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An Effect of Heat Exchanger Pipe Diameters on the Condensation Heat Transfer for PCCS Designs

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1. Introduction

The PCCS is a system considered for the containment cooling in case of accidents like LOCA or MSLB in order to control the pressure of the concrete containment by installing cooling units connected to the heat sink located outside of the containment. To estimate the cooling capacity of the PCCS, it is important to predict the condensate heat transfer rate at the surface of the pipes of the cooling units. The existence of the liquid film on the outer surface of the cooling pipe in the PCCS plays an important role in condensation of steam, as it is between the cooling wall and steam. The present work was performed to investigate the effect of the surface curvature to the thickness of the condensation and heat transfer coefficient through the theoretical and numerical studies.

2. Liquid Film Thickness on the Curved Surface

The liquid film thickness is determined by the force balance between the gravitational force and the frictional force as given

$$\mu \frac{1}{r}\frac{\partial}{\partial r} r \frac{\partial u_z}{\partial r} = -\rho g \ . \tag{1}$$

It is subjected to no slip condition at the surface of the pipe and the no gradient of the downward liquid velocity at the liquid film interface as

$$u_z = 0 \, at \ r = R \, and \ \frac{\partial u_z}{\partial r} = 0 \, at \, r = R + \delta \tag{2}$$

The downward liquid velocity profile is determined from Eq.(1) and boundary conditions, Eq.(2) and put $\eta = r/R \; and \; x = \delta / R$. Then, we have

$$u_z(r) = \left(\frac{\rho g R^2}{4\mu}\right)\left(1-\eta^2\right) + \left(\frac{\rho g R^2}{2\mu}\right)\left(1+x\right)^2 \ln\left(\eta\right) \tag{3}$$

The area averaged liquid velocity is given by truncating the higher order terms than third of r/R as

$$\tilde{u}_z = \frac{2\pi R^2 \int_1^{1+x} u(\eta)\eta d\eta}{\pi R^2 (2x+x^2)}$$
$$= \frac{2}{2x+x^2}\left(\frac{\rho g R^2}{4\mu}\right)\left[\int_1^{1+x}\left[(1-\eta^2)\eta + 2(1+x)^2 \eta \ln(\eta)\right] d\eta\right]$$
$$= \frac{+\rho g R^2 x^3}{2\mu(2x+x^2)} f(x) \tag{4}$$

Using the definition of the Reynolds number, we have

$$\mathrm{Re} = \frac{\rho \tilde{u}_z}{\mu}\frac{4A}{P} = \frac{\rho \tilde{u}_z}{\mu}\frac{4\pi(2R\delta+\delta^2)}{2\pi R}$$
$$= \left(\frac{\rho}{\mu}\right)^2 g R^3 x^3 f(x) = \frac{3}{2}\left(\frac{\rho}{\mu}\right)^2 g \delta^3_{Nusselt} \tag{5}$$

The film thickness is proportional to the square root of Re/R as

$$\delta = \frac{\delta_{Nusselt}}{\sqrt[1/3]{f(x)\left(\frac{2}{3}\right)\left(\frac{4}{3}\right)}} = \frac{\delta_{Nusselt}}{\left(\frac{8}{9}\right)^{1/3}\left(1+x\right)^{1/3}} \approx \frac{\delta_{Nusselt}}{1+\frac{1}{3}x} \tag{6}$$

It is a very interesting result that when we increase diameter of the pipe to make the infinite curvature like a flat plate, the film thickness converged into the film thickness of the Nusselt. However, for a pipe with a certain diameter, the film thickness is always smaller than the prediction of Nusselt. Furthermore, as the radius decreases, the film thickness decreases which guide us to enhance the condensation heat transfer as noted by Nusselt:

$$h_c = \frac{k}{\delta} Nu \tag{7}$$

As shown in Fig. 1, the normalized film thickness of the curved surface is much thinner than that of the flat plate and it increases as the radius of the pipe increases.

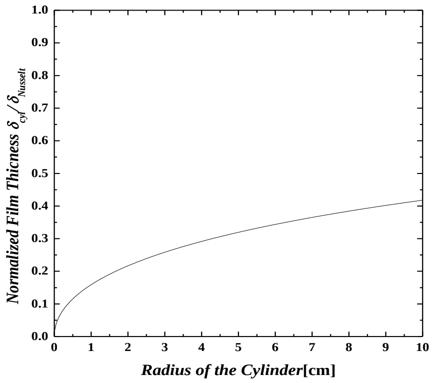

Fig. 1 The normalized film thickness

Also, the condensation heat transfer shows large variation for the pipe of the radius less than 1cm, but for the pipe of larger than 2cm radius the heat transfer coefficient varies small. Therefore, it can be recommended for the sized of the cooling pipe to be around 4cm in diameter.

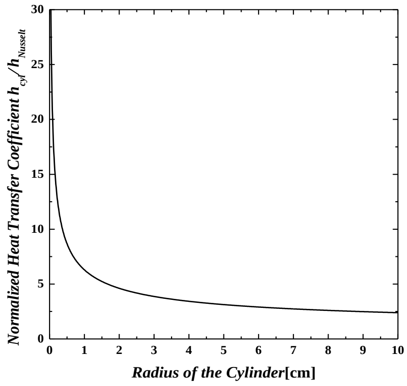

Fig. 2 The normalized heat transfer coefficient

3. Three dimensional Numerical Analysis

CFX, a 3 D computer code for the computational fluid dynamics, is employed to determine the film thickness on the surface of the pipe. As known, the condensation heat transfer simulation is not easy due to the imperfection in the mathematical modeling of the multiphase flow and the rapid mass change in the process of the condensation. Therefore, in the present study, we set a simple vertical cooling channel with radii of 2cm, 4cm, and 6cm in the center of the rectangular steam and air duct as shown in Figure 3. Simulation was performed in the transient mode with the dispersed liquid droplet mode in multiphase simulation mode. The steam and air mixture velocity set to be 0.1 m/sec.

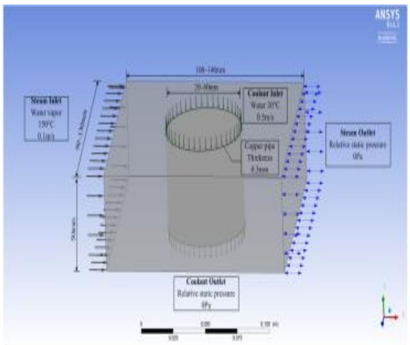

Fig.3 Test meshes for the CFD analysis

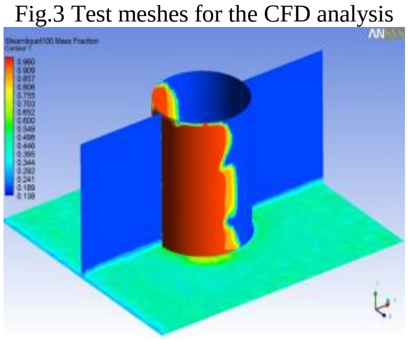

Fig.4 The cross sectional view of the liquid fraction distribution and the velocity vector

Simulation results are shown in Fig. 4 for the liquid fraction distribution near the cooling pipe and the velocity vectors. The liquid fraction was found that the front side against the steam flow was thicker than the rear side which represents that the massive condensation is made in the front surface in a short period of the simulation. Also, the velocity vector clearly supports it because the direction of the vector is heading to the surface of the heat in the front side but out of the surface in the rear side. The change of the film thickness with respect to the pipe diameter is depicted in Fig.5. Near the pipe surface, the liquid fraction increases suddenly at a certain distance from the surface which indicates the formation of the condensate film.

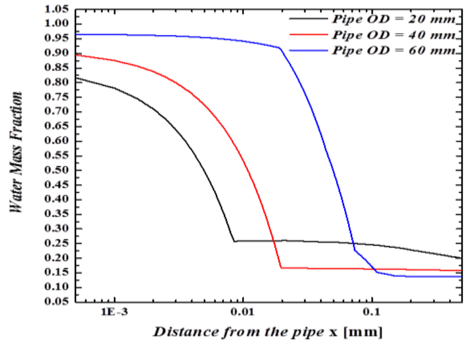

Fig. 5 The film thickness vs. pipe diameters

4. Conclusions

We investigated the effect of the pipe diameter on the condensation film thickness and heat transfer rate based on the analytical approach of the force balance and the 3-dimensional computational fluid dynamics. Both approaches made an agreement that the film thickness increases as the pipe diameter increases, which can lead to an insight that condensate heat transfer on the curved surface may be enhanced as the curvature of the cooling surface becomes larger. However, the practical application of the 3D-CFD code for condensate process needs further studies to firmly demonstrate this conclusion, as we found that the numerical results were sensitive to the models and computational time step management.

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