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Forced and combined convection of water in a vertical seven-rod bundle with $P/D = 1.38$

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Abstract—Heat transfer experiments of forced turbulent and laminar, and combined laminar downflows of water are conducted in a uniformly heated, triangularly arranged, seven-rod bundle having a P/D ratio of 1.38. In the forced flow experiments Re ranged from 1200 to 24 800 and Pr from 6.8 to 9.0, while in the combined convection experiments Re varied from 148 to 3800, Gr_q from 1.3×10^5 to 3×10^6, and Ri from 0.01 to 9. The data in the forced turbulent and the laminar flow regimes are in good agreement with the upflow correlations (within $\pm 10\%$). Also, the transition between these two regimes, occurring at $Re = 3800$, is the same as that for the upflow condition. In the laminar flow regime, the flow entering the heated section is hydrodynamically developing while the flow in the heated section is thermally developed. The transition from forced laminar to combined convection occurred at $Ri = 0.1$, which is an order of magnitude lower than that for upflow. The combined convection data are correlated by superimposing the correlations for forced laminar and natural laminar flows as:

$$Nu_{C,L} = [Nu_{F,L}^3 + Nu_{N,L}^3]^{1/3}, \quad \text{for upflow}$$

$$Nu_{C,L} = [Nu_{F,L}^2 - Nu_{N,L}^2]^{1/2}, \quad \text{for downflow.}$$

These correlations are within ± 11 and $\pm 15\%$ of the upflow and downflow data, respectively.

INTRODUCTION

CONVECTIVE heat transfer for both upflow and downflow of water in multi-rod bundles at low Reynolds number ($Re < 10^4$) is of particular importance to many engineering applications, where both forced and combined flow conditions are encountered. Examples of these applications include the design and operation of heat exchangers for low grade energy recovery and solar and geothermal energy systems, steam generators in nuclear and fossil power plants, and the operation and safety of open-pool type nuclear research reactors.

Most heat transfer data for forced convection of water in rod bundles with either a square or a triangular geometry have been limited to upflow conditions at $Re > 6000$ [1–4]. At lower Reynolds numbers only a few data has been reported for upflow in rod bundles with either an isothermal or a uniformly heated wall [5–7]; no experimental data are available for downflow.

Although numerous theoretical studies of fully developed forced laminar flow and buoyancy assisted and opposed combined convection in multi-rod bundles have been performed [8–15], only limited comparison with upflow data has been reported [16]. Because these studies are based on idealized subchannel geometries and flow conditions, where the effects of inter-subchannel cross-flow and buoyancy induced flow mixing are excluded [17, 18], the use of theoretically developed heat transfer correlations is restricted to steady, fully developed flow conditions, which are of limited practical application. Therefore, there is a need to perform heat transfer experiments and develop design correlations for both upflow and downflow conditions in rod bundles at low Re, but at high Ra_q.

Recently, heat transfer data were collected and correlated for forced turbulent and laminar and combined laminar upflows, as well as natural laminar flow of water in uniformly heated, triangularly arrayed, seven-rod bundles with P/D ratios of 1.25, 1.38 and 1.5 [18, 19]. In these experiments, Re ranged from 80 to 5×10^4, Ra_q from 8.5×10^6 to 4.5×10^{11}, and Pr from 3 to 8.5. The forced upflow data fell into two basic flow regimes—turbulent flow and laminar flow. The Reynolds numbers at the transition between these two regimes, Re_T, increased linearly with P/D ratio. The turbulent flow data for $Re \geqslant Re_T$ were in good agreement with the fully developed turbulent flow correlation of Weisman [20] (within $\pm 15\%$). Also, the transition between forced laminar and combined laminar upflows occurred at $Ri = 1$. The data for natural laminar flow ($8.5 \times 10^6 \leqslant Ra_q \leqslant 2.5 \times 10^8$ and $260 \leqslant Re \leqslant 2000$) were correlated as [18, 19]

$$Nu_{N,L} = 0.272 Ra_q^{0.25}. \quad (1)$$

To extend the data base of Kim and El-Genk, this research performed heat transfer experiments of

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NOMENCLATURE

C_p	specific heat [J kg^{-1} K^{-1}]
D	heated rod diameter [m]
D_e	hydraulic equivalent diameters (4 × cross-sectional flow area/wetted perimeter) [m]
D_{eh}	heated equivalent diameter, $D[(2\sqrt{3}/\pi)(P/D)^2-1]$ [m]
g	acceleration of gravity [m s^{-2}]
Gr_q	Grashof number based on heat flux, $(g\beta qD_{eh}^4/k\nu^2)$
k	thermal conductivity [W m^{-1} K^{-1}]
L	total heated length [m]
l_h	hydrodynamic entry length
Nu	Nusselt number, $qD_{eh}/k(T_w-T_b)$
$Nu_{C,L}$	Nusselt number for combined laminar flow
$Nu_{F,L}$	Nusselt number for forced laminar flow
$Nu_{F,T}$	Nusselt number for forced turbulent flow
$Nu_{N,L}$	Nusselt number for natural laminar flow
P	pitch between adjacent rods [m]
p	power [W]
Pr	Prandtl number, $(C_p\mu/k)$
P/D	pitch-to-diameter ratio
q	surface heat flux, $(p/\pi DL)$ [W m^{-2}]
Ra_q	Rayleigh number, $(Gr_q\,Pr)$
Re	Reynolds number based on the heated equivalent diameter, $(\rho vD_{eh}/\mu)$
Re_{D_e}	Reynolds number based on the hydraulic equivalent diameter, $(\rho vD_e/\mu)$
Re_T	Reynolds number at the transition from forced laminar to turbulent convection
Ri	Richardson number, (Gr_q/Re^2)
T_b	mean bulk temperature of water based on heat balance [K]
T_w	mean wall 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}]
ν	kinematic viscosity [m^2 s^{-1}]
ρ	density [kg m^{-3}].

forced turbulent and laminar, and combined laminar downflows using the same apparatus and test section as for the upflow experiments [18, 19]. The downflow data in the respective flow regimes are compared with those of Kim and El-Genk [18, 19] for upflow conditions to quantify the effects of P/D ratio and flow direction on heat transfer.

EXPERIMENTAL SETUP

As shown in Fig. 1, the test section consisted of seven, uniformly heated, type 304 stainless steel tubes, 1.27 cm o.d., 90.44 cm long, and 0.089 cm in wall thickness, enclosed in a hexagonal Plexiglas shroud. Because of the low thermal conductivity of the Plexiglas wall (1.27 cm thick) and the small temperature difference across the wall in the experiments (<10 K), the thermal losses through the shroud walls were insignificant. To provide the same flow area per heated rod, the side length of the hexagonal shroud's inner wall was taken equal to $\sqrt{((7/3)P)}$, and the spacing between the rods was maintained using Plexiglas spacers at the top and the bottom of the test section. The heated section was silver welded at the bottom and top to unheated brass rods, except for the three instrumented rods, which had brass tubes at the top. The unheated section at the top (51.3 cm) served as an entry length for the hydrodynamic development of the flow before entering the heated section, while the unheated section at the bottom (51.3 cm) served to minimize the exit effects on the heat transfer in the heated section [18, 19].

The shrouded rod bundle was supported at the bottom by a structure consisting of two concentric Plexiglas cylinders. The inner cylinder had eight longitudinal openings (each is 17 cm long and 2.54 cm wide) to allow flow circulation through the test section during the downflow experiments. The containment tank, which housed the shrouded test section, was a Plexiglas cylinder, 61 cm o.d. and 244 cm long. During the experiments, the water from the containment tank entered from the top, flowed through the heated section, then through the openings in the lower support structure to the lower cavity. The flow from the lower cavity was then circulated through an external loop by two centrifugal pumps, which were connected in series, providing a maximum flow rate of 1.52 kg s^{-1}. The loop was equipped with a 500 kW heat exchanger for rejecting the heat into a chilled water line. The returned flow entered the containment tank through a diffuser assembly at the bottom of the tank (see Fig. 1). More details on the design and instrumentation of the test section are available elsewhere [18, 19].

EXPERIMENTAL MEASUREMENTS

The stainless steel tubes in the test section was electrically heated using direct current supplied by a 200 kW power generator. The actual power produced in the heated section (93.91% of the total electric power input) was determined from the electric current and the voltage measurements, after subtracting the losses in the connecting electric cables, the electric con-

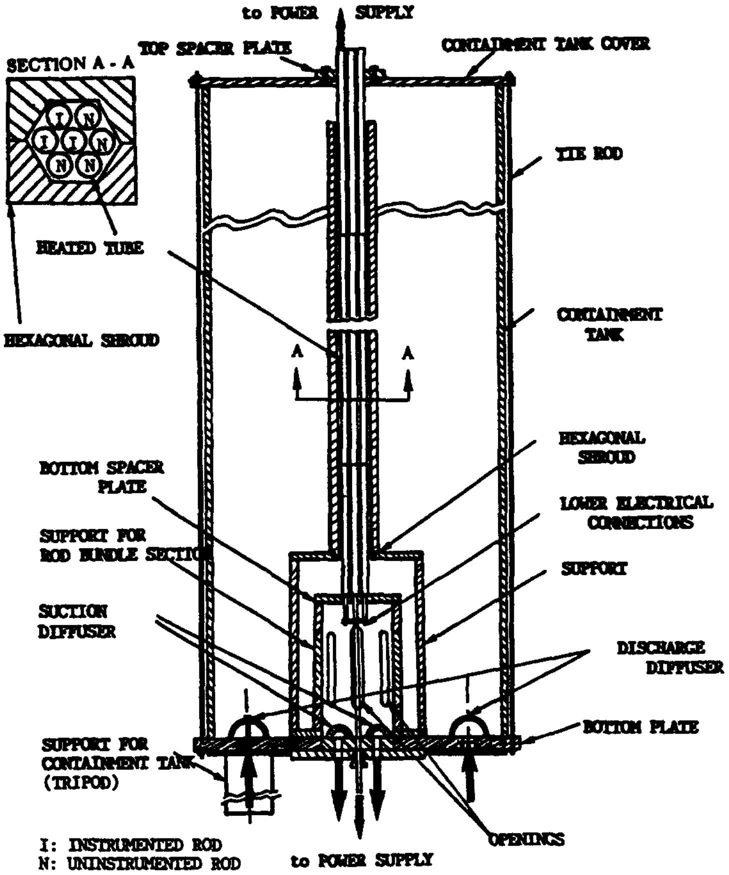

FIG. 1. Longitudinal view of the containment tank and the rod bundle test section.

nectors in the circuit, and the upper and lower brass rods/tubes. Pretesting at lower power levels (50 W) in air showed the axial power to be uniform along the length of the heated section with a maximum variation of about 10% [18, 19].

The flow measurements were obtained by a total of three flow meters covering a range of flow rates extending from 0.025 to 1.52 kg s^{-1}. Two Signet MK515 turbine flow meters were connected in parallel for flow measurements from 0.126 to 1.52 kg s^{-1}, while a Dwyer rotameter was used for flow rates below 0.126 kg s^{-1}. In the experiments, demineralized water with electric resistivity of 15–18 MΩ-cm was used to minimize the mineral deposits on the surface of the heated rods.

As shown in Fig. 1, 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 using a specially designed probe [18, 19]. The heated wall temperature was measured at eight axial locations from the top of the heated section: 4.53, 16.16, 27.79, 39.42, 51.05, 62.68, 74.31, and 85.94 cm. The inlet water temperature to the test section was monitored using two thermocouples placed 4 cm from the top of the heated section. The water bulk temperatures at the different axial locations in the heated section were determined from the overall heat balance using the measured flow rate, actual power produced in the heated section, and the measured water temperature at the entrance of the test section. The arithmetic average of the water bulk temperatures was used to evaluate the physical properties of water in the dimensionless quantities such as Nu, Re, Pr, and Gr_q.

In the experiments, because the hexagonal shroud is insulated, the water around the peripheral rods was non-uniformly heated, resulting in non-uniform heat transfer from the heated rods to the water. In addition to the non-uniform heat transfer, the cross-flow between subchannels in the rod bundle caused a variation of up to 6 K in the measured wall temperatures at the same axial location between the center rod and the two peripheral rods, and between the two peripheral rods. When the average of the wall temperatures in the three instrumented rods was used to evaluate Nu at each axial location, it resulted in a large scattering in the heat transfer data. To reduce the scattering in the heat transfer data, the average wall temperature in the heated section was taken equal to the arithmetic average of the measured wall temperatures at the various axial locations in the center rod. Then, the difference between the average wall surface temperature and the average water bulk temperature was used to determine the average values of Nu in the heated section as functions of inlet Re and axial heat flux.

The uncertainties in the power, temperature, and the flow rate measurements were approximately ± 0.63, ± 2, and $\pm 5\%$, respectively. These uncertainties resulted in approximately $\pm 5.3\%$ uncertainty in Nu, ± 5 in Re, $\pm 0.3\%$ in Pr, from ± 1.8 to $\pm 2.3\%$ in Gr_q, and $\pm 10.3\%$ in Ri [18, 19].

EXPERIMENTAL CONDUCT

In the experiments, the electric power was increased in increments of approximately 0.25 kW, beginning from as low as 0.5 kW, to 12.0 kW. At each power level, the flow rate was reduced in increments of approximately 0.012 kg s^{-1}. At each flow rate, after steady state was reached, the temperature, electric power, and flow rate measurements were collected and analyzed using a data acquisition system consisting of an IBM-CS 9000 computer and an HP-3497A data logger. These procedures continued until a flow rate of 0.0126 kg s^{-1} was reached or a release of dissolved air in the test section occurred, causing oscillations in the temperature measurements. Such air release, which was encountered in forced laminar flow and combined convection experiments at low flow rates, affected the reproducibility of the data (in excess of 15%). Hence, only the data with a reproducibility of less than or equal to $\pm 15\%$ were included in the downflow data base for developing the heat transfer correlations presented in the following section.

RESULTS AND DISCUSSION

The hydrodynamic entry length, l_h, for the flow to be hydrodynamically developed is estimated by the formula suggested by White [21] for various flow channels, including rod bundles, $\{l_h = D_e[0.04Re_{D_e}+0.5]\}$. The entry length estimated at the highest Reynolds number in the laminar flow regime, $Re = 3800$, was (74 cm) much longer than the length of the unheated section at the top of the test section (51.3 cm), suggesting that the flow will be hydrodynamically developing before entering the heated section. Also, as with the upflow experiments [18, 19], the measured values of Nu at the eight axial locations along the heated section were almost uniform (maximum variation was about 12%), suggesting that the flow in the heated section could have been thermally developed. This small variation in Nu with axial location could partially be attributed to cross-flow mixing among subchannels in the bundles.

A total of 419 data points were collected for forced turbulent and laminar flows as well as for combined laminar flow. In the forced convection experiments, Re ranged from 1200 to 24800, and Pr from 6.4 to 9, while in the combined convection experiments Re ranged from 148 to 3800, Gr_q from 1.3×10^5 to 3×10^6 and Ri from 0.01 to 9. In the experiments, the mean wall temperature ranged from 286 to 301 K, and from 286 to 319 K, for forced turbulent and laminar flows and combined flow, respectively. The corresponding water mean bulk temperatures varied from 284 to 296 K, and from 284 to 294 K, respectively.

Forced turbulent and laminar convection data

The data for forced turbulent and laminar downflows were compared with those for upflow conditions [18, 19] to verify the soundness of the experimental setup for the former. Figure 2 compares the forced downflow data with the correlations of Kim and El-Genk [18, 19] for forced turbulent and laminar upflows and that of Weisman [20] for fully developed turbulent upflow. As expected, the forced downflow data were in good agreement with the upflow correlations. As for upflow, these data were classified into two basic flow regimes—turbulent and laminar flow—where the transition between these two regimes, occurring at $Re = 3800$, is the same as that for upflow conditions [18, 19].

In the turbulent downflow regime ($Re \geqslant 3800$) the data are within $\pm 15\%$ of Kim and El-Genk's correlations for upflow [18, 19], and within 15 to -6% of Weisman's correlation [20] for fully developed turbulent upflow (see Fig. 2). Also, the forced laminar downflow data were in good agreement with Kim and El-Genk's correlations for forced laminar upflow (within $\pm 13\%$). The heat transfer in these two flow regimes is mainly due to forced convection and the effect of buoyant forces is insignificant. As demonstrated in Fig. 3, Nu values for forced turbulent and laminar downflows for $P/D = 1.38$, are independent of Gr_q, which explains the good agreement between the downflow data and the upflow correlations in these flow regimes. This agreement, though expected in the forced turbulent and laminar flow regimes, verifies the soundness of the experimental setup and the accuracy of the measurements for downflow combined convection, where flow direction would affect the heat transfer rate.

Combined laminar downflow data

Combined convection occurs when the heat transfer process is governed by the combined effects of inertial and buoyant forces; however, the effect of the latter on heat transfer depends on the flow direction. For example, in the upflow, buoyant forces accelerate the flow next to the heated wall, hence enhancing the heat transfer rate and causing Nu to increase with Gr_q [18, 19]. Conversely, in downflow the buoyant forces act in the opposite direction of the flow, causing Nu to decrease with Gr_q. As indicated in Fig. 4, the reduction in Nu for combined downflow increases with Gr_q/Re, which is in qualitative agreement with the fully developed flow solutions of Yang [13, 14] and Iannello et al. [16]. However, because of the flow mixing between subchannels in the rod bundle the transition from laminar to combined convection occurred over

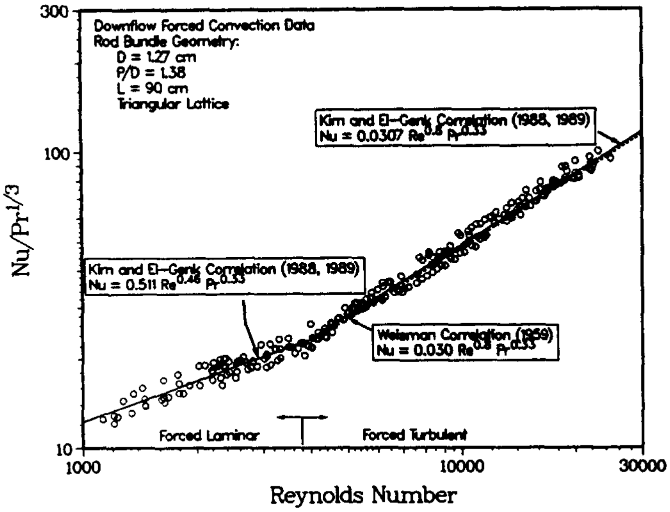

FIG. 2. Comparison of heat transfer data for forced turbulent and laminar downflows with upflow correlations for $P/D = 1.38$.

an extended range of Gr_q/Re, rather than at a single value as suggested by the fully developed flow solutions [13, 14, 16]. As the insert in Fig. 4 shows, the transition from laminar to combined convection occurred at Gr_q/Re values ranging from 100 to 300, where the corresponding values of Re varied from 148 to 3800, respectively. In addition to the dependency of Gr_q/Re at the transition from laminar to combined convection on Re, the use of Gr_q/Re has introduced a large scattering in the combined convection data (see the inset in Fig. 4). However, when Gr_q/Re was replaced by Ri the data scattering was reduced and the transition from laminar to combined convection occurred at a single Ri value, regardless of the value of Re (see Fig. 4). These results suggest that because of the expected flow mixing among subchannels in the bundle geometry the combined convection regime is best described using Ri, rather than Gr_q/Re, as recommended for fully developed flow conditions [13, 14, 16].

As indicated in Fig. 4, the transition from forced laminar to combined laminar downflow occurred at $Ri = 0.1$, which is an order of magnitude lower than that for upflow conditions in the same rod bundle [18, 19]. At $Ri > 0.1$, Nusselt numbers for laminar downflow were independent of Ra_q (or $Nu_{N,L}/Nu_{F,L}$), but increased proportionally to Re raised to the 0.46 power (see Figs. 2–5). As shown in Fig. 5, Nusselt

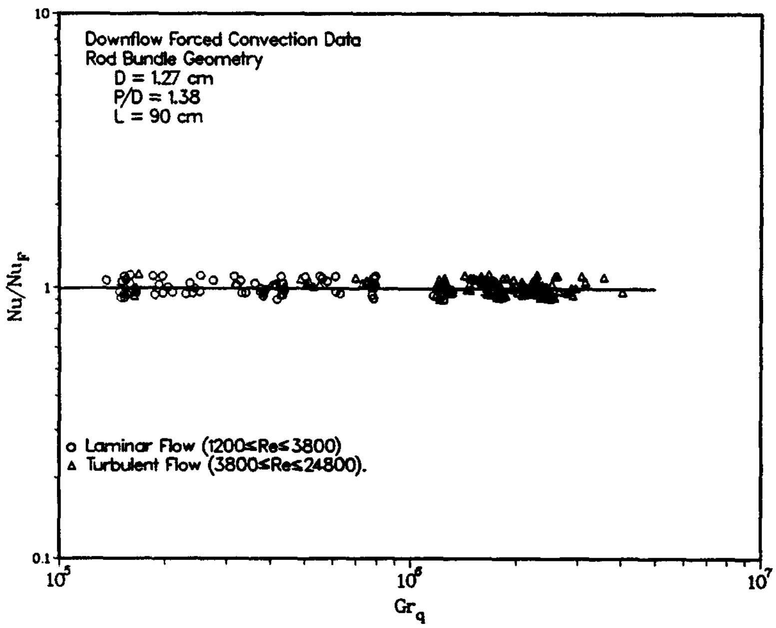

FIG. 3. Heat transfer data for forced turbulent and laminar downflows for $P/D = 1.38$.

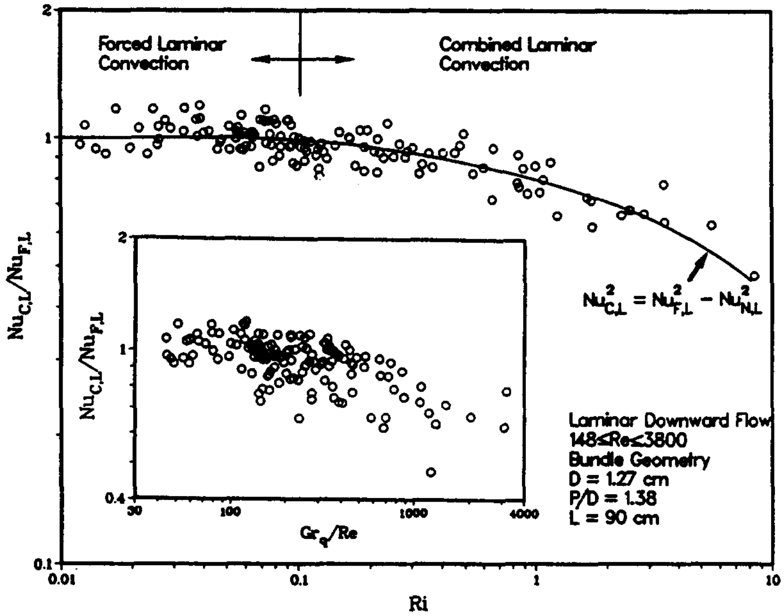

FIG. 4. Heat transfer data for forced laminar and combined downflows in rod bundle with $P/D = 1.38$.

number values for combined downflow decreased with Ra_q (or $Nu_{N,L}/Nu_{F,L}$), while those for combined upflow increased with Ra_q. This figure also shows that the value of $Nu_{N,L}/Nu_{F,L}$ (or Ra_q) at which the contribution of buoyant forces to heat transfer becomes measurable is much lower in upflow than in downflow, which is in agreement with the results of other investigations of laminar and combined flows in vertical tubes and annuli [22–24]. The results show that in buoyancy opposed laminar flow, buoyant forces begin to affect the heat transfer process in rod bundles at a significantly lower buoyant-to-inertia forces ratio than in upflow (see Figs. 4 and 5).

A general correlation for both combined laminar upflow and downflow is developed by superimposing the correlations for forced laminar and natural laminar flows, as was first suggested by Churchill [22]

$$Nu_{C,L} = [Nu^n_{F,L} \pm Nu^n_{N,L}]^{1/n} \tag{2}$$

where the positive and the negative sign of $Nu_{N,L}$ corresponds to buoyancy assisted and opposed flows, respectively. The general form of equation (2) had

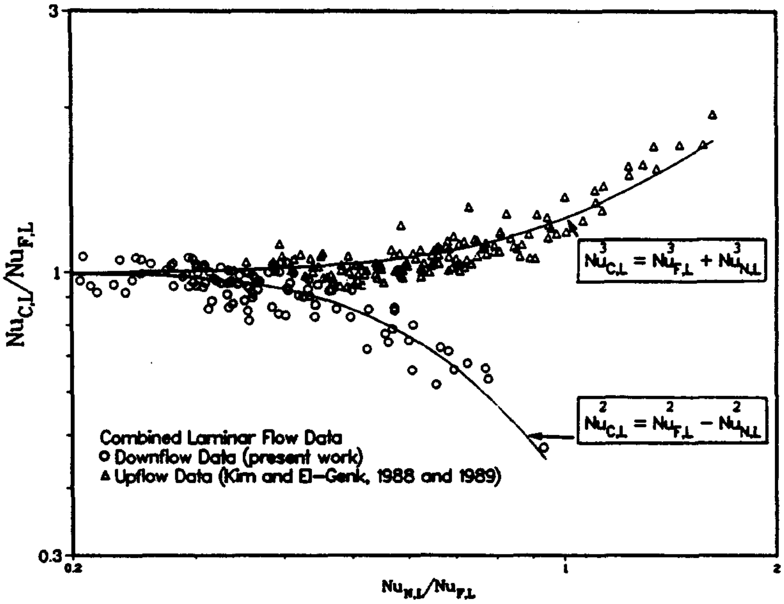

FIG. 5. Heat transfer data and correlations for both forced laminar and combined downflows in rod bundle with $P/D = 1.38$.

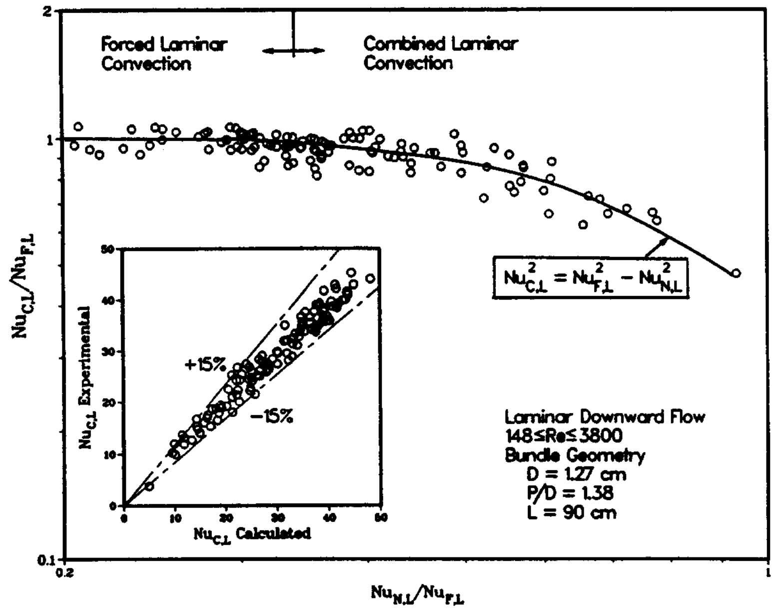

FIG. 6. A comparison of experimental and calculated Nu for combined downflow in a rod bundle with $P/D = 1.38$.

been used successfully by several investigators to correlate combined laminar flow data for isothermal and uniformly heated vertical tubes and annuli [22–24]. Values of the exponent ranging from 2 to 6 have been rationalized for achieving the best fit with experimental data.

The validity of equation (2) for describing combined convection in vertical, multi-rod bundles has never been examined before this work. The appropriate values of the exponent in equation (2) for correlating the heat transfer data for upflow and downflow conditions in rod bundles were determined based on the best fit of the data in each case. As shown in Figs. 5 and 6, while a cubic exponent of Nusselt numbers gave the best correlation with the upflow data, the downflow data were best correlated using a square exponent as

$$Nu_{C,L} = [Nu^3_{F,L} + Nu^3_{N,L}]^{1/3}, \quad \text{for combined upflow} \quad (3)$$

and

$$Nu_{C,L} = [Nu^2_{F,L} - Nu^2_{N,L}]^{1/2}, \quad \text{for combined downflow.} \quad (4)$$

These correlations were within ± 11 and $\pm 15\%$ of the data for combined laminar upflow and downflow, respectively. In equations (3) and (4), $Nu_{N,L}$ and $Nu_{F,L}$ are given, respectively, by the natural convection correlation (equation (1)) and the forced laminar upflow correlation of Kim and El-Genk [18, 19] as

$$Nu_{F,L} = 0.511 Re^{0.46} Pr^{1/3}. \quad (5)$$

SUMMARY AND CONCLUSIONS

Heat transfer experiments were conducted for forced turbulent and laminar and combined laminar downflows. The experiments, with water as the working fluid, employed a uniformly heated, triangularly arranged, seven-rod bundle with a P/D ratio of 1.38. To assess the effects of flow direction on combined convective heat transfer, the experimental data are compared with those for upflow obtained using the same apparatus. Both combined laminar upflow and downflow data were correlated (equations (3) and (4)) using the general form suggested by Churchill [22] (equation (2)). Results show that while a square exponent of Nusselt numbers for forced laminar and natural laminar downflows gave the best correlation of the combined convection data (within $\pm 15\%$), a cubic exponent was more appropriate for correlating the combined upflow data (within $\pm 11\%$). Also, the sign of the Nusselt number for natural laminar flow was negative and positive for downflow and upflow, respectively, reflecting the effect of buoyant forces on heat transfer in each case.

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CONVECTION FORCEE ET MIXTE D'EAU DANS UNE GRAPPE VERTICALE DE SEPT CYLINDRES AVEC $P/D = 1{,}38$

Résumé—Des expériences thermiques de convection descendante turbulente, laminaire et laminaire mixte d'eau sont conduites pour une grappe de 7 cylindres uniformément chauffés, arrangés en triangle, avec $P/D = 1{,}38$. Dans les expériences de convection forcé Re varie entre 1200 et 24800 et Pr entre 6,8 et 9, tandis que pour celles de convection mixte Re varie de 148 à 3800, Gr_q de $1{,}3 \times 10^5$ à 3×10^6 et Ri de 0,01 à 9. Les résultats pour les régimes d'écoulement forcé turbulent et laminaire sont en bon accord avec les formules d'écoulement ascendant (à $\pm 10\%$). La transition entre ces deux régimes apparait à $Re = 3800$ comme dans le cas de l'ascension du fluide. Dans le régime d'écoulement laminaire, l'écoulement à l'entrée de la section chauffée est hydrodynamiquement en développement alors que l'écoulement dans la section chauffée est thermiquement developpé. La transition entre convection laminaire forcée et convection mixte se produit à $Ri = 0{,}1$ qui est un ordre de grandeur plus faible que pour le mouvement ascendant. Les données de convection mixte sont corrélées en supposant les formules de convection laminaire forcée et naturelle:

$$Nu_{C,L} = [Nu_{F,L}^3 + Nu_{N,L}^3]^{1/3} \text{ pour l'ascension}$$

$$Nu_{C,L} = [Nu_{F,L}^2 + Nu_{N,L}^2]^{1/2} \text{ pour la descente.}$$

Ces formules donnent à $\pm 11\%$ et $\pm 15\%$ près les résultats expérimentaux respectivement pour l'ascension et la descente.

ERZWUNGENE UND MISCHKONVEKTION VON WASSER IN EINEM VERTIKALEN BÜNDEL VON SIEBEN ROHREN MIT $P/D = 1{,}38$

Zusammenfassung—Untersucht wird der Wärmeübergang bei der Abwärtsströmung von Wasser in einem Rohrbündel für den Bereich der laminaren und turbulenten erzwungenen Konvektion. Die sieben Einzelrohre in Dreiecksanordnung ($P/D = 1{,}38$) werden dabei gleichförmig beheizt. Bei den Versuchen zur erzwungenen Konvektion wird Re von 1200 bis 24 800 and Pr von 6,8 bis 9,0 variiert, während bei den Versuchen zur gemischten Konvektion Re im Bereich von 148 bis 3800, Gr_q von $1{,}3 \times 10^5$ bis 3×10^6 und Ri von 0,01 bis 9 variieren. Die Meßergebnisse bei erzwungener laminarer und turbulenter Abwärtsströmung stimmen innerhalb einer Streubreite von $\pm 10\%$ gut mit den Ansätzen für die Aufwärtsströmung überein. Auch der Übergang zwischen den beiden Strömungsformen, der bei $Re = 3800$ auftritt, stimmt mit dem Übergang bei Aufwärtsströmung überein. Im laminaren Bereich ist die Strömung beim Eintritt in die Heizzone bereits thermisch ausgebildet, während sie sich hydraulisch erst noch entwickelt. Der Übergang von der erzwungenen laminaren zur gemischten Konvektion tritt bei $Ri = 0{,}1$ auf, was um eine Größenordnung tiefer liegt als bei der Aufwärtsströmung. Die Meßwerte der gemischten Konvektion werden mit Beziehungen korreliert, die durch Überlagerung der Beziehungen für die erzwungene und die natürliche Konvektion entstehen. Diese sind:

$$Nu_{C,L} = [Nu_{F,L}^3 + Nu_{N,L}^3]^{1/3} \text{ für die Aufwärtsströmung und}$$

$$Nu_{C,L} = [Nu_{F,L}^2 + Nu_{N,L}^2]^{1/2} \text{ für die Abwärtsströmung.}$$

Die Meßwerte können mit diesen Gleichungen innerhalb einer Streubreite von $\pm 11\%$ für die Aufwärtsströmung und $\pm 15\%$ für die Abwärtsströmung beschrieben werden.

ВЫНУЖДЕННАЯ И СМЕШАННАЯ КОНВЕКЦИЯ ВОДЫ В ВЕРТИКАЛЬНОМ ПУЧКЕ ИЗ СЕМИ СТЕРЖЕНЕЙ С $P/D = 1{,}38$

Аннотация—Проведены эксперименты по теплопереносу при вынужденных турбулентном и ламинарном, а также смешанных (свободных и вынужденных) ламинарных нисходящих течениях воды в однородно нагретом и расположенном в виде треугольника пучке из семи стержней с отношением P/D, равным 1,38. В случае вынужденного течения значение Re изменялось от 1200 до 24 800, а Pr—от 6,8 до 9,0, В случае же смешанной конвекции значение Re варьировалось от 148 до 3800, Gr_q—от $1{,}3 \times 10^5$ до 3×10^6 и Ri—от 0,01 до 9. Данные, полученные для режимов вынужденного турбулентного и ламирного течений, хорошо согласуются с соотношениями для восходящего течения (с точностью до $\pm 10\%$). Переход от одного из этих режимов к другому, наблюдавшийся при $Re = 3800$, аналогичен происходящему в условиях восходящего течения. При режиме ламинарного течения поток, входящий в сечение нагрева, является гидродинамически неустановившимся и развивающимся, в то время как поток на участке нагрева является термически развитым. Переход от вынужденной ламинарной к смешанной конвекции имел место при $Ri = 0{,}1$, что на порядок величины ниже, чем в случае восходящего течения. Зависимость между данными, полученными для смешанной конвекции, устанавливается с помощью соотношений для вынужденного и свободного ламинарных течений

$$Nu_{C,L} = [Nu_{F,L}^3 + Nu_{N,L}^3]^{1/3} \quad \text{для восходящего течения}$$

$$Nu_{C,L} = [Nu_{F,L}^2 - Nu_{N,L}^2]^{1/2} \quad \text{для нисходящего течения.}$$

Данные соотношения являются точными в пределах ± 11 и $\pm 15\%$ соответственно для восходящего и нисходящего течений.