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Introduction

The mixed convection regime is a transitional heat transfer regime between forced convection and natural convection, where both the forced component of flow, and the buoyancy induced component are important. Aiding flow is when buoyancy forces act in the same direction as the forced flow (heated upflow or cooled downflow), while opposing flow is when the buoyancy force is in the opposite direction of the forced flow (cooled upflow or heated downflow). For opposing flow the buoyancy always increases the rate of heat transfer over the forced convection value. For aiding flow, as the heat flux increased, a reduction in heat transfer is encountered until a condition known as laminarization occurs, where the heat transfer is at a minimum value. Further increases in the wall heat flux causes re-transition to turbulence, and increased heat transfer. In this paper, for the first time, experiments were performed to characterize the effect of surface roughness on heat transfer in mixed convection, for the case of aiding flow. A correlation was developed to allow calculation of mixed convection heat transfer coefficients for rough or smooth tubes.

Experiment

Experiments were performed in rough and smooth tubes with an inside diameter (ID) of 15.75 mm (0.62 inches), and a heated length of 1.422 m (56 inches). Unheated runs were first made to determine test section heat loss and isothermal pressure drop. The test section was heated using a DC power supply. Heated runs were performed varying the heat flux and flow rate. Outside tube wall temperatures were measured using thermocouples (TC) welded to the tube OD equally spaced along the axial length of the tube. For each outside wall thermocouple, the inside wall temperature was determined by solving the heat conduction equation in the tube, accounting for internal heat generation and heat losses from the outside wall. Temperatures were corrected for voltage drop across the TC bead by reversing the test section polarity for several test runs. Both rough and smooth tubes were tested to determine the influence of surface roughness as the flow transitions from the forced convection regime into mixed convection. Roughness elements approximately 0.1 mm (0.004 inches) high were produced using a knurling tool. The mean height of the roughness was determined using profilometry. The relative roughness (ε/D) was .00645. A smaller smooth tube (10.14 mm or 0.40 inch ID) was also tested to provide a tube diameter effect. Data were obtained over a Reynolds number ranging from 2,600 to 70,000. The inlet temperature was 93°C (200°F) and the pressure was 4.14 MPa (600 psia).

Pressure Drop Results

Pressure drop for single phase forced convection in smooth tubes can be calculated using the Blasius equation:

$$f_{smooth} = \frac{.316}{Re^{.25}} \quad (1)$$

For rough tubes, the Colebrook-White Equation [1] may be used:

$$\frac{1}{\sqrt{f}} = -2\, log_{10}\left[\frac{\varepsilon / D}{3.7} + \frac{2.51}{Re\sqrt{f}}\right]$$

An alternate rough tube friction factor formulation, based on equivalent sand grain roughness, is due to Nikuradse [2]. A fit has been derived from his data to determine f_{rough}/f_{smooth}. The isothermal smooth tube data was well represented by the Blasius Equation, and the rough tube data followed the Colebrook and Nikuradse correlations when ε=0.1 mm (0.004 inches) is used for the roughness. For non-isothermal flow, the buoyancy forces increased the friction factor over the smooth and rough tube correlations, as was observed in [3].

Heat Transfer Correlation

The data for the three tubes are shown in Figure 1, which plots the Nusselt number ratio (Nu/Nu_{smooth}) versus the buoyancy number ($Bo=Gr/Re^3Pr^{.5}$). In this analysis, the Dittus-Boelter correlation is used for Nu_{smooth}:

$$Nu_{smooth} = .023\, Re^{.8} Pr^{.4}$$

The smooth tube data of Parlatan et. al. [3], (with tube ID=1.05 in.) are also shown for comparison. The correlation which gave the best fit the smooth tube data is:

$$\frac{Nu}{Nu_{fc}} = \frac{Nu}{Nu_{smooth}} = \left[\left(\frac{\left(\frac{5.9\times 10^{-6}}{Bo}\right)^{2.174}}{1+\left(\frac{5.9\times 10^{-6}}{Bo}\right)^{2.174}}\right)^{5} + \left(76.08\, Bo^{.426}\right)^{5}\right]^{1/5} \quad (4)$$

The heated flow runs in the forced convection regime indicated about a 20% increase in the heat transfer coefficient. It was found that the increase in the heat transfer coefficient was in good agreement with Nunner's Equation [4].

$$\frac{Nu_{rough}}{Nu_{smooth}} = \sqrt{\frac{f_{rough}}{f_{smooth}}} \quad (5)$$

Both the Nikuradse fit and the Colebrook equation (Eq. 2) were used to evaluate f_{rough}, and the Blasius correlation (Eq. 1) is used to obtain f_{smooth}. For the highest flow tested, roughness increased the pressure drop by a factor of 1.43, and the Nusselt number by a factor of 1.20. In the mixed convection regime, the effect of roughness gradually decreases with increasing buoyancy number, as indicated by the dashed line in Figure 1. The data indicated that as the buoyancy number increases, and the mixed convection regime is entered, the effect of roughness gradually diminishes. At the heat transfer minimum, the change in the heat transfer coefficient due to roughness is negligible. This is because the flow becomes laminar at the heat transfer minimum, and roughness does not influence laminar flow heat transfer. Figure 2 shows the same data after application of Nunner's equation to account for roughness. A fit to Nikuradse's data was used to evaluate f_{rough}, but the Colebrook correlation works almost as well. Nunner's Factor collapses the rough tube data back onto the smooth tube data for high flow, but the factor has a small effect on the low flow data, which closely follows the smooth tube mixed convection results.

Criterion for the Onset of Mixed Convection

It is noted in Figure 2 that the transition from the forced convection regime to mixed convection occurs at $Bo=10^{-6}$. This same mixed convection criterion was proposed by Inagaki et. al. [5]. If this criterion is plotted on the flow regime map of Metais and Eckert [6], it is well above their onset of mixed convection line, as shown in Figure 3. This regime map, which appears in many heat transfer text books, significantly underestimates the mixed convection regime.

Summary

A correlation has been developed for the heat transfer coefficient for the mixed convection regime, accounting for the first time for the effects of surface roughness. The procedure is to evaluate f_{rough}/f_{smooth} using the isothermal rough and smooth tube correlations, then multiply the square root of this ratio by the Dittus-Boelter Nusselt number to obtain Nu_{fc}. This forced convection Nusselt number is then substituted into Equation 4 to obtain the mixed convection Nusselt number, Nu:

$$\frac{Nu}{Nu_{fc}} = \frac{Nu}{Nu_{smooth}\sqrt{\dfrac{f_{rough}}{f_{smooth}}}} = \left[\frac{\left(\dfrac{5.9\times 10^{-6}}{Bo}\right)^{2.174}}{1+\left(\dfrac{5.9\times 10^{-6}}{Bo}\right)^{2.174}} + \left(76.08\,Bo^{.426}\right)^{5} \right]^{1/5} \qquad (6)$$

The correlation matches the rough and smooth tube data of Figure 2 to within +/-20%. It has been shown that the flow regime map for heat transfer in vertical pipes and tubes [6] underestimates the extent of the mixed convection regime. The onset of mixed convection is the simple criterion $Bo>10^{-6}$.

References

1) J. Nikuradse, Stromungsgesetze in rauhen Rohren. Forsch. Arb. Ing.-Wes. No. 361, (1933); see also Boundary Layer Theory, 6th ed., pp 580, H. Schlichting, McGraw-Hill, (1968).

2) C.F. Colebrook, Turbulent flow in pipes with particular reference to the transition region between smooth and rough pipe laws, J. Institute of Civil Engineers, (1939); see also Marks Standard Handbook of Mechanical Engineers, 8th ed., pp3-54, McGraw-Hill, (1978).

3) Parlatan, Y., N.E. Todreas and M.J. Driscoll "Buoyancy and Property Variation Effects in Turbulent Mixed Convection of Water in Vertical Tubes", Journal of Heat Transfer, Vol. 118, pp. 381-387 1996.

4) Nunner, W., Z. ver. Deut. Ing., Forshungsh., Vol 255, 1(1956), or Convective Heat and Mass Transfer, W.M. Kays, McGraw-Hill Book Company, 1966. pp 197.

5) Inagaki, T. (1996) "The Criterion for Turbulent Combined Forced and Natural Convection in a Vertical Flow System", Journal of Heat Transfer, Vol. 118, pp 213-215.

6) Metais, B. and E.R.G. Eckert, "Forced, Mixed and Free Convection regimes", Journal of Heat Transfer, Vol. 86, 1964, pp. 295-296.

Nomenclature

Dimensional Parameters:

D	Tube inside diameter (ID, mm)
G	Mass Flux (Kg/m^2sec)
g	Acceleration due to gravity (9.8 m/sec^2)
h	Heat Transfer Coefficient ($W/m^2\ ^oC$)
k	Liquid Thermal Conductivity (W/m^oC)
P	Pressure (MPa)
q	Heat Flux (W/m^2)
T_w	Wall Temp (^oC)
T_b	Bulk Temperature (^oC)
ρ	Density (Kg/m^3)
β	Thermal expansion coefficient of liquid ($1/^oK$)
μ	Dynamic viscosity (Kg/m-sec)
υ	Kinematic viscosity (m^2/sec)

Dimensionless Parameters:

$Gr = \frac{g\beta(T_w - T_b)D^3}{\upsilon}$ (Grashof Number)

$Re = \frac{GD}{\mu}$ (Reynolds Number)

$Pr = \frac{\mu C_p}{k}$ (Prandtl Number)

$Nu = hD/k$ (Nusselt Number)

$Bo = Gr/(Re^3 Pr^{0.5})$ (Buoyancy Number)

Figure 1
Nusselt Number Ratio vs Buoyancy Number

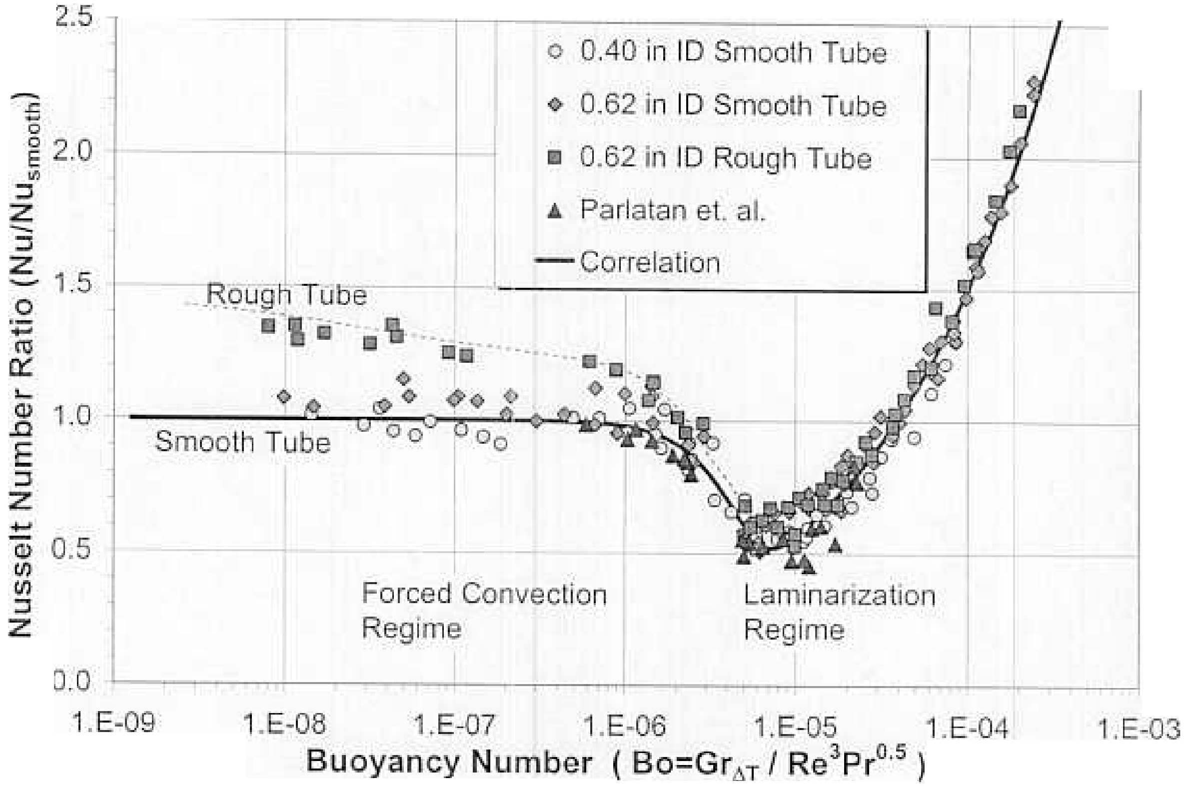

Figure 2
Nusselt Ratio vs Buoyancy Number,
with Nunner's Correction to the Rough Tube Data

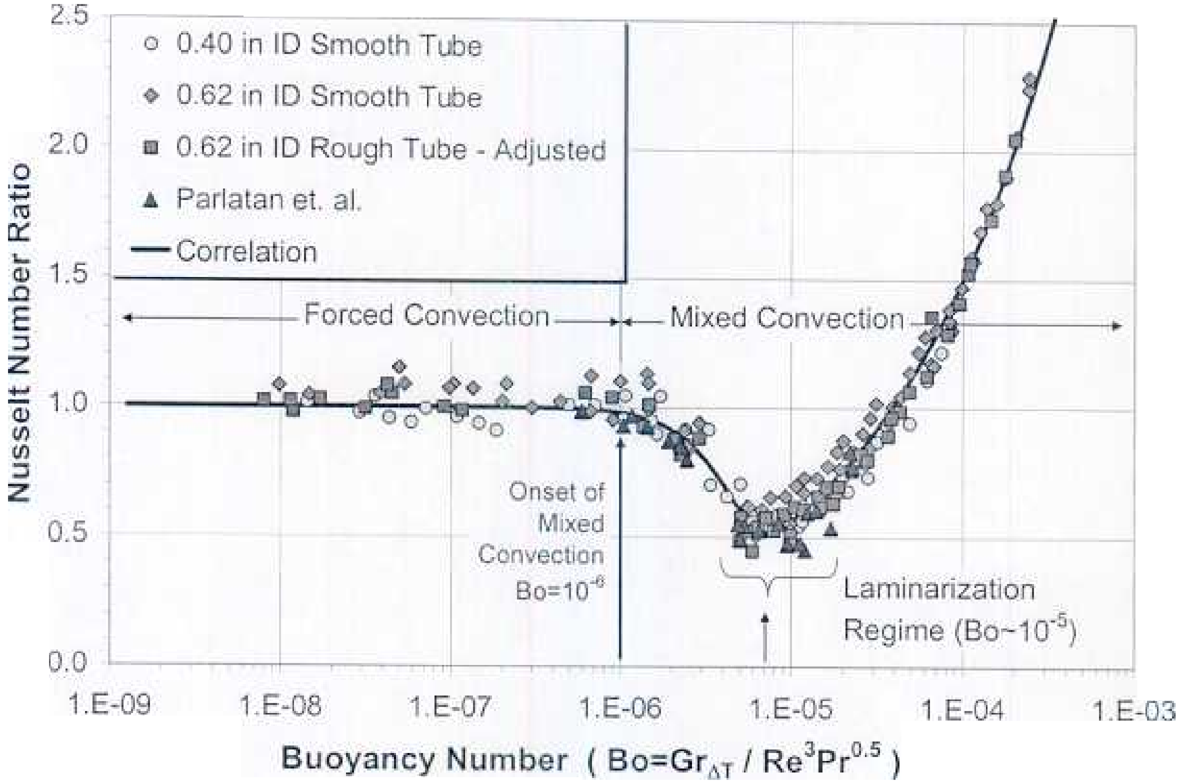

Figure 3
Flow Regime Map of Metais and Eckert [6],
with Onset of Mixed Convection Criterion based on the Present Data