Impairment of Local Heat Transfer on the Buoyancy-aided Flow of the Mixed Convection

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1. Introduction

The concept of passive safety is emphasized in design of nuclear power plant to accomplish the enhanced safety goal [1]. The passive cooling system uses the natural convection cooling and heat transfer under the accident conditions to remove the decay heat out of the containment.

The natural convection flow is developed in the passive cooling system devices such as Passive Containment Cooling System by buoyancy forces. Due to the duct flow condition, the flow regime becomes similar to the forced convection. Thus, the flow forms a complex mixed convection [2, 3]. Thus, the local heat transfer along the heated pipe can be partially impaired because of the development of the mixed convective phenomena.

In this paper, the laminar and turbulent mixed convection experiments were performed to investigate the impairment of the local heat transfer along the vertical circular pipe. Based on the analogy concept between heat and mass transfer process, mass transfer experiments replace heat transfer ones. It could be achieved with reasonable test facility heights for the sake of large *Rayleigh numbers*.

2. Previous Studies

Mixed convection occurs when the driving forces of both forced and natural convection are of comparable orders of magnitude [4]. It is not intermediate phenomena with natural convection and forced convection but independent complicated phenomena.

In a vertical pipe, the direction of the forced convection can be either upward or downward flow, the buoyancy force is only upward. When the direction of the forced convection and buoyancy force are the same, the flow is a buoyancy-aided flow. While, the direction of forced convection and buoyancy force are opposite, the flow is a buoyancy-opposed flow. As the exchange of the momentum mechanism, the flow condition is classified into laminar and turbulent flows.

2.1 Laminar Mixed Convection

For the laminar mixed convection, the approximate theoretical model and numerical prediction were successfully understood as physical basis for behavior of laminar heat transfer [5]. In laminar mixed convection, the heat transfer coefficient of buoyancy-aided flow increases relative to the corresponding forced convection due to increasing velocity near the wall [5]. While, the heat transfer coefficient of buoyancy-opposed flow decreases as the flow velocity affected the near wall by the buoyancy forces. This effect leads to the distortion of the velocity profiles. These results which carried out buoyancy-aided and opposed flow were theoretically and experimentally presented, and the numerical studies has been well agreed with theoretical solution to the problem and experimental results [6, 7].

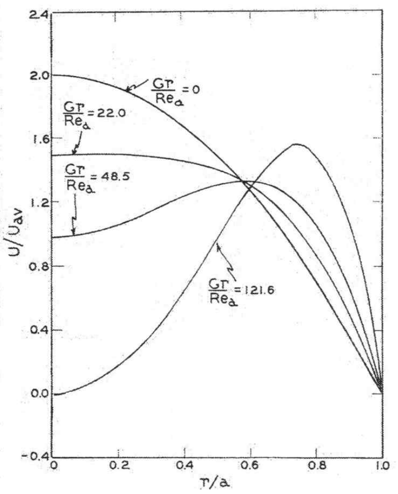


Fig. 1. Velocity profiles inside a vertical pipe at buoyancy-aided flow [5].

2.2 Turbulent Mixed Convection

The phenomena of turbulent mixed convection was complicated and it is difficult to predict the heat transfer [8]. The buoyancy coefficient (B_o) as governing parameter in the mixed convection denotes the relative influence of forced convection and natural convection. Depending on the magnitude of natural convection component in relation to the forced convection one, buoyancy forces can act either to impair or enhance turbulent heat transfer.

In buoyancy-aided flow, the heat transfer shows an impairment at small buoyancy forces as compared with pure forced convection due to the laminarization. And then, the heat transfer presents a gradational enhancement for large buoyancy. While, buoyancy-opposed flow indicates enhanced heat transfer due to increased turbulence production.

There have been experimental and numerical studies investigating local heat transfer coefficient under condition where natural convection has significant impact on forced convection flow [9-14]. The results of existing studies showed the non-monotonous behavior of heat transfer along the vertical pipe appeared. The local heat transfer of the axial position pipe was partly impaired in certain *Gr* and *Re* range correspond to the forced convection [9-12]. Because the flow condition is influenced by locally buoyancy forces. And then, the local heat transfer is recovered or enhanced by buoyancy forces depending on their magnitude. This can be associated with the recovery of turbulence concerned with buoyancy influences. Some studies reported that the double impaired heat transfers along the pipe were presented.

3. Experiments

3.1 Mass Transfer Method

Based on the analogy between heat and mass transfer, heat transfer problem can be solved using mass transfer experiments [15]. The mass transfer coefficient can be calculated from the mass flux (electric current) and the mass concentration difference. The mass transfer coefficient is similar to that of heat transfer. Measurements were made using a cupric acid-copper sulfate (H_2SO_4-$CuSO_4$) electroplating system [16].

3.2 Test Matrix and Apparatus

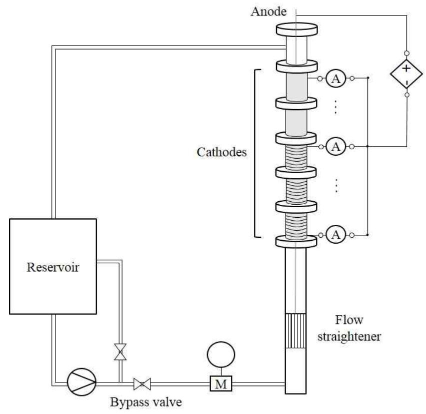
Table 1 shows the test matrix. The concentrations of $CuSO_4$ and H_2SO_4 are 0.1 *M* and 1.5 *M* respectively, which determine the material properties. Thus, the *Prandtl number* was 2,094. The diameter of cathodes was 0.063 *m*. The *Rayleigh number* is adjusted depending on the height of the pipe. *Reynolds numbers* are to be varied from 210 to 16,050 covering the laminar and turbulent mixed convection.

Figures 2 shows the experimental circuit and the test facility. It is a closed loop consisted of a reservoir, a chemical pump, bypass valves, an electromagnetic flow meter and an acryl circular duct included a test section. Fluid flows from the reservoir through the pump and electromagnetic flow meter and then passes through the test section. In order to achieve the hydraulic fully developed condition before entering the test section, the unheated section of enough length of about 1 *m* included 0.2 *m* straighteners was employed.

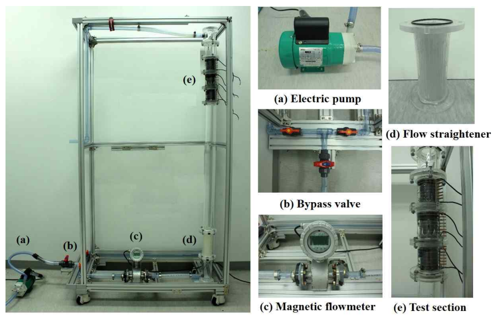
Table 2. Test matrix for the present test facility.

Pr	***D* (*m*)**	***H* (*m*)**	Ra_H	Re_D
2,094	0.063	0.01, 0.10, 0.20, 0.30, 0.50	1.7×10^{8}, 5.7×10^{11}	210 - 16,050

Figure 3 shows an image of the test section as the heated wall. In order to measure the local average *Nusselt number*, the electrode should be segmented. The piecewise electrodes were composed of several single electrodes length of 0.01 *m* and very thin epoxy (0.001 *m*) between the electrodes for insulating their cathodes.



(a) Experimental circuit



(b) Test facility

Fig. 2. Experimental circuit and test facility.

Fig. 3. The photograph of experimental set-up.

4. Results and discussion

4.1 Comparison with Forced Convection Correlations

In order to quantify buoyancy effect in the mixed convective flows, the forced convection experiments were performed at a small *Grashof number* (*H*=0.01 *m*) as weakness of buoyancy forces. These were compared with the existing forced convection correlations under the corresponding to *Reynolds number* conditions. These correlations have hydrodynamically fully developed and thermally developing flow conditions.

Figure 4 (a) indicates the test results together with the laminar forced convection heat transfer correlations (1) of Kays (1955). The measured *Nusselt numbers* have an average error of 14.42% compared with that of Kays's correlation. Therefore, the fitted correlation was suggested in the current test-rig-specific forced convection using the test results for *Reynolds numbers* as shown in eq. (2). Fig. 4 (b) shows the measured *Nusselt numbers* and the turbulent forced convection correlation eq. (3) by Petukhov et al. *Nusselt numbers* were in reasonable agreement with the correlation. The average error was 1.71% with regard to the correlation.

- Laminar flow

$$Nu_D = 3.66 + \frac{0.0668Gz}{1+0.04Gz^{2/3}}, \quad Pr \geq 5 \tag{1}$$

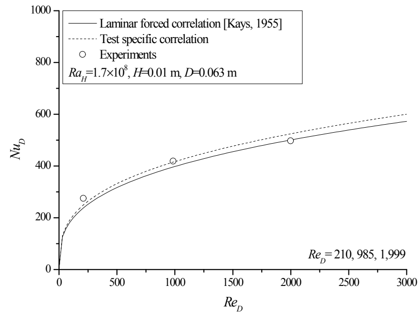
$$Nu_D = 3.66 + \frac{0.075Gz}{1+0.04Gz^{2/3}} \tag{2}$$

- Turbulent flow

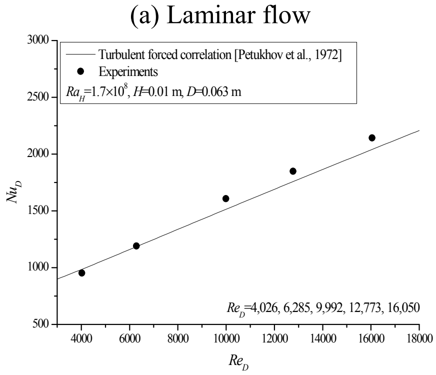
$$Nu_D = C_{therm.} \frac{RePr\left(\frac{f}{8}\right)}{1.07+\left(\frac{900}{Re}\right)-\left(\frac{0.63}{1+10Pr}\right)+12.7\sqrt{\frac{f}{8}}\left(Pr^{3/2}-1\right)} \tag{3}$$

$$C_{therm.} = 1+0.48\left(1+\frac{3600}{Re\sqrt{x/D}}\right)\frac{e^{-0.17(x/D)}}{(x/D)^{0.25}}$$

$$f = \left[1.82log\left(\frac{Re}{8}\right)\right]^{-2}, \quad 4{,}000 \leq Re \leq 5\times10^6$$



(a) Laminar flow

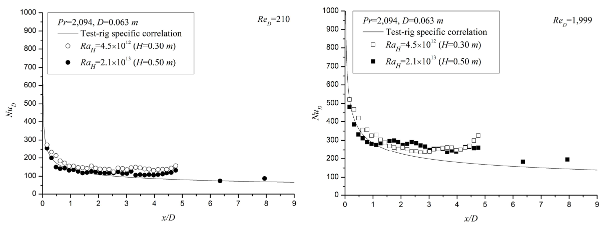


(b) Turbulent flow

Fig. 4. Comparison of laminar and turbulent forced convection in test results and correlations.

4.2 Local Heat Transfer in Laminar Flow

Figure 5 (a), (b) shows the local average *Nusselt numbers* according to the *x*/*D* for the laminar flow in the *Reynolds number* of 210 and 1,999. The results were compared with fitted forced convection correlation (Solid line). The experimental results indicated that the measured *Nusselt number* obtained with the piecewise electrodes were in good agreement with the test specific forced convection correlation at upstream region. While, when the *x*/*D*'s are further increased, the measured *Nusselt numbers* were enhanced compared to the pure forced convection. Because in laminar flow, the velocity on the near wall increased as supplemented the buoyancy force, enhancing the heat transfer.



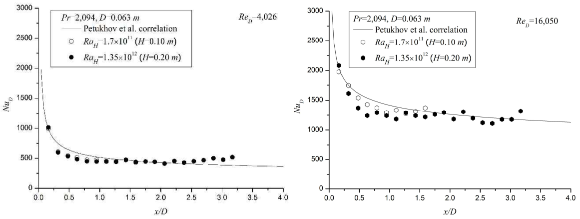
(a) Re_D=210 (b) Re_D=1,999

Fig. 5. *Nu* according to *x*/*D* in a laminar mixed convection.

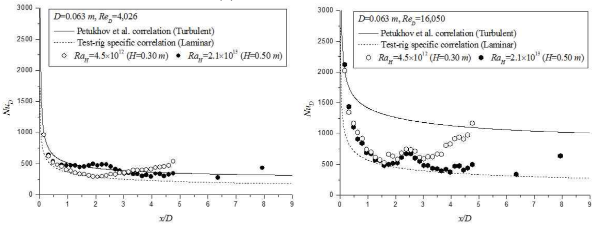
4.3 Local Heat Transfer in Turbulent Flow

Figure 6 (a), (b) shows the local *Nusselt numbers* according to the *x/D*'s for the turbulent flow in the *Reynolds number* of 4,026 and 16,050 at the height ranged from 0.1 *m* to 0.5 *m*. The solid line denotes the forced convection correlation established by Petukhov et al. And the dashed line is test specific correlation of laminar flow.

Figure 6 (a) indicated that *Nusselt number* showed similar with those of correlation for small and large *Reynolds numbers*. However, in Fig. 6 (b) at Re=16,050, the local heat transfer presented non-monotonous behavior as increasing *x/D*. The impairment of heat transfer was due to the laminarization of the turbulent flows under the influence of buoyancy. And then, enhancement of heat transfer was associated with the recovery of turbulence intensity by interacting buoyancy influences. The impairment of heat transfer was intensified with increasing *Reynolds numbers*. Also, the *x/D* position apparent minimum *Nusselt number* moves upward of the vertical pipe as increasing *Reynolds numbers*. The local maximum impairment of the *Nusselt number* was as large as 50 – 60% of that for pure forced convection.



(a) H=0.10 *m*, 0.20 *m*



(b) H=0.30 *m*, 0.50 *m*

Fig. 6. *Nu* according to *x*/*D* in a laminar and turbulent mixed convection.

5. Conclusions

The mixed convection experiments were performed to investigate the impairment of local heat transfer in the vertical pipe. Based on the analogy concept between heat and mass transfer process, the mass transfer experiments were carried out varying *x*/*D*, *Reynolds numbers*, and height of pipe.

In the sufficiently high passive cooling devices, the natural convection flow was developed by buoyant flow. The flow regime becomes similar to the mixed convection due to the duct flow condition. Thus, the main duct flow together with the buoyant flow near the heated wall derives a mixed convection. In buoyancy-aided flow, the local *Nusselt number* distribution under the turbulent mixed convection exhibited the non-monotonic behavior as *x/D*. The impairment of heat transfer is influenced by locally laminarization. And then, recovered or enhanced the heat transfer can be associated with the increase of turbulence production by interacting buoyancy effect and main stream.

This study was concluded that the heat transfer impairment can occur in the passive cooling devices.

ACKNOWLEDGEMENTS

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