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Experimental Study of Differences in Single-Phase Forced-Convection Heat Transfer Characteristics between Upflow and Downflow for Narrow Rectangular Channel

Yukio SUDO, Keiichi MIYATA, Hiromasa IKAWA,
Masami OHKAWARA and Masanori KAMINAGA

*Japan Atomic Energy Research Institute**

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The differences in the single-phase forced-convection heat transfer characteristics between upflow and downflow were investigated experimentally with a narrow vertical rectangular channel. The objectives of the experiment were to investigate in both laminar and turbulent flow regions the applicability of existing correlations to and the effects of buoyant force on the heat transfer characteristics in the narrow vertical rectangular channel, which is simulating a subchannel of 2.25 mm in gap and 750 mm in length in the fuel element of the research reactor, JRR-3 to be upgraded at 20 MWt. As the results, it was revealed that (1) by use of equivalent hydraulic diameter, existing correlations are applicable to a channel as narrow as 2.25 mm in gap for turbulent flow though the precision and critical Reynolds number are different among the correlations, and (2) in the laminar flow, the difference in heat transfer characteristics arises between upflow and downflow with Reynolds number less than about 700 and Grashof number larger than about 1,000, giving smaller Nusselt number for downflow than for upflow as the effect of buoyant force. New heat transfer correlations for channel heated from both sides are proposed as lower limits for upflow and downflow, respectively, in the laminar flow.

KEYWORDS: single-phase flow, forced-convection, heat transfer, narrow rectangular channel, upflow, downflow, turbulent flow, laminar flow, buoyant force, research reactors, JRR-3

I. INTRODUCTION

The problem which is addressed to in this study is the differences in the single-phase forced-convection heat transfer characteristics between upflow and downflow for a vertical rectangular channel simulating a subchannel in the fuel element of the research reactor, JRR-3, which is scheduled to be upgraded from 10 to 20 MWt at the Japan Atomic Energy Research Institute. Each fuel element of the JRR-3 has twenty flat fuel plates arranged in parallel with 2.25 mm gap. The configuration of a subchannel consisting of two adjacent fuel plates is rectangular with 750 mm in length, 66.6 mm in width and 2.25 mm in gap. For the condition of normal operation at 20 MWt, fuel plates are cooled by downward flow at the velocity of about 6.2 m/s without allowing the nucleate boiling anywhere in the core. It is considered that there are no differences in the single-phase forced-convection heat transfer characteristics between upflow and downflow for such a high velocity at the normal operation condition of the JRR-3. The effects of buoyant force should, however,

* *Tokai-mura, Ibaraki-ken 319-11.*

enhance the heat transfer for upflow and on the contrary, decrease the heat transfer for downflow at low velocities, which occur during the operational transients and accidents. Therefore, the understanding of the differences in the forced-convection heat transfer characteristics between upflow and downflow at low velocities is required for the safety analysis of the operational transients and accidents of the JRR-3, because some of operational transients and accidents assumed for the safety analysis of JRR-3 has a history of core flow in which the downward flow decreases to zero velocity and after that upward flow is established at low velocity as a cooling mode of natural circulation to remove the decay heat after scram.

It has not been made clear experimentally nor theoretically on what conditions the differences should arise between upflow and downflow in the single-phase forced-convection heat transfer characteristics, that is, on what condition the buoyant force should make differences between upflow and downflow(1). Reynolds number Re is a key parameter for a turbulent forced-convection heat transfer in a circular tube, and Grashof number Gr shows the magnitude of the effects of buoyant force. Therefore, Re and Gr are key parameters which are considered to give the condition on which the buoyant force should make the differences between upflow and downflow. It should be mentioned here that heat transfer correlations are different between circular tubes and rectangular channels in the laminar forced-convection even in case of no effects of buoyant force(5)(7), so that it is not necessarily certain if the heat transfer correlations derived for circular tubes are applicable directly to a rectangular channel of which gap is as narrow as 2.25 mm in case of JRR-3 for both laminar and turbulent flows.

The objectives of this study are, therefore, to investigate the following items for forced-convection heat transfer in a vertical rectangular channel simulating a subchannel of fuel element of the JRR-3.

(1) Applicability of existing heat transfer correlations derived for circular tubes to a rectangular channel whose gap is as narrow as 2.25 mm for such high velocities as there are no effects of buoyant force on the heat transfer, for upflow and downflow.

(2) Conditions on which the differences arise between upflow and downflow in forced-convection heat transfer characteristics at relatively low velocities as the effects of buoyant force for the rectangular channel.

(3) Evaluation of heat transfer coefficients for both upflow and downflow in the rectangular channel under the conditions where the effects of buoyant force are dominant.

To investigate the above items, two rectangular channels were used, one is heated from both sides and another is heated from one side. The differences in heat transfer characteristics between channels heated from both sides and from one side are also investigated to help in the understanding of the effects of differences of heating conditions.

It should be added here that the knowledge obtained from this experiment of the differences in heat transfer characteristics between upflow and downflow should be applicable to the plate-type fuel elements of other research reactors than JRR-3 though this experiment treats only the flow channel whose dimension and configuration are simulating the fuel element of the JRR-3.

II. Experiment

1. Schematic of Test Rig

Figure 1 shows a schematic diagram of the test loop. The test loop is composed of a coolant storage tank, a recirculation line and a test section simulating a subchannel of a JRR-3 standard fuel element. As a coolant, water is used in this experiment.

The coolant storage tank is 0.2 m^3 in volume and has a cooling line of copper coil in it to keep constant the coolant temperature at the inlet of test section during a test. The tank is open to the atmosphere.

The recirculation line has a pump, a bypass line, a rotor flow meter, an electromagnetic flow meter, stop valves and regulating valves. Both of upflow and downflow can be selected for the test section. The flow paths are shown with dotted lines for downflow and with solid lines for upflow in Fig. 1.

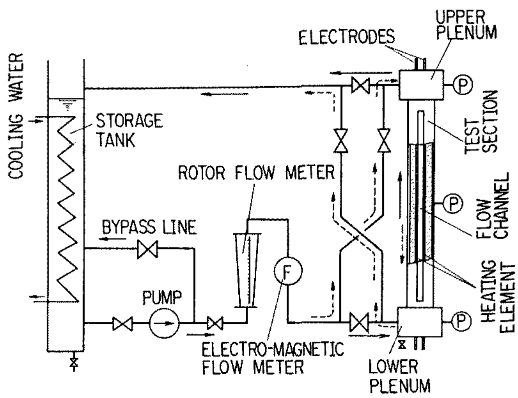
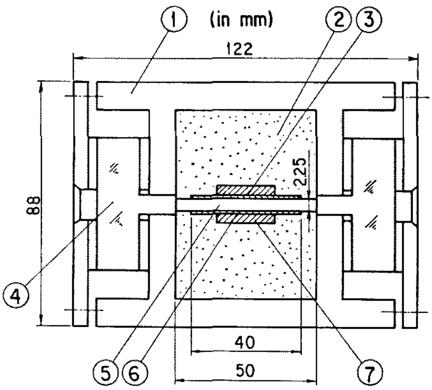


Fig. 1 Schematic diagram of test rig

The test section has a lower plenum and an upper plenum at the lower end and the upper end, respectively. The configuration of flow channel is rectangular with 50 mm in width, 2.25 mm in gap and 750 mm in length, and the flow channel is composed of adjacent two heating plates apart from each other at 2.25 mm. The heating plates are made of Inconel 600 with 1.0 mm in thickness and 40 mm in width. The lengths of heating plates are 750 mm for channel heated from both sides and 370 mm for channel heated from one side. A detailed cross section of the test section is shown in **Fig. 2** for channel heated from both sides. As shown in Fig. 2, the inside of flow channel can be observed through the window made of lucite and the heating plates are mounted on the thermal insulator in the frame of the test section.



① Frame ⑤ Flow channel
② Insulator ⑥ Heating element
③ Thermocouple ⑦ Guide hole for thermocouple
④ Window

Fig. 2 Details of test section

Sheathed thermocouples of 0.5 mm O.D. are attached on the back surface of the heating plates, 20 for one plate and 10 for the other along the longitudinal direction of heating plates, to measure the temperature of heating plates.

As the key items for instrumentation are flow rate, heat input into the flow channel, coolant temperatures at the inlet and outlet of the flow channel, pressures of coolant in the flow channel and surface temperatures of heating plates. The heating plates are heated by direct current and heat input into the flow channel is obtained by measurements of current and voltage for each of heating plates. The inlet and outlet coolant temperatures are measured with 1.6 mm O.D. thermocouples installed in the upper and lower plena.

2. Major Parameters and Test Procedure

Major parameters in this experiment are flow direction (upflow or downflow), heat input, flow rate, inlet coolant temperature for the flow channel and the heating condition, that is, if the channel is heated from one side or both sides. The ranges investigated in this study of these parameters are listed in **Table 1.**

As shown in Table 1, the experiment covers a fairly wide range for *Re* of 100~50,000 corresponding to the velocities of about 0.03~7.0 m/s for both upflow and downflow to investigate the differences in heat transfer characteristics between upflow and downflow for a narrow rectangular vertical channel.

Table 1 Test conditions

Flow direction	Upflow, Downflow
Heated length	0.37 m (one side heated), 0.75 m (both sides heated)
Heat input	0.28~30 kW
Flow rate	0.14~45 l/min
Inlet coolant temperature	281~315 K
Reynolds number	100~50,000
Surface temperature	288~377 K

Test procedure is as follows: (1) The flow direction, upflow or downflow, is determined with a specified flow rate. Flow rate is regulated with regulating valves installed before the flow meters and at the bypass line. (2) Specified electric power is supplied to the heating plates. (3) Coolant temperature at the inlet of flow channel is kept constant by regulating the flow rate of the cooling copper coil in the storage tank. After the steady-state condition is established, required items are measured such as flow rate, pressure, inlet and outlet coolant temperatures, heat input and surface temperatures of heating plates.

III. Experimental Results and Discussion

1. Introductory Remarks for Data Reduction

Fundamental variables required to identify heat transfer characteristics in the single-phase forced-convection heat transfer are heat flux q(kW/m^2), local surface temperature T_{wi}(K) of heating plate on the flow channel side, local bulk temperature T_b of coolant and average coolant velocity u in the flow channel.

Heat flux q is uniform over the heating plates because thickness of heating plates is uniform at 1.0 mm, and is defined with the total heat input Q(kW) into the flow channel, width W(m) and length L(m) of heating plates as follows because there are two plates in case of flow channel heated from both sides:

$$q=Q/2WL\ .$$

Total heat input Q is given with the product Q_1 of current I(A) and voltage V(V) applied to the heating plates or with the product Q_2 of coolant flow rate G(kg/h), coolant temperature increase ΔT_b(K) over the flow channel and specific heat of coolant C_P(J/kg·K), where $Q_1=IV\times10^{-3}$ and $Q_2=GC_P\Delta T_b/3.6\times10^6$.

In order to investigate the validity for evaluation of Q, Q_1 is compared with Q_2 in **Fig. 3**. It is considered that the heat loss can be neglected because the agreement of Q_1 with Q_2 is fairly good as shown in Fig. 3, and then any of Q_1 and Q_2 gives a precise value of Q.

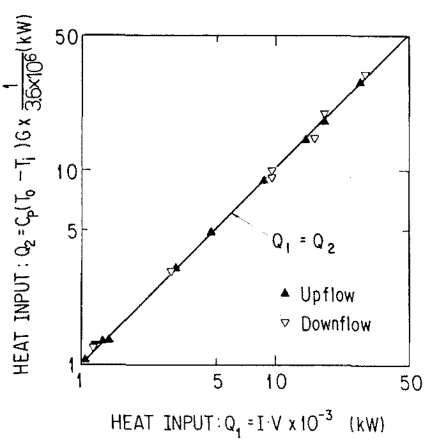


Fig. 3 Comparison of heat input calculation by electric power Q_1 with enthalpy increase of coolant Q_2

As already described, the temperature of heating plates required to identify the heat transfer characteristics is the surface temperature T_{wi} on the flow channel side, but thermocouples are attached on the surface on the thermal insulator side because it is difficult to attach the thermocouples on the flow channel side. Therefore, under the assumption of no heat loss into the thermal insulator, which is confirmed in Fig. 3, T_{wi} was obtained as follows with the heat generation rate $\dot{q}$(kW/m^3) in the heating plates, local surface temperature T_{wo} of heating plates measured on the thermal insulator side, thickness S(m) and thermal conductivity k(kW/m·K) of heating plates.

$$T_{wi}=T_{wo}-\frac{\dot{q}\cdot S^2}{2k}, \qquad \text{where} \quad \dot{q}=Q/(2SWL).$$

Local bulk temperature T_b of coolant at the distance x from the inlet of flow channel is obtained with the inlet and outlet temperatures T_i and T_o and length L of heating plates as follows:

$$T_b=T_i+(T_o-T_i)(x/L).$$

Figure 4 shows an example of test results of T_{wo} and T_b along the longitudinal direction of flow channel for the test conditions of Q=2.0 kW, G=179 kg/h for upflow, and Q=1.95 kW, G=181 kg/h for downflow. The flow channel is heated from both sides. For this case the temperature difference between T_{wo} and T_{wi} is about 1.1 K.

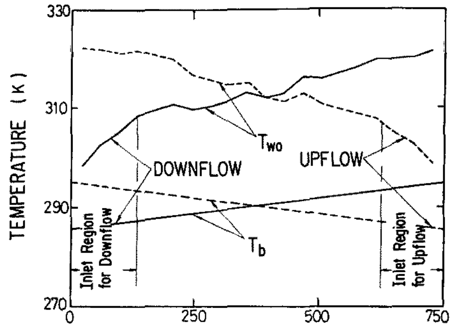


Fig. 4 Illustration of profiles of surface temperature T_{wo} and bulk temperature T_b of coolant along longitudinal direction of heating plates

It should be noted here that there are inlet transition regions for x/De (x: Distance from the inlet of flow channel, De: Equivalent hydraulic diameter) of 0~25 in the neighborhood of the inlet of flow channel for both upflow and downflow. Therefore, the data for x/De <25 are omitted for data reduction of turbulent forced-convection heat transfer. It should be mentioned here on the experimental error that the change of T_{wo} along the heating plate is not smooth a little. This should be primarily because the conditions of welding of thermocouples on the heating plate are different from one by one. In Fig. 4 the maximum deviation of T_{wo} from the average change along the heating plate is about ±1 K, and the temperature difference between T_{wi} and T_b is about 20 K. Therefore, the experimental error due to the uncertainty of T_{wo} is about ±5% with respect to the estimate of heat flux. In this experiment, the errors of heat flux due to the uncertainty of T_{wo} are within ±5%.

2. Heat Transfer Characteristics for Turbulent Flow Region

Figures 5 and 6 show the comparison of the experimental results for Re larger than 1,000 with the existing correlations for turbulent forced-convection heat transfer. The existing correlations compared with are those proposed by Dittus-Boelter[2], Sieder-Tate[3] and Colburn[4], which are shown below.

Dittus-Boelter : $Nu=0.023Re_b^{0.8}Pr_b^{0.4}$, (1)

Sieder-Tate : $Nu=0.027Re_b^{0.8}Pr_b^{1/3}(\mu_b/\mu_w)^{0.14}$, (2)

Colburn : $Nu=0.023Re_f^{0.8}Pr_f^{0.3}$, (3)

where Nu is the Nusselt number, Pr the Prandtl number and μ the dynamic viscosity. Subscripts b, f and w indicate that physical properties are evaluated at bulk temperature T_b of coolant, film temperature T_f $(=(T_b+T_{wi})/2)$ of coolant and surface temperature T_{wi} of heating plate on flow channel side, respectively. The Nu and Re are defined as

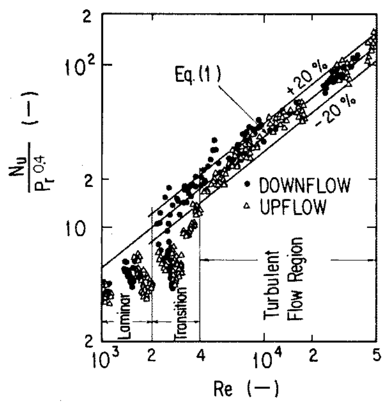
$Nu \equiv h \cdot De/k$ and $Re \equiv u \cdot De/\nu$, where h is the heat transfer coefficient, De the equivalent hydraulic diameter of flow channel ($=2sW'/(s+W')$), k the thermal conductivity of coolant, u the velocity of coolant in the flow channel and ν the kinematic viscosity, and h is defined as $h=q/(T_{wi}-T_b)$.

Figure 5 shows the comparison of experimental results for upflow and downflow with Eq. (1) in both cases of flow channels heated from both sides and from one side. Pointed out from Fig. 5 are the following:

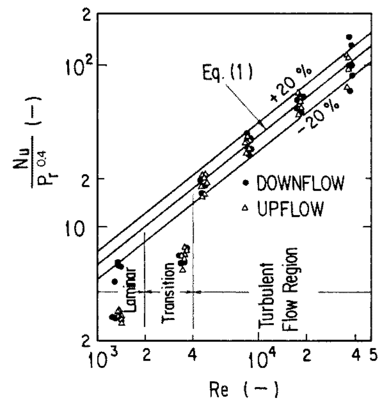
(1) Lowest Reynolds number Re_c for turbulent forced-convection heat transfer is 4,000 though Re_c is a little different between upflow and downflow in the case of flow channel heated from both sides. Equation (1) gives good prediction within the error of $\pm 20\%$ for Re larger than 4,000 for both upflow and downflow in any case of flow channels heated from both sides and from one side.

(2) No significant difference is observed between upflow and downflow in heat transfer characteristics for turbulent flow region, that is, for Re larger than 4,000 where it is considered that no effects of buoyant force are observed.

(3) It is recognized that the transition region between laminar and turbulent flows exists for Re of 2,000~4,000.



(a) Both sides heated



(b) One side heated

Fig. 5(a), (b) Comparisons of experimental results on Nu with Dittus-Boelter's correlation for $Re>10^3$ in upflow and downflow in cases of flow channels which are heated from one side and both sides

Experimental results for Re of 1,000~10,000 are compared with Eqs. (2) and (3) in **Fig. 6** for the flow channel heated from both sides. It is known from Fig. 6 that Eq. (2) gives good prediction with the error of about $-30 \sim +25\%$ and Eq. (3) with the error of about $-20 \sim +50\%$ though the precision of Eqs. (2) and (3) for the experimental results is a little worse than that of Eq. (1) for both upflow and downflow. It should be noted here that Re_c is a little

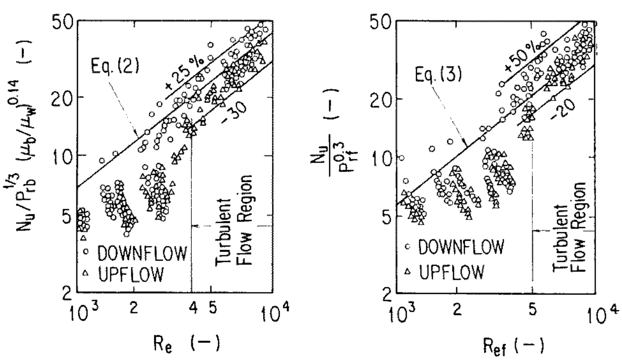


Fig. 6 Comparisons of experimental results on Nu with Sieder-Tate's and Colburn's correlations at $Re>10^3$ in upflow and downflow in cases of flow channel which is heated from both sides

different among the existing correlations when the experimental results are compared with the correlations, 4,000 for Eqs. (1) and (2) and 5,000 for Eq. (3), respectively as seen in Figs. 5 and 6.

From the described above, the characteristics of the turbulent forced-convection heat transfer for narrow vertical flow channel are summarized for both upflow and downflow as follows:

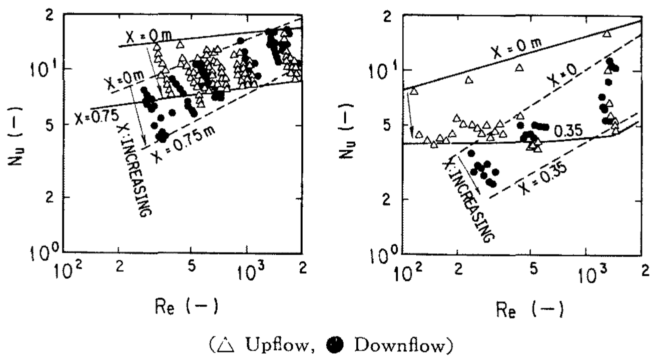
(1) By use of equivalent hydraulic diameter for rectangular channel, any existing heat transfer correlation of Dittus-Boelter, Sieder-Tate and Colburn is available for both upflow and downflow even to the channel whose gap is as narrow as 2.25 mm.

(2) It should be noted that the precision and Re_c for turbulent flow are a little different among the correlations, Eqs. (1)~(3).

(3) No significant differences were observed between upflow and downflow in turbulent forced-convection heat transfer characteristics and therefore, it is considered that there are no effects of buoyant force for the turbulent flow.

3. Heat Transfer Characteristics for *Re* less than 2,000

For *Re* less than 2,000 at which the flow is laminar, **Fig. 7** shows the comparison of experimetal results on *Nu* for upflow with those for downflow in both cases of flow channel heated from both sides and flow channel heated from one side. It is obviously recognized that (i) *Nu* at downflow are smaller than those at upflow for *Re* less than about 700 and (ii) *Nu* for the channel heated from one side are much smaller than *Nu* for the channel heated from both sides for the same *Re* in both upflow and downflow. Especially, it is pointed out that for the same *Re*, *Nu* are scattered in the fairly wide range to the average for both upflow and downflow, as shown with dotted lines for downflow and with solid lines for upflow, respectively, as the upper and lower limits in Fig. 7. Upper limits of *Nu* correspond to *Nu* at the inlet of channel ($x=0$ m) and lower limits to *Nu* at the end of heating plates ($x=0.75$ m for channel heated from both sides and $x=0.35$ m for channel heated from one side). Therefore, for almost the same *Re*, *Nu* decrease with the increase of distance x from the inlet of channel for both upflow and downflow. This result implies that the distance x from the inlet of channel is another key parameter for characterizing the laminar forced-convection heat transfer.



(a) Both sides heated (b) One side heated

Fig. 7(a), (b) Comparison of experimental results on *Nu* for $Re<2,000$ in upflow and downflow between flow channels heated from one side and both sides

Figure 8 shows the analytical results obtained by Hwang *et al.*[7] of *Nu* in the laminar forced-convection heat transfer for a channel between infinite parallel plates heated from both sides. In the analysis, the effects of buoyant force are not taken into account. The results show that *Nu* is strongly dependent on *Pr* and a dimensionless number $(x/De)\cdot(1/Re)$, in which the effect of x is included. The comparisons of experimental results already shown in Fig. 7(a) with the analytical results of Hwang *et al.* along with those of

Kays[6] for circular tube are shown in Fig. 9(a) and (b) for upflow and downflow, respectively. These comparisons are shown only for the channel heated from both sides. In the case of upflow shown in **Fig. 9**(a), analytical results of Hwang *et al.* give a rather good prediction for the experiment though experimental results are larger than analytical ones of Hwang *et al.* for $2/Gz$ less than 0.02 and some are smaller for $2/Gz$ larger than 0.02. In the case of downflow shown in **Fig. 9**(b), on the other hand, almost all experimental results for low velocities less than about 0.3 m/s are smaller than the analytical ones of Hwang *et al.* This indicates that the effects of buoyant force become significant and suppress the heat transfer at the velocities less than about 0.3 m/s in the downflow. It is also pointed out that Nu do not always have a systematic tendency with respect to velocity. This is considered to be because effects of buoyant force are not included in Fig. 9(a) and (b). Therefore, it is strongly suggested that Gr should be adopted as a key parameter representing the effects of buoyant force.

As the lower limits of Nu for channel heated from both sides, which are important to the safety analysis related to the thermohydrodynamics, the following correlations are proposed for upflow and downflow, respectively.

Upflow : $Nu=2.0Gz^{0.3}$ for $Gz \geqq 40$, (4)

$Nu=6.0$ for $16<Gz<40$, (5)

Downflow : $Nu=0.915Gz^{0.4}$ for $Gz \geqq 40$, (6)

$Nu=4.0$ for $16<Gz<40$. (7)

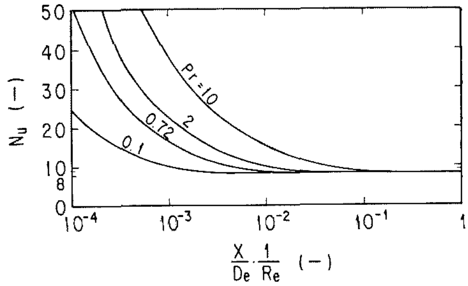
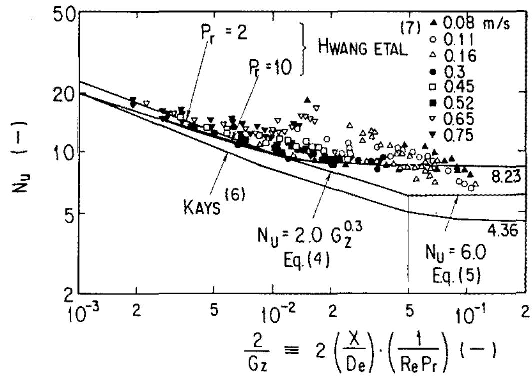
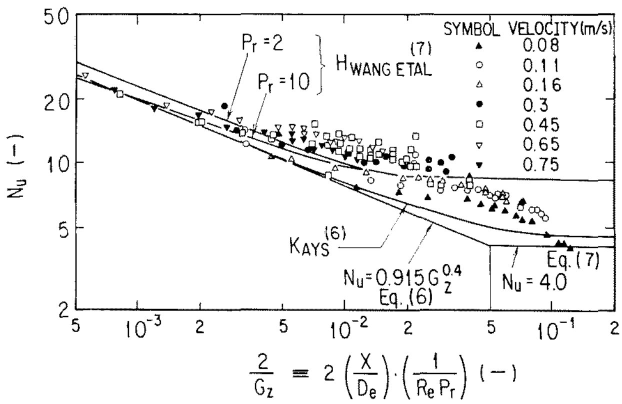


Fig. 8 Theoretical results of Nu for laminar forced-convection heat transfer in rectangular channel heated from both sides, in which no effects of buoyant force are considered (after Hwang *et al.*[7])



(a) UPFLOW



(b) DOWNFLOW

Fig. 9(a), (b) Comparisons of experimental results on Nu for laminar flow with theoretical results of Kays[6] for circular tube and Hwang *et al.*[7] for rectangular channel heated from both sides, in which no effects of buoyant force are considered

The effects of Gr on Nu are shown in **Fig. 10** by adopting Gr on the abscissa and

$Nu/Gz^{0.268}$ on the ordinate, in order to make clear why Nu for upflow is larger than that for downflow as shown in Eqs. (4)~(7). The reason why $Nu/Gz^{0.268}$ is adopted on the ordinate is because Nu is, on the average, proportional to $Gz^{0.268}$ at the constant velocity for all experimental data, as seen in Fig. 9(a) and (b). As shown in Fig. 10(a) and (b), the following correlations are obtained as the lower limits for upflow and downflow, respectively,

$$\text{Upflow}\quad : \; Nu=1.43Gr^{0.1}Gz^{0.268}$$

$$\text{for } 300\leqq Gr\leqq 6{,}000,\; 16\leqq Gz\leqq 4{,}000, \quad (8)$$

$$\text{Downflow}: \; Nu=2.25Gz^{0.268}$$

$$\text{for } 200\leqq Gr\leqq 6{,}000,\; 16\leqq Gr\leqq 4{,}000. \quad (9)$$

As far as the lower limit given with Eq. (8) is concerned, $Nu/Gz^{0.268}$ increases with an increase of Gr for upflow, showing that the increase of Gr enhances heat transfer. On the other hand, Eq. (8) for upflow does not give the lower limit for downflow but Eq. (9) does, which gives lower value of $Nu/Gz^{0.268}$ than Eq. (8) for the same Gr, as shown in Fig. 10(b).

In order to make clearer the effects of buoyant force and differences between upflow and downflow and also between rectangular channel heated from both sides and from one side, **Figs. 11** and **12** are presented by adopting Gr for the abscissa and Nu/Nu_o for the ordinate. Here, Nu_o is the Nusselt number obtained for the rectangular channel heated from both sides from the analysis by Hwang *et al.* under the same Re, Gz and Pr as those for Nu of the experiment. Because the effects of buoyant force are not included in Nu_o and Gr indicates the magnitude of buoyant force, Nu/Nu_o gives the effects of buoyant force as the function of Gr in Figs. 11 and 12. The effects of buoyant force are evident for downflow in Fig. 11(b), that is, Nu/Nu_o is less than 1 and decreases with the increase of Gr in the downflow for Re less than about 700 and Gr larger than about 1,000. Nu/Nu_o is about 0.5 at Re of 250~400 and Gr of 2×10^3~5×10^3. In the upflow shown in Fig. 11(a),

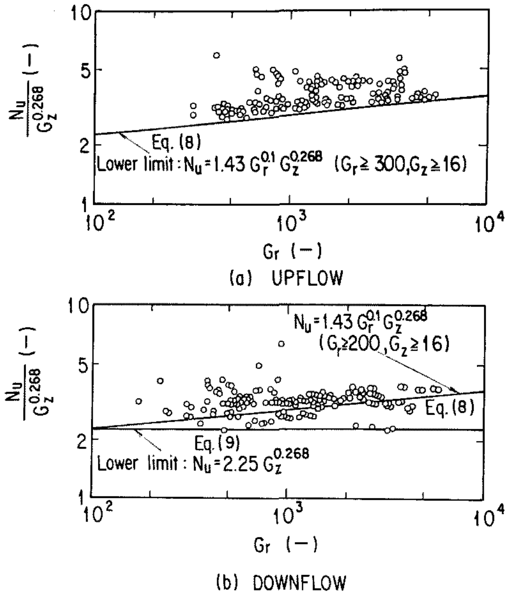


Fig. 10(a), (b) Effects of Graetz number Gz and Grashof number Gr on Nu for rectangular channel heated from both sides in laminar upflow and downflow

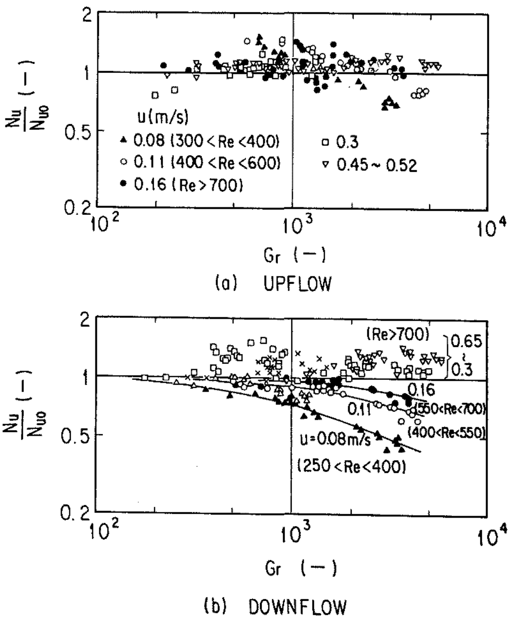


Fig. 11(a), (b) Effects of buoyant force on heat transfer characteristics in laminar upflow and downflow for rectangular channel heated from both sides

on the other hand, the effects of buoyant force are not clear because Nu/Nu_0 is scattered at random around 1.0, although the buoyant force is expected to increase Nu at velocities as low as about 0.1 m/s. However, it is clear that Nu/Nu_0 for upflow is larger than that for downflow at the same Gz and Gr.

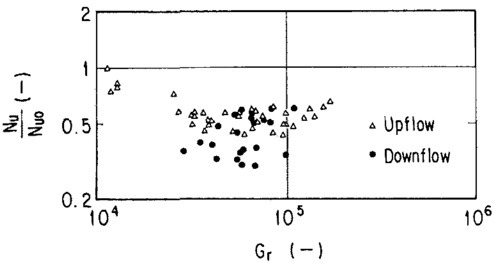


Fig. 12 Effects of buoyant force on heat transfer characteristics in laminar upflow and downflow for rectangular channel heated from one side

Figure 12 shows the relationship of Nu/Nu_0 *vs.* Gr at Re less than 1,000 for the rectangular channel heated from one side and compares the effects of Gr on Nu between upflow and downflow. The ratio of Nu/Nu_0 is less than 1 for both upflow and downflow, and this is a quite different tendency from that for channel heated from both sides. On the other hand, Nu/Nu_0 for downflow is smaller than for upflow and this is the same tendency as that for channel heated from both sides. It should be pointed out that Nu for upflow is about twice as large as that for downflow in case of channel heated from one side with Gr as large as $3\times10^4\sim2\times10^5$. This is almost the same magnitude as the maximum difference between upflow and downflow for channel heated from both sides with Gr of $2\times10^3\sim5\times10^3$, and thus, the corresponding Gr is different between the channels heated from both sides and from one side. The reason for this is not clear at present but it is understood that the flow behavior should be different between the channel heated from one side and from both sides. It is, therefore, suggested that the minute difference of velocity and temperature profiles across the section of flow channel between channel heated from both sides and channel heated from one side should be investigated.

IV. Conclusions and Recommendation

This study investigated the differences in the single-phase forced-convection heat transfer characteristics between upflow and downflow experimentally with a vertical rectangular channel, in order to make clear the applicability of existing correlations to and the effects of buoyant force for the narrow vertical rectangular channel which is simulating a sub-channel in the fuel element of the research reactor, JRR-3.

As the results, the following were made clear:

(1) By use of equivalent hydraulic diameter De $(=2sW'/(s+W'))$, the existing correlations of Dittus-Boelter, Sieder-Tate and Colburn are applicable to a vertical rectangular channel as narrow as 2.25 mm in gap of channel for a turbulent flow region. It should, however, be noted that the precisions of the correlations and critical Reynolds number to the turbulent flow are different among these correlations when compared to the experimental results.

(2) In the laminar flow region, the differences in heat transfer characteristics arise with Re less than about 700 between upflow and downflow. The Nu for downflow is smaller than that for upflow in the range of Re described above. It is considered that this is due to the effects of buoyant force.

(3) As the lower limits of Nu in the laminar flow region, Eqs. (4) and (5) are proposed for upflow and Eqs. (6) and (7) for downflow of the rectangular channel heated from both sides for the conditions investigated in this study $(Gz>16)$.

(4) The differences in heat transfer characteristics between channel heated from both sides and channel heated from one side are observed in the laminar flow at Re less than about 700, with respect to the effects of Gr on Nu. The reason for this is not clear at present but it is recommended that the effects of differences in flow behavior between the channels heated from both sides and from one side should be investigated, related to the velocity and temperature profiles across the section of flow channel.

[NOMENCLATURE]

C_p:	Specific heat of water	(J/kg·K)
De:	Equivalent hydraulic diameter $=2sW'/(s+W')$	(m)
g:	Acceleration of gravity	(m/s²)
G:	Mass flow rate of water	(kg/h)
Gr:	Grashof number $=g\beta De^3(T_{wi}-T_b)/\nu^2$	
Gz:	Graetz number $=Re\cdot Pr\cdot De/x$	
h:	Heat transfer coefficient	(kW/m²·K)
I:	Current	(A)
k:	Thermal conductivity	(kW/m·K)
L:	Length of heating plate	(m)
Nu:	Nusselt number $=hDe/k$	
Nu_o:	Theoretical Nusselt number obtained by Hwang *et al.* for rectangular channel heated from both sides	
Pr:	Prandtl number	
q:	Heat flux	(kW/m²)
$\dot{q}$:	Heat generation rate	(kW/m³)
Q:	Heat input	(kW)
Q_1:	Heat input $=IV\times10^{-3}$	(kW)
Q_2:	Heat input $=GC_p\Delta T_b/3.6\times10^6$	(kW)
Re:	Reynolds number	
Re_c:	Critical Reynolds number	
s:	Gap of rectangular channel	(m)
S:	Thickness of heating plate	(m)
T_b:	Bulk temperature of water	(K)
T_f:	Film temperature of water	(K)
T_i:	Inlet water temperature	(K)
T_o:	Outlet water temperature	(K)
T_{wi}:	Surface temperature of heating plate on channel side	(K)
T_{wo}:	Surface temperature of heating plate on thermal insulator side	(K)
ΔT_b:	Bulk temperature increase through channel $=T_o-T_i$	(K)
u:	Velocity	(m/s)
V:	Voltage	(V)
W:	Width of heating plate	(m)
W':	Width of rectangular channel	(m)
x:	Distance from the inlet of channel	(m)
β:	Volumetric expansion coefficient	(1/K)
μ:	Dynamic viscosity	(Pa·s)
ν:	Kinematic viscosity	(m²/s)

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