Journal of the Korean Nuclear Society
Volume 26, Number 4, December 1994

An Experiment of Natural Circulated Air Flow and Heat Transfer in the Passive Containment Cooling System

S.H. Ryu, S.M. Oh, and G.C. Park
Seoul National University
(Received May 9, 1994)

격납용기 피동냉각계통내 자연순환 공기유량 및 열전달 실험연구

류석희 · 오승민 · 박군철
서울대학교
(1994. 5. 9 접수)

Abstract

Since the TMI and Chernobyl accidents, many passive safety features are suggested in advanced reactors in order to enhance the safety in future nuclear power plants. In order to verify the effectiveness and provide the data for detailed design of passive cooling system, in the present work, the effects of air inlet position and external condition on the natural circulated air flow rate and the natural and forced convective heat transfer coefficients have been investigated for the one-side heated closed path such as the passive containment cooling system of the Westinghouse's AP-600. A series of experiments have been performed with the 1/26th scaled segment type test facility of the AP-600 passive containment. Under natural and forced convection, the air velocities and temperatures are measured at several points of the air flow path. The experimental results are compared with a simple one-dimensional model and it shows a good agreement.

요 약

TMI 및 Chernobyl 사고이후 향후 원전에 대한 안전성 향상을 강화하기위해 개량형 원전에 대해 여러가지 피동형 안전설비가 제안되고 있다. 피동냉각계통의 타당성을 검증하고 상세 설계자료를 제공하기 위해, 본 연구는 웨스팅하우스사의 AP-600 피동격납용기와 같은 한쪽 가열면을 갖는 폐쇄유로에 대한 공기 유입구 위치 및 외부영향이 자연순환 공기유량에 미치는 영향과 자연 및 강제대류하에서 대류열전달계수를 조사하였다. 본 실험은 AP-600 격납용기를 1/26로 축소한 segment 유형의 실험장치를 토대로 수행되었다. 자연 및 강제대류 조건하의 공기유로내 특정 위치에서 공기의 속도 및 온도를 측정하였다. 실험결과는 일차원 단순 모델과 비교하였으며, 좋은 일치점을 보였다.

1. Introduction

Since the TMI and Chernobyl accidents, a significant improvement in the safety of the future nuclear power plants has been demanded. Thus, suggested alternative is passive reactors, which adopt passive safety features operated by gravity or natural circulation for accident conditions. Those systems should be simple, reliable, and minimize the operator action. The AP-600 is one of such reactors developed by Westinghouse. In the AP-600 [1], several passive safety systems are to be installed including the passive containment cooling system(PCCS) as ultimate heat sink to prevent the containment shell from exceeding its design pressure, as shown in Figure 1.

The system uses natural air circulation between the steel shell containment and the air baffle, whose cooling is enhanced by draining water onto the steel shell. In order to verify the effectiveness of this system and to provide the data for detailed design, many experimental and analytical works have been performed or still ongoing for the heat transfer, air flow, water distribution and so on [2~6].

The AP-600 has the air inlet at the top of shielding building to exclude the effect of surrounding buildings on the inlet air flow, instead of the bottom inlet as in the Ebasco's NPR-HWRF [3, 4] containment system and the B&W's ASPWR which has more benefit in air natural circulation. The Westinghouse has performed the wind tunnel test to determine the optimum location for the PCCS air inlet and the effect of wind direction.

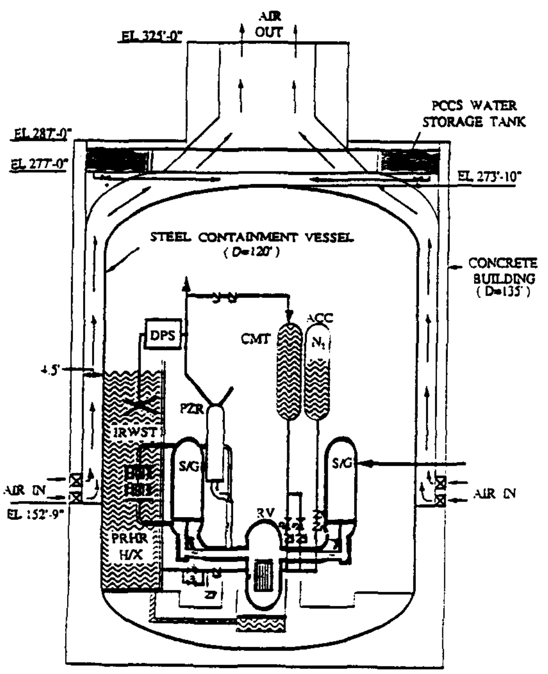

Fig. 1. AP600 Containment

In this study, the effects of air inlet position and external conditions on natural circulated air flow rate were investigated with the 1/26th length scaled AP-600 containment. The test facility is composed of 150cm high and 30cm wide steel containment including up and downward air paths, heating chamber, circulating pump and blower. The gap sizes of the inner and outer air paths are 1.5 and 3cm, respectively. The natural circulated air flow velocity is measured at the inner air path with hot-wire anemometer. Also, the temperatures are measured in the air path and at the wall surface with thermocouples.

The experiments have been performed with varying the size of air inlet, air inlet temperature, and air inlet position. Additionally, the effect of external wind has been investigated. As the second task in this study, the convective heat transfer coefficients are measured for such geometry as one-side heated closed path of the PCCS to provide the design data. Measurements are implemented under natural and forced convection. The experimental correlations fitted as a function of Re and compared with other correlations.

Finally, to compare with experimental results, numerical calculations are performed with a one dimensional steady state air natural circulation model.

2. Experimental Facility and Measurements

The objective of the present experiment is to investigate the characteristics of the PCCS cooling by natural circulated air flow in the AP-600. Before manufacturing the test facility, a scaling analysis has been performed [7, 8].

2.1. Scaling Analysis

In this study, the scale analysis is performed to predict the flow characteristcs. The governing equations are set to consider the effects of buoyancy force and shear force with convective heat transfer from the outside wall of the containment to air in the air path. Nondimensional forms of the governing equations are given as [9~13],

Governing Equations:

–Continuity equation

$$\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}=0 \tag{1}$$

–Momentum equation

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-g\beta\Delta T+\nu\left[\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right] \tag{2}$$

–Energy equation

$$\rho c_p\left[u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right]=k\left[\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right] +\mu\left[2\left(\frac{\partial u}{\partial x}\right)^2+2\left(\frac{\partial v}{\partial y}\right)^2+\left(\frac{\partial v}{\partial x}+\frac{\partial u}{\partial y}\right)^2\right] \tag{3}$$

Nondimensional Forms of the Governing Equations

–Continuity equation

$$U\frac{\partial \rho}{\partial X}+\rho\frac{\partial U}{\partial X}=0 \tag{4}$$

–Momentum equation

$$U\frac{\partial U}{\partial X}=\frac{GrLe}{2Re^2}\theta+\frac{2Le}{Re}\left[\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\Big|_w\right] \tag{5}$$

–Energy equation

$$U\frac{\partial \theta}{\partial X}=\frac{2Le}{PrRe}\frac{\partial^2\theta}{\partial Y^2}+\frac{2EcLe}{Re}\left(\frac{\partial U}{\partial Y}\right)^2 \tag{6}$$

From the above equations, the scaling factor, Gr/Re^2, is chosen in this study because this factor is dominant in natural circulation and other nondimensional parameters such as Pr and Ec have relatively small effects in the case of air. According to this relation, the test facility is manufactured with the same shape of the 1/26th length scaled AP600 containment segment.

2.2. Test Facility

The test facility is a loop, which is composed of containment, heating chamber, air path including chimney, air baffle, blower, electric heater and so on, as shown in Figure 2. The dimensions of the steel containment, which has the same shape as AP-600, are 150cm in height, 30cm in width, and 72cm in radius. The gap sizes of the inner and outer air paths are 1.5 and 3cm, respectively and they are separated by an acryle air baffle.

The heating chamber of a rectangular duct with 50cm ×50cm ×50cm has 2 heaters with the capacity of 3 kw each, where the water filled in the loop is heated to maintain constant wall temperature of the containment and circulated by a pump.

And at the top, the blower is installed at the opposite position of the air inlet and blows air to investigate the effect of external wind on the top of chimney. In the forced convection test this blower is located at the bottom air inlet and blows air with controlled speed. To prevent heat loss from the containment to atmosphere, the sides except for the containment wall are insulated by ceramic fiber.

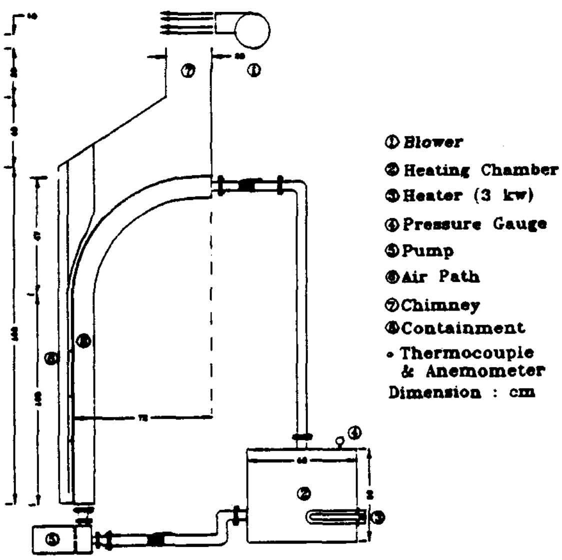

Fig. 2. Configuration of Test Facility

2.3. Measurements

First, in order to investigate the natural air flow, the air velocity at the inner air path and wall temperature are measured by the hot wire anemometer located at the center of the air flow path and stainless steel sheathed thermocouple welded by soldering paste, repectively. In this study, the smoke wire[14] is also used for cross-checking because the reliability of hot-wire detection is known to be questionable for such a low velocity near 1m/sec in the present experiments. However, quantitative comparisons are failed due to the turbulent dispersion for low velocity and difficulty of high-speed photographing.

During experiments, the wall temperature is controlled to maintain constant and recorded. The ambient, inlet, and exit temperatures are measured as well as the external wind velocity as the boundary conditions for various experimental conditions. Under the experiment, the air inlet temperature is controlled because the test facility heats steadily ambient air.

For heat transfer coefficient measurements, the air inlet position is fixed at the bottom both for natural and forced convection. In this experiment, the air velocities and bulk temperatures are measured at the upper and lower positions 10cm apart from the center where the wall surface temperature is measured. Measured positions are located in 30cm, 50cm, and 70cm vertically aprat from the bottom, and they are located in the center of the flow cross section. Then the heat transfer coefficient can be obtained from Equation (7). Those 3-point measurements are conducted in 3 sectors of inner gap and measured values are averaged to obtain the heat transfer coefficient for given Re. For forced convection tests, the gap size is varied to 2.5, 3.0, 3.5, and 5.5cm.

3. Numerical Calculations

In this study, in order to compare with experimental results and use in the P/T calculation for the future, numerical calculations are performed with a one dimensional steady state air natural circulation model. For the sake of simplification, the constant wall temperature and insulated conditions are assumed. In heating region, the air density is not constant and thus expressed as a function of the air temperature. Therefore, the air temperature variation in this region is calculated from the heat balance equation given as,

$$\dot{m} C_p (T_i - T_{i-1}) = h(T_w - T_{bi})\Delta A_i \tag{7}$$

where, $T_{bi} = \dfrac{T_i + T_{i-1}}{2}$

and the heat transfer coefficient is the experimental correlation obtained in this study. Then, the inlet temperature of subsequent node can be obtained from Equation (7)

$$T_i = \frac{(\dot{m} C_p - \frac{h\Delta A_i}{2}) T_{i-1} + h T_w \Delta A_i}{\dot{m} C_p + \frac{h \Delta A_i}{2}} \tag{8}$$

Now, with the density variation at each heated node calculated from the equation of state, one can calculate the air velocity in the natural condition where the total pressure drop and the total buoyancy driven force should be equal. In this calculation, the pressure drops from the friction loss and local loss are obtained as follows,

$$\Delta P_{total} = \sum_i K_i \frac{1}{2} \rho_i u_i^2 \tag{9}$$

In this equation, the loss coefficient for each node is calculated from Ref.[15]. And the buoyancy driven force can be calculated from the following equation,

$$F_{b,total} = \sum_i (\rho_a - \rho_i) g \Delta Z_i \tag{10}$$

Finally, calculations are repeated by modifying the initial air velocity until satisfying the steady state natural circulation condition. The nodalization scheme and calculation flow chart used in the above calculation are shown in Figures 3 and 4.

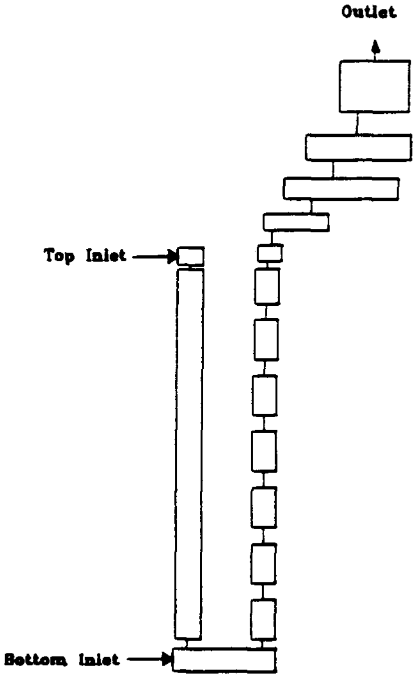

Fig. 3. Nodalization of PCCS

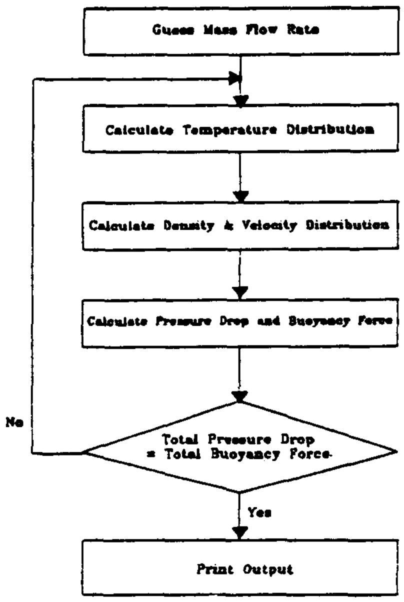

Fig. 4. Flow Chart for Calculation of Natural Circulated Air Flow Rate in PCCS

4. Results

In this study, experiments are performed varying the position(top vs bottom) and size (280mm × 40mm vs 280mm × 14mm) of the air inlet, air inlet temperature(10−25°C) and the velocity(1.20−3.45m/s) of external wind. The results are shown in Figures 5 to 7.

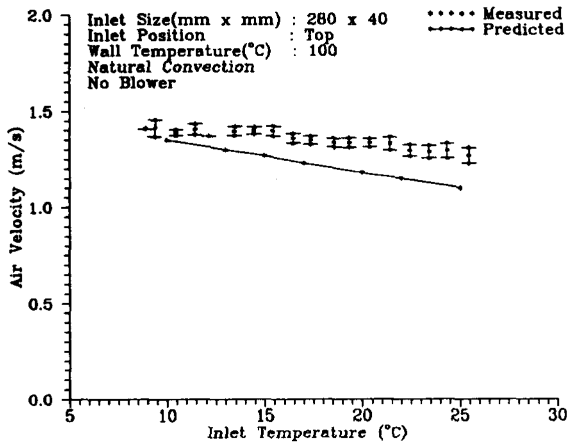

Fig. 5(a). Measured and Predicted Air Velocity vs Air Inlet Temperature(Top)

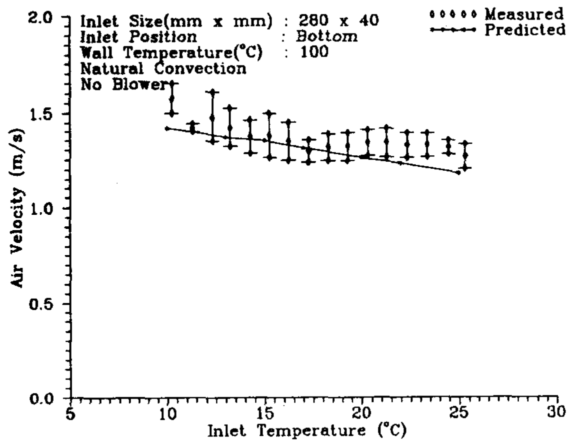

Fig. 5(b). Measured and Predicted Air Velocity vs Air Inlet Temperature(Bottom)

As shown in Figure 5, even though the range of the ambient temperature is narrow due to the difficulty of generating such experimental conditions, the trend is obvious that the air velocity decreases as the inlet air temperature increases because the driving force decreases as the temperature difference between air and wall become smaller. As a result, the effect of the inlet location shows that the bottom inlet has the advantage of heat removal than the top for the same inlet air temperature. Because the air the top inlet is heated by the air baffle as long as air flows downward in the outer air path. The calculations predict quite well the measured velocities for the case of the bottom inlet, but underpredict for the top inlet due to uncertainties of the loss coefficients for such a complicated geometry. From these results, the inlet air temperature greatly affects the natural circulated air flow rate and thus, when the inlet air temperature at the bottom is higher than that at the top (about 7°C in experiment), the bottom position may lose its benefit for natural air circulation. For the bottom inlet, the air velocity increases as inlet temperature decrease. But for the top inlet, the air velocity dose not increase more than the bottom because of the outer air path. As a result, the air velocity pattern has different trend due to the buoyancy force added to air. This will be important in the design of the nuclear power plants where the surrounding buildings may block the air flow and increase the ambient temperature. Calculations for the AP-600 show that such temperature difference is up to 5°C.

Figure 6 shows that the air velocity increases due to the decrease of the flow restriction as the air inlet size increases. Increase of the external wind velocity increases the air velocity as shown in Figure 7 due to the suction effect, but not much.

Next, Figure 8 shows that the heat transfer coefficient increases as the Grashof or Reynolds number increases. It is generally known that the heat transfer coefficient depends mainly on the Grashof number in natural convection. Also, comparisons are made

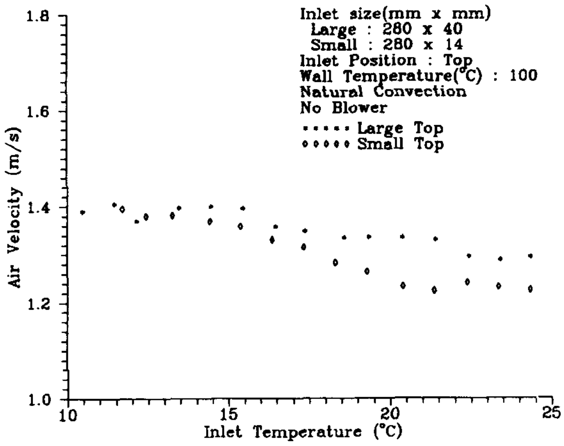

Fig. 6(a). Air Velocity vs Air Inlet Temperature for Inlet Size(Top)

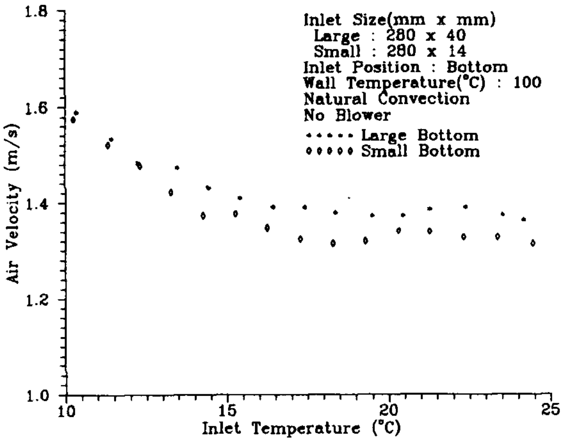

Fig. 6(b). Air Velocity vs Air Inlet Temperature for Inlet Size(Bottom)

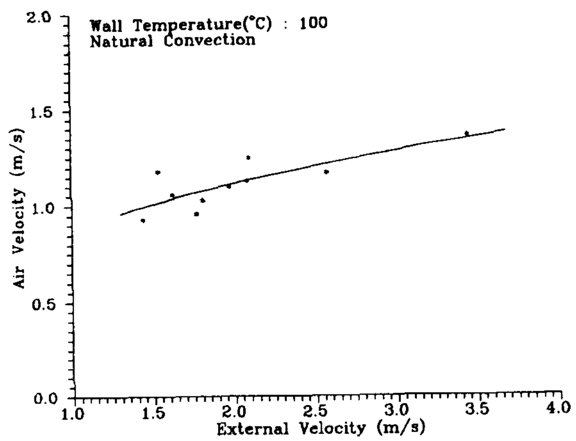

Fig. 7(a). Air Velocity vs External Velocity

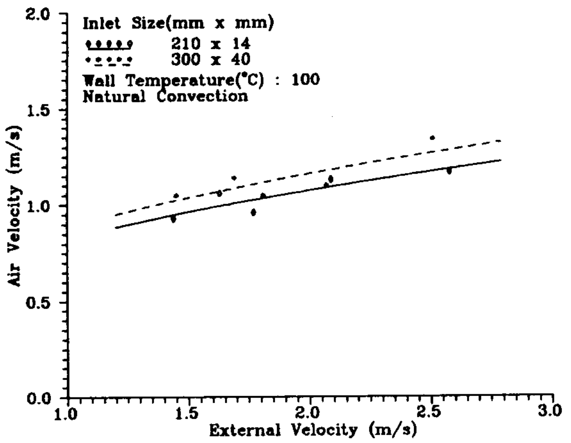

Fig. 7(b). Air Velocity vs External Velocity for Inlet Size

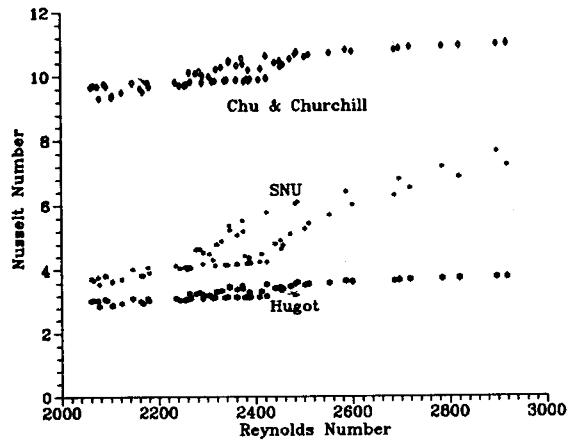

Fig. 8(a). Comparison of Measurement and Other Correlations for Nusselt Number vs Reynolds Number

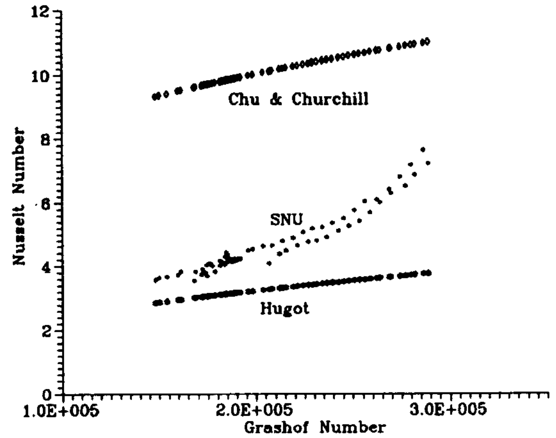

Fig. 8(b). Comparison of Measurement and Other Correlations for Nusselt Number vs Grashof Number

with two other well-known data; Hugot's[16] and Churchill & Chu's[17] experimental data which are obtained in the rectangular both-side heated closed path and the one-side heated open path, respectively. As shown in this figure, the result of this study which is more applicable to the PCCS lies between those data.

Figure 9 shows the final form of the heat transfer coefficient correlations as a function of Re for natural and forced convection, which are obtained by fitting the data, as

$$Nu = 2.494E\text{-}6 \times Re^{1.860} \quad (11)$$

for natural convection

$$Nu = 0.0269 \times Re^{0.728} \quad (12)$$

for forced convection

As the gap size of the air path increases for forced convection in a closed loop, the heat transfer coefficient increases until the opposite wall no longer affects the temperature profile as shown in Figures 10 and 11. Table 1 shows the comparison of the measured and predicted temperatures and it show a good agreement. The radiation effect of the air baffle is out of question because the temperature difference of the bulk and air baffle temperature is small.

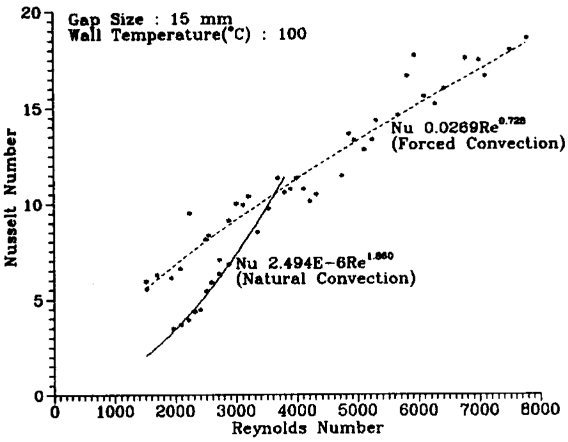

Fig. 9. Nusselt Number vs Reynolds Number in Natural and Forced Convection

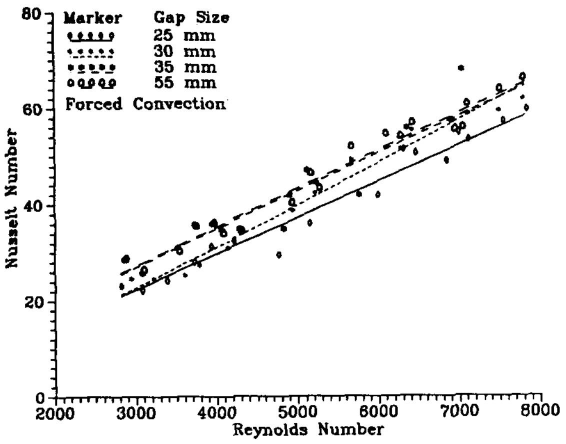

Fig. 10. Nusselt Number vs Reynolds Number for Gap Size

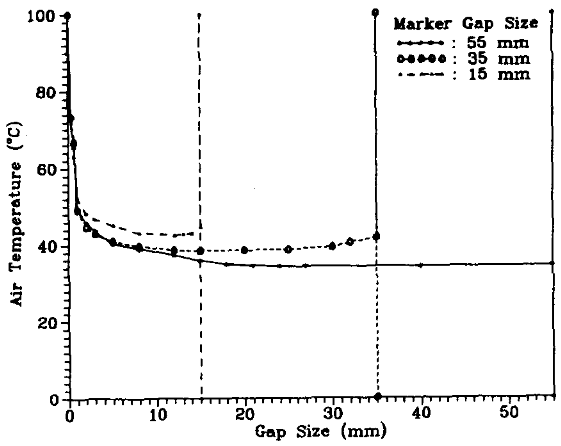

Fig. 11. Temperature Distribution vs Gap Size

Table 1. Comparison of Experiment and Numerical Calculation

Case	Gap Size (mm)	Reynolds Number	Tcmp (Height: 30cm) EXP.	CAL.	Tcmp (Height: 50cm) EXP.	CAL.	Tcmp (Height: 70cm) EXP.	CAL.	Tcmp (Height: 90cm) EXP.	CAL.
1	25	3100	31.9	28.3	35.5	32.7	39.7	36.9	42.6	40.7
2	25	4000	30.7	27.9	35.5	32.2	38.4	36.1	42.7	39.8
3	25	5000	29.7	27.6	33.9	31.6	37.2	35.4	40.8	40.0
4	25	6000	29.2	27.3	33.5	31.3	36.1	34.9	39.0	38.4
5	25	6800	28.4	27.1	32.6	31.0	35.0	34.6	37.9	37.9
6	30	3000	31.8	28.3	36.0	32.7	39.5	36.8	43.4	40.6
7	30	4000	31.3	27.8	35.5	32.0	37.7	35.9	41.5	39.5
8	30	5000	30.6	27.5	34.2	31.5	36.4	35.3	40.7	38.8
9	30	6000	29.2	27.3	33.0	31.1	36.2	34.7	39.6	38.2
10	30	7000	28.7	27.1	32.5	30.8	35.2	34.3	38.8	37.6
11	36	3000	30.8	28.4	34.8	32.7	40.9	37.1	43.5	41.0
12	35	3900	31.3	28.2	35.0	32.5	38.1	36.6	41.8	40.3
13	35	5000	31.0	28.0	34.7	32.2	37.0	36.3	41.9	40.0
14	35	6000	29.8	27.7	33.5	31.9	36.7	35.7	40.4	39.4
15	35	7100	27.9	27.5	31.5	31.5	34.3	35.3	38.4	38.8
16	55	3000	29.2	26.7	32.5	30.3	35.6	33.7	39.1	36.9
17	55	4000	29.1	25.9	32.3	29.0	34.0	31.9	36.5	34.7
18	55	5200	27.5	25.2	32.0	27.9	34.3	30.4	35.8	32.9
19	55	6300	27.9	24.8	31.7	27.2	33.3	29.6	34.6	31.8
20	55	7000	27.6	24.6	31.3	26.9	32.7	29.1	33.7	31.3

5. Conclusions and Discussion

In this study, experiments with the 1/26th scaled segment-type test facility have been performed to investigate the natural circulated air flow and the heat transfer coefficient in the PCCS of the AP600.

The experimental results show that the air velocity increases as the inlet air temperature decreases and the external wind velocity increases. Also, reduction of the air inlet area decreases the natural circulated air flow. Finally, it can be concluded that the effect of outlet steam on the air inlet due to the external wind is small and the bottom position will lose its benefit in natural circulation if the surrounding buildings block the air flow and increase the ambient temperature. In real design, the expected temperature difference may be 5°C instead of 7°C in experiments.

The measured heat transfer coefficient correlations for air flow in the one-side heated closed loop such as in the PCCS are obtained as the final forms of a function of Re for natural and forced convection, which are obtained by fitting the data. The study of the optimum gap size determination will be carried out in the future.

Also, a simple one-dimensional air flow model developed in this study shows a good agreement with experimental results.

Acknowledgement

The authors express great appreciation to the Center for Advanced Reactor Research(CARR) and the Korea Science and Engineering Foundation for their financial support.

Nomenclature

- b : gap size of the air path
- c_p : specific heat of air at constant pressure
- $F_{b,total}$: total byoyancy force in the air path
- g : acceleration of gravity
- h : heat transfer coefficient(w/m2°C)
- K : loss coefficient
- k : thermal conductivity
- L : reference length
- m : mass flow rate of air
- $\triangle P_{total}$: total pressure drop in the air path
- T_{bi} : the bulk temperature of node i
- T_i : the outlet temperature of node i
- T_{i-1} : the inlet temperature of node i
- T_w : heating wall temperature
- T_o : reference temperature
- u : axial velocity
- U : dimensionless axial velocity
- v : transverse velocity
- V : dimensionless transverse velocity (v/U_o)
- x : coordinate in the flow direction
- X : dimensionless coordinate in the axial direction (x/L)
- y : coordinate in the transverse direction
- Y : dimensionless coordinate in the transverse direction
- θ : dimensionless temperature $(T-T_o)/(T_w-T_o)$
- $\triangle A_i$: the heat transfer area of node i
- $\triangle Z_i$: height of node i
- ρ_a : ambient air density
- ρ_i : air density of node i
- μ : dynamic viscosity coefficient
- v : kinematic viscosity coefficient
- β : thermal expansion coefficient
- Gr : Grashof Number ($8g\beta(T_w-T_o)b^3/nu^2$)
- Re : Reynolds Number ($2bU/v$)
- Pr : Prandtl Number ($\mu c_p/k$)
- Le : Length Ratio Number (L/b)
- Ec : Eckert Number ($U^2/c_p(T_w-T_o)$)

References

1. S.N. TOWER, T.L. SCHULZ, and R.P. VIJUK, "Passive and Simplified System Features for the

Advanced Westinghouse 600 MWe PWR," Nucl. Eng. & Design, Vol. 109, pp. 147−154, 1988.

2. PIPLICA, E., "The Westinghouse AP600 PCC Test Analysis Program," Intl' Conf. on Design and Safety of ANP, Vol II, Tokyo, Japan, 1992.
3. WRIGHT, R.F., E.P. PIPLICA, W.A. STEWART, and F. DEIOSE, "Heavy Water Reactor Facility Containment Cooling Verification Program," ANS Trans. Vol. 64, 1991.
4. "Passive Cooling of Containment Shells for the NPR/HWR," EBASCO, July, 1991.
5. MENAKER, B.J., R.E. ALLEN, and H.E. PAINTER, "Passive Containment Cooling for an Advanced Small Pressurized Water Reactor," Nuclear Technology, Vol. 91, 1990.
6. "Simplified Passive Advanced Light Water Reactor," SIR. Program Summary, ABB Combustion Engineering Nuclear Power, June, 1990.
7. T.K. LARSON and R.A. DIMENNA, "Preservation of Natural Circulation Similarity Criteria in Mathematical Models," Nucl. Science & Eng., Vol. 100, pp. 21−32, 1988.
8. WILLIAM M. KAYS and MICHAEL E. CRAWFORD, Convective Heat and Mass Transfer-2ed, McGraw-Hill, 1966.
9. M. AL-ARABI and B. SAKR, "Natural Convection Heat Transfer from Inclined Isothermal Plate," Int. J. Heat Transfer Vol. 31, No. 3, pp. 559−566, 1988.
10. A. LA PICA, G. RODONO, and R. VOLPES, "An Experimental Investigation on Natural Convection of Air in a Vertical Channel," Int. J. Heat Mass Transfer, Vol. 36, No. 3, pp. 611, 1993.
11. V.M. SOUNDALGEKAR, H.S. TAKHAR and N. V. VIGHNESAM, "Combined Free and Forced Convetion Flow Past a Semi-infinite Vertical Plate with Variable Surface Temperature," Nuclear Engineering and Design 110, pp. 95−98, 1988.
12. R, CHEESEWRIGHT, "Natural Convection from a Plane, Vertical Surface in Non-isothermal Surroundings," Int. J. Heat Mass Transfer Vol. 10, pp. 1847−1859, 1967.
13. T.S. CHEN, H.C. TIEN and B.F. ARMALY, "Natural Convection on Horizontal, Inclined and Vertical Plates with Variable Surface Temperature or Heat Flux," Int. J. Heat Mass Transfer, Vol. 29, No. 10, pp. 1465+ −1478, 1986.
14. RICHARD J. GOLDSTEIN, Fluid Mechanics Measurement, Univ. Minn., 1969
15. I.E. Idel'Chik, Hand Book of Hydraulic Resistance, Coefficients of Local Resistance and Friction, 1960.
16. G. HUGOT, "Study of the Natural Convection Between Two Plane, Vertical, Parallel and Isothermal Plates," Ph. D Thesis Univ. Paris, 1972.
17. STUART W. CHURCHILL and HUMBERT H.S. CHU, "Correlating Equation for Laminar and Turbulent Free Convection from a Vertical Plate," 1975.