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Heat transfer experiments for low flow of water in rod bundles

SUNG-HO KIM and MOHAMED S. EL-GENK
Department of Chemical and Nuclear Engineering, University of New Mexico, Albuquerque, NM 87131, U.S.A.

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Abstract—Heat transfer experiments are conducted for fully developed forced and natural flows of water through triangularly arrayed, seven uniformly heated rod bundles with $P/D = 1.25$, 1.38 and 1.5. For forced circulation experiments, Re ranges from 80 to 50 000 and Pr from 3 to 8.5; while in natural circulation, Re varies from 260 to 2000, and Ra_q from 8×10^6 to 2.5×10^8. The forced flow data fall into two basic flow regimes: turbulent and laminar flow. At the transition between these regimes, Re_T, which varies from 2200 for $P/D = 1.25$ to 5500 for $P/D = 1.5$, increases linearly with P/D. The turbulent heat transfer data is in good agreement ($\pm 15\%$) with Weisman's correlation, designed for fully developed turbulent flow in rod bundles at $Re > 25\,000$. However, the laminar flow data shows the dependency of Nu on Re to be weaker than for turbulent flow. Natural circulation data indicates that rod spacing insignificantly affects heat transfer; for $P/D = 1.38$ and 1.5, Nu is correlated as $Nu = 0.272 Ra_q^{0.25}$

INTRODUCTION

THE STUDY of convective heat transfer in rod bundles at low Reynolds number ($Re < 10\,000$) is important for many engineering applications including: the design of heat exchangers for low grade energy, the coolability of nuclear reactors following a loss-of-core cooling accident, the decay heat removal from spent fuel storage tanks, and steam generators in nuclear power plants [1]. In addition, the heat transfer in rod bundles at low Reynolds numbers is also essential to the design and operation of open-pool type research reactors, such as the Annular Core Research Reactor (ACRR) at Sandia National Laboratories (SNL) [2] and TRIGA type reactors. These reactors are cooled by natural convection of water at Reynolds numbers ranging from 3000 to 6000, and Rayleigh numbers, Ra_q, up to 2.1×10^7. Because heat transfer correlations for water flow in rod bundles at these low Reynolds numbers are not available, turbulent flow correlations, based on data for $Re > 10\,000$, have been used to predict the heat transfer coefficient in low Reynolds number applications. However, the validity of extrapolating turbulent convection correlations to low Reynolds number flow has never been quantified before this work.

Although extensive experimental studies have focused on collecting and/or correlating heat transfer data in rod bundles with either a square or triangular lattice, most of the data was limited to flow conditions with high Reynolds numbers, in excess of 6000 [3–7]. Although few results have been reported at lower Reynolds numbers, some studies do exist. For example, Kalinin *et al.* [8] correlated heat transfer data for forced flow of water in triangularly arrayed 7-rod bundles of $1.2 \leqslant P/D \leqslant 1.5$ at $2000 < Re < 70\,000$, and Inayatov [9] reported the heat transfer data in a triangularly arrayed nineteen-rod bundle with $P/D = 1.22$ at $1000 < Re < 12\,000$. In these investigations, because only the central rod was heated, an internal circulation due to buoyancy at low flow could have affected the heat transfer process in the test section; consequently, the applicability of the results for fully heated rod bundles may be limited. Recently, experimental studies of natural convection of water in 7- and 21-rod bundles were conducted by Gruszczynski and Viskanta [10] and Hallinan and Viskanta [11]. In addition to being limited to a narrow range of Reynolds numbers ($70 < Re < 500$) and low surface heat flux (7.84 kW m^{-2}), these studies used an isothermal heated wall rather than an isoflux heated wall, which is more common in rod bundle experiments. In theoretical studies of fully developed laminar flow in rod bundles, Dwyer and Berry [12] showed that although the Nusselt number for an isothermal boundary condition is slightly higher than that for an isoflux condition for small P/D values, the Nusselt numbers were essentially the same for both boundary conditions when $P/D > 1.5$. Additionally, the heat transfer correlation by Inayatov [7] for $6000 < Re < 10^6$ showed no definite difference in heat transfer in rod bundles between isothermal and isoflux conditions. Although theoretical studies of heat transfer in rod bundles at low Reynolds numbers have also been reported for fully developed forced laminar flows [12–15] and for mixed forced and natural laminar flows [16–19], the results have never been confirmed by experimental data. In general, there have been numerous studies of heat transfer in rod bundles, but

NOMENCLATURE

C_p	specific heat [J kg^{-1} K^{-1}]
D_e	hydraulic equivalent diameter (4 × cross-sectional flow area/wetted perimeter) [m]
D_{eh}	heated equivalent diameter (4 × cross-sectional flow area/heated perimeter) [m]
g	acceleration of gravity
Gr_q	Grashof number based on heat flux, $g\beta q'' D_{eh}^4/kv^2$
$Gr_{q,D}$	Grashof number based on heat flux and diameter, $g\beta q'' D^4/kv^2$
h	heat transfer coefficient [W m^{-2} K^{-1}]
k	thermal conductivity [W m^{-1} K^{-1}]
l_h	hydraulic entry length [m]
Nu	Nusselt number, $q'' D_{eh}/k(T_w - T_b)$
Pr	Prandtl number, $C_p\mu/k$
P/D	pitch-to-diameter ratio
q''	surface heat flux [W m^{-2}]
Ra_q	Rayleigh number, $Gr_q Pr$
$Ra_{q,D}$	Rayleigh number based on heated diameter, $Gr_{q,D} Pr$
Re	Reynolds number, $\rho v D_{eh}/\mu$
Re_D	Reynolds number based on heated diameter, $\rho v D/\mu$
Ri	Richardson number, Gr_q/Re^2
T_b	mean bulk temperature of water [K]
T_w	mean well temperature of heated section [K]
v	flow velocity [m s^{-1}]

Greek symbols

β	volumetric thermal expansion coefficient [K^{-1}]
μ	dynamic viscosity [kg m^{-1} s^{-1}]
v	kinematic viscosity [m^2 s^{-1}]
ρ	density [kg m^{-3}]

no heat transfer correlations and little experimental data are available for water at Reynolds numbers below 10 000.

To extend the heat transfer data base at low flow of water in rod bundles, heat transfer experiments were conducted for both forced and natural flows in triangularly arrayed 7-rod bundles. The heat transfer data was correlated and compared with fully developed turbulent flow correlations [5, 6, 8] to examine their applicability to low flow conditions.

Figure 1 shows the ranges of Reynolds number and P/D of the experimental work reported for forced and natural flow of water in triangularly arrayed rod bundles as well as those of the present study. As shown in this figure, the present experiments cover a wide range of Reynolds numbers and extend the heat transfer data base to Reynolds numbers as low as 80.

EXPERIMENTAL SETUP

Longitudinal views of the containment tank and the rod bundle test section are shown in Fig. 2. All seven rods in the bundle were electrically heated and connected in series to maximize the surface heat flux in the experiments. Electric power to the test section was supplied by a d.c., Sel.Rex Model B2D80A generator with a maximum power output of 200 kW (2000 A and 100 V). In the experiments, demineralized water with an electric resistivity in excess of 18 MΩ-cm, which corresponds to a salt content less than 20 p.p.m., was used to minimize the deposition of minerals on the surface of the heated rods. In the forced flow experiments, the water was circulated through the test section by two centrifugal pumps, which were connected in series, and provided a maximum flow rate of 2.2 kg s^{-1}.

The shrouded rod bundle was supported at the bottom by a structure which consisted of two concentric plexiglas cylinders. Each cylinder had eight longitudinal openings (17 cm long and 2.54 cm wide) for controlling the flow circulation during forced and natural circulation experiments. During forced flow experiments the openings in the outer cylinder were closed, but were left open during the natural circulation experiments. In natural circulation experiments, the water from the tank flowed radially through the openings in the lower support structure and then circulated through the test section; no flow was allowed to circulate through the external loop. However, during the forced circulation experiments, the openings in the outer cylinder were closed and the flow was allowed to circulate through the external loop, where the waste heat was rejected through a 500 kW capacity heat exchanger into a chilled water line. The inlet water temperature to the test section was adjusted using a 1.5 kW auxiliary immersion heater which was installed in the loop between the heat exchanger and the discharge line from the containment tank. The containment tank, which housed the test section, was a plexiglas cylinder, 61 cm o.d. and 244 cm long, with top and bottom cover plates. Before the experiments, the containment tank was filled with demineralized water and secured on its base by four steel tie rods.

As shown in Fig. 2, the test section consisted of seven electrically heated rod bundles enclosed in a thermally insulated hexagonal plexiglas shroud. To provide the same flow area per heated rod, the side

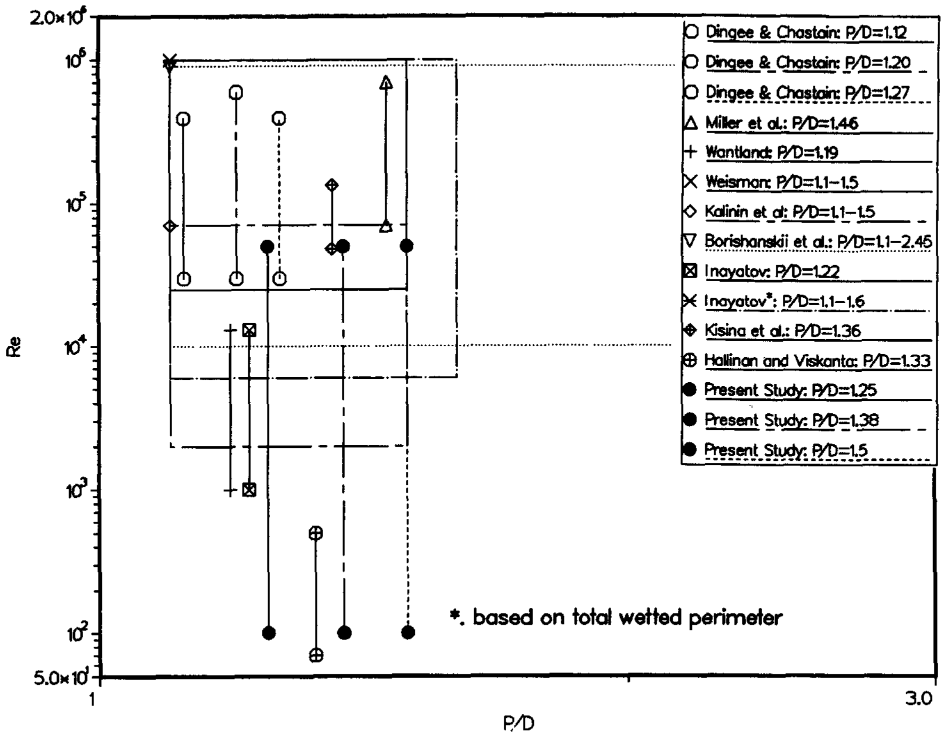

FIG. 1. Heat transfer data base for water flow in triangularly arrayed rod bundles.

length of the hexagonal shroud's inner wall was taken as equal to $\sqrt{((7/3)P)}$, and the spacing between the rods was maintained using spacer plates above and below the heated section. Each rod consisted of a type 304 stainless steel tube (1.27 cm in diameter, 90.44 cm long and 0.089 cm in wall thickness) which was welded at the bottom and the top to unheated brass rods, except for the three instrumented rods which had a brass tube at the top. The unheated section at the bottom of the test section (51.3 cm) served as an entry length for the hydrodynamic development of the flow [20], while the unheated section at the top of the heated section minimized the exit effects on the heat transfer in the test section.

INSTRUMENTATION AND MEASUREMENT UNCERTAINTY

As mentioned earlier, three out of the seven rods in the bundle (the central and two peripheral rods) were instrumented with type-K thermocouples to measure the inner surface temperature of the heated wall at different axial locations. A specially designed temperature probe, with eight chromel–alumel thermocouples, was inserted inside these rods. The temperature probe consisted of a Teflon rod, 0.985 cm o.d., with eight radial holes (each was 0.4 cm in diameter and 0.4 cm in depth). These holes were placed axially 11.63 cm apart, and each hole was rotated 45° counterclockwise from the location above it to keep the thermocouple leads separated. At each measuring location, the thermocouple junction was housed by a spring loaded Teflon plunger. The pressure exerted by the compression spring kept the thermocouple junction touching the inner surface of the heated tube during the experiments. The leads of thermocouples were inserted in longitudinal grooves (each was 0.1 cm in width and 0.1 cm in depth) starting at the measuring point and terminating at the top end of the probe. The thermocouple leads were then exited through the top of brass tubes of the rod and connected to the data acquisition system. Because the temperature probe measured the temperatures at the inner surface of the heated wall, the surface temperatures were taken as equal to the value measured by the thermocouples plus the temperature drop, due to conduction in the heated wall (<0.5 K) and the contact resistance between the thermocouple junction and the heated wall (<0.1 K). Details on the experimental setup and the design of the temperature probe can be found elsewhere [21].

The water temperature in the test loop was monitored at eight locations: two at the exit and four at the inlet of the test section and two in the immersion heater. Thermocouples measuring the inlet water temperature were placed radially 90° apart and the inlet temperature was taken as the average of the measured temperatures by the four thermocouples at the inlet of the test section. The variation between temperatures measured by these thermocouples ranged up to 1 K. The bulk temperature of the water at the bottom of the heated section was equal to that at the inlet of the

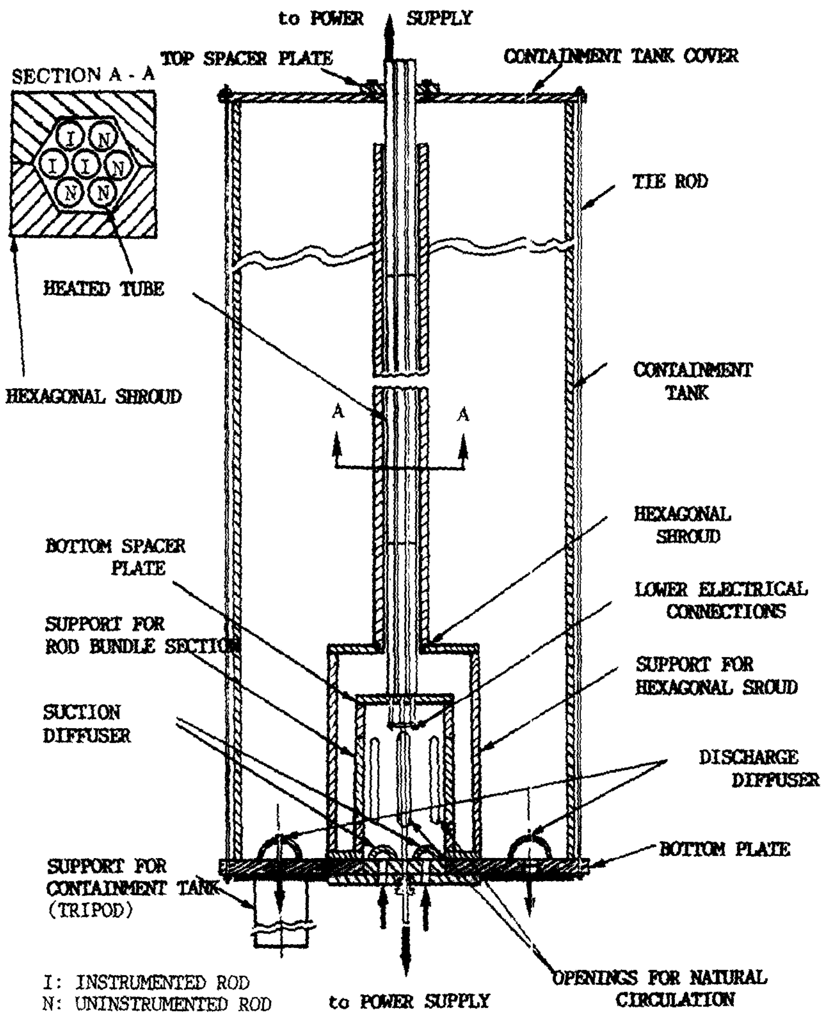

FIG. 2. Longitudinal views of the containment tank and the rod bundle test section.

test section plus the temperature rise due to the heat deposited in the brass section at the bottom. In addition, the water temperature at the exit of the heated section was taken as the arithmetic average of the two thermocouples' readings at this location. The water bulk temperature at the different axial locations in the heated section was then calculated from the overall heat balance in the test section using the measured flow rate, operating power, and the water bulk temperature at the bottom of the heated section.

The flow measurements in the experiments were carried out using a total of five flowmeters covering a wide range of flow rates extending from 0.0019 to 2.2 kg s^{-1}. Various flow rates, however, were measured using appropriate flowmeters, thus for flow rates from 0.126 to 2.2 kg s^{-1}, two Signet MK515 turbine flowmeters were connected in parallel; for flow rates from 0.0315 to 0.189 kg s^{-1}, a Flow Technology FT-6-8N5-LJ turbine flowmeter was used; for flow rates from 0.019 to 0.126 kg s^{-1}, a Dwyer rotameter was used; and for flow rates from 0.0019 to 0.019 kg s^{-1}, a Gilmont No. 5 micrometer controlled rotameter was used. All flowmeters were precalibrated before each experiment. In the natural flow experiments, only the overall heat balance across the test section was used to determine the water flow rate. To verify the accuracy of the heat balance method, the flow measurements during the forced flow experiments were compared with those obtained from the overall heat balance in the test section. At high flow rates, the difference between the flow rates determined using the heat balance method and the measured values was as high as 35%. This large difference between the two methods was the result of two major uncertainties. Because the temperature rise across the test section at high flow rate was only a few degrees (1–2 K), the percentage error based on the measured uncertainty of the thermocouples increased. Also, at high flow rates, the error in the water temperature measured at the outlet of the test section increased because it was difficult to keep the thermocouple junctions in the small flow gap between rods and prevent them from touching the surface of the heated section.

Conversely, the flow rate determined from the heat balance at low flow was within $\pm 5\%$ of the measured values. This result suggested that since the flow rate in the natural circulation experiments was low (<0.095 kg s^{-1}) and the temperature rise across the test section was large (4–26 K), the accuracy of the flow rate measurements using the heat balance method was expected to be within $\pm 5\%$.

The power to the test section was determined from the electric current and voltage measurement. As indicated in the Nomenclature, the Nusselt number in the heated test section was based on the measured difference between the wall and water bulk temperatures as well as the actual power produced in the stainless steel tubes (93.9% of electric power input). The uncertainties in the power and the flow rate measurements were approximately ± 0.63 and $\pm 5\%$, respectively. The uncertainty in the temperature measurement was about $\pm 2\%$. These uncertainties in the power, flow rate, and temperature measurements resulted in approximately $\pm 5.3\%$ uncertainty in Nusselt number, $\pm 5\%$ in Reynolds number, $\pm 0.3\%$ in Prandtl number, ± 1.8 to $\pm 2.3\%$ in Grashof number, and $\pm 10.3\%$ in Richardson number.

EXPERIMENTAL CONDUCT

Before the flow experiments, the uniformity of the surface heat flux along the heated section was examined. The test section was heated in air at low power, and the wall temperature at the different axial locations was measured as a function of time. Then, the rate of temperature rise was used to determine the energy deposited at these locations. The results indicated that the axial flux profile was quite uniform and the maximum variation between the energy depositions at the different axial locations was about 10%.

In the present experiments, the electric power was increased incrementally, beginning as low as 0.2 kW. At each power level, after the steady-state was reached, the data for flow rate, temperature, and power were collected and analyzed using a data acquisition system, which consisted of a HP-3497A data logger, a HP-7470A plotter, and an IBM-9000 CS computer. This procedure continued until the exit water temperature reached approximately 348 K. No data were collected at a higher temperature to avoid a boiling on the heated wall or a bowing of the heated rods due to the axial thermal expansion of the stainless steel tube. Also, above 348 K, dissolved air was released and the air bubbles were formed. These bubbles caused a large oscillation in the measured wall temperatures.

EXPERIMENTAL RESULTS

As mentioned earlier, the unheated section at the bottom of the heated rods was introduced to allow the flow to be hydrodynamically developed before entering the heated section. Using the expressions for hydrodynamic entry length in channels by White [20]:

$$l_h/D_e = 0.04Re_{D_e} + 0.5$$

the hydrodynamic entry lengths for the rod bundles were less than the unheated section at the bottom of the test section (51.3 cm), thus indicating that the flow in the experiments was hydrodynamically developed. However, the question of whether the flow in the experiments was thermally developed was examined based on the measured temperatures in the heated section. The local differentials between the heated wall and the bulk water temperatures at the eight axial locations corresponding to the locations of the wall thermocouples (4.5, 16.13, 27.76, 39.39, 51.02, 62.65, 74.28 and 85.91 cm from the bottom of the heated section) were used to determine the local Nusselt number. Figure 3 presents a typical profile of Nusselt numbers as a function of axial location for the rod bundle test section with $P/D = 1.38$. Similar results were also obtained for $P/D = 1.25$ and 1.5 [21]. As shown in Fig. 3, the axial variation of the local Nusselt number was insignificant, suggesting that the flow through the rod bundles was thermally developed. In this study, then, since the axial heat flux was uniform, the mean arithmetic values for the heated wall and the water bulk temperatures were used to evaluate the dimensionless heat transfer parameters. In addition, the physical properties of the water were evaluated at the arithmetic mean bulk temperature in the test section.

In these experiments, heat transfer data were collected for both forced and natural circulation. In the forced circulation experiments, the Reynolds numbers, Re, ranged from 80 to 50 000; the Rayleigh numbers, Ra_q, ranged from 8.5×10^6 to 4.5×10^{11}, and the Prandtl numbers, Pr, ranged from 3 to 8.5. In the natural circulation experiments, the Reynolds numbers varied from 260 to 2000 and the corresponding Rayleigh numbers ranged from 8×10^6 to 2.5×10^8, but the range of Prandtl numbers was similar to the range in the forced flow experiments. In these experiments, the mean wall temperature of the heated section ranged from 290 to 340 K; in contrast the mean bulk temperature of water varied between 285 and 328 K. In the natural flow experiments, the mean wall temperature and the mean bulk temperature of water varied from 300 to 340 K, and from 298 to 310 K, respectively.

Forced circulation results

Figures 4–6 plot the forced circulation data collected for the rod bundles of $P/D = 1.25$, 1.38 and 1.5, respectively. These figures indicate that regardless of the value of P/D, the forced circulation data can be classified into two basic flow regimes: turbulent flow and laminar flow. The Reynolds number at the transition from the turbulent to the laminar regime,

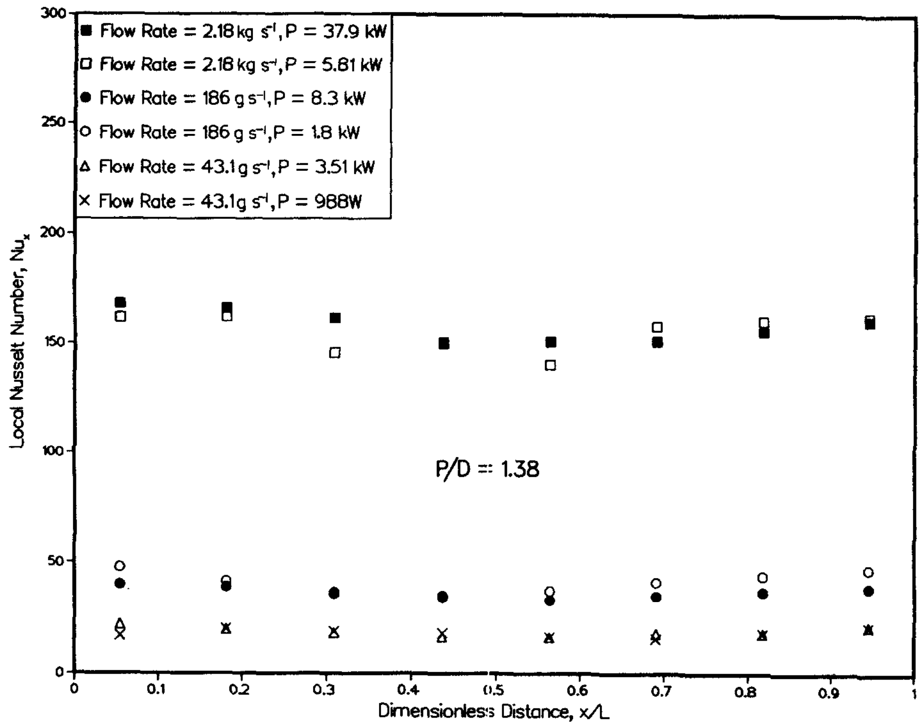

FIG. 3. Typical axial profile of Nusselt number for the rod bundle with $P/D = 1.38$.

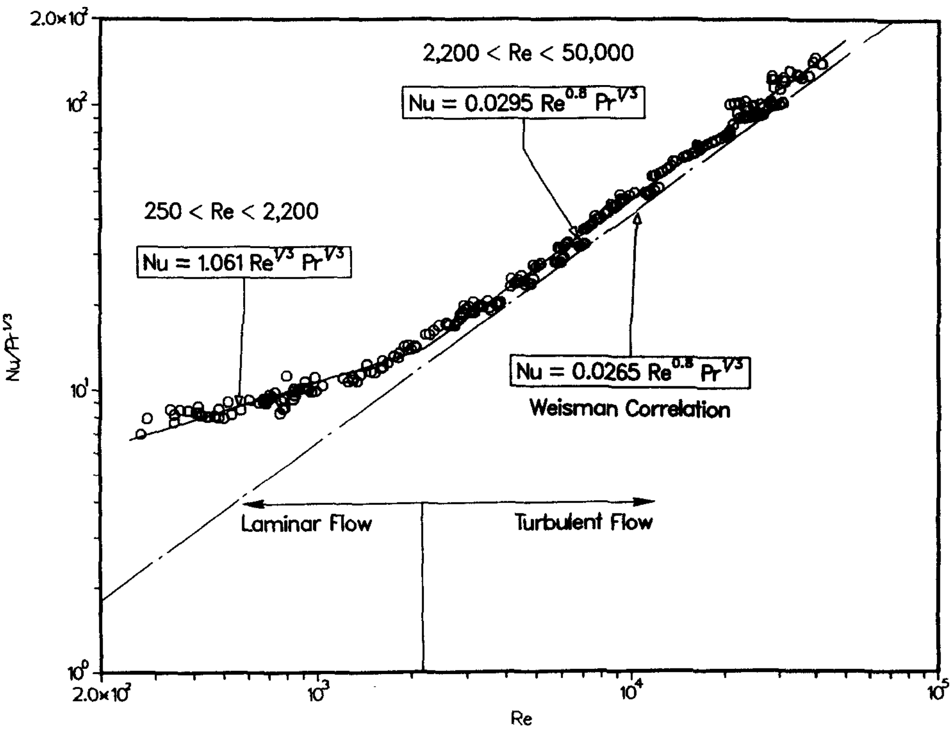

FIG. 4. Heat transfer data for forced flow in rod bundle with $P/D = 1.25$.

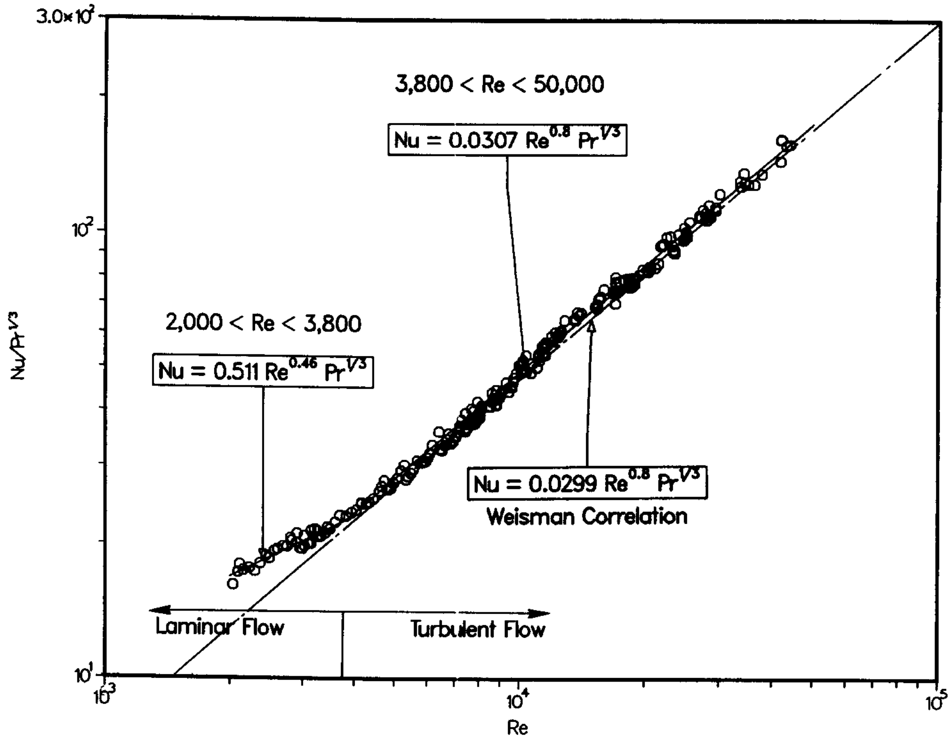

FIG. 5. Heat transfer data for forced flow in rod bundle with $P/D = 1.38$.

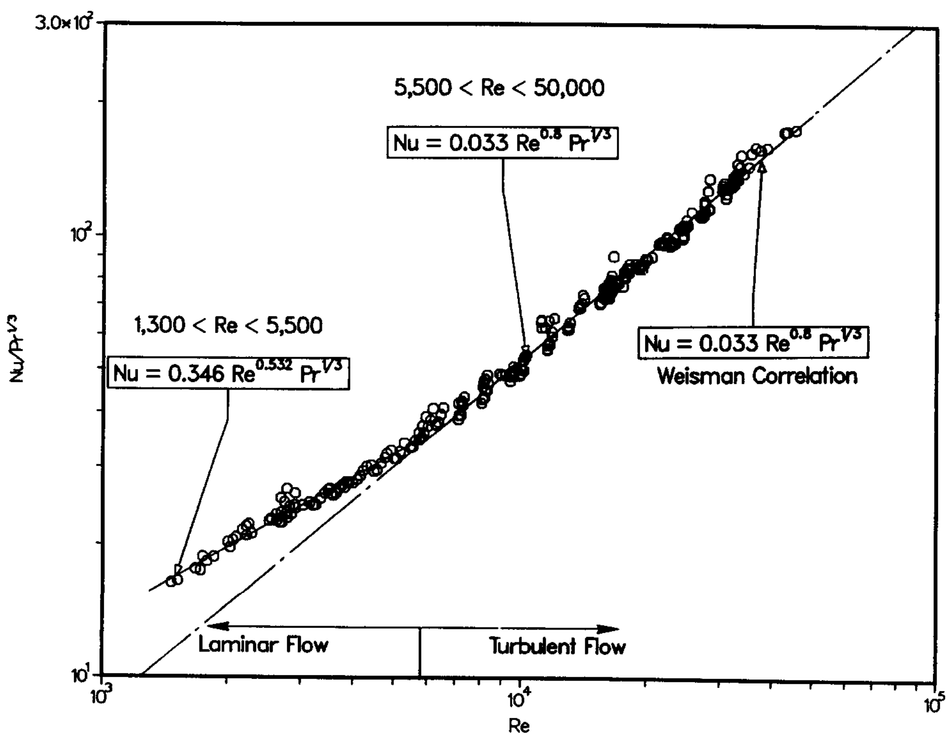

FIG. 6. Heat transfer data for forced flow in rod bundle with $P/D = 1.5$.

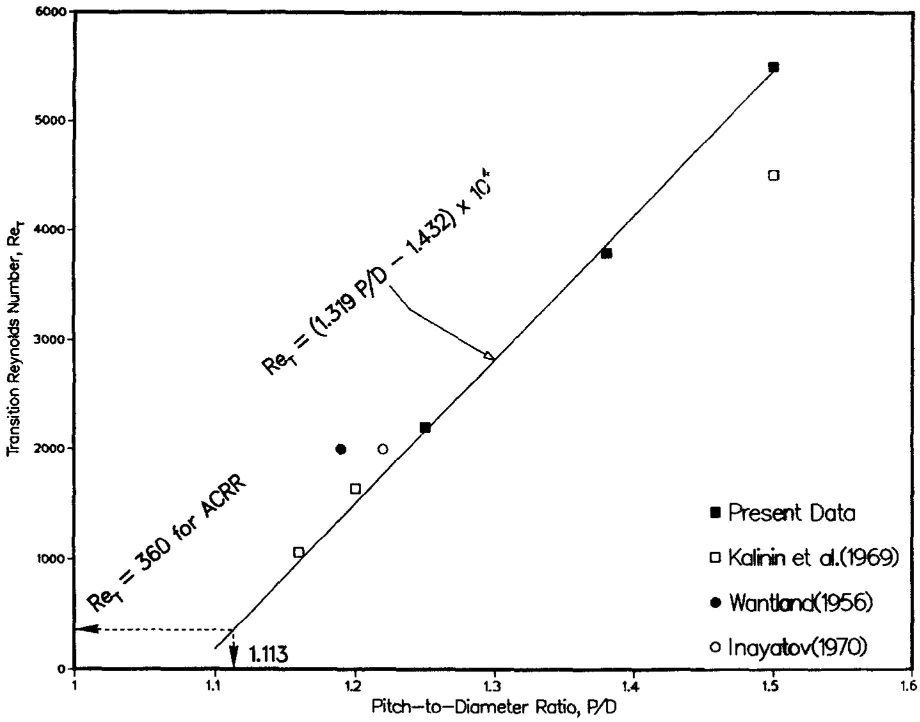

FIG. 7. Reynolds number for flow transition between turbulent and laminar regimes as a function of P/D.

referred to as Re_T, increased linearly with P/D as follows:

$$Re_T = (1.319P/D - 1.432) \times 10^4. \qquad (1)$$

Figure 7 compares equation (1) with the present data and with the data of Wantland [22], Kalinin *et al.* [8], and Inayatov [9]. Although equation (1) was developed based only on the present data, it was within $\pm 10\%$ of the data reported in the literature for Re_T, except for a few data points. For example, the value for Re_T given by Kalinin *et al.* [8] for $P/D = 1.5$ was about 43% higher and that by Wantland [22] for $P/D = 1.19$ was about 30% lower than equation (1).

Turbulent flow data

The turbulent flow data in Figs. 4–6 agreed with the fully developed turbulent flow heat transfer correlation in rod bundles by Weisman [5]. Other correlations were also reported in the literature for fully developed turbulent flow by Kalinin *et al.* [8], Borishanskii *et al.* [6] and Inayatov [7]. These correlations had the following form:

$$Nu = C\, Re^{0.8}\, Pr^{1/3} \qquad (2)$$

where the coefficient C depends on P/D and the type of rod lattice (triangular or square). Weisman's correlation for C in triangularly arrayed rod bundles was given as

$$C = 0.026P/D - 0.006 \quad \text{for} \quad 1.1 \leqslant P/D \leqslant 1.5 \qquad (3)$$

and Kalinin *et al.*'s correlation for C was given as

$$C = 0.032P/D - 0.0144 \quad \text{for} \quad 1.1 \leqslant P/D \leqslant 1.5. \qquad (4)$$

In Weisman's heat transfer correlation, the physical properties of water were evaluated at the mean bulk temperature; while in Kalinin *et al.*'s correlation, the properties of water were evaluated at the mean film temperature. For the purpose of comparison, the values of coefficient C given in equation (4) were modified by evaluating the physical properties of water at the mean bulk temperature. In Fig. 8, Kalinin *et al.*'s modified correlation was compared with Weisman's and with other correlations reported in the literature. As this figure shows, Weisman's and Kalinin *et al.*'s correlations for coefficient C provided the best comparison with the present data. The deviation between Weisman's correlation for the coefficient C and the present data was less than $\pm 11\%$ and that between Kalinin *et al.*'s correlation and the data was less than $\pm 11.6\%$.

Laminar flow data

As indicated in Fig. 9, in the laminar flow regime, where $Re < Re_T$ and the Richardson number, Ri, was less than 1.0, the Nusselt number values were independent of the Richardson number but increased with Reynolds number. The data in this regime was correlated as follows:

$$Nu = A\, Re^B\, Pr^{1/3}. \qquad (5)$$

In equation (5), coefficient A = 1.061, 0.511 and 0.346

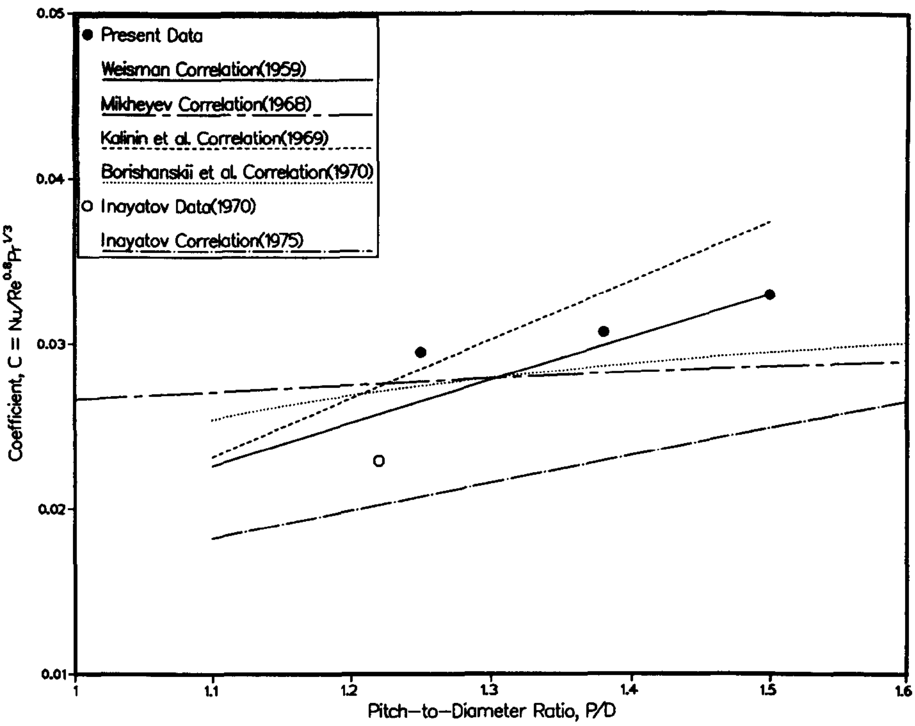

FIG. 8. Dependency of the coefficient C for turbulent convection in rod bundles as a function of P/D (equation (2)).

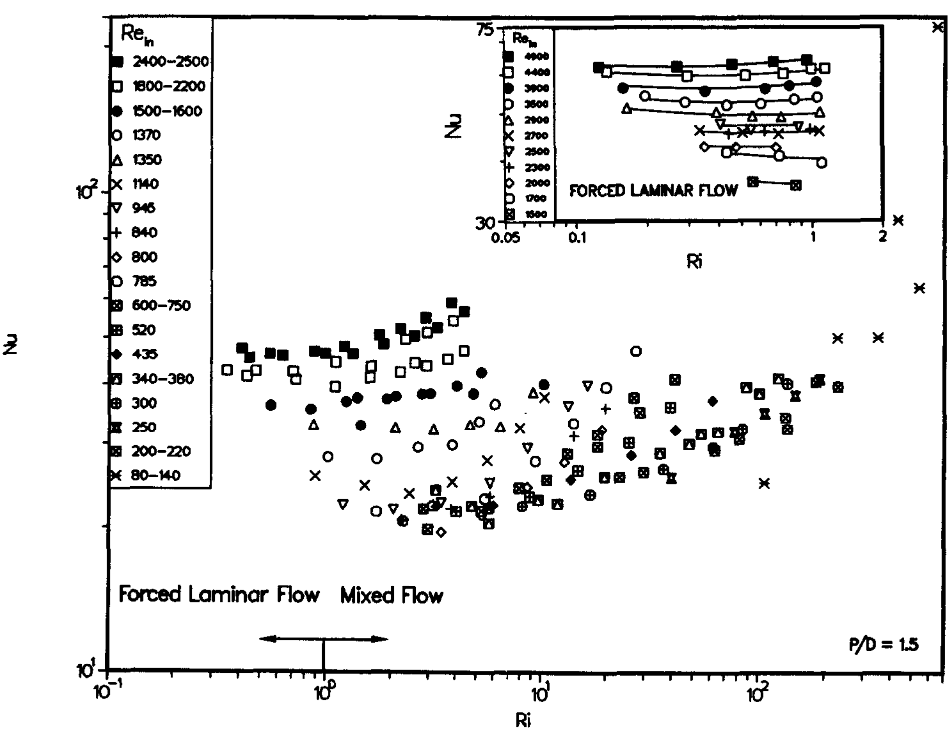

FIG. 9. Nusselt number for forced laminar and mixed flow as a function of Richardson number for $P/D = 1.5$.

for $P/D = 1.25$, 1.38 and 1.5, respectively, while exponent B increased linearly with P/D

$$B = 0.797P/D - 0.656. \quad (6)$$

This exponent, ranging from 0.34 to 0.532, is smaller than that for the Reynolds number in the turbulent flow regime. This small exponent suggests that the dependency of Nusselt number on Reynolds number in the forced laminar flow is weaker than that in the turbulent flow regime.

Mixed flow data

In the laminar flow regime, when the Richardson number was greater than unity, the Nusselt number not only depended on Reynolds number but also increased as the Richardson number increased (see Fig. 9). In this regime, the heat transfer data for $P/D = 1.38$ and 1.5 were correlated as follows:

$$Nu = a\,Ri^b\,Re^c \quad \text{when} \quad Ri \geqslant 1.0 \quad (7)$$

where a, b, c were 2.762, 0.163, 0.282 and 0.95, 0.25, 0.404 for $P/D = 1.38$ and 1.5, respectively. The deviation between the above correlation and the data was within ± 15 and $\pm 10\%$ for $P/D = 1.38$ and 1.5, respectively. The positive exponents of the Richardson number in equation (7) indicate that in the mixed flow, the heat transfer rate increases as the ratio of buoyancy to inertia forces increases. This finding agrees with the results of the theoretical studies of mixed convection in rod bundles by Iqbal *et al.* [16] and Yang [17, 18]. It is also important to note that the increase in the exponent of the Richardson number with P/D demonstrates the enhanced effect of buoyancy on heat transfer as the P/D ratio increases. Similarly, equation (7) shows that the dependency of the Nusselt number on the Reynolds number increased as P/D increased; this dependency is indicated by the increase in the value of the Reynolds number exponent with P/D.

Natural circulation results

Natural circulation experiments were also conducted in the rod bundles with $P/D = 1.38$ and 1.5. Since buoyancy was the primary driving force in these experiments, the inertia force, expressed in terms of Reynolds number, depended on the buoyancy force, expressed in terms of Rayleigh number. In Fig. 10, Re_D is plotted as a function of $Ra_{q,D}$ to examine the effect of buoyancy on flow circulation by natural convection. As this figure indicates, at low $Ra_{q,D}$ values ($< 10^7$), Re_D increased with $Ra_{q,D}$, regardless of the value of P/D. However, at higher $Ra_{q,D}$ values, the value of Re_D increased with P/D. The results suggest that at low heating rate, where the influence of buoyancy on the flow was limited to the liquid layer next to the heated wall, the P/D ratio had no effect on flow circulation.

Results also suggest that at high heating rate, where the effect of buoyancy extended across the flow channel, the flow resistance due to the spacing between rods suppressed the flow circulation as P/D decreased. This suppression explains whey Re_D increased faster with $Ra_{q,D}$ for $P/D = 1.5$ than for $P/D = 1.38$.

By contrast, the spacing between the heated rods insignificantly affected the heat transfer rate with natural circulation; the values of Nusselt number for both $P/D = 1.38$ and 1.5 were correlated in terms of Rayleigh number as follows:

$$Nu = 0.272\,Ra_q^{0.25} \quad \text{for } 8 \times 10^6 \leqslant Ra_q \leqslant 2.5 \times 10^8. \quad (8)$$

This correlation was compared with the experimental data in Fig. 11; the correlation was within $\pm 4\%$ of the data. In equation (8) the proportionality of Nu to Ra_q raised to the one-fourth power is similar to that reported in the literature for flow in tubes [23] and in annuli [24]. In Fig. 12, equation (8) was compared with the correlation reported for isothermally heated rod bundles [10, 11] to determine the effect of the heated boundary condition on the heat transfer by natural convection. Although the previous studies [9, 12] showed that for forced flow the type of the heated boundary had minimal effects on the heat transfer rate, results in Fig. 12 show that for natural flow, the Nusselt number increased faster with the Rayleigh number for the isothermal wall than for the isoflux wall.

Flow regime maps

To compare the different flow regimes identified in both the forced and natural circulation experiments, the present data were plotted on a flow regime map similar to that of Metais and Eckert [25], which was developed for fluid flow inside tubes (see Fig. 13). Metais and Eckert's map for rod bundles was reproduced by substituting the tube diameter by the heated equivalent diameter of the rod bundles. The Grashof number in Fig. 13 is based on the temperature differential between the heated wall and the water bulk, and the flow regimes indicated by the solid lines are those defined by Metais and Eckert [25].

As shown in Fig. 13, although Metais and Eckert's map has predicted the flow regimes for the forced turbulent flow data, it failed to accurately predict the flow regimes for low flow conditions. For example, the transition regime between turbulent and laminar flows was not detected in the present experiments, the transition occurred directly from turbulent to laminar flow. The data also shows that because of the cross flow between subchannels in the rod bundles, the boundary between the forced flow and the mixed flow regime occurred at lower Reynolds numbers than those suggested by Metais and Eckert [25]. Also, this figure shows that some of the mixed flow data fell into the forced laminar regime in Metais and Eckert's map and the natural circulation data fell into the mixed flow regime suggesting that a Metais and Eckert type map is inappropriate for describing the heat transfer regimes on rod bundles. Therefore, a flow regime map

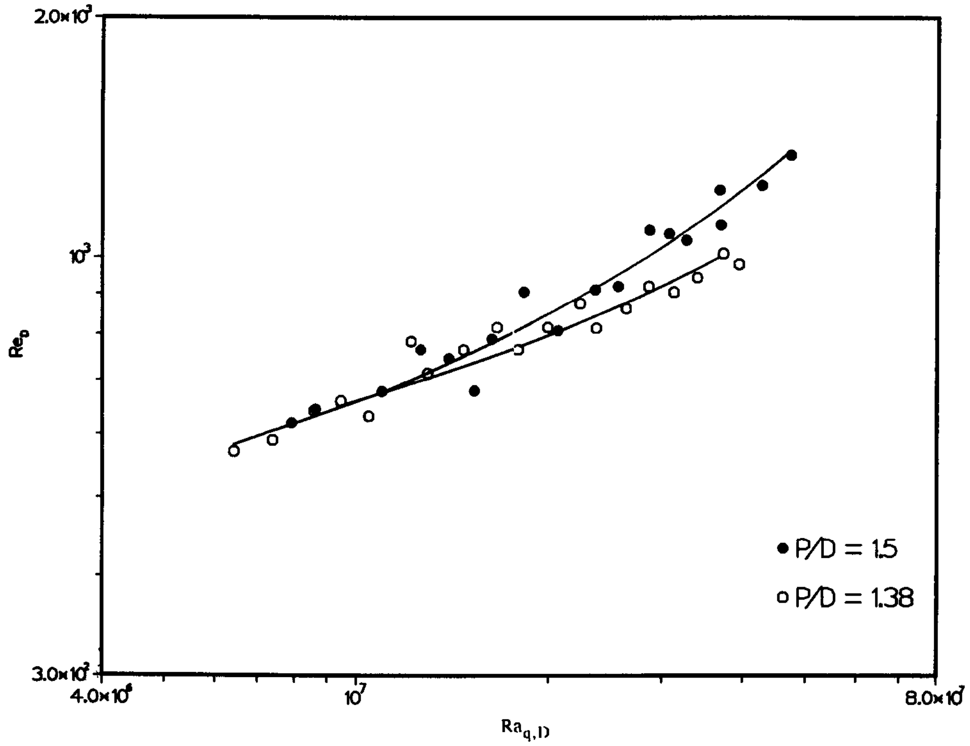

FIG. 10. Relation between $Ra_{q,D}$ and Re_D in natural circulation experiments for $P/D = 1.38$ and 1.5.

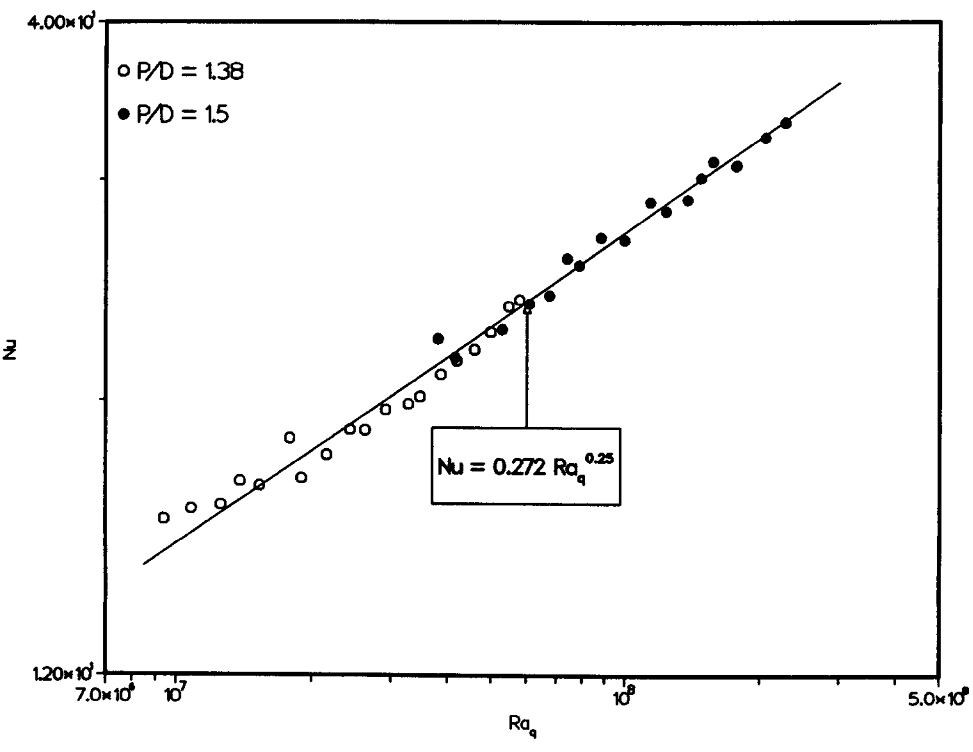

FIG. 11. Heat transfer data and correlation for natural circulation of water in rod bundles for $P/D = 1.38$ and 1.5.

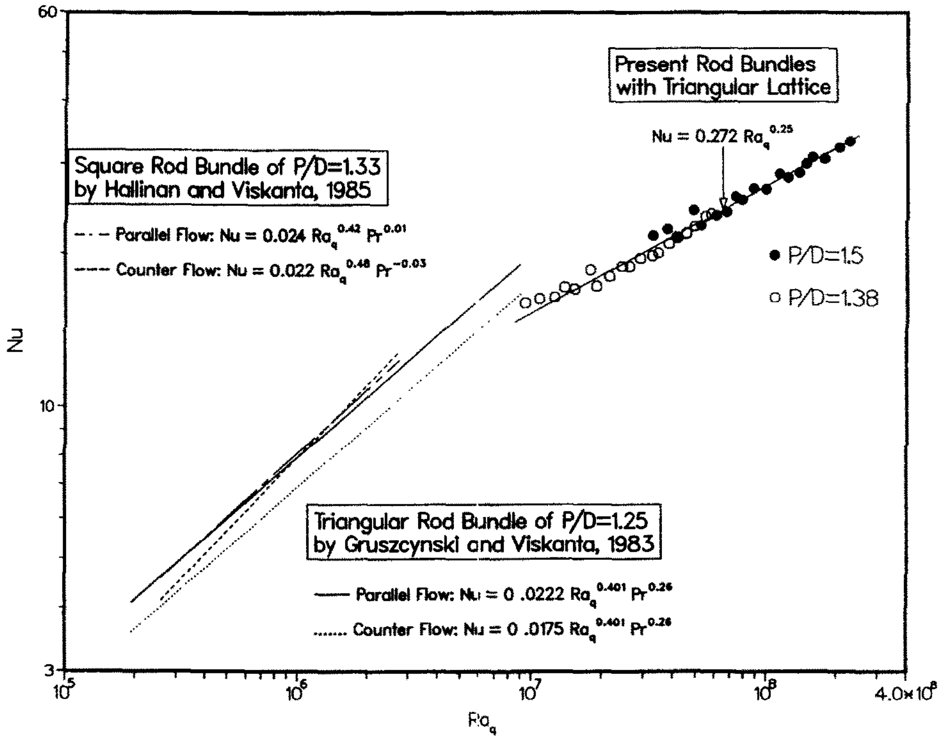

FIG. 12. Comparison of natural circulation data for isoflux and isothermal heated conditions.

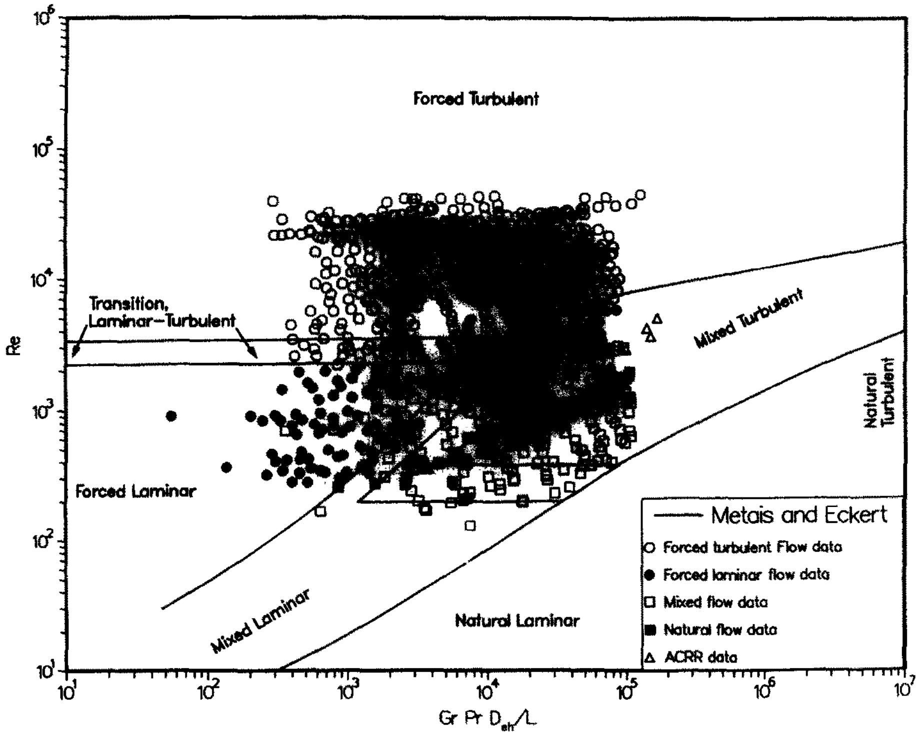

FIG. 13. Comparison of the present data with Metais and Eckert's flow regime map [24].

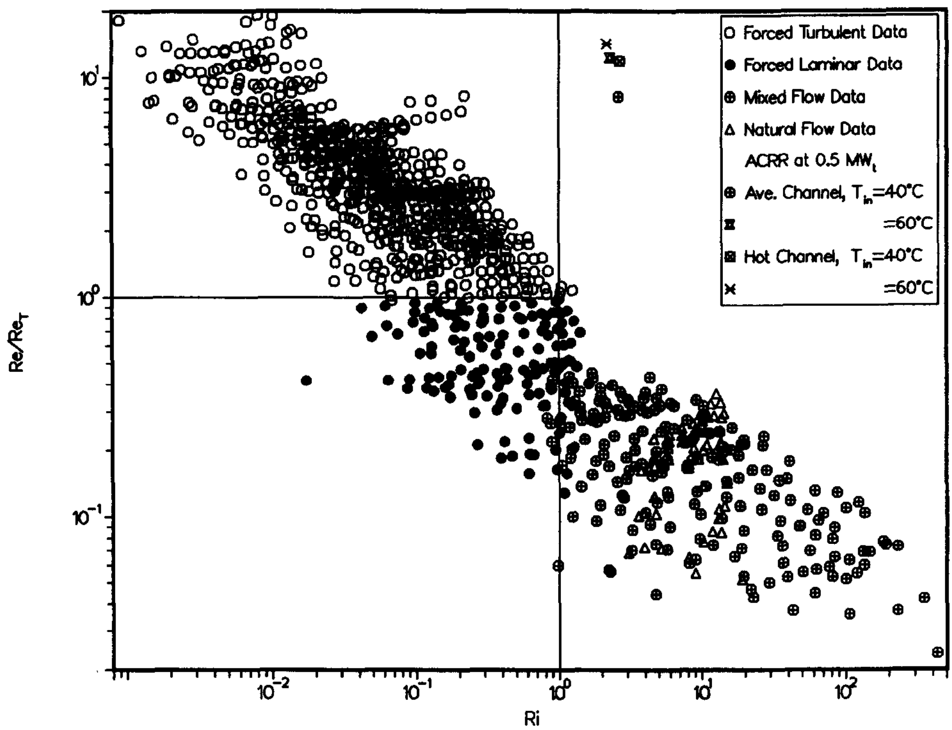

FIG. 14. A convective flow regime map in rod bundles developed in the present study.

for the rod bundles, based on the present data, was developed and presented in Fig. 14.

In this map, the values of Re/Re_T and Ri were employed to describe the effect of the inertia and buoyancy forces on the convective flows in rod bundles. The Re_T at the transition between forced turbulent and laminar flow regimes is given by equation (1). The ordinate in Fig. 14, represented by Re/Re_T, serves as an indicator to whether the forced flow for $Ri < 1$ is turbulent or laminar as follows: if $Re/Re_T < 1$, the convective flow regime is forced laminar; otherwise, it is forced turbulent flow. On the other hand, the Richardson number on the abscissa, serves to distinguish forced laminar convection data from mixed convection data for $Re/Re_T < 1.0$: if $Ri < 1$, the convective flow regime is forced laminar flow; otherwise, it is mixed flow. Because the Richardson number is the ratio of the buoyancy to inertia forces, the high Richardson number reflects the dominance of the buoyancy forces on the heat transfer process. In summary, Fig. 14 shows that when $Re/Re_T \geqslant 1.0$ and $Ri < 1.0$, the data were in the forced turbulent flow regime; when $Re/Re_T < 1.0$ and $Ri \leqslant 1.0$, the data were in the forced laminar flow regime; and when $Re/Re_T < 1.0$ and $Ri > 1.0$, the data were in the mixed flow regime. The natural circulation data, however, fell into the mixed flow regime similar to the case of Metais and Eckert's map.

Application to ACRR and TRIGA type reactors

As shown in Fig. 7, the Reynolds numbers at the transition from forced turbulent flow to laminar flow in the ACRR core is estimated by equation (1) to be 360, which is much lower than the Reynolds number range of the ACRR during normal operation ($3000 < Re < 5000$). However, since the flow through the ACRR core is primarily driven by the buoyancy forces, the flow will be a natural turbulent rather than a forced turbulent flow.

As shown in Fig. 14, the data for the ACRR at the operating power of 0.5 MW were also plotted for comparison. The data for the ACRR were generated by the ACRR thermal–hydraulic model, developed by Rao *et al.* [26]. When the natural flow correlation (equation (8)) was incorporated into the model, the model's predictions of the temperature differences across the core were in excellent agreement with the in-core measurements. An examination of this figure also reveals that although the flow in the ACRR is driven solely by the buoyancy forces, the heat transfer data during normal operating conditions are located in the upper right hand quarter of Fig. 14. This suggests that the heat transfer regime in the ACRR could be natural turbulent rather than forced turbulent, since the Richardson number in the ACRR is greater than unity.

However, since the Reynolds numbers in the ACRR are much higher than Re_T for the transition from the forced turbulent to the forced laminar flow, and the Richardson number is only slightly greater than unity (2–3), it could be argued that the flow and heat transfer in the ACRR core may also be characterized by the forced turbulent flow. To investigate this argument, the Nusselt numbers in the ACRR, calculated using

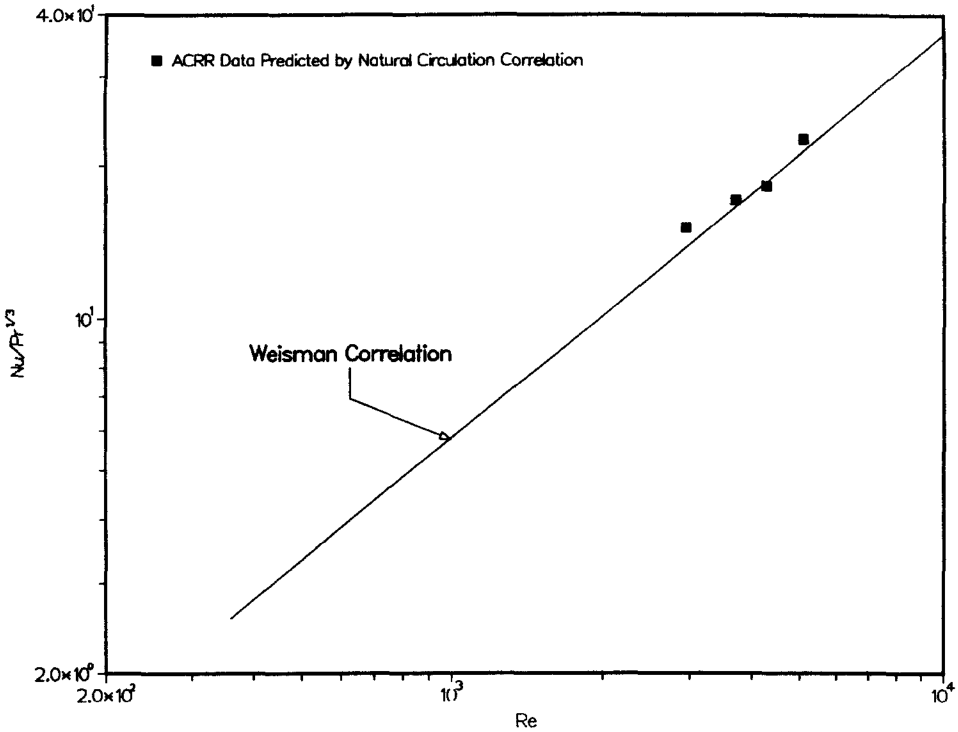

FIG. 15. Comparison of the ACRR data, predicted by the natural circulation correlation (equation (8)), with Weisman's correlation (equation (2)).

both the forced turbulent flow correlation (equation (2)) and the natural flow correlation (equation (8)), were compared in Fig. 15, which clearly shows that the Nusselt numbers predicted by the forced turbulent flow correlation were similar to those calculated by using the natural flow correlation. These heat transfer data for the ACRR are also plotted on Metais and Eckert's flow regime map (Fig. 13) [25]. As this figure indicates, the heat transfer regime in the ACRR during normal operation is at the boundary between the turbulent and mixed flow regimes. These results explain why either the forced turbulent flow, equation (2), or the natural flow, equation (8), correlation can accurately predict the single phase heat transfer coefficient in the ACRR.

SUMMARY AND CONCLUSIONS

This study conducted heat transfer experiments for fully developed forced and natural flows of water in triangularly arrayed rod bundles with $P/D = 1.25$, 1.38 and 1.5. Results showed that the forced circulation data can be classified into two basic flow regimes: turbulent flow and laminar flow. The Reynolds number at which the flow transition from turbulent to laminar flow occurred, Re_T, was found to increase linearly with P/D.

The turbulent flow data, where $Re \geqslant Re_T$, agreed with fully developed turbulent flow correlations, in rod bundles, which were developed based on data for $Re > 10\,000$. These results also indicate that the heat transfer coefficients in rod bundles at low Reynolds numbers (as low as Re_T) can be predicted by the turbulent flow correlations, such as that of Weisman [5].

The Nusselt number for forced laminar flow, where $Re < Re_T$ and $Ri < 1$, increased with the Reynolds number, while the exponent of Reynolds number increased linearly with P/D. However, the Nusselt numbers for the mixed flow, where $Ri \geqslant 1$, were correlated in terms of the Richardson number and the Reynolds number.

Unlike the forced flow experiments, the rod spacing only insignificantly affected the heat transfer in the natural circulation experiments. In these experiments, the Nusselt numbers increased proportionally to Ra_q raised to the one-fourth power. The comparison of the natural circulation results with those for isothermally heated rod bundles revealed that the dependency of the Nusselt number on the Rayleigh number was higher for the isothermal heated boundary than for the isoflux heated boundary. Table 1 lists the correlations developed in this work and the range of data in the different flow regimes.

The application of the present results to the SNL's ACRR operation suggested that although the reactor core is cooled by natural circulation, the single phase convective Nusselt number in the ACRR could be accurately predicted either by the natural flow correlation or the forced turbulent flow correlation.

Table 1. Summary of correlations for different flow regimes

Flow regime	Range of parameter	Correlation
Forced turbulent	$Re \geqslant Re_T$†	$Nu = C\,Re^{0.8}\,Pr^{1.3}$‡
Forced laminar	$Re < Re_T$,† $Ri < 1$	$Nu = A\,Re^{B}\,Pr^{1/3}$§
Mixed	$Ri \geqslant 1$	$Nu = a\,Ri^{b}\,Re^{c}$‖
Natural	$8\times10^{6} \leqslant Ra_q \leqslant 2.5\times10^{8}$	$Nu = 0.272Ra_q^{0.25}$

† $Re_T = (1.319P/D - 1.432)\times10^{4}$.

‡ $C = 0.026P/D - 0.006$.

§ $A = 1.061$, 0.511, 0.346 for $P/D = 1.25$, 1.38, 1.5, and $B = 0.797P/D - 0.656$.

‖ a, b, $c = 2.762$, 0.163, 0.282 and 0.95, 0.255, 0.404 for $P/D = 1.38$ and 1.5.

Results indicated that this reactor operates near the boundary between the mixed and turbulent flow regimes.

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EXPERIENCES DE TRANSFERT DE CHALEUR POUR DES FAIBLES DEBITS D'EAU DANS DES GRAPPES DE TUBES

Résumé—Des expériences de transfert de chaleur sont réalisées pour des écoulements forcés établis et naturels d'eau à travers sept grappes de tubes uniformément chauffés et triangulairement arrangés, avec $P/D = 1{,}25$, 1,38 et 1,5. Pour les expériences à circulation forcée, *Re* varie entre 80 et 50 000 et *Pr* entre 3 et 8,5 ; alors qu'en circulation naturelle *Re* varie depuis 260 jusqu'à 2000 et Ra_q depuis 8×10^6 jusqu'à $2{,}5 \times 10^8$. Les données d'écoulement forcé correspondent aux deux régimes fondamentaux : turbulent et laminaire. A la transition entre ces régimes Re_T qui varie depuis 2200 pour $P/D = 1{,}25$ jusqu'à 5500 pour $P/D = 1{,}5$, croît linéairement avec P/D. Les données en écoulement turbulent s'accordent bien ($\pm 15\%$) avec la formule de Weisman établie pour l'écoulement turbulent établi dans des grappes à $Re > 25\,000$. Néanmoins, les données en écoulement laminaire montrent que la dépendance de *Nu* vis-à-vis de *Re* est plus faible que pour l'écoulement turbulent. Les données de circulation naturelle montrent que l'espácement des tubes affecte très peu le transfert thermique ; pour $P/D = 1{,}38$ et 1,5, *Nu* est tel que $Nu = 0{,}272 Ra_q^{0,25}$.

MESSUNG DES WÄRMEÜBERGANGS BEI GERINGER STRÖMUNG VON WASSER IN STABBÜNDELN

Zusammenfassung—Messungen des Wärmeübergangs wurden durchgeführt bei voll ausgebildeter erzwungener und natürlicher Strömung von Wasser durch Bündel aus 7 gleichmäßig beheizten Stäben, die in Dreiecksteilung mit $P/D = 1{,}25$; 1,38 und 1,5 angeordnet waren. Bei den Versuchen zur erzwungenen Strömung lag die *Re*-Zahl zwischen 80 und 50 000, die *Pr*-Zahl zwischen 3 und 8,5. Bei natürlicher Strömung variierte die *Re*-Zahl von 260 bis 2000, die *Ra*-Zahl von 8×10^6 bis $2{,}5 \times 10^8$. Die Meßergebnisse für erzwungene Strömung unterteilen sich in zwei Bereiche : turbulente und laminare Strömung. Die *Re*-Zahl, Re_T, am Übergang der beiden Bereiche steigt linear von 2200 für $P/D = 1{,}25$ bis 5500 für $P/D = 1{,}5$ an. Die Meßwerte bei turbulenter Strömung stimmen gut ($\pm 15\%$) mit der Korrelation von Weismann überein, die für vollausgebildete turbulente Strömung in Stabbündeln bei $Re > 25\,000$ aufgestellt wurde. Trotzdem zeigen die Meßwerte bei laminarer Strömung eine schwächere Abhängigkeit der *Nu*-Zahl von der *Re*-Zahl als bei turbulenter Strömung. Die Daten bei natürlicher Strömung zeigen, daß der Wärmeübergang nur unwesentlich vom Abstand der Stäbe beeinflußt wird. Für $P/D = 1{,}38$ und 1,5 gilt folgende Korrelation : $Nu = 0{,}272 Ra_q^{0,25}$.

ЭКСПЕРИМЕНТАЛЬНОЕ ИССЛЕДОВАНИЕ ТЕПЛОПЕРЕНОСА ПРИ НИЗКИХ СКОРОСТЯХ ТЕЧЕНИЯ ВОДЫ ЧЕРЕЗ ПУЧКИ СТЕРЖНЕЙ

Аннотация—Проведено экспериментальное изучение теплопереноса при полностью развитых вынужденном и свободном течениях воды через семь расположенных в форме треугольника и равномерно нагреваемых стержневых пучков с $P/D = 1{,}25$; 1,38 и 1,5. В случае вынужденного движения *Re* изменялось от 80 до 50000, а число *Pr*—от 3 до 8,5, в то время как при свободной циркуляции число *Re* варьировалось от 260 до 2000, а число Ra_q от 8×10^6 до $2{,}5 \times 10^8$. Данные для вынужденного течения классифицируются в соответствии с двумя основными режимами течени: турбулентным и ламинарным. Число Re_T, характеризующее переход от одного режима к другому и изменяющееся от 2200 при $P/D = 1{,}25$ до 5500 при $P/D = 1{,}5$, увеличивается линейно с P/D. Данные для турбулентного теплопереноса хорошо согласуются ($\pm 15\%$) с соотношением Вайсмана, выведенным для полностью развитого турбулентного течения в пучке стержней при $Re > 25\,000$. Однако, данные для ламинарного течения показывают, что зависимость числа *Nu* от числа *Re* является более слабой, чем в случае турбулентного течения. Из результатов для свободного циркуляционного течения следует, что расстояния между стержнями незначительно влияют на теплоперенос; при $P/D = 1{,}38$ и 1,5 для числа *Nu* справедливо соотношение $Nu = 0{,}272 Ra_q^{0,25}$