Optimization methodology for fin geometry on Pool-dry PDHR (Passive Decay Heat Removal system) to enhance air cooling using CFD

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1. Introduction

After the Fukushima accident, passively operated safety system has been reemphasized for decay heat removal even under the station blackout (SBO). Therefore, innovative air cooled condenser design is suggested as shown in Fig. 1 to replace presenting auxiliary feedwater system (PAFS). The system is called Pool-dry PDHR (Passive Decay Heat Removal system). Firstly, the heat exchangers are submerged in a water pool to deal with the high decay heat within 3days because heat transfer rate in a pool boiling is effective. After the water level is decreased because of the evaporation, air path has been opened as shown in the right figure in Fig. 1. The air cooling method is totally passive and permanent way, so the decay heat can be removed even under the station blackout with a proper design of the Pool-dry PDHR.

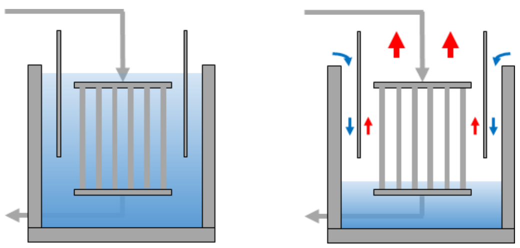

Fig. 1. Schematic of Pool-dry PDHR in early period (Left) and late period (Right).

In order to maximize the heat removal through air cooling, therefore, extended surface so-called fin is adopted. By using fins, heat transfer rate from the same base area is enhanced by increasing surface area. Meanwhile, the extended surface may deteriorate heat transfer rate due to the increase of the resistance of air flow. Hence, the sensitivity of fin geometric parameters such as fin spacing, fin thickness and fin height is investigated to check their influence on thermal performance herein, and the optimizing methodology for fin geometry on Pool-dry PDHR is presented.

2. Methods and Results

In this section unit cell concept used to simulate finned surface is described and overall effectiveness is introduced as a fin performance parameter to investigate the effect of fin geometric parameters.

2.1 Description of a unit cell for CFD simulation

ANSYS Fluent 17.0 was used as a solver. CFD simulation was performed under steady state. RNG k-epsilon model with enhanced wall treatment and DO model were used for turbulent air flow and radiation heat transfer, respectively. Fig. 2 shows a unit cell geometry from top view and side view. Among numerous fins, we defined a unit cell including only one fin and applied symmetry boundary condition for both sides of the unit cell to represent the whole heat exchanger (HX) [1].

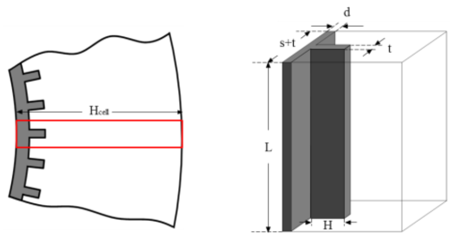

Fig. 2. Unit cell geometry from top view (Left) and side view (Right). [Figure in courtesy of Kim et al., 2015] .

HX inside temperature was 563.15K that is the saturation temperature at 7.4MPa of steam generator and ambient temperature was 319.26K. The height of HX was 10m and the material of HX and fin was stainless steel and aluminum respectively.

In order to check the effect of heat transfer rate enhancement by fins, overall effectiveness (ε_o) is introduced as a fin performance parameter. It is defined as a ratio of the total heat transfer rate from a finned surface to the total heat transfer rate from bare surface with the same base area of a unit cell.

2.2 Optimum spacing

If fin spacing is very small, the air flow between two fins becomes fully developed vertical channel flow. On the other hand, if fin spacing is so large, the boundary layer develops independently without interferences. According to Yazicioğlu and Yüncü [2], convection heat transfer rates from a fin array increases firstly and reaches a maximum and then it decreases as the fin spacing increases. The trend of the convection heat transfer rates along the increase of fin spacing was validated through fin array experiments. Therefore, they defined an optimum spacing as a fin spacing that maximizes the convection heat transfer rates and shows that the optimum spacing is only dependent on fin length and base-to-ambient temperature through order-of-magnitude analysis. It means optimum spacing of our HX has a constant value regardless of fin geometry because the fin length and base-to-ambient temperature of our HX is fixed. Fig. 3 shows obtained overall effectiveness from various fin geometries as a function of fin spacing by using CFD. Fig. 3 shows the fin spacing between 3cm and 4cm maximizes overall effectiveness regardless of fin geometry. However, the overall effectiveness value between 3cm and 4cm is almost the same, and the smaller fin spacing needs the more number of fins that results in the more cost for manufacture and installation of the HXs. Therefore 4cm is selected for the optimal spacing considering the overall effectiveness and the economics in this case.

$$s_{opt} = 1.97 L Ra_L^{-2/9} \quad (1)$$

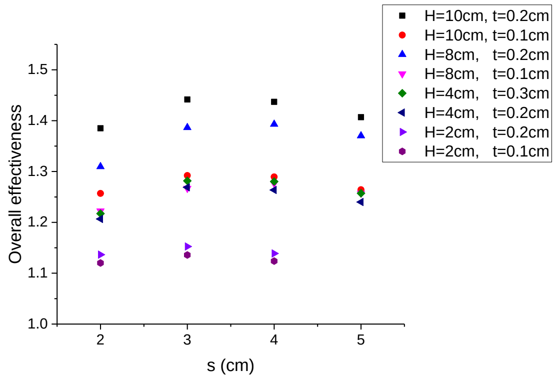

Fig. 3. Overall effectiveness as a function of fin spacing.

2.2 Overall effectiveness correlation

Overall effectiveness correlation is suggested based on the definition of it given in Eq. (1) as follows :

$$\varepsilon_o = \frac{\bar{h}_t}{\bar{h}_{t,o}}\left(\frac{s+t+2H}{s+t}\right)\eta_o \quad (2)$$

$$\eta_o = 1 - \frac{t+2H}{s+t+2H}\left(1 - \frac{\tanh mH_c}{mH_c}\right) \quad (3)$$

Eq. (3) is overall efficiency that corrects the temperature decrease along fin. Using the correlation, overall effectiveness for various fin geometry can be obtained without numerous times of CFD calculation and it is helpful for estimating the configuration of HXs and overall design of Pool-dry PDHR. Heat transferred from a vertical fin array consists of natural convection and radiation. Hence, the total heat transfer coefficient is expressed as the sum of the natural convection and the radiation contribution. Because there are a lot of heat transfer coefficient correlations for bare surface, heat transfer coefficient correlations for finned surface is needed to complete the overall effectiveness correlation.

2.2.1. Natural convection heat transfer coefficient correlation.

Natural convection heat transfer from a fin array has been investigated in many studies. Tari et al. [3] considered a buoyancy driven air flow between two fins and presented governing equations which were continuity, momentum equation and energy equation. Based on the study, they proposed a Nusselt number correlation as a simple form:

$$\overline{Nu_s} = \frac{\bar{h}_c s}{k_a} = C(Gr'\mathrm{Pr})^n \quad (4)$$

They used n=1/3 for 250< $Gr'\mathrm{Pr}$ <10^6 and they obtained the empirical coefficient C based on their experiment data. In this study, the exponential term in Gr' is modified as $\exp(-ak_aH/k_ft)$ by multiplying an empirical constant a to reflect our CFD results well. We found the optimum thickness of the overall effectiveness correlation is well matched with the CFD results when a is 90.

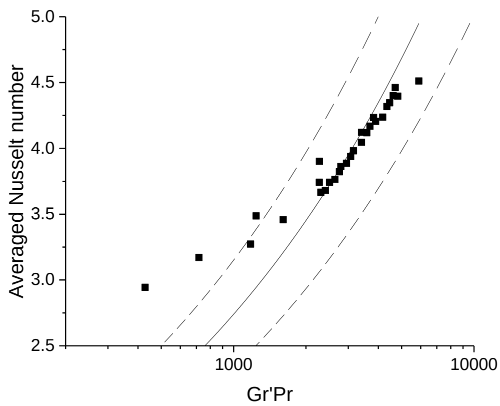

Fig. 4. CFD results of averaged Nusselt number and fitting curve in the form of Eq. (4).

We plotted the averaged Nusselt number obtained from CFD results as a function of $Gr'\mathrm{Pr}$ and found that C is 0.274 by fitting the graph as a functional form of Eq. (4). Fig. 4 shows the averaged Nusselt number correlation is well matched with the CFD results within 15% error that is presented as a dashed line.

2.2.2. Radiation heat transfer coefficient correlation.

In order to estimate the radiation heat transfer coefficients, view factor study was conducted. A view factor F_{ij} is defined as the fraction of the radiation leaving surface i that is intercepted by surface j. For a calculation of a view factor of each channel, surface numbers were assigned as shown in Fig. 5. The view factors from a fin side to outside and from a base to outside were calculated respectively with assumptions that fin length is infinitely long and surface temperature of fins is the same as T_w [4].

$$F_{21} = 1 - F_{23} - F_{24} = 1 - \left(\frac{1+\bar{L}-\sqrt{1+\bar{L}^2}}{2}\right) - \left(\sqrt{1+\bar{L}^2} - \bar{L}\right) = \frac{1+\bar{L}-\sqrt{1+\bar{L}^2}}{2} \quad (5)$$

$$F_{31} = 1 - 2F_{32} = 1 - \frac{2}{\bar{L}} F_{23} = 1 - \frac{2}{\bar{L}}\left(\frac{1+\bar{L}-\sqrt{1+\bar{L}^2}}{2}\right) \quad (6)$$

The view factor analysis shows that the radiation heat transfer rate from the same base area remains the same regardless of the change of fin geometry. Therefore, the radiation heat transfer coefficient is obtained as follows:

$$\bar{h}_{rad} = \frac{s+t}{s+t+2H} \varepsilon\sigma \frac{T_w^{\ 4} - T_a^{\ 4}}{T_w - T_a} \quad (7)$$

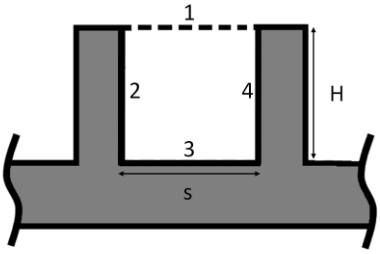

Fig. 5. Allocated surface number for view factor analysis [Figure in courtesy of Kim et al., 2015]

Finally, we can suggest total heat transfer coefficient which is the sum of natural convection and radiation heat transfer coefficient from Eqs. (4) and (7) respectively for a given fin geometry. Furthermore, the overall effectiveness correlation is also established as a form of Eq. (2) based on the Eqs. (3), (4) and (7). Fig. 6 shows our overall effectiveness correlation well predicts the overall effectiveness value for various fin geometry.

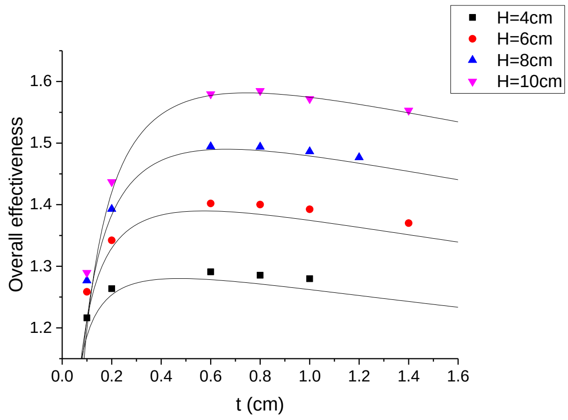

Fig. 6. Comparison of the overall effectiveness correlation with CFD results.

Utilizing the correlation, overall effectiveness can be estimated for a given fin geometry and the information of the overall effectiveness value and the fin geometry is used to design the Pool-dry PDHR because an overall effectiveness is closely related to the sensitivity study of the number of HXs that is needed for decay heat removal, the configuration of HXs, the tank size and the number of the tank and so on.

3. Conclusions

Optimization methodology for fin geometry is established using CFD simulation. Firstly optimum spacing is obtained because it has a unique value for our Pool-dry PDHR. Then, overall effectiveness correlation considering both of natural convection and radiation has been completed comparing with CFD results. Based on the correlation, overall effectiveness for various fin geometry can be estimated and it is utilized to design the Pool-dry PDHR. Furthermore, the optimization methodology presented in this study can be applied to other large scale vertical structure using fin to evaluate the enhancement of heat transfer.

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