OPTIMIZATION METHODOLOGY FOR LARGE SCALE FIN GEOMETRY ON THE STEEL CONTAINMENT OF A PUBLIC ACCEPTABLE SIMPLE SMR (PASS)

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ABSTRACT

Heat removal capability through a steel containment is important in accident situations in order to preserve the integrity of a nuclear power plant which adopts a steel containment concept. A heat transfer rate will be enhanced by using fins on the external surface of the steel containment. The fins, however, create an increase in flow resistance that can deteriorate the heat transfer rate at the same time. This study investigates an optimization methodology of large scale fin geometry for a vertical base where a natural convection flow regime is turbulent. Rectangular plate fins adopted in the steel containment of a Public Acceptable Simple SMR (PASS) is used as a reference. The heat transfer rate through the fins is obtained from CFD tools. A heat transfer coefficient correlation of a vertical fin array considering both natural convection and radiation is suggested. The general functional form of a natural convection heat transfer coefficient is used as the fin effectiveness term considering temperature decrease along the fin height is modified. This is compared with our CFD results. A radiation heat transfer coefficient is obtained analytically by a view factor study. A total heat transfer coefficient is expressed as the sum of the convection heat transfer coefficient and the radiation heat transfer coefficient. Scaling analysis is conducted to show the existence of an optimum spacing which turns out to be 7cm in the case of PASS. In order to optimize fin geometry, an overall effectiveness concept is introduced as a fin performance parameter. The overall effectiveness is expressed as a function of fin geometric parameters, showing that an optimum thickness exists. However, the optimum thickness is changed as a fin height varies. Therefore, optimal fin geometry is obtained as a function of a fin height. With the assumption that the heat removal rate from the finned steel containment is the same as that from the original steel containment, we found out that containment volume and material volume can be reduced as a function of the overall effectiveness; the reduction is 43% of the containment volume and 13% of the material volume as a least-material containment.

KEYWORDS

Rectangular plate fin, Steel containment, Natural convection, Overall effectiveness, Fin optimization methodology

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1. INTRODUCTION

Nuclear safety systems to remove decay heat in an accident situation are important issue for preserving integrity of nuclear power plant. Moreover, passive safety systems have gained attention after the Fukushima accident because their can even operate in station blackout. In the viewpoint of passive containment cooling, a steel containment concept can be introduced. In an accident situation, decay heat is removed by external natural convection of air through the steel containment. It has advantage of being passively operated and it has an infinite amount of the ultimate heat sink which is air. Due to advantages of the steel containment, it is adopted in several nuclear power plants such as AP1000, NuScale and Low-pressure Inherent heat sink Nuclear Desalination plant (LIND) [1]. Public acceptable simple SMR (PASS) [2, 3] also applies the steel containment design concept, which is the reference reactor of this study.

A steel containment should be capable enough to remove decay heat to prevent containment failure. Also, its compact size is desirable to reduce material cost for economics. These two requirements are satisfied by adopting extended surface, so-called fin. By using fins, heat transfer rate from the same base area is enhanced by increasing surface area. It implies the smaller base area is needed to remove the same amount of heat with fins. Then the size of a steel containment becomes smaller and the material cost for construction is also reduced. Meanwhile, extended surface may cause to increase resistance of air flow due to boundary layer interference. In this case, the extended surface would deteriorate heat transfer rate and increase material cost because of the material used for the fin itself. Therefore an optimizing process for fin geometry is needed to enhance heat transfer rate and reduce size of containment and material cost simultaneously, which is the way to improve the economics of PASS.

The objective of this study is to establish an optimization methodology for large scale fin geometry. In order to optimize the fin geometry, we changed fin geometric parameters such as fin spacing, fin thickness and fin height to check their influence on thermal performance. Also, we suggest a heat transfer coefficient correlation of a fin array and optimization methodology for large scale fin geometry by introducing the fin performance parameter, overall effectiveness. At last, optimal fin geometry for the PASS containment is presented and estimation of the reduction of containment volume and material volume is also conducted for evaluating the economics of PASS.

2. PREVIOUS STUDIES

Among the various types of extended surfaces, rectangular plate fins are widely used because of their ease of manufacture and effective cooling capability. Since Elenbaas [4] examined it first, there have been several experimental investigations about natural convection heat transfer of a fin array and optimum spacing as tabulated in Table I. Bar-Cohen et al. [5] assessed optimization of fin geometry, but the limitation of the studies was that radiation heat transfer was neglected by using low-emissivity material while radiation contributes to heat removal from steel containment. Y. Shabany [6] and Y.K. Khor et al. [7] investigated radiation heat transfer from a fin array, and they focused on the calculation of analytical view factor and the effect of radiation on the thermal performance of a fin array. However they did not assess fin geometry optimization for higher heat removal. Abdullah et al. [8] adopted rectangular plate fin on steel containment to compare the heat removal rate with a bare steel containment. They adopted specific fin geometry which is 40m of fin length, 50cm of fin height, 2mm of fin thickness and 50cm of fin spacing. Conway et al. [9] just suggested fin geometry with 0.79cm fin spacing, 4.45cm of fin height to increase heat transferred surface. However, the reason of selecting the fin geometry was not mentioned at all, and fin geometry optimization strategy for improving heat transfer rate or economics was not presented in the studies. Moreover, most of the above studies used small scale fin array with an air flow in the laminar regime. Therefore, they are not directly applicable to a steel containment because fin length for containment is around tens of meter and the air flow regime is turbulent. Therefore, this study suggests a fin geometry optimization methodology for large scale fin array where the flow regime is turbulent considering both natural convection and radiation.

Table I. Ranges of Parameters Investigated in Previous Studies

Name	Fin length (mm)	Fin height (mm)	Fin thickness (mm)	Fin spacing (mm)	Optimum spacing (mm)	Base-to-ambient temperature difference (K)
[10]	250	60	3	3-33	9-11	20-80
[11]	150	10, 17	3	3-45	9, 9.5	20-40
[12]	150-500	30-90	1-19	3-45	9, 9.5	20-40
[13]	250, 340	5-25	3	5.75-85.5	10.4-11.9	30-150
[14]	100-500	5-90	1-19	2.85-85.5	-	14-162
[15]	100	5-25	3	4.5-58.75	7	30-60
[16]	25-49	13.5	1	3-11	15-20	15-22
[17]	250, 340	5-25	3	5-85.5	11.75	14-185

3. METHOD

3.1. Description of a Unit Cell for CFD

PASS is a SMR which applies design characteristics of high temperature gas-cooled reactors (HTGRs) into water-cooled reactors to achieve inherent safety features [2]. Among the design characteristics, we focus on a steel containment concept. The upper part of the containment is a hemisphere shape and the lower part of the containment is a cylindrical shape. In PASS, its reference diameter is 15m and its containment height from a bottom to the top of the cylinder part is 20m [3].

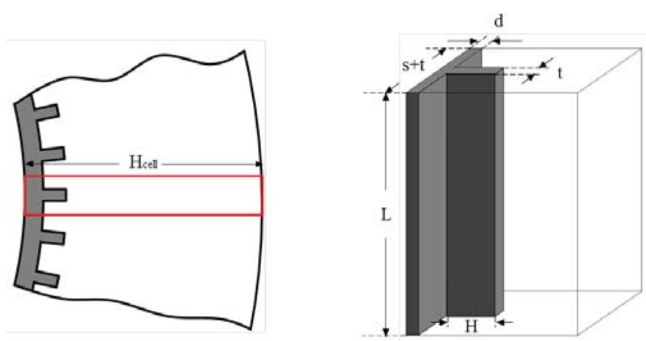

Figure 1. Unit Cell Geometry from Top View (Left) and Side View (Right).

In order to obtain heat transfer rates from various fin geometries, Design Modeler in ANSYS Workbench 15.0 was used for three-dimensional model construction and ANSYS Fluent 15.0 was used as a solver. CFD calculation 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. 1 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 containment while reducing computational load. We assumed that the unit cell is a hexahedron shape because the width (s+t) of the unit cell is extremely small compared with the perimeter of the containment. The fins were only adopted in the vertical region of the containment. Height of the unit cell (H_{cell}) was determined as 2.5m through sensitivity study to simulate opened atmosphere. Detail information of fin geometric parameters is shown in Table II. Fin length was fixed as the same as containment height and we varied other geometric parameters to optimize fin geometry. Material of the containment and fins were carbon steel with 0.8 of emissivity. Working fluid for external natural convection was air, which is assumed as an ideal gas. The air was assumed as a stagnant state as the initial condition. Ambient temperature (T_a) was given as 46.11℃ conservatively [18]. Containment inside wall temperature (T_w) was a constant boundary temperature of 80℃ as a reference temperature.

Table II. Fin Geometric Parameters in This Study

Unit cell height H_{cell} (m)	2.5
Fin length L (m)	20
Base thickness d (cm)	4.44
Fin spacing s (cm)	3-12
Fin thickness t (cm)	0.5-5
Fin height H (cm)	5-25

3.2. Overall Effectiveness and Containment Diameter Reduction

In order to check the effect of heat transfer rate enhancement by fins on reduction of containment volume and material cost, overall effectiveness is introduced as a fin performance parameter. It is defined as follows:

$$\varepsilon_o = \frac{\dot{Q}_{total}}{\dot{Q}_{no\ fin}} \tag{1}$$

The overall effectiveness is 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. Therefore, fin geometry that has the maximum overall effectiveness is defined as optimal fin geometry. Also, the overall effectiveness is used to calculate the containment volume and the material volume. The total heat transfer rate through the external surface of the PASS containment is given as Eq. (2) with neglecting containment base thickness which is relatively small. Meanwhile, the total heat transfer rate through the containment after adopting fins is given as Eq. (3).

$$\bar{h}_{t,o}\left[\frac{1}{2}4\pi(\frac{D_o}{2})^2\right]\Delta T_w + \bar{h}_{t,o}\left[\pi D_o H_{con,o}\right]\Delta T_w \tag{2}$$

$$\bar{h}_{t,o}\left[\frac{1}{2}4\pi(\frac{D}{2})^2\right]\Delta T_w + \varepsilon_o \bar{h}_{t,o}\left[\pi D H_{con,o}\right]\Delta T_w \tag{3}$$

Containment height remains 20m whether there are fins or not because reducing diameter is more effective in reducing containment volume. If we assume that the total heat transfer rate from the containment is the same regardless of using fins, Eq. (2) and Eq. (3) should be equal. Therefore, reduced containment diameter after adopting fins is expressed as follows:

$$D=\sqrt{400\varepsilon_o{}^2+825}-20\varepsilon_o \tag{4}$$

Based on the reduced diameter, we calculated containment volume and material volume as follows:

$$V_{con}=\frac{1}{2}\frac{4}{3}\pi\left(\frac{D}{2}\right)^3+\pi H_{con,o}\left(\frac{D}{2}\right)^2 \tag{5}$$

$$V_{steel}=\frac{\pi}{12}\left[\left(D+d\right)^3-D^3\right]+\frac{\pi}{4}H_{con,o}\left[\left(D+d\right)^2-D^2\right]+\frac{\pi\left(D+d\right)}{s+t}HtH_{con,o} \tag{6}$$

We assumed that material cost is proportional to material volume not considering additional cost for installing fins on the containment surface. The containment volume and the material volume is a function of a containment diameter. It means that high overall effectiveness is desired to reduce the containment volume and the material volume. Therefore, maximizing overall effectiveness is important for improving economics of PASS.

3.3. Effect of Fin Geometric Parameters

3.3.1. Effect of fin spacing change

Fin spacing is an important parameter that affects heat transfer rate from a fin array. If fin spacing is very small, the boundary layer starting from each side of neighbor fins interferes with each other and the air flow between two fins becomes fully developed vertical channel flow. This is called a small-s limit. On the other hand, if fin spacing is so large, the boundary layer develops independently without interferences. This is called a large-s limit. According to Yazicioğlu and Yüncü [13], convection heat transfer rates from a fin array increases firstly and reaches a maximum and then it decreases as the fin spacing increases. Therefore, they defined an optimum spacing as a fin spacing that maximizes the convection heat transfer rates. The optimum spacing is determined where two extreme limits become equal. Thus, the optimum spacing considering turbulent natural convection is obtained as follows:

$$s_{opt}\approx LRa_L{}^{-2/9} \tag{7}$$

Eq. (7) shows that the optimum spacing is only dependent on fin length and base-to-ambient temperature. Therefore, we predict that the optimum spacing of the PASS containment has a constant value regardless of fin geometry because the fin length and the base-to-ambient temperature of the PASS containment are fixed as 20m and 80℃, respectively.

3.3.2. Effects of fin thickness and fin height change

Many previous studies, which investigated fin geometry, set fixed fin thickness around 1~3mm. It was known that the 1~3mm thickness is the most conventional one for electronic device cooling where natural convection flow regime is laminar. However, fin thickness is needed to be optimized also. If fin thickness is too thin, thermal resistance from a base to a fin tip is increased too much. Then, the surface temperature is highly decreased along fin height and it makes overall effectiveness low. On the other hand, if fin thickness is too thick, an increasing effect of surface area by fins is lowered in a unit cell and it decreases overall effectiveness. Therefore, optimum thickness should be obtained.

Fin height also affects overall effectiveness. The longer fin height leads to the higher overall effectiveness because the longer fin height increases heat transfer area. However, an increasing rate of overall effectiveness with an increase in fin height keeps decreasing and finally the overall effectiveness is saturated. This is because surface temperature decrease becomes severer if fin height is too long and the increased surface area is not used in heat transfer effectively any more.

3.4. Heat Transfer Coefficient Correlation

Heat transferred from a vertical fin array consists of natural convection and radiation. Therefore, the total heat transfer coefficient is expressed as the sum of the natural convection heat transfer coefficient and the radiation heat transfer coefficient.

$$\overline{h}_t = \overline{h}_c + \overline{h}_r \tag{8}$$

3.4.1. Natural convection heat transfer coefficient correlation

Natural convection heat transfer from a fin array has been investigated in many studies. Leung et al. [10] and Tari et al. [17] 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:

$$\overline{Nu}_s = \frac{\overline{h}_c s}{k_a} = C(Gr'\mathrm{Pr})^n \tag{9}$$

They used $n = \frac{1}{3}$ for $250 < Gr'\mathrm{Pr} < 10^6$. Leung et al. obtained that C is 0.423 based on their experiment data. And the modified Grashof number includes an exponential term for fin effectiveness, $\exp\left(-k_a H / k_f t\right)$. However, in this study, the exponential term is modified as $\exp\left(-a k_a H / k_f t\right)$ by multiplying an empirical constant a to reflect our CFD results well.

3.4.2. Radiation heat transfer coefficient correlation

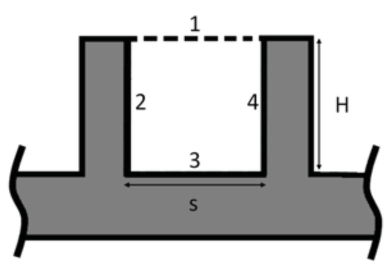

Figure 2. Allocated Surface Numbers (Top View).

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. 2. A normalized length $\overline{L} = s / H$ was defined for simplicity of an expression. 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 [19].

$$F_{21} = 1 - F_{23} - F_{24} = 1 - \left(\frac{1+\bar{L} - \sqrt{1+\bar{L}^2}}{2} \right) - \left(\sqrt{1+\bar{L}^2} - \bar{L} \right) = \frac{1+\bar{L} - \sqrt{1+\bar{L}^2}}{2} \tag{10}$$

$$F_{31} = 1 - 2F_{32} = 1 - \frac{2}{\bar{L}} F_{23} = 1 - \frac{2}{\bar{L}} \left(\frac{1+\bar{L} - \sqrt{1+\bar{L}^2}}{2} \right) \tag{11}$$

Based on the view factors, total radiation heat transferred from a unit cell is estimated as follows:

$$\dot{Q}_r = \bar{h}_{r,o} \Delta T_w (t + 2HF_{21} + sF_{31}) L = \bar{h}_{r,o} \Delta T_w (s+t) L \tag{12}$$

where $\bar{h}_{r,o}$ is radiation heat transfer coefficient of a bare surface.

It shows that the amount of radiation heat transferred from a unit cell of a finned surface is equal to the radiation from a bare surface with the same base area of the unit cell. It means 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:

$$\bar{h}_r = \frac{s+t}{s+t+2H} \bar{h}_{r,o} = \left(\frac{s+t}{s+t+2H} \right) \varepsilon \sigma \frac{T_w^{\,4} - T_a^{\,4}}{T_w - T_a} \tag{10}$$

4. RESULTS AND DISCUSSION

4.1. Optimum Spacing

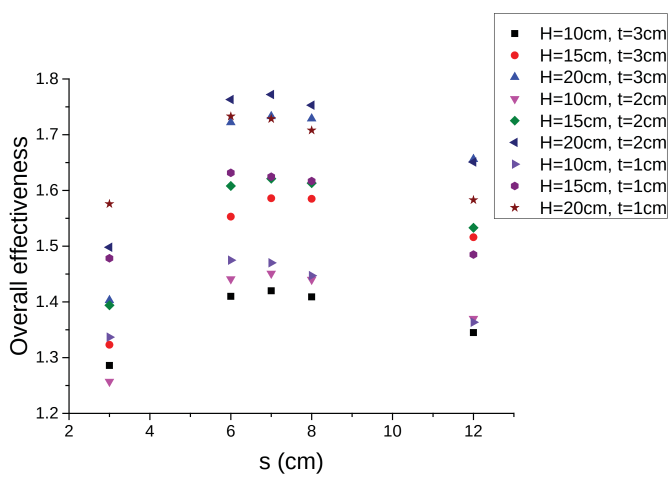

Figure 3. Overall Effectiveness as a Function of Fin Spacing.

We obtained overall effectiveness from various fin geometries as a function of fin spacing by using CFD, and the results are given in Fig. 3. We observed that overall effectiveness increases firstly until reaching a maximum and it keeps decreasing. Fig. 3 shows the fin spacing around 7cm maximizes overall effectiveness regardless of fin geometry. In order to obtain accurate optimum spacing value, polynomial curve fitting was used. By differentiating the polynomial curve, we found that optimum spacing is 7cm within 11% error and this result corresponds with the scale analysis result that optimum spacing for PASS containment has a unique value. Therefore, 7cm is suggested as an optimum spacing for PASS containment and an optimum spacing correlation is presented based on Eq. (7).

$$s_{opt} = 2.9 L Ra_L^{-2/9} \text{ for H} \geq 5\text{cm} \tag{11}$$

Moreover, Fig. 3 proved that the longer fin height gives the higher overall effectiveness.

4.2. Optimum Thickness with Fixed Fin Spacing

We investigated overall effectiveness along fin thickness increase while fixing fin spacing as the optimum spacing, 7cm by using CFD. Fig. 4 shows that overall effectiveness increases firstly until reaching a maximum and it keeps decreasing. It is observed that optimum thickness exists in given fin height respectively. Meanwhile, the optimum thickness is shifted as fin height changes. The reason is that if fin height is increased, thermal resistance along the fin height increases. As a result, thermal degradation becomes severer. Therefore, in order to avoid severe thermal degradation, optimum thickness becomes larger as fin height becomes longer.

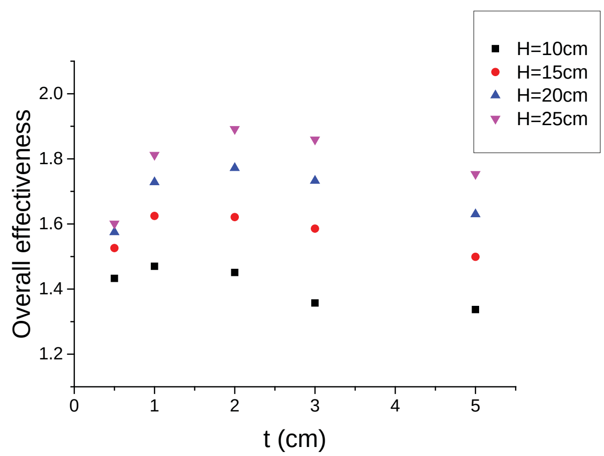

Figure 4. Overall Effectiveness as a Function of Fin Thickness.

Table III. Comparison of Optimum Thickness from CFD Results and Overall effectiveness correlation

Fin height H (cm)	Optimum thickness (cm)	
	CFD result	Overall effectiveness correlation
10	0.9	1.0
15	1.2	1.3
20	1.6	1.6
25	1.8	1.8

4.3. Overall effectiveness correlation

In order to estimate optimum thickness, the overall effectiveness correlation is used based on the definition of overall effectiveness given in Eq. (1). The overall effectiveness correlation is expressed using $\dot{Q}_{total}$ and $\dot{Q}_{no\ fin}$

$$\dot{Q}_{total} = \bar{h}_t A_f \Delta T_w \eta_o \tag{12}$$

$$\dot{Q}_{no\ fin} = \bar{h}_{t,o} A_b \Delta T_w \tag{13}$$

Here, η_o is overall efficiency which is defined as a ratio between total heat transfer rate to total heat transfer rate from the same surface without temperature degradation at all. The overall efficiency is expressed as follows where m is $(2\bar{h}_t / k_f t)^{1/2}$ and H_c is $H + (t/2)$ in rectangular plate fin [19].

$$\eta_o = 1 - \frac{t + 2H}{s + t + 2H}\left(1 - \frac{\tanh mH_c}{mH_c}\right) \tag{14}$$

In order to obtain the empirical constant a in the exponential term of $\bar{h}_c$, we obtained optimum thickness from the overall effectiveness correlation and compared it with CFD results. We found that optimum thickness is well matched as tabulated in Table III when a is 25.

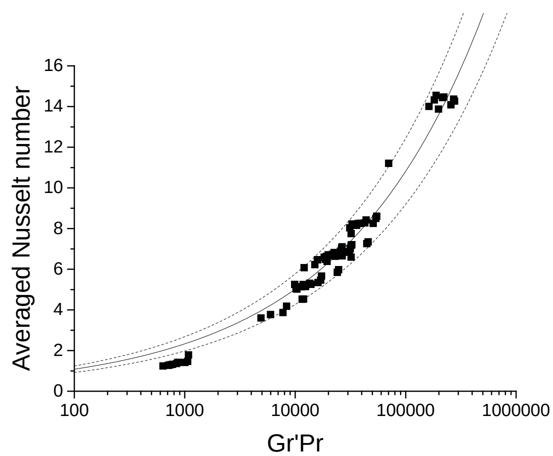

Figure 5. CFD Results of $\overline{Nu}_s$ and Fitting Curve in the Form of Eq. (9).

We plotted the averaged Nusselt number obtained from CFD results as a function of $Gr'\mathrm{Pr}$ and found that C is 0.2331 by fitting the graph as a functional form of Eq. (9). Fig. 5 shows the averaged Nusselt number correlation is well matched with the CFD results within 15% error. As a result, the following averaged Nusselt number correlation is suggested and the natural convection heat transfer coefficient is obtained based on Eq. (15):

$$\overline{Nu}_s = \frac{\bar{h}_c s}{k_a} = 0.2331(Gr'\mathrm{Pr})^{\frac{1}{3}} \text{ for } 400 < Gr'\mathrm{Pr} < 3\cdot 10^5,\ \mathrm{H} \geq \text{ 5cm} \tag{15}$$

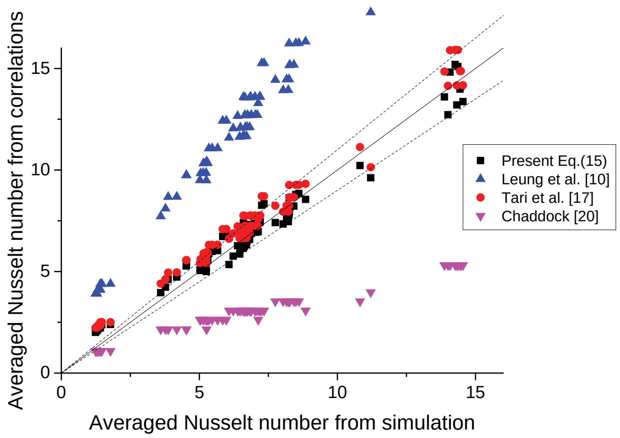

Figure 6. Comparison of $\overline{Nu}_s$ Parity Plots Using CFD Results for Eq. (15), Leung et al. [10] Correlation, Tari et al. [17] Correlation and Chaddock [20] Correlation.

Fig. 6 shows parity plots of the averaged Nusselt number using CFD results for the present correlation and other correlations [10, 17, 20]. The Leung et al. correlation highly overestimates the averaged Nusselt number while Chaddock correlation highly underestimates it. Our correlation and Tari et al. correlation are located near parity line within 10% of error range. The two correlations predict the averaged Nusselt number very well, but the Tari correlation is limited to predict optimum thickness because it does not consider an exponential term in the modified Grashof number. Finally, we can suggest total heat transfer coefficient which is the sum of natural convection heat transfer coefficient from Eq. (15) and radiation heat transfer coefficient from Eq. (10) for a given fin geometry.

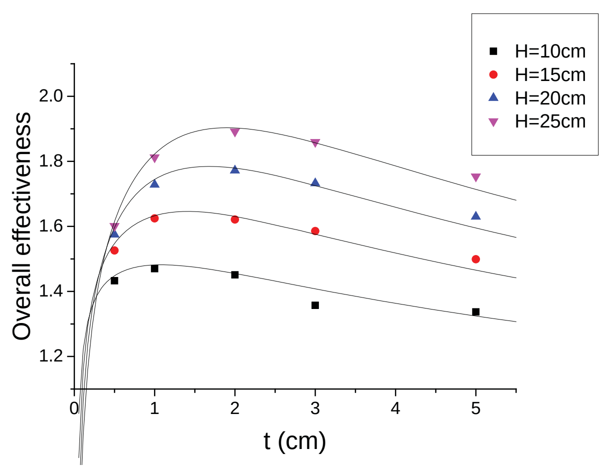

Figure 7. Comparison of the Overall Effectiveness Correlation with CFD Results.

Furthermore, the overall effectiveness correlation is established based on Eq. (1), Eq. (12), Eq. (13) and Eq. (14) as follows:

$$\varepsilon_o = \frac{\overline{h}_t}{\overline{h}_{t,o}}\left(\frac{s+t+2H}{s+t}\right)\left[1-\frac{t+2H}{s+t+2H}(1-\frac{\tanh mH_c}{mH_c})\right] \tag{16}$$

From the correlation, we can predict optimum thickness for a given fin height. Fig. 7 shows our overall effectiveness correlation predicts CFD results well. Overall effectiveness gives us a reduced diameter after adopting fins by Eq. (4). Then containment volume and material volume are also estimated by Eq. (5) and Eq. (6) for considering containment design and the economics of the containment.

4.4. Fin Geometry Optimization Results

Table IV. Optimal Fin Geometry and Its Overall Effectiveness for a Given Fin Height

Fin height (cm)	Fin spacing (cm)	Fin thickness (cm)	Overall effectiveness
10	7	0.9	1.49
15	7	1.2	1.69
20	7	1.6	1.85
25	7	1.8	1.93

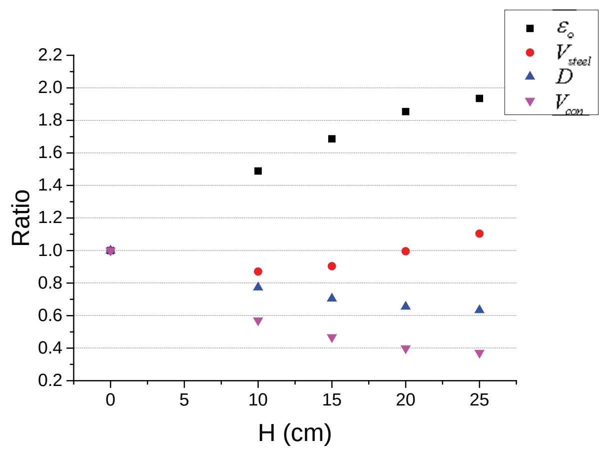

Figure 8. Ratios of Overall Effectiveness, Containment Diameter, Containment Volume and Material Volume of Optimized Fin-type Containment to those of Original PASS Containment without Fins as a Function of Fin Height.

Optimal fin geometry and its overall effectiveness are obtained for given several fin heights as tabulated in Table IV. The information in Table IV is used to check how much we reduce containment volume and material volume by using Eq. (5) and Eq. (6). The ratios of the changed values to the original ones of overall effectiveness, containment diameter, containment volume and material volume are plotted in Fig. 8. As we predict, the overall effectiveness keeps rising as fin height increases. However the increasing rate is gradually decreasing because temperature decrease takes place along fin height. Containment diameter and containment volume are monotonously decreased because they are decreasing function as the overall effectiveness increases. However the decreasing rate is kept decreasing as fin height increases. On the other hand, the material volume is decreased firstly and then it is increased. This is because the material volume of fin itself becomes considerable when fin height is long, although the containment volume is reduced. Therefore, if we use fin height shorter than 20cm, the containment volume and the material volume are reduced simultaneously. Meanwhile, fin height longer than 25cm needs more material than the original PASS containment. The least-material containment of PASS, which minimizes material volume, is obtained when we use fin geometry of fin height 10cm, fin spacing 7cm and fin thickness 0.9cm. In this case, we can reduce 13% of material volume and 43% of containment volume simultaneously.

Fig. 8 gives practical suggestions for designers how to choose the fin-type containment design option: the least-material containment design concept or compact containment design with small diameter for small containment size and volume. Designers can select proper fin geometry to meet their need by utilizing Fig. 8. Moreover, the optimization methodology of large scale fin geometry can be utilized not only for PASS containment but also for other large scale structure to give practical design options to meet their purposes.

5. CONCLUSIONS

We established an optimization methodology for large scale fin geometry by using CFD simulation. The steel containment of PASS was used as a reference of this study. First, the optimum spacing is obtained because it has a constant value in containment as a result of scaling analysis. Second, the optimum thickness given fin height is obtained with the optimum fin spacing. In this step, the total heat transfer coefficient correlation is suggested and the overall effectiveness correlation is also established. Then, optimal fin geometry is determined as a function of fin height. After that, we correlated the enhanced heat removal capability to containment diameter by using overall effectiveness. Through the relationship between overall effectiveness and containment diameter, we can suggest several practical options of fin geometry in accordance with designer’s needs such as least-material containment or compact containment and so on. In the case of the PASS steel containment as a reference, fin spacing of 7cm, fin thickness of 0.9cm and fin height of 10cm are the least-material fin geometry that can reduce 13% of material volume and 43% of containment volume simultaneously. While the fin structure with fin spacing of 7cm, fin thickness of 1.6cm and fin height of 20cm leads to a compact containment that can reduce 60% of containment volume with the same amount of material volume. The fin optimization methodology presented in this study can be applied to other large scale vertical structure to enhance heat transfer rate.

NOMENCLATURE

A	surface area (m^2)
D	containment diameter (m)
d	base thickness (cm)
F	view factor
Gr'	modified Grashof number, $Gr' = \left(\frac{g\beta\Delta T_w s^3}{\nu^2}\right)\left(\frac{s}{H}\right)\left(\frac{H}{L}\right)^{1/2} \exp\left(-a\frac{k_a H}{k_f t}\right)$
g	gravitational acceleration (m/s^2)
H	fin height (cm)
H_{con}	containment height (m)
$\bar{h}_t$	averaged total heat transfer coefficient (W/m^2K)
$\bar{h}_c$	averaged natural convection heat transfer coefficient (W/m^2K)
$\bar{h}_r$	averaged radiation heat transfer coefficient (W/m^2K)
k	thermal conductivity (W/mK)
L	fin length (m)

$\overline{L}$	normalized length, $\overline{L} = s / H$
$\overline{Nu_s}$	averaged Nusselt number based on s
Pr	Prandtl number
$\dot{Q}_{total}$	total heat transfer rate from a unit cell with fin (W)
$\dot{Q}_{\text{no fin}}$	total heat transfer rate from a unit cell without fin (W)
$\dot{Q}_r$	radiation heat transfer rate from a unit cell (W)
Ra_L	Rayleigh number based on L, $Ra_L = (g\beta\Delta T_w L^3)/(\nu\alpha)$
s	fin spacing (cm)
s_{opt}	optimum spacing (cm)
T	temperature (K)
t	fin thickness (cm)
ΔT_w	base-to-ambient temperature difference (K)

Subscripts

a	air
b	without fin, bare
f	fin
o	original, without fin
w	wall

Greek symbols

α	thermal diffusivity (m^2/s)
β	volumetric thermal expansion coefficient $(1/K)$
ε	emissivity
ε_o	overall effectiveness
η_o	overall efficiency
ν	kinematic viscosity (m^2/s)
ρ	air density (kg/m^3)
σ	Stefan-Boltzmann constant

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