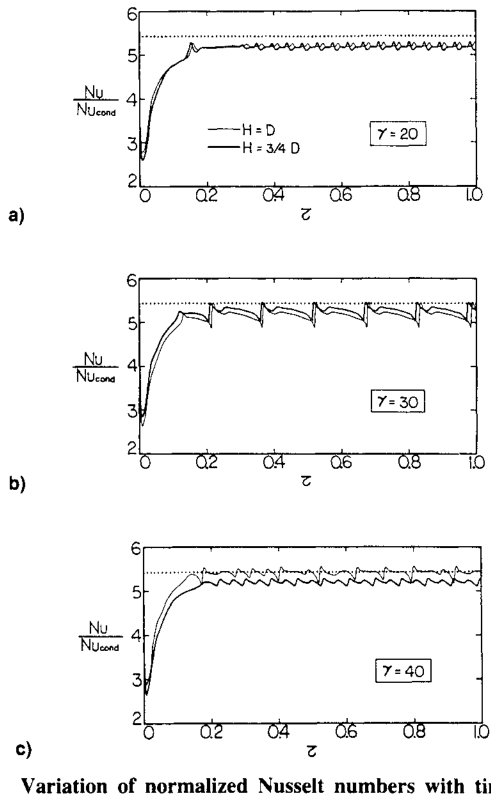

Fig. 5 Variation of normalized Nusselt numbers with time for an annulus with three outer baffles: a) γ = 20 deg, b) γ = 30 deg, and c) γ = 40 deg.

averaged Nusselt number for these cases is still less than that for an annulus with no baffles.

Although the relation between the Nusselt number and baffle length is complicated by the orientation angle, it is clear that for most cases, baffles with a length of $H/D = \frac{1}{2}$ produce the best result. In addition, it is found that inner baffles generally work better than outer baffles.

To reveal the nature of the oscillatory flows the variation of normalized Nusselt numbers with time is shown in Fig. 5 for an annulus with three outer baffles. The corresponding cases of full baffles are also included for comparison. For $\gamma = 20$ and 30 deg, the variation curves not only have a similar pattern but they also have a same period. However, there is a time lag between them. For $\gamma = 40$ deg, no similarity can be found in their patterns. In comparison to that of the full baffles, the period for the outer baffles is shorter and the averaged Nusselt number is also smaller.

Conclusions

In this study the feasibility of using radial baffles to reduce convective heat losses from pipe insulation has been successfully demonstrated. From the results obtained, it shows that the effectiveness of a baffle is not directly proportional to its length. It is found that a baffle with a length of $H/D = \frac{1}{2}$ gives the best result in heat loss reduction. For a given baffle the efficiency is also dependent on its orientation. Therefore, when applying radial baffles to pipe insulation, it is of equal importance to adjust the baffle orientation than to simply emphasize the baffle length.

Acknowledgment

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Heat Transfer in Turbulent Boundary Layers with a Short Strip of Roughness

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Introduction

THIS note presents the results of an experimental investigation of the influence of a short strip of roughness on the heat transfer and fluid flow in the zero pressure gradient flat-plate turbulent boundary layer. The primary motivation for this work is to gain insight into the heat transfer and fluid flow in a turbulent boundary layer with a wide variation in surface roughness. Of particular interest are inservice gas turbine blades which have been shown to have a wide variation of both the magnitude and statistical character of the roughness from point-to-point around the blades.[1] The average roughness ranged from about 10–1.5 μm. This is very rough considering that the thickness of the boundary layer is on the order of 1 mm. A particular feature of this data was the large variation in the character of the roughness around the blade.

A good review of the literature on surfaces that contain an interface between rough and smooth surfaces is given by Smits and Wood.[2] Andreopoulas and Wood[3] reported extensive measurements of velocity profiles, turbulence quantities, and skin friction distribution for flow over a smooth surface which was roughened in one narrow strip at about midplate using sandpaper. References 4 and 5 reported heat transfer data for the case where the first one-third of their flat plat was rough and the remainder was smooth. They also presented predictions using the discrete element method roughness model.

In this note, the experimental results are presented for Stanton number distributions, mean velocity profiles, and turbulence intensity profiles for a turbulent boundary layer with a short strip of surface roughness. These results are compared with the results of the rough-to-smooth experiments of Taylor et al.[5] and the all-smooth experiments of Coleman et al.[6]

Experimental Apparatus and Measurements

The experiments were performed in the turbulent heat transfer test facility which is a closed loop wind tunnel designed for boundary-layer heat transfer experiments. The description and qualification of this facility are given in the open literature (Coleman et al.[7]), and are not repeated here because of the space limitations of a technical note. In Ref. 7, the smooth-wall experiments were shown to be in excellent agreement with the accepted classical experiments and correlations, the flow was shown to be a proper two-dimensional turbulent boundary layer, and the uncertainties associated with the data collection and reduction were discussed in detail.

Figure 1a shows a schematic diagram for the test surface used in these experiments. The first 0.7 m of the test surface is smooth, the next 0.2 m is roughened with 1.27-mm-diam hemispheres, and the remaining 1.5 m is smooth. This is accomplished using 7 smooth plates, 2 rough plates, and 15

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