EXPERIMENTAL STUDY OF LAMINAR MIXED CONVECTION IN A ROD BUNDLE WITH MIXING VANE SPACER GRIDS

Lokanath Mohanta* **and Fan-Bill Cheung**
Department of Mechanical and Nuclear Engineering, Pennsylvania State University
University Park, PA 16802, USA
lxm971@psu.edu

Stephen M. Bajorek, Kirk Tien and Chris L. Hoxie
Office of Nuclear Regulatory Research, U.S. Nuclear Regulatory Commission
Washington, DC 20555-0001, USA

ABSTRACT

Heat transfer by mixed convection in a rod bundle occurs when convection is affected by both the buoyancy and inertial forces. Mixed convection can be assumed when the Richardson number ($Ri = Gr/Re^2$) is on the order of unity, indicating that both forced and natural convection are important contributors to heat transfer. In the present study, data obtained from the Rod Bundle Heat Transfer (RBHT) facility was used to determine the heat transfer coefficient in the mixed convection regime, which was found to be significantly larger than those expected assuming forced convection based on the inlet flow rate. The inlet Reynolds (Re) number in the tests ranged from 500 to 1300 while the Grashof (Gr) number varied from 1.5×10^5 to 3.8×10^6 yielding $0.25 < Ri < 4.3$. Using results from RBHT test along with the correlation from the FLECHT-SEASET test program for laminar forced convection, a new correlation for mixed convection in a rod bundle is proposed. The new correlation accounts for the enhancement of heat transfer relative to laminar forced convection.

KEYWORDS
Mixed Convection, Rod Bundle, Spacer Grid, Linear Heat Flux, Natural Convection

1. INTRODUCTION

In a hypothetical loss of coolant accident following the quenching of the bundle, the heat transfer regime can be mixed convection in the lower portion of the reactor core. For long-term cooling of the reactor core, mixed convection would be the prevailing mode of heat transfer. It should be noted that the new generation of small modular reactors with passive safety cooling systems highly depend on mixed convection heat transfer. In addition to nuclear reactor applications, mixed convection heat transfer mode also occurs in other engineering fields including, for example, cooling of electronic equipment, solar energy systems, drilling of oil wells, and removal of pollutants from confined spaces [1].

Mixed convection heat transfer has been extensively studied for vertical surfaces [2-6] and vertical channels [7-11]. In the existing mixed convection studies (for both laminar and turbulent flows), the Nusselt number is correlated as a function of several non-dimensional parameters including $Gr, Pr, Re, Ra,$ and Gz. Laminar mixed convection in partially blocked vertical channel was studied by Habchi and Acharya [12]. In their study, the Nusselt number was found to decrease in the streamwise direction with a local maximum at the blockage. Jackson *et al.* [13] provided a detailed review of mixed convection in vertical tubes. They found that the Nusselt number correlates well as a function of Gr/Re. Jackson and Fewster [14] correlated the enhancement factor (Nu/Nu_0) for circular tubes as a function of $Gr/(Re^{21/8})$ for opposing turbulent flows. Swanson and Catton [15] presented an enhancement factor for vertical parallel plates in terms of the Richardson number. A later work on laminar mixed convection in a vertical circular tube with uniform heat flux for both opposed and assisted flow in the laminar Reynolds number range of 400-1600 was presented by Mohammed [16]. They found that the Nusselt number is high at the inlet and decreases axially downstream, but the mixed convection remained higher than the laminar forced convection. The Nusselt number was correlated for buoyancy assisted flow as a function of Gr/Re.

In addition to simple geometries, the study of mixed convection has been extended to annulus and rod bundle geometries. Maitra and Raju [17] showed that the Nusselt number varies with $Ra^{1/4}$ in mixed convection in a vertical annulus. Maudou *et al.* [1] studied mixed convection in vertical annular channels with and without eccentricity. An increase in the Nusselt number with increasing Re was observed for both concentric and eccentric channels. Dawood *et al.* [18] presented a review of experimental and numerical work on mixed convection heat transfer in concentric as well as eccentric annulus geometry. Yang [19] in a theoretical study analyzed mixed convection in both square and triangular array rod bundles, and showed that the Nusselt number increases with Gr/Re. An analytical solution was proposed by Iannello *et al.* [20] for fully developed laminar mixed convection flows in an annular and in rod bundle geometries with a triangular array. They obtained a relation for the Nusselt number as a function of Gr/Re. Suh *et al.* [21] correlated the subchannel pressure drop in laminar and transitional mixed convection as a function of $ln(1 + (Gr/Re)$. Kim and El-Genk [22] investigated single-phase heat transfer in a triangular array rod bundle. Based on the experimental data it was concluded that the flow can be treated as forced laminar flow for $Ri < 1$ and mixed convection for $Ri \geq 1$. In mixed convection regime, they correlated the Nusselt number as

$$Nu = a\, Ri^b Re^c \tag{1}$$

The values of a, b and c depend on the pitch to diameter ratio of the bundle. A similar study was conducted by El-Genk *et al.* [23] in a nine-rod square array rod bundle. They concluded that the mixed convection regime is significant when $Ri \geq 2$ for a square array rod bundle. The Nusselt number in the mixed convection regime was obtained by superimposing the correlations for forced and natural convection. Kim [24] presented a Nusselt number for infinite rod bundle with square pitch of 1.33 as given below

$$\frac{Nu}{Pr^{0.33}} = 7.86 \tag{2}$$

In the FLECHT-SEASET test program [25], steam cooling tests were conducted in a 161-rod unblocked bundle with square array having pitch to diameter ratio of 1.33. A minimum Reynolds number for transition was observed to be 2500 from the test data. They recommended the Nusselt number given in Eq. (2) for laminar forced convection.

Previous studies demonstrated that in mixed convection the Nusselt number is a function of Gr/Re^n, with the value of n being different from different studies. The effect of mixed convection is found to be important in the range of $0.3 < Ri < 5$ for a vertical plate [26]. Sudo *et al.* [27] defined the mixed convection regime to be in the range of $0.0001 < Gr/(Re^{21/8}) < 0.01$. For a rod bundle having a uniform heat flux, El-Genk *et al.* [23] concluded that the flow can be treated as forced laminar for $Ri < 2$ in a square array. In the present work, experiments were conducted in a 7x7 rod bundle that has mixing vane type spacer grids. The rod bundle has linearly increasing heat flux. Experimental data obtained in the present work shows that mixed convection is important even at very low Ri ($\sim$0.3). The local Nusselt numbers are found to be higher than those reported by El-Genk *et al.* [23]. An enhancement factor to account for the effect of mixed convection has been developed in this study in the form of $ln(1 + (Gr/Re^2))$.

2. EXPERIMENTAL METHOD

2.1. Test Facility Description

The Rod Bundle Heat Transfer (RBHT) test section consists of 7x7 full-length electrically heated rods with a diameter of 9.5 mm (0.374 in) and pitch of 12.6 mm (0.496 in) with seven spacer grids placed in a square flow housing of 90.2 mm (3.55 in) [28, 29]. The test section simulates a portion of a commercial size 17x17 rod bundle. There are 45 electrically heated rods and 4 support rods in the corners. The length of the heated portion of the rod bundle is 3.66 m (144 inch) with a top-skewed axially linear power profile having a peak power of 1.5 times the average power at 2.74 m (108 inch) elevation and 0.5 times of the average power at both ends. A schematic of the flow loop and a 3-D drawing of the facility is shown in Figure 1. The flow loop consists of a water supply tank, injection line with a centrifugal pump with the capacity of 0.946 m^3/min, a flow meter having a range 0-454 kg/min, lower plenum, test section, upper plenum, carryover tanks, and pressure oscillation damping tank connected to the exhaust line.

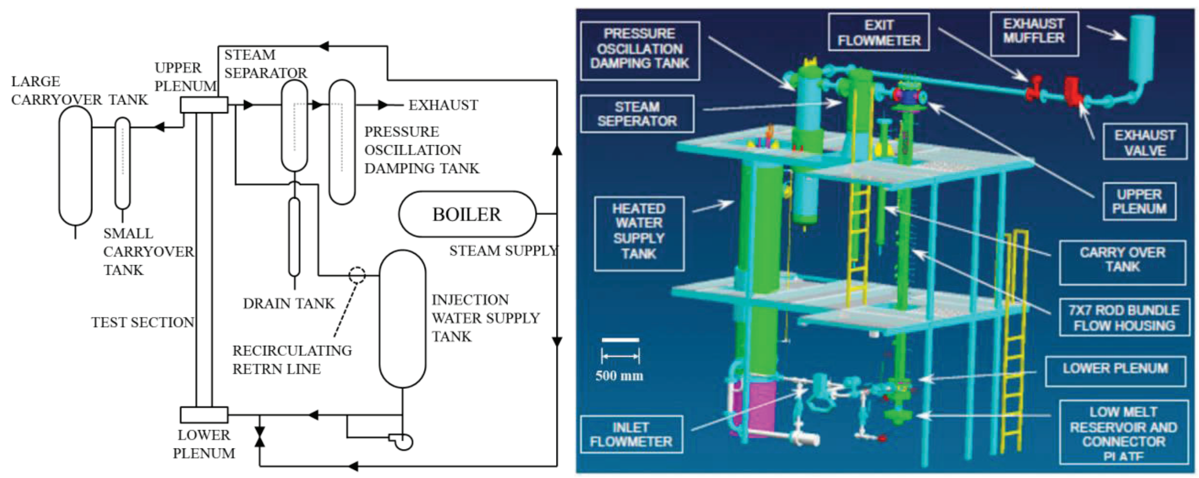


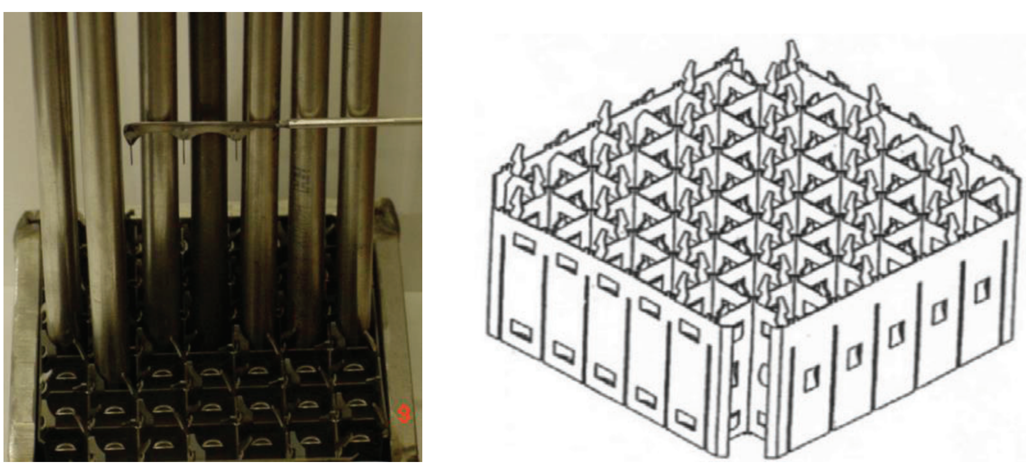
Figure 1. Schematic of the Flow Loop and 3-D Drawing of the Test Facility.

To measure the temperature in the rod bundle, 256 K-type thermocouples are located on the inside surface of the cladding with a distribution that covers the entire length of the test section. The bundle has seven mixing vane spacer grids with a design prototypical of a commercial fuel bundle. Figure 2 (a) shows an actual photograph of the spacer grid with a few rods and a traversable temperature probe. An isometric view of the spacer grid used in the rod bundle is shown in Figure 2 (b). The first spacer grid is located at a distance of 0.102 m from the inlet and the second spacer grid is at a distance of 0.588 m from the first. The other spacer grids are separated by 0.522 m subsequent to Grid 2. These spacer grids have the same blockage ratio of 36.22%, when viewed from the top. Several thermocouples are mounted on the spacer grid. In order to measure the centerline temperature, 13 traversable probes having three thermocouples each (as shown in Figure 2 (a)) are installed at various axial locations. The RBHT facility has 23 differential pressure (DP) cells mounted on the flow housing. Sixteen DP cells having a span length of 76.2 to 127 mm (3 to 5 in) are used to provide a detailed axial pressure drop near the middle of the bundle. High-resolution digital cameras along with an infrared laser are used to observe the flow field in the bundle. There is also appropriate instrumentation to measure the input power, inlet flow rate, exhaust steam flow rate, inlet liquid temperature, upper plenum pressure, flow housing temperature, and liquid level.

2.2. Test Procedure

Several level swell tests were carried out in the RBHT facility using deionized water [30, 31]. The system pressure, regulated using a Nitrogen line connected to the supply tank, was varied from 138 to 414 kPa (20 to 60 psia). To carry out a test, water was fed to the test section through the lower plenum having a flow straightener, which ensures a uniform flow distribution in the test section. The power was applied to the bundle using a DC power supply. Flow rates were varied from 2.5 to 40.6 mm/s. When the flow rate was varied the bundle was allowed to reach steady state, which took few hundred seconds; then temperature measurements were recorded for more than 100 sec at a frequency of 1 Hz. The inlet liquid subcooling was varied from 11 to 56 K. The power of the bundle was varied from 54 kW to 144 kW.

In all the level swell tests, the lower portion of the bundle exhibited single-phase cooling. However, to evaluate mixed convection effects, only tests in which single-phase convection was observed at least up to 1.5 m from the inlet are considered. Twenty three of the tests showed the single-phase convection in the first 1.5 m section of the bundle. The possibility of subcooled boiling could be ruled out when cladding temperatures (surface temperatures) were below the saturation temperature. To ensure that the heat transfer was by single-phase convection, the region of interest is between the inlet of the rod bundle and the elevation, where the cladding temperature remains at least 5 K below the saturation temperature. Any temperature below saturation rules out the possibility of boiling, however to be conservative, the criterion of 5 K below subcooling is chosen. Twenty three tests qualify the aforesaid criteria for evaluation. Test conditions for these tests are shown in Table I. The letter after the flow rate is an identifier for a Test. For these tests, based on the test conditions, the inlet Reynolds number ranged from 500 to 1300 and the bundle power ranged from 72 to 144 kW. The heat flux at the inlet was 7320 kW/m^2 for a test with 72 kW input bundle power and the heat flux linearly varied to 21970 kW/m^2 at the peak power location (2.74 m). The local Reynolds numbers are calculated based on the mass flux and the local fluid properties evaluated at the local bulk mean temperature. The local Grashof number (Gr) varied from 1.5×10^5 to 3.8×10^6. These variations in Gr and Re generated data for the Richardson number ranging from 0.25 to 4.3.



(a) (b)

Figure 2. (a) Photograph of a Spacer Grid with Heater Rods, Flow Housing and Traversable Temperature Probes (b) Isometric View of the Spacer Grid Used in the RBHT Facility.

Table I. Tests Conditions that showed Mixed Convection in the Bundle

Test No	Pressure (kPa)	Inlet Temperature (K)	Bundle Power (kW)	Flow rate (cm/s)	Inlet Subcooling (K)
1578	138	328.5	69.5	2.52(A), 3.13(B), 3.06(C)	53
1651	276	348.2	72.3	3.07(A), 3.05(B), 3.03(C), 2.52(E)	55
1659	414	361.7	72.3	3.05(A), 3.06(B), 3.03(C), 2.54(D), 2.03(F), 2.1(G), 2.03(H), 1.53(I)	56
1678	138	327.2	71.9	3.1(A), 3.04(B), 2.54(D), 2.54(E)	54
1679	138	328.1	72.3	2.03(A)	53
1683	276	348.4	144.4	4.05(A), 4.1(B), 4.12(C)	55

2.3. Data Reduction Method and Uncertainties

Thermocouples embedded in the heater rods were used to measure the temperature inside the cladding, which has a thickness of 0.71 mm. The outer cladding temperature and the surface heat flux are calculated by solving a transient, one-dimensional inverse heat conduction problem [29]. The heat transfer coefficient is defined as

$$h(z) = \frac{q_w''(z)}{[T_w(z) - T_b(z)]} \tag{3}$$

where q_w'' is the wall heat flux, $T_w(z)$ is the cladding temperature at a given location z, and $T_b(z)$ is the local bulk mean temperature. A one-dimensional energy balance method is used to calculate the local bulk mean temperature. The heat flux obtained from the inverse heat conduction solution is verified with the steady state heat flux. The Nusselt number is defined as

$$Nu = \frac{hD_h}{k} = \frac{q_w'' D_h}{(T_w - T_b)k} \tag{4}$$

where D_h is the hydraulic diameter of the subchannel and k is the thermal conductivity of the liquid calculated at the local bulk mean temperature. The Grashof number is defined as

$$Gr = \frac{g\beta\Delta T D_h^3}{\nu^2} \tag{5}$$

where, ΔT is the temperature difference between the cladding and the centerline temperature, measured using the temperature probes, ν and β are kinematic viscosity and thermal expansion coefficient, respectively, evaluated at the bulk mean temperature.

The uncertainty due to the thermocouple measurement was estimated to be ±1.11 K. The uncertainty in the flow measurement was ±0.0237 kg/min. The error in the bundle power measurement was ± 1.29 kW. Compared to the bundle power of 72 kW, the uncertainty in the power measurement is 1.8 % and for the bundle power of 144 kW it was 0.9 %. Using these uncertainties as input, the uncertainty in Nu was less than ± 2%. Considering a simple heat conduction network through the flow housing wall with insulation and assuming free convection from the outer surface of the insulation the heat loss was found to be less than 0.1 %.

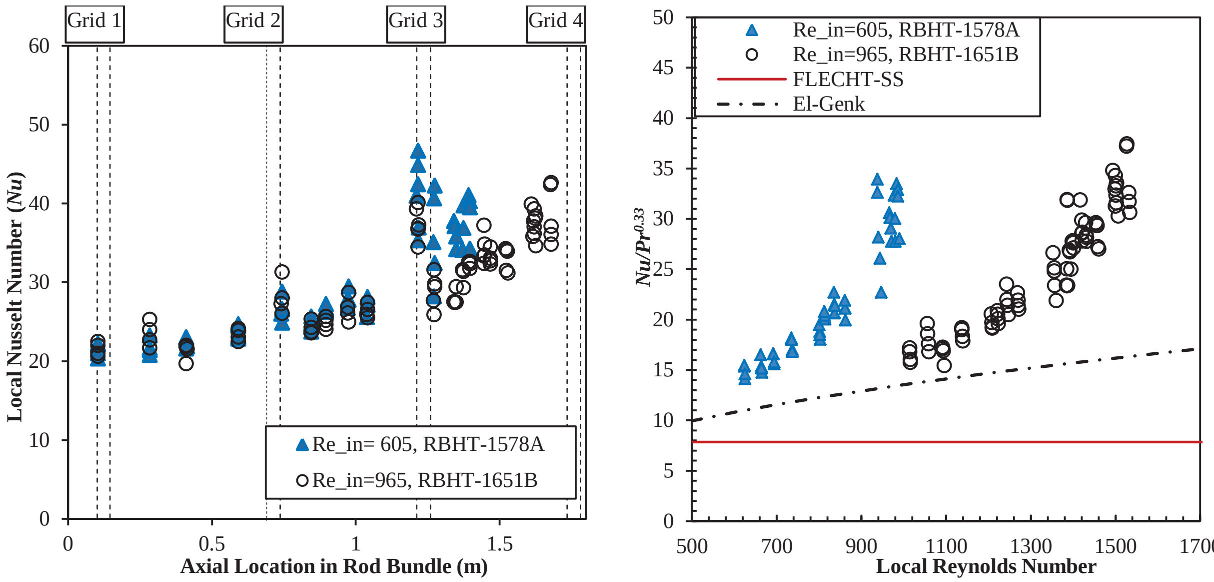
3. RESULTS AND DISCUSSION

The critical Reynolds number for flow transition from laminar to turbulent for the rod bundle geometry is 2500 according to Wong and Hochreiter [25]. According to El-Genk *et al.* [23], however, the critical Reynolds number is given by

$$Re_T = 1.33 \times 10^4 \left[\left(P_r / D \right) - 1 \right] \tag{6}$$

where P_r and D are the pitch and diameter of the heater rods, respectively. For the RBHT bundle, the pitch to diameter ratio is, $P_r/D = 1.326$. The transition Reynolds number for RBHT bundle according to El-Genk *et al.* [23] is therefore 4340. The inlet Reynolds number for the RBHT tests varied from 500 to 1320. The local Reynolds numbers were below 2000. Checking the transition from a natural convection perspective, the Raleigh numbers for these tests were below 10^7. This confirms that the flow was laminar for these data.

Figure 3 (a) shows the variation of the Nusselt number for Tests 1578A and 1651B, having an inlet Reynolds number of 605 and 965, respectively. These Nusselt numbers are presented with the local Reynolds number in Figure 3(b). Note that, in Figure 3(b), data points at the spacer grids are not presented. There are several thermocouples at the same axial elevation in different heater rods which resulted in different Nusselt numbers at the same axial location. In general, the Nusselt number is found to increase with increasing elevation. This behavior is different from that observed in a vertical tube with uniform heat flux [16]. The reason for the increase is due to the linearly increasing heat flux with elevation. The local Reynolds number along with the Richardson number increases with elevation because of the axially increasing heat flux. The variation of Ri in the rod bundle is shown in Figure 4. A similar observation, i.e., increase in the Nusselt number with increasing Ri at a given axial location, has been reported in literature for vertical parallel surfaces [32] and for a vertical tube [16]. The local Reynolds number increases because of a lower viscosity of vapor at a higher temperature. Despite different inlet Reynolds numbers in Tests 1578A and 1651B, the Nusselt numbers near the inlet, i.e. between Grids 1 and 2 are the same. As the heat flux is low near the inlet of the bundle the Richardson number is small, hence the flow behaves like purely forced laminar flow. However, near Grid 3, the Nusselt number is higher for Test 1578A having a lower local Reynolds number. This observation can be corroborated by observing the slope of the data points for the two tests shown in Figure 3(b). This is due to the fact that the Richardson number is higher in Test 1578A. Another important observation is that the rate at which the Nusselt number increases is higher between Grids 3 and 4 compared to those between Grids 1 and 2 and Grids 2 and 3. The heat flux linearly increases with elevation due to the axially skewed power profile. With increasing heat flux, Ri increases, which causes stronger buoyancy effect. Figure 3(b) also compares the fully developed forced laminar Nusselt numbers using the correlations by Wong and Hochreiter [25] and El-Genk *et al.* [23]. According to Wong and Hochreiter, $Nu/Pr^{0.33}$ is a constant having the value of 7.86 in the laminar forced convection regime. Their model under-predicts the RBHT data. El-Genk *et al.* [23] derived their correlation assuming the flow to be forced laminar when the Reynolds number is below the transition Reynolds number and the Richardson number is less than 2. Hence, their correlation had actually accounted for the mixed convection implicitly. Because of this reason, the El-Genk correlation predicts a higher Nusselt number than the Wong and Hochreiter correlation, but under-predicts the RBHT data.



(a) (b)

Figure 3. (a) Axial Variation of the Nusselt Number in the Rod Bundle (b) Variation of the Nusselt Number with the Local Reynolds Number

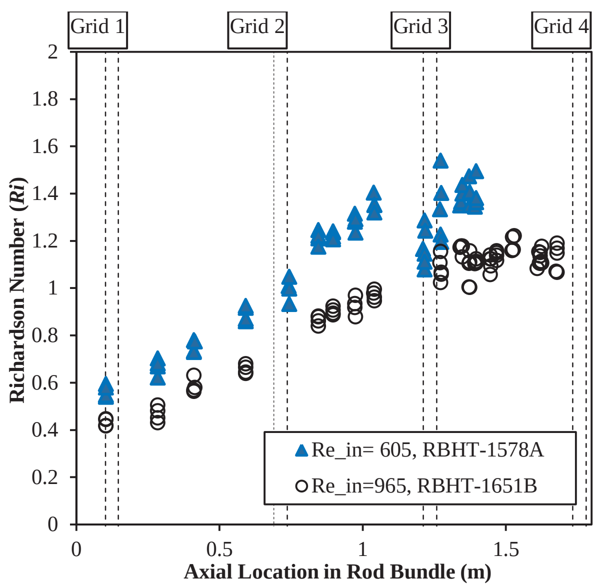


Figure 4. Axial Variation of the Richardson Number in the Rod Bundle

3.1. Effect of Spacer Grid

The effect of spacer grid on heat transfer can also be observed from Figure 3 (a). The Nusselt number is higher at Grids 2 and 3. The effect of spacer grid at Grid 1 is not evident from the figure, because this grid is very close to the inlet of the rod bundle and the entrance effects would be dominant near this grid. Downstream of the spacer grids, the Nusselt number increases in the flow direction. This behavior is different from the spacer grid effects observed in turbulent single-phase steam flow [33, 34, 35], when the effect of natural convection is not significant. In turbulent flow over a spacer grid, the swirling effect decays downstream of the spacer grid resulting in a decay in the heat transfer enhancement. In the mixed convection regime, a local maximum is observed at the spacer grid and the Nusselt number increases downstream of the spacer grid with the Richardson number, as shown in Figures 3(a) and 4. A similar observation pertaining to local maximum has been reported by Habchi and Acharya [12], using a numerical model of laminar mixed convection in a partially blocked vertical channel. In their work, a local maximum in the Nusselt number was observed where flow reattaches downstream of the blockage. The present measurement does have closely spaced thermocouples, to examine reattachment, however, local maxima are observed at the spacer grid locations. The enhancement at the spacer grid is due to flow acceleration as a result of blockage in the flow area. In addition to that the local flow structure changes due to the presence of the grid.

3.2. Comparison with Forced Laminar Correlation

To compare with the fully developed forced laminar correlation, the RBHT data at a distance of $z/D_h >$ 38 from the first and third grids and at a distance of $z/D_h >$ 32 from the second grid are considered. For laminar forced convection, a fully developed flow is observed at a distance $(z/D_h)/Re > 0.05$ [36]. The redevelopment length from the trailing edge of the grid for an inlet Reynolds number of 500 would be $z > 25D_h$, however, for Re=1000, $z > 50\ D_h$. The RBHT data at a distance of $z/D_h >$ 32 can be assumed to be fully developed for comparison with forced convection correlation. Two correlations for forced laminar flow in rod bundle having square array are compared with the RBHT data. Figure 5 (a) and (b) compare the current data (from 23 tests) with the Wong and Hochreiter [25] and El-Genk [23] correlations, respectively. According to Wong and Hochreiter [25], the Nusselt number is independent of the Reynolds number, but is a function of Prandtl number. In the current data, the Prandtl number does not vary significantly. That explains why the Nusselt number is nearly constant using the Wong and Hochreiter [25] model. Note that the El-Genk correlation implicitly accounts for the mixed convection. As such, the El-Genk Correlation compares better with the RBHT data although it still appreciably under-predicts the RBHT data.

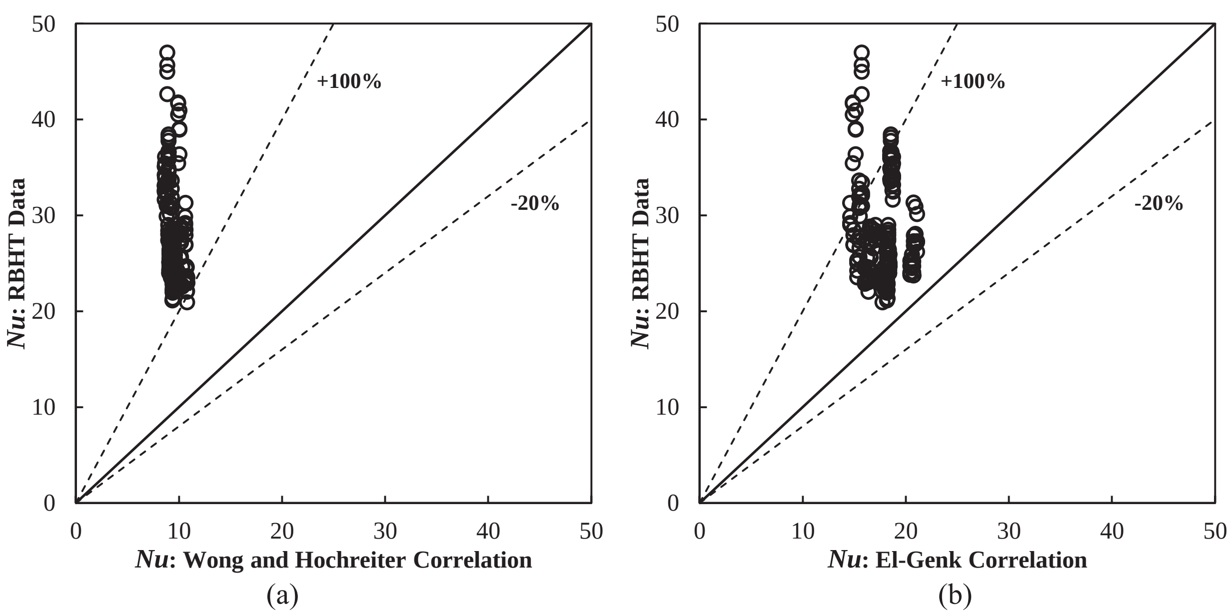


Figure 5. Comparison with Single-Phase Forced Laminar Convection (a) Wong and Hochreiter Correlation [25] (b) El-Genk *et al.* Correlation [23]

3.3. Enhancement due to Mixed Convection

There are several correlations for the Nusselt number based on the Reynolds, Prandtl, Richardson and Grashof numbers for mixed convection heat transfer for buoyancy assisted flow [16, 22, 26, 32, 37]. One of the disadvantages in using these forms of the correlation is that when the effect of buoyancy is negligible, the correlation does not reduce to the condition for forced laminar flow. The correlation by Huang et al. [32] includes the non-dimensional distance from the inlet. Another way to capture the enhancement due to mixed convection over forced laminar flow is to define an enhancement factor using the fully developed forced laminar convection Nusselt number as a baseline case for comparison. This is similar to the works by Jackson and Fewster [14] and by Swanson and Catton [15] for turbulent mixed convection in an opposed flow. Swanson and Catton presented an enhancement factor for the mixed convection regime in between two vertical heated surfaces. In the present study, the correlation by Wong and Hochreiter [25] is chosen as the baseline correlation for calculating the heat transfer enhancement factor. The latter is expressed as a function of Ri. Following Swanson and Catton's [15] work, the Nusselt number Nu for mixed convection can be written in terms of Nu_0, the baseline case using the Wong and Hochreiter correlation. The correlation coefficients have been calculated using the present RBHT data as shown in Figure 6. The final form of the new correlation is given below

$$\frac{Nu}{Nu_0} = 1 + 2.55\left[ln\left(\frac{Gr}{Re^2}+1\right)\right]^{0.89} \tag{7}$$

Figure 6 also shows the variation of the enhancement factor developed by Swanson and Catton [15]. The enhancement factor is significantly higher for the present work as compared to that reported by Swanson and Catton. Predicted Nusselt numbers using these values are compared with the experimental data in Figure 7. The observed mean absolute percentage error is 12.5% and root mean square of percentage error is 15.7 %.

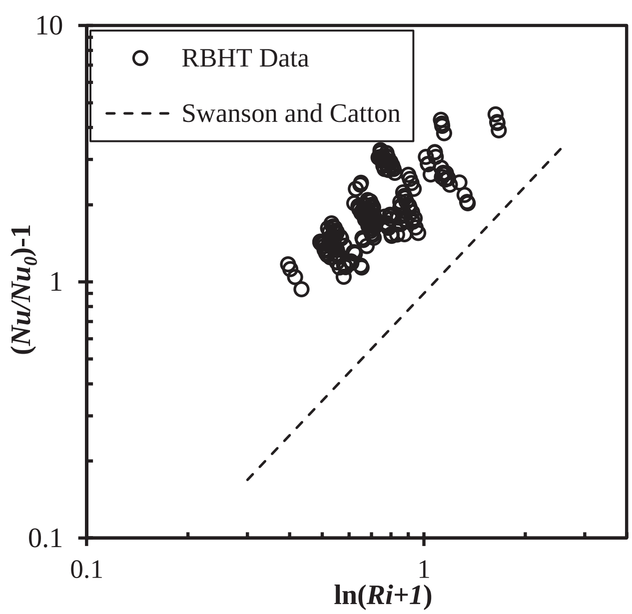


Figure 6. Variation of the Heat Transfer Enhancement Factor with the Richardson Number

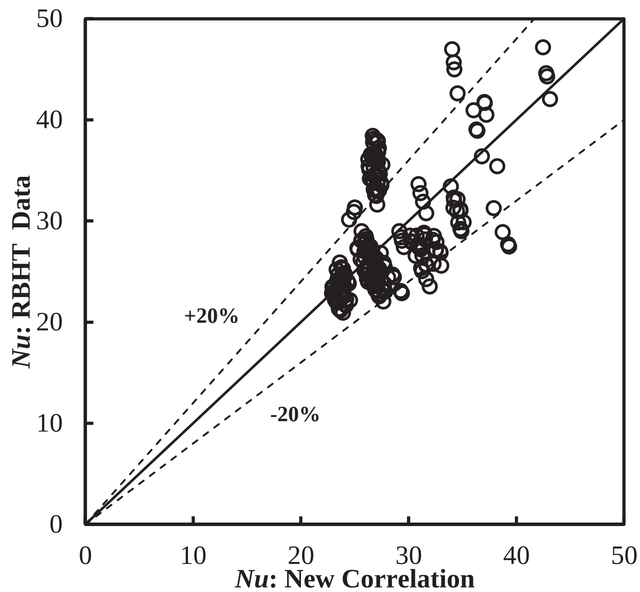


Figure 7. Comparison of the Experimental Data with the Predicted Results Using the Correlated Enhancement Factor

4. CONCLUSION

Steady-state level swell tests conducted at the RBHT test facility exhibited the mixed laminar convection heat transfer regime in the lower portion of the test section. Using the steady state level swell data, the Nusselt number for mixed laminar flow is determined. It is found that the Nusselt number increases with elevation in the rod bundle due to increase in the Richardson number over the range of inlet Reynolds number explored. The local maxima were observed to occur at Grids 2 and 3. The Nusselt number did not vary with the inlet Reynolds number near the inlet of the rod bundle. However, the Nusselt number varied at the higher elevations as the local Richardson number differed due to an increase in the local heat flux with elevation. The Nusselt numbers based on the RBHT data were found to be significantly higher than the fully developed, single phase laminar flow Nusselt number for rod bundles. An enhancement factor is developed using the fully developed forced laminar to account for the effect of mixed convection.

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REFERENCES

1. L. Maudou, G.H. Choueiri, and S. Tavoularis, “An Experimental Study of Mixed Convection in Vertical, Open-Ended, Concentric and Eccentric Annular Channels,” *Journal of Heat Transfer*, **135**, pp. 072502-1-9 (2013).
2. E.M. Sparrow, and J.I. Gregg, "Buoyancy Effects in Forced Convection Flow and Heat Transfer," *ASME J Appl Mech,* **81**, pp. 133–134 (1959).
3. J.R. Lloyd, and E.M. Sparrow, "Combined Forced and Free Convection Flow on Vertical Surfaces," *Int. Journal of Heat Mass Transfer*, **13**, pp. 434–43 (1970).
4. G. Wilks, "Combined Forced and Free Convection Flow on Vertical Surfaces," *Int. Journal of Heat Mass Transfer*, **16**, pp. 1958–1964 (1973).

5. P.H. Oosthuizen, and M. Bassey, "An Experimental Study of Combined Forced and Free Convection Heat Transfer from Flat Plates to Air at Low Reynolds Numbers," *Journal of Heat Transfer*, **95**, pp. 120–121 (1973).
6. N. Ramachandran, B.F. Armaly, and T.S. Chen, "Measurements and Predictions of Laminar Mixed Convection Flow Adjacent to a Vertical Surface," *Journal of Heat Transfer*, **107**, pp. 636–641 (1985).
7. W. Aung, and G. Worku, ''Theory of Fully Developed, Combined Convection Including Flow Reversal,'' *Journal of Heat Transfer*, 108 (2), pp. 485–488 (1986).
8. A. Barletta, and E. Zanchini, "On the Choice of the Reference Temperature for Fully-Developed Mixed Convection in a Vertical Channel," *Int. Journal of Heat Mass Transfer,* **42**, pp. 3169-3181 (1999).
9. K. Boulama, and N. Galanis, "Analytical Solution for Fully Developed Mixed Convection between Parallel Vertical Plates with Heat and Mass Transfer," *Journal of Heat Transfer*, **126 (3)**, pp. 381-388 (2004).
10. G. Desrayaud, and G. Lauriat, "Flow Reversal of Laminar Mixed Convection in the Entry Region of Symmetrically Heated, Vertical Plate Channels," *Int. Journal of Thermal Sciences*, **48**, pp 2036-2045 (2009).
11. C. Gau, K. A. Yih, and W. Aung. "Measurements of Heat Transfer and Flow Structure in Heated Vertical Channels," *Journal of Thermophysics and Heat Transfer*, **6** (4), pp. 707-712 (1992).
12. S. Habchi, and S. Acharya, "Laminar Mixed Convection in a Partially Blocked, Vertical Channel," *Int. Journal of Heat and Mass Transfer*, **29** (11), pp. 1711-1722 (1986).
13. J.D. Jackson, M.A. Cotton, and B.P. Axcell, "Studies of Mixed Convection in Vertical Tubes," *Int. Journal of Heat and Fluid Flow*, **10** (1), pp. 2-15 (1989).
14. J.D. Jackson, and J. Fewster, "Enhancement of Turbulent Heat Transfer due to Buoyancy for Downward flow of Water in Vertical Tubes," *Heat Transfer and Turbulent Buoyant Convection*, ed., D.B. Spalding, pp. 759 (1977).
15. L.W. Swanson, and I. Catton, "Enhanced Heat Transfer due to Secondary Flows in Mixed Turbulent Convection," Journal of Heat Transfer, **109**, pp. 943-946 (1987).
16. Mohammed A. H., "Laminar Mixed Convection Heat Transfer in a Vertical Circular Tube Under Buoyancy-Assisted and Opposed Flows," *Energy conversion and management*, **49**, pp. 2006-2015 (2008).
17. D. Maitra, and K. S. Raju, "Combined Free and Forced Convection Laminar Heat Transfer in a Vertical Annulus," *Journal of Heat Transfer,* 97(1), pp. 135-137 (1975).
18. H.K. Dawood, H.A. Mohammed, N.A.C. Sidik, K.M. Munisamy, and M.A. Wahid, "Forced, Natural and Mixed-Convection Heat Transfer and Fluid Flow in Annulus: A Review," *Int. Communications in Heat and Mass Transfer*, **62**, pp. 45-57 (2015).
19. J. W. Yang, "Analysis of Combined Convection Heat Transfer in Infinite Rod Arrays," *Proceedings of Int. Heat Transfer Conference*, **1**, Toronto, Canada, pp. 49-54 (1978).
20. V. Iannello, K.Y. Suh, and N.E. Todreas, "Mixed Convection Friction Factors and Nusselt Numbers in Vertical Annular and Subchannel Geometries", *Int. Journal of Heat and Mass Transfer*, **31**, No. 10, pp. 2175-2189, 1988
21. K. Y. Suh, N.E. Todreas, and W.M. Rohsenow, "Mixed Convective Low Flow Pressure Drop in Vertical Rod Assemblies: I-Predictive Model and Design Correlation" *Journal of Heat Transfer,* **111**(4), pp. 956-965 (1989).
22. S.H. Kim, and M. S. El-Genk, "Heat Transfer Experiments for Low Flow of Water in Rod Bundles," *Int. Journal of Heat and Mass Transfer*, **32** (7), pp. 1321-1336, (1989).
23. M.S. El-Genk, B. Su, and Z. Guo, " Experimental Studies of Forced, Combined and Natural Convection of Water in Vertical Nine-Rod Bundles with Square Lattice," *Int. Journal of Heat and Mass Transfer*, **36 (**9), pp. 2359-2374 (1993).

24. J.H. Kim, "Heat Transfer in Longitudinal Laminar Flow Along Cylinders in Square Array," in *Fluid Flow and Heat Transfer Over Rod or Tube Bundles*, American Society of Mechanical Engineers, New York, pp. 155-161 (1979)
25. S. Wong, and L.E. Hochreiter, "Analysis of the FLECHT SEASET Unblocked Bundle Steam Cooling and Boiloff Tests," *Tech. Report*, NUREG/CR-1533, United States Nuclear Regulatory Commission (1981).
26. G. Venugopal, C. Balaji, and S. P. Venkateshan, "A Correlation for Laminar Mixed Convection from Vertical Plates Using Transient Experiments," *Heat and Mass Transfer*, **44**, pp. 1417-1425, (2008).
27. Y. Sudo, M. Kaminaga, and K. Minazoe, "Experimental Study on the Effects of Channel Gap Size on Mixed Convection Heat Transfer Characteristics in Vertical Rectangular Channels Heated from Both Sides," *Nuclear Engineering and Design*, **120** (2–3), pp. 135-146(1990).
28. E. R. Rosal, T.F. Lin, I.S. McClellan, and R. C. Brewer, "Rod Bundle Heat Transfer Test Facility Description," *Tech. Report*, NUREG/CR-6976, United States Nuclear Regulatory Commission (2010).
29. L. E. Hochreiter, F. B. Cheung, T. F. Lin, J. P. Spring, S. Ergun, A. Sridharan, A. Ireland, and E. R. Rosal, "RBHT Reflood Heat Transfer Experiments Data and Analysis," *Tech. Report*, NUREG/CR-6980, United States Nuclear Regulatory Commission (2012).
30. L.E. Hochreiter, F. B. Cheung, T. F. Lin, and D.J. Miller, "Rod Bundle Heat Transfer Facility Two-phase Mixture Level Swell and Uncovery Experiments Data Report," *Tech. Report*, NUREG-XX, United States Nuclear Regulatory Commission (XXXX).
31. Welter K.B., Kelly J.M. and Bajorek S.M., "Assessment of TRACE Code Using Rod Bundle Heat Transfer Mixture Level-Swell Tests," *Proceedings of 14th International Conference on Nuclear Engineering*, Miami, FL, USA, pp. 785-794 (2006)
32. T.M. Huang, C. Gau, and W. Aung, "Mixed Convection Flow and Heat Transfer in a Heated Vertical Convergent Channel," *Int. Journal of Heat and Mass Transfer*, **38**, No. 13, pp. 2445-2456, (1995).
33. J. Marek, and K. Rehme, "Heat Transfer in Smooth and Roughened Rod Bundles Near Spacer Grids," *ASME Winter Annual Meeting*, December 2–7, pp. 163–170, (1979).
34. M. V. Holloway, D. E. Beasley, and M. E. Conner, "Single-Phase Convective Heat Transfer in Rod Bundles", *Nuclear Engineering and Design*, **238** (4), pp. 848-858 (2008).
35. D.J. Miller, F.B. Cheung, and S.M. Bajorek, "On the Development of a Grid-Enhanced Single-Phase Convective Heat Transfer Correlation," *Nuclear Engineering and Design*, **264**, pp. 56-60 (2013).
36. Y.A. Cengel and J.M. Cimbala, "*Fluid Mechanics, Fundamentals and Application*," pp. 352-353, third edition, McGraw Hills (2014).
37. N. Gnanasekaran, and C. Balaji, "A Correlation for Nusselt Number Under Turbulent Mixed Convection Using Transient Heat Transfer Experiments," *Proceedings of 14th International Heat Transfer Conference* (IHTC 14), 22428, Washington, DC, USA, August (2010).