The Natural Convection Heat Transfer in a Vertical Pipe varying length, diameter

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1. Introduction

The passive containment cooling system (PCCS) driven by natural forces convection is gaining research interests after Fukushima NPP accident [1]. Many passive cooling components are being considered for an ultimate heat sink in containment vessel. Thus phenomenological research of PCCS is needed.

The considered passive cooling system can be conceived as the heat exchanger consist of several pipe arrangements which are connected to cooling tank outside the containment, The performance of the PCCS is affected by the geometries of the PCCS, containment atmospheric conditions, accident scenarios, etc.

This study aimed at investigating the influence of the length, diameter for the heat exchanger pipe of the PCCS. The length and diameter of the pipe was varied from 0.1 to 1.0m and from 0.002 to 0.03m, respectively.

Based on the analogy, heat transfer experiments were replaced by mass transfer experiments using sulfuric acid-copper sulfate (H_2SO_4-$CuSO_4$) electroplating system.[2] Numerical analyses were carried out using the commercial CFD program FLUENT 6.3 [3].

2. Previous studies

2.1 Natural convection on an inside vertical cylinder

The buoyancy causes the upward flow in a vertical pipe. Boundary layers developed along the inner wall do not meet at center of large diameter, short length pipe and the heat transfers are similar to that at vertical plate of open channel [4].

Le Fevre suggested a correlation (1) for the laminar natural convection heat transfer on a vertical plate. Also Fouad suggested a correlation (2) for the turbulent natural convection on a vertical plate

$$Nu_L = 0.67(Gr_L Pr)^{0.25} \quad \text{at} \quad Gr_L < 10^9 \qquad (1)$$

$$Nu_L = 0.31(Gr_L Pr)^{0.28} \quad \text{at} \quad Gr_L > 10^9 \qquad (2)$$

Meanwhile, for small D/L vertical pipes, the boundary layers formed by two walls merge into a single buoyant stream rising through the chimney. [5] Flows through the buoyancy were accelerated by the chimney effect, and the heat transfer is enhanced with the resulting increase in flow rate. [6]

3. Experiments

3.1 Test apparatus and matrix

The experimental apparatus and schematic diagram are presented in Fig. 1. The anode rod is located inside the vertical cathode pipe. The test matrix is shown in Table I. The diameter (D) was varied from 0.005 to 0.03m for all lengths (L).

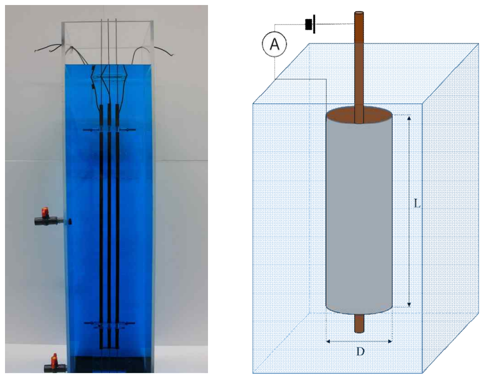

Fig. 1. Test apparatus and circuit.

Table I: Test matrix of experiments for vertical pipe.

Pr	***D*(10^{-3}m)**	***L*(m)**	***Gr$_L$***
2,014	5, 10	0.2, 0.3, 0.4, 0.5, 0.6, 0.8, 1.0	$4.2\times10^7 \sim 4.2\times10^{10}$

3.2 Numerical analysis

In order to investigate the thinner diameter cases of under 0.005m for Prandtl number is 2,014 and the all cases for Prandtl number is 7. Numerical simulations were performed using commercial CFD program FLUENT 6.3[2]. The test model geometry and material properties are presented Table II and III.

Table II: Test matrix of experiments for vertical pipe.

Pr	***D*(10^{-3}m)**	***L*(m)**	***Gr$_L$***
7, 2,014	3, 5, 10, 30	0.2, 0.3, 0.4, 0.5, 0.6, 0.8, 1.0	$4.2\times10^7 \sim 4.2\times10^{10}$

Table III : The test model material properties.

Density (kg/m^3)	1,096.6
Specific heat ($J/kg\text{-}K$)	3.5, 1,000
Thermal conductivity ($W/m\text{-}K$)	6.23×10^{-4}
Viscosity ($kg/m\text{-}s$)	1.253×10^{-3}
Pr	7, 2,014

The simulation was carried out using Boussinesq approximation. Sparart-Allmaras turbulence model under the length range from 0.3m to 1.0m to solve the eddy-viscosity. The y+ ranges from 0.7 to 0.8. The temperature gradient between wall and fluid was 100K to establish a constant temperature condition. A second-order upwind scheme for energy and momentum equation and SIMPLE algorithm were employed.

4. Results and Discussion

Fig. 2 shows the measured and calculated Nu_L with the predictions from natural convection heat transfer correlations. The measured Nu_L's agree with LeFevre laminar natural convection heat transfer correlations for short pipe of Grashof number under 10^9. As the Grashof number increase, the measured Nu_L's agree with Fouad turbulent natural convection heat transfer correlations.

Calculation results also agree with the correlations within 5% error. This explains that the natural convection heat transfer inside the vertical pipe is very similar to that on vertical wall.

For Pr=7, only calculated results for D=0.03m agree with the correlations only. Others results were all lower than the correlations. This is due to the relatively faster boundary layer development in Pr=7 rather than Pr=2,014. It cause more boundary layer interference and heat transfer impairs, as the length of the pipe increase.

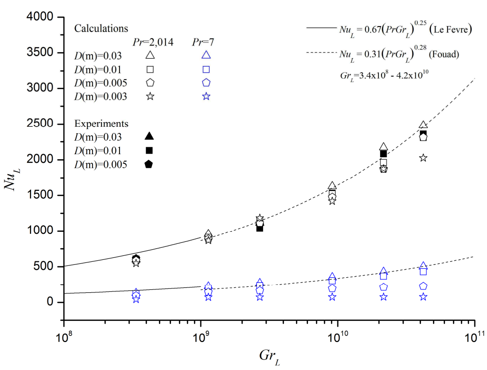

Fig. 2. Comparison of test results to correlations.

Fig. 3 shows heat transfer coefficient according to the diameter at Pr=2,014. The heat transfer rates increase sequentially with the pipe diameter for all length. As the thin boundary layers do not interfere at the high Prandtl number.

The heat transfer coefficient were highest for length 0.2m pipe cases regardless of diameters as the boundary layer develops from the leading edge where heat transfer is most effective.

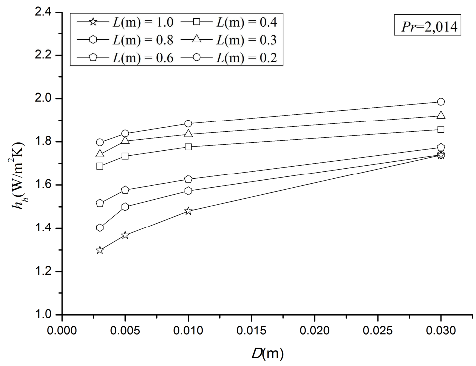

Fig. 3. h_h according to the diameter of vertical pipe(Pr=2,014).

Fig. 4 shows heat transfer coefficient according to the diameter at Pr=7. As the diameter decrease, heat transfer coefficient reduces steeply it. The chimney effect seems not appeared. For narrow pipe cases, the boundary layers interfere each other and seem to increase friction loss.

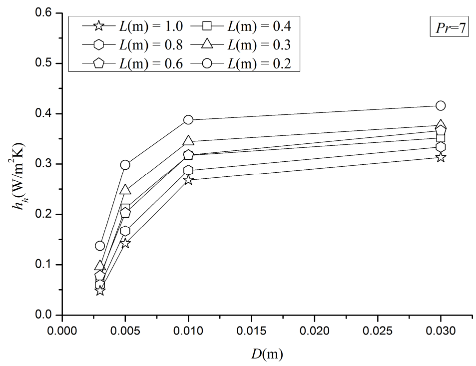

Fig. 4. h_h according to the diameter of vertical pipe(Pr=7).

5. Conclusions

Natural convection heat transfer rates inside vertical pipes were measured. Based on the analogy concept between heat and mass transfer system and mass transfer experiment were carried out using sulfuric acid-copper sulfate (H_2SO_4-$CuSO_4$) electroplating system.

For thinner pipes of diameter under 0.005m and changing Prandtl number, numerical analyses were performed.

We expect that heat transfer rate enhanced as diameter being smaller. But there is little difference of heat transfer between diameter 0.003m and 0.03m at Prandtl number is 2,014. It shows that there is no boundary layer interference. The shorter the length of pipe, the higher the heat transfer rate as the relative portion of the leading edge is increased where the heat transfer is high due to thinner thermal boundary layer.

When Prandtl number is 7, the heat transfer rates according to the pipe diameter show the knee values: As the diameter of the pipe decrease, the heat transfer coefficient reduces a little until the knee point. Further decrease of the diameter, the heat transfer coefficient reduces sharply. It seems that the friction effect from the interfered boundary layer is more dominant than chimney effect after the knee point.

ACKNOWLEDGMENT

This study was sponsored by the Ministry of Science, ICT and future Planning(MSIP) and was supported by Nuclear Research and Development program grant funded by the National Research Foundation (NRF) (Grant code:2012M2A8A2025679)

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