Natural convection phenomena of the air in the vertical parallel plates under high wall temperature conditions

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1. Introduction

Nowadays, the heat removal performance of air natural convection under the high wall temperature conditions is an interesting issue in the passive cooling system in Generation-IV Reactor such as Reactor Cavity Cooling System in Very-High-Temperature Reactor, Reactor Vessel Auxiliary Cooling System (RVACS) in Sodium-cooled Fast Reactor.

In RVACS, to predict the heat removal performance in vertical parallel plates under high wall temperature conditions accurately, we must thoroughly understand the natural convection coupled with the thermal radiation phenomenon, which is influential in the high wall temperature conditions.

However, little experimental data of the natural convection heat transfer in the vertical parallel plates under the high wall temperature conditions were presented. M. Miyamoto *et al.* [1] was investigated the turbulent effect in vertical parallel plates. They presented various parameters in natural convection such as wall temperature and the air temperature distribution, turbulent intensity, and induced flow rate. But the radiation effect was not considered because the operating condition was less than 60℃. X. Cheng *et al.* [2] was investigated the turbulent natural convection phenomenon coupled with the thermal radiation effect numerically and experimentally. The target system was AP-600, which is the type of Pressurized Water Reactor (PWR). Therefore, the conducted experimental condition of wall temperature was less than 300 ℃. In addition, the heat transfer analysis was considered only for the heated wall, not for the opposite side wall facing the heated wall (Opposite side wall) that were indirectly heated by the radiation heat transfer.

For understanding the natural convection heat transfer phenomena in the vertical parallel plates, we conduct the air natural convection experiment in vertical parallel plates under high wall temperature conditions up to 500 ℃ on steady state. The heat transfer performance of the heated wall and the opposite side wall are analyzed in terms of localized nondimensional groups (Ra #, Nu #), respectively.

The experiment result is compared to the conventional natural convection heat transfer correlations [3.4]. The effects of the thermal physical properties of the air and the inertial flow near the inlet should be reflected additionally in the correlating equation to predict local heat transfer performance under high wall temperature conditions.

2. Experimental apparatus and data reduction method

2.1 Experimental apparatus

The heated section was made of two vertical stainless-steel walls (Heated wall and opposite side wall), and the nichrome-based heater is attached to the backside of the heated wall. The heated section was divided into five units for providing the electric power uniformly in each section. The temperatures of both walls and air temperature in the vertical channel were measured using K-type thermocouples and the flow rate was measured at the downcomer using thermal mass type velocimetry.

Detailed information of the experimental apparatus is described in Ref. [5].

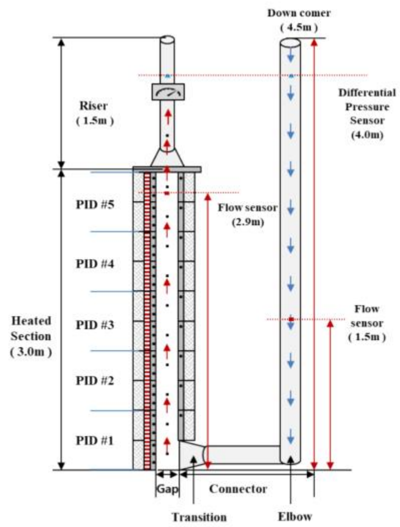

Fig. 1. Air natural convection experimental apparatus

2.2 Data reduction method

In this section, the data reduction methods are described. All thermal physical properties of air were considered based on the film temperature (T_{film}) using each wall temperature and the ambient temperature.

To evaluate the heat removal performance in each case, the enthalpy rise in the channel was calculated based on the energy balance equation using the measured flow rate, inlet, and outlet temperatures.

$$Q_{removal} = \dot{m}C_p\left(T_{Air,Out} - T_\infty\right) \quad (1)$$

Based on the assumption that the net radiation heat transported from the heated wall to the opposite side wall is equal to the convective heat transported from the opposite side wall to the air in the vertical channel on steady state, the energy balance equation was established as follows.

$$Q_{removal} = Q_{Conv,W1} + Q_{Rad,W1} = Q_{Conv,W1} + Q_{Conv,W2} \quad (2)$$

The radiation heat was calculated using the radiation network approach [6] as follows.

$$Q_{Rad,i} = \frac{\sigma T_i^4 - J_i}{(1-\varepsilon_i)/\varepsilon_i A_i} = \sum_{j=1}^{N} \frac{J_i - J_j}{\left(A_i F_{ij}\right)^{-1}} \quad (3)$$

The conducted experimental results are noted in Table 1.

Table 1: Experimental results in each case

	Case 1	Case 2	Case 3	Case 4
$Q_{removal}$ (kW)	1.04	1.97	3.18	4.51

3. Results and conclusions

3.1 Experimental results on the heated wall

As shown in Fig. (2), the local Nusselt number in case 1 is in agreement with the predicted results from laminar [3] to turbulent regime [4] closely. However, as the operating condition increases, the Nusselt number in the other cases are not matched with the predicted results. Some previous studies were reported that additional consideration for the ratio between the wall temperature and ambient temperature (T_W/T_∞) should be required to predict the heat transfer performance in the natural convection of gas [7, 8].

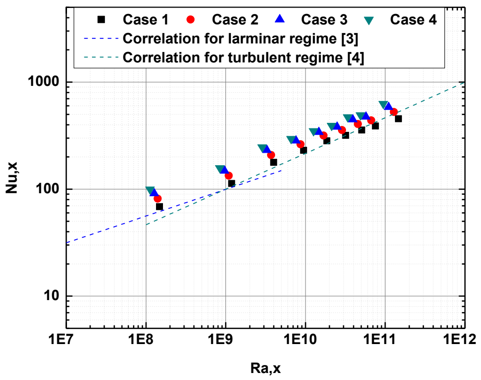

Fig. 2. Comparison between normalized data on the heated wall and natural convection correlations

3.2 Experimental results on the opposite side wall

In Fig. (3), the local Nusselt number in cases 3 and 4 is well matched with the predicted result of heat transfer correlation for the turbulent regime in range of the Rayleigh number over 10^{10} (Corresponded height from 1.5 m to 3.0 m). However, the other two cases are mismatched with the predicted result in the same range. It could be explained the effect of the temperature ratio between the wall and bulk described above.

Interestingly, in the range of the Rayleigh number less than 10^{10} (Corresponded height from 0 to 1.5 m), the local Nusselt number is extensively higher than the predicted results. It might be explained by the additional effect caused by the inertia flow near the inlet. We were expected the inertial flow effect near the wall reported in Ref. [5] based on the measured temperature distributions of both walls and horizontal air temperature profile. Although the flow in the channel could be oriented by the natural convection in vertical walls, the inertial flow in the unheated region (Downcomer, Connector) generated by the suction in the heated channel. This could accelerate the flow originated by the natural convection near the opposite side wall by the forced convection effect caused by the inertial flow near the inlet.

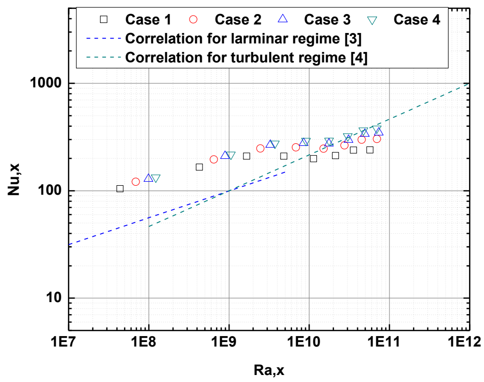

Fig. 3. Comparison between normalized data on the opposite side wall and natural convection correlations

3.3 Conclusions

In this study, the natural convection of the air in the vertical parallel plates under the high wall temperature conditions is investigated to predict the heat transfer performance accurately.

The heat transfer performances of the heated wall and the opposite side wall are analyzed, respectively, in terms of nondimensional groups (Ra#, Nu#). As a result, the temperature ratio effect should be considered in the conventional heat transfer correlations additionally on both walls. In addition, the effect of the inertial flow near the inlet, which was expected in our previous research results [5], also should be considered to predict the local heat transfer performance well.

For our further research work, the mixed convection analysis, which could consider the two different effects (the temperature ratio and the inertial flow near the inlet) simultaneously, will be conducted to develop the correlating equation for the prediction of local heat transfer performance of both walls simultaneously.

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NOMENCLATURE

A : Heat transfer area
T_∞ : Ambient temperature
T_W : Wall temperature
T_{film} : Film temperature ($=(T_W+T_\infty)/2$)
$T_{Air,Out}$: Air temperature at the outlet
ΔT : Temperature difference ($=T_{Air,Out}-T_\infty$)
C_P : Specific heat
F : View factor
h : Convective heat transfer coefficient
J : Radiosity
k : Thermal conductivity
$\dot{m}$: Mass flow rate
Nu_x : Local Nusselt number ($=hx/k$)
$Q_{removal}$: Heat removal rate
Q_{Conv} : Convection heat rate
Q_{Rad} : Radiation heat rate
q'' : Heat flux
Ra_x : Local Rayleigh number ($=g\beta(T_W-T_\infty)x^3/\nu\alpha$)
α : Thermal diffusivity
β : Thermal expansion coefficient
ε : Emissivity
ν : Kinematic viscosity
σ : Plank constant

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