Transactions of the Korean Nuclear Society Autumn Meeting
Gyeongju, Korea, October 27-28, 2016

CFD Analysis of the Inclination Effects on Pool Heat Exchanger

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1. Introduction

Pool heat exchangers are important elements of advanced passive safety systems for most advanced nuclear power reactors. In the Advanced Power Reactor Plus (APR+), the pool heat exchanger is associated with Passive Condensation Cooling Tank (PCCT) and Passive Auxiliary Feed Water Systems (PAFS). This is used to remove decay heat from the reactor core by cooling down the secondary system of the steam generator using a condensation heat exchanger installed in the PCCT. Pool heat exchangers are also associated with Passive Residual Heat Removal (PRHR) system, Isolation Condenser Systems (ICS) and Passive Containment Cooling System (PCCS).

The mechanism of nucleate boiling has been widely investigated in the past due to its ability to enhance heat transfer in a limited space. However, recent suggestions have been made that heat transfer can be enhanced further by inclining the heat exchanger tube. This is because the resulting bubble sliding motion improves thermal mixing and mitigate thermal stratification. A brief literature survey on the previous work is given below.

Kang (2000) carried out an experimental study for the effects of tube inclination on pool boiling heat transfer of water at atmospheric pressure. Seven angles (0, 15, 30, 45, 60, 75 and 90^o) were considered using two tubes (12.7 mm and 19.1 mm diameter) of 540 mm in length. It was concluded that horizontal tubes (0^o inclination) had maximum heat transfer coefficient while vertical tubes (90^o inclination) had minimum heat transfer coefficient. The reason that was given for this is the decrease in bubble slug formation on the tube surface and easy liquid access to the surface of the tube.

Influence of wall orientation angle on boiling heat transfer has also been established in the work of Kang (2013) in which local heat transfer coefficient has been shown to decrease as azimuthal increases from the bottom to the top of the tube circumference.

Furthermore, in the experimental study carried out by Sateesh et al. (2009) to determine the effect of tube inclination on nucleate pool boiling heat transfer for saturated liquid, it was found that when tube was tilted from vertical to horizontal, the temperature at the top and bottom increases and decreases, respectively. The increase and decrease in liquid temperature at the top and bottom balanced out and resulted in little variation in average heat transfer coefficient with inclination angles.

Most recently, Minocha et al. (2016) carried out two phase 3D CFD simulations using mixture model (based on Euler-Euler approach) to investigate the effect of inclination of condenser tube on sliding bubble dynamics and associated heat transfer coefficient. Seven angles were considered (0, 15, 30, 45, 60, 75 and 90^o) and it was found that the heat transfer was maximum when inclination angle was 30^o and minimum when inclination angle was 75^o.

It has to be pointed out that most of the previous studies did not consider the in-tube condensation process because electrical heaters were used in most of the experimental studies. It was thought desirable to develop a simulation tool (validated within a certain region of approximation) for performing 3D CFD simulations of pool boiling which is driven by 1D calculation of in-tube condensation process. This work studies the effect of inclination angles on both boiling and condensation heat transfer simultaneously and also seeks to find an optimum inclination angle for both boiling and condensation phenomena.

In order to predict the major phenomena in a pool heat exchanger accurately, a sophisticated two-phase thermal hydraulic analysis is required. Therefore, coupling of a multi-dimensional (CUPID) code with a one-dimensional system analysis (MARS) code has been identified as an attainable way to predict both the boiling and condensation phenomena in the pool heat exchanger.

In this study, the coupled CUPID-MARS code was used for the simulation of a pool heat exchanger. This paper presents the description of the heat exchanger, boundary conditions and the simulation results using the coupled code.

2. Methods and Results

In this study, simulation of pool heat exchanger was carried out using CUPD-MARS coupled code. The heat exchanger consists of two separated systems. The primary system is the steam supply system in the inner tube while the secondary system is the pool tank. The interface between the two systems is defined by the wall of the tube which prevents flow interaction but allows heat transfer. In the first instance, MARS solves hydrodynamic equations and the conduction equations with given boundary conditions including the inner tube outer wall temperatures. Afterwards, the second outmost solid temperatures (T_{solid}) is transferred from MARS to CUPID as shown in Fig. 2. With this solid temperature, CUPID uses flow variables inside the outer tube (fluid temperature (T_{fluid}), liquid velocity etc.) calculated by itself to solve the energy balance equations to obtain the outer wall temperature (T_{wall}) as shown in Fig. 1.

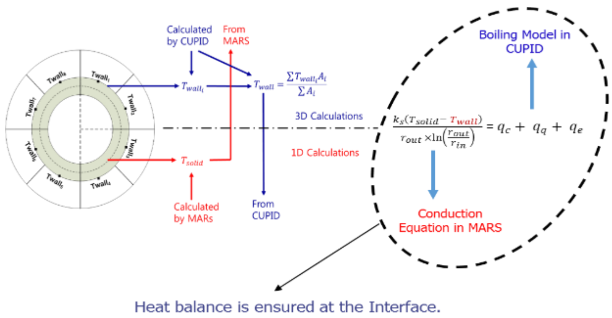


Fig. 1 CUPID-MARS Coupling Approach

The energy balance equation at the fluid-solid interface is given in Eq. (1) which is the combination of the conduction equation in MARS and RPI boiling heat partitioning model in CUPID.

$$\frac{k_s(T_{solid}-T_{wall})}{r_{out}\times\ln\left(\frac{r_{out}}{r_{in}}\right)} = q_c'' + q_q'' + q_e'' \qquad (1)$$

2.1 Geometry and Boundary Conditions

Boiling phenomenon was studied inside the pool which contains condenser tube at a position below the center. Likewise, condensation phenomenon was also studied but in one-dimension. The model of the pool tank and the condenser tube are shown in Fig. 2 and Fig. 3

Detailed dimensions of the pool and heating tubes are given in Table 1. The pool and the heating tubes were rotated together about the same point to give different inclination angles. The inclination angles (θ) varied in the range 0° to 90° (θ=0°, 15°, 30°, 45°, 60°, 75° and 90°). The sidewalls, the bottom and top of the tank pool were considered as adiabatic with no slip boundary condition. An outlet pressure boundary condition was defined for the topmost edge of the pool

Table 1. Geometry and Flow Conditions

Parameters	Values
Inner Tube	
Inner Diameter	16.05 mm
Outer Diameter	19.05 mm
Inlet Quality	0.938987
Inlet Velocity	3 m/s (0.001313 kg/s)
Pressure	0.4 MPa
Rectangular Pool	
Pool Size	0.5 m × 0.5 m × 0.15 m
Heat Transfer length (z)	0.1 m
Initial Temperature	372.76 K
Outlet Pressure	0.1 MPa
Inclination Angles	3°,15°,30°,45°,60°,75°, 90°

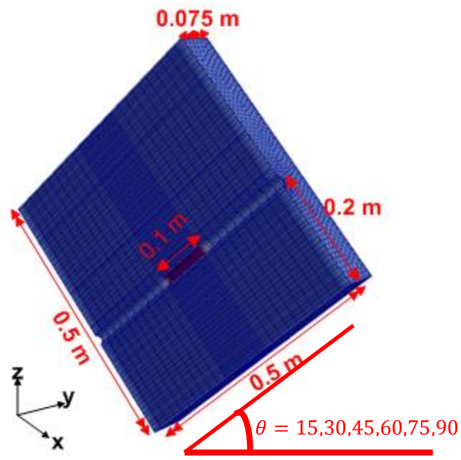


Fig. 2 Model of the pool for CUPID mesh generation

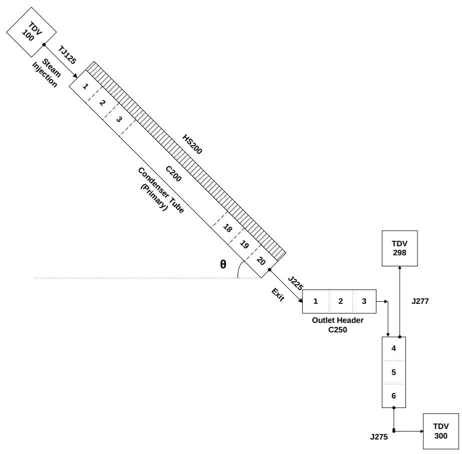


Fig. 3 Nodalization of the condenser tube for MARS

2.2 Physical Models and Correlations

In addition to the governing equations, physical models and correlations were selected to simulate the pool heat exchanger based on the identified major phenomena (free surface, subcooled boiling, single phase natural convection, two phase natural convection, and boil-off) in the pool. The default condensation models in MARS (Nusselt, Shah and Chato) were used to predict condensation phenomenon in the tube. It is pertinent to note that no inclination effect has been implemented in MARS.

The RPI model for heat partitioning was used to model subcooled boiling in CUPID as shown in Equations 2 - 4 and this was coupled with the conduction equation for heat structure in MARS.

The total wall heat flux is partitioned into three components: convective heat flux, quenching heat flux and evaporative heat flux.

$$q_w'' = q_c'' + q_q'' + q_e'' \qquad (2)$$

Each component of the heat flux is defined as follow in Eqs. (3)-(5).

Single phase convective heat flux:

$$q_c'' = h_c A_{1f}\left(T_{wall} - T_{fluid}\right) \qquad (3)$$

Quenching heat flux:

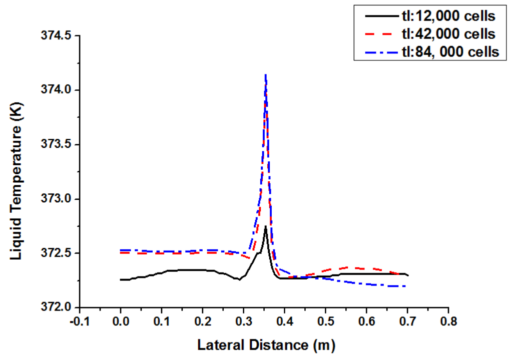
$$q_q'' = \left(\frac{2}{\sqrt{\pi}}\sqrt{t_w k_l \rho_l C_{pl} f}\right) A_{2f}\left(T_{wall} - T_{fluid}\right) \qquad (4)$$

Evaporative heat flux:

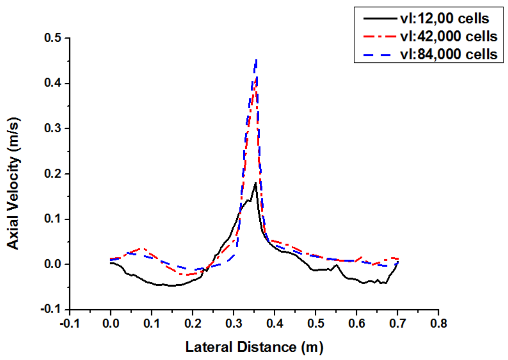
$$q_e'' = N'' f\left(\frac{\pi}{6} D_{b,depart}^3\right)\rho_g h_{fg} \qquad (5)$$

2.2 Grid Independence Test

In-house geometry and mesh generation code in CUPID was used for the geometry and mesh generation. The grid independence of the heat exchanger system at 45° inclination angle was investigated by considering three different grid cases: (a) 12,000; (b) 42,000; (c) 84,000. A non-uniform hexahedral grid was used which was finer near the tube wall where gradient are more important than those away from the wall. Axial temperature and axial velocity after t = 20s of simulation were compared for the three grid cases as shown in Fig. 3. All chosen grids predict the mean flow pattern effectively but for further simulations a grid size of 42000 was selected as only minor differences have been observed between 42000 and 84000.



(a)



(b)

Fig. 3 Grid test results: (a) axial temperature at t = 20 s, (b) axial velocity at t = 20 s

2.3 Azimuthal Variation of Fluid Temperature

The azimuthal (0°–360°) variation of fluid temperature at 1 mm away from the condenser tube in horizontal position are shown in Fig. 4.

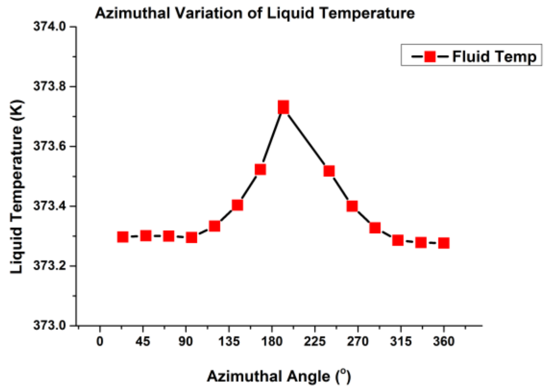


Fig. 4 Azimuthal variation of fluid temperature of condenser tube in horizontal position

The fluid temperature increases from bottom of the tube to the top of the tube. A nominal temperature variation was observed in the region ($0^0 \leq \alpha \leq 90^o$). However a rapid increase in fluid temperature was observed in the region ($120^0 \leq \alpha \leq 180^o$). Fluid moves from bottom to the top of the tube due to buoyancy force acting in the vertical direction. Bubbles lift off happened in the region ($160^0 \leq \alpha \leq 180^o$) which created a low pressure and thereby increase the fluid temperature at the top part of the condenser tube. Cluster of bubbles as a result of bubble lift off at the top of the condenser tube is responsible for plume formation at the top part of the condenser tube as shown in Fig. 5. The fluid accelerates in the region ($0^0 \leq \alpha \leq 90^o$) due to favorable pressure gradient ($\frac{\partial P}{\partial \alpha} > 0$)

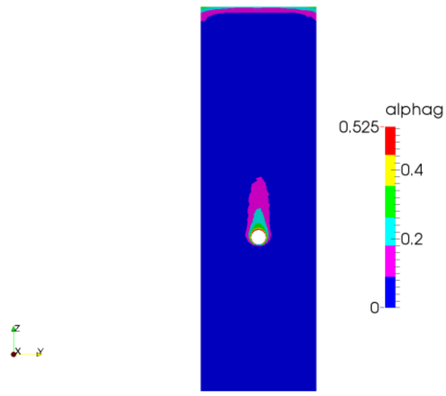


Fig. 5 Azimuthal variation of Vapor Fraction around the Condenser Tube at t =500 s

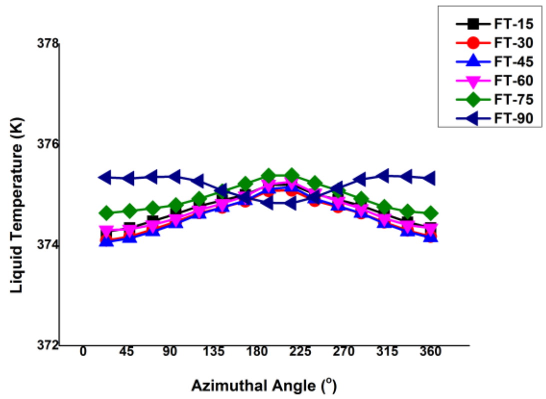
2.4 Inclination Effect on Boiling Heat Transfer Coefficient

The pressure gradient in the region ($120^0 \leq \alpha \leq 180^o$) generates the primary fluid flow along the length of the condenser tube and result in increased axial velocity.

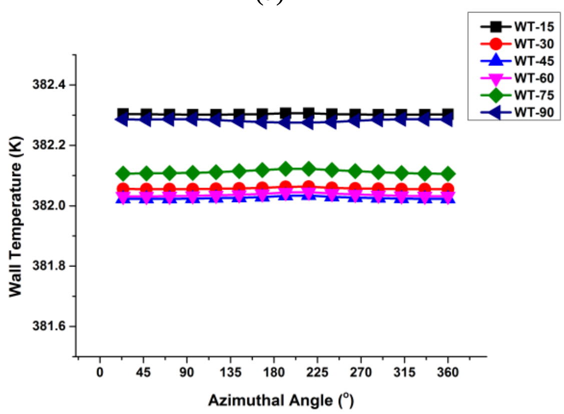
With increase in inclination angle (θ), the primary flow (along the tube length due to pressure gradient) becomes stronger and the secondary flow (in vertical direction due to buoyancy gets weaker. This result in the increase of fluid temperature as shown in Fig. 6(a).

The combined effect of both secondary flow and primary flow provide kinetic energy to overcome the adverse pressure gradient near the tube periphery. And minimum fluid temperature is found at 30° inclination

It is pertinent to note that the fluid temperature around the condenser tube is barely nominal when the inclination angle is 90° (vertical). This is shown in Fig. 6(a). The inclination of the system also has slight effect on the azimuthal wall temperature profile. Fig. 6(b) showed that approximately the same wall temperature distribution was predicted for both 30° and 45° which is minimum.



(a)



(b)

Fig. 6 (a) Azimuthal Variation of Fluid Temperature, (b) Azimuthal Variation of Wall Temperature at Inclination Angles of (15°, 30°, 45°, 60°, 75°, 90°)

The average total boiling heat transfer coefficient (h) was calculated from the components (convective, evaporative and quenching) of the heat partitioning model as shown in Eqs. (6) – (10).

$$Q_{tot} = q_c + q_e + q_q \tag{6}$$

Average liquid pool temperature at time t:

$$\bar{T}_{l,t} = \frac{\iiint_{r,\theta,z}^{R,\theta,Z} T_{l,t} dr d\theta dz}{\iiint_{r,\theta,z}^{R,\theta,Z} dr d\theta dz} \tag{7}$$

Average tube surface temperature at time t:

$$\bar{T}_{s,t} = \frac{\iint_{l,\theta}^{L,\theta} T_{s,t} dl d\theta}{\iint_{l,\theta}^{L,\theta} dl d\theta} \tag{8}$$

$$Q_{tot} = hA(\bar{T}_{s,t} - \bar{T}_{l,t}) \tag{9}$$

$$h = \frac{Q_{tot}}{A(\bar{T}_{s,t} - \bar{T}_{l,t})} \tag{10}$$

As the angle of inclination increases, the primary flow (along the tube length due to pressure gradient) becomes stronger while the secondary flow (along vertical direction due to buoyancy forces) decreases. These flows distribution account for the variation of boiling heat transfer coefficient with inclination angle and the maximum value is found at inclination angle of 30° as shown in Fig. 8. The difference between minimum and maximum is ~5%.

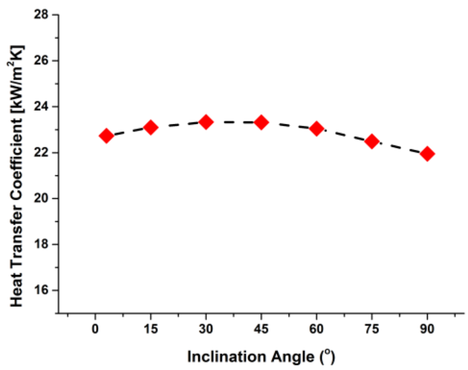


Fig. 8 Inclination effect on boiling heat transfer coefficients

2.5 Inclination Effect on Condensation Heat Transfer Coefficient

It is pertinent to note that no numerical implementation of inclination effects on condensation heat transfer coefficient has been done in MARS. This explains why there was no distinctive variation of condensation heat transfer coefficient with respect to inclination angles as shown Fig. 9. The condensation heat transfer coefficient that was predicted by the default model showed general increase of condensation heat transfer coefficient as inclination angle increases and the difference between minimum and maximum value is

~1%. This contrasted with the experimental findings of Olivier et al. (2016).

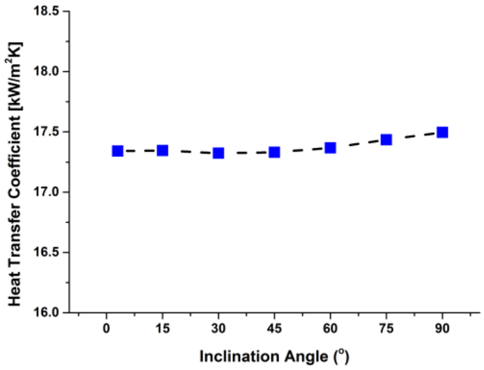


Fig. 9 Inclination Effect on Condensation Heat Transfer Coefficient

2.6 Inclination Effect on Heat Transfer Rate in Pool Heat Exchanger

By and large, the heat transfer rate in the pool heat exchanger showed no significant variation with the inclination angle as shown in Fig. 10. The difference between maximum at 90° inclination and minimum at 15° is just about 1%.

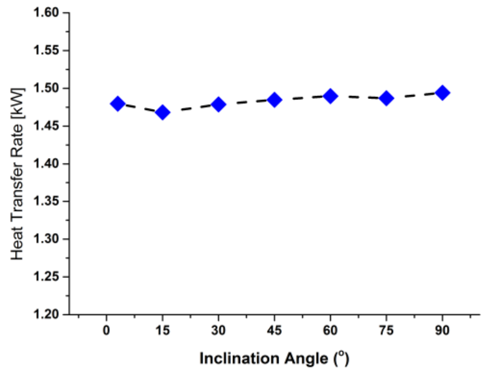


Fig. 10 Inclination Effect on Heat Transfer Rate in the Pool Heat Exchanger

3. Conclusions

The effects of inclination angles on the performance of pool heat exchanger have been studied for seven inclination angles using the default boiling and condensation models in CUPID and MARS. The main conclusions are:

Firstly, based on the predictions of default subcooled boiling model in CUPID and default condensation model in MARS, there was no significant change in the performance of the pool heat exchanger with respect to inclination angles. In fact, the difference between the minimum and maximum heat transfer rate over the range of inclination angles is ~ 1%

Secondly, boiling heat transfer coefficient was found to be maximum when the inclination angle is 30°. And the difference between minimum and maximum is ~5%. The combined effects of primary flow along the condenser tube length (pressure gradient driven) and secondary flow in the vertical direction (buoyancy force driven) with respect to inclination angles is responsible for the variation of heat transfer with respect to inclination angles.

Thirdly, the condensation heat transfer coefficient generally increases with the increase in inclination angle. However, the variation in the heat transfer coefficient was not very distinct as reported in the previous experimental studies. The reason for this is that the inclination effects on condensation heat transfer coefficient has not be numerically implemented in MARS code.

Thirdly, for more accurate prediction of boiling model, there is need to consider some other important bubble dynamics such as sliding bubble and bubble merger in the boiling model. There is also need to incorporate the effect of inclined surfaces, flow and pressure field on bubble parameters such as bubble departure diameter and bubble nucleation site density.

REFERENCES

[1] Anderson, N., Hassan, Y., Schultz, R., 2008. Analysis of the hot gas flow in the outlet plenum of the very high temperature reactor using coupled RELAP5-3D system code and a CFD code. Nucl. Eng. Des. 238, 274–279

[2] Aumiller, D.L., Tomlinson, E.T., Bauer, R.C., 2001. A coupled RELAP5-3D/CFD methodology with a proof of principle calculation. Nucl. Eng. Des. 205, 83–90

[3] Papukchiev, A., Lerchl, G., Waata, C., Frank, T., 2009. Extension of the simulation capabilities of the 1D system code ATHLET by coupling with the 3D CFD soft- ware package ANSYS CFX. In: Proceedings NURETH-13, Kanazawa, Japan, Sep. 27–Oct. 2

[4] Park, I.K., Lee, J.R., Lee, S.W., Yoon, H.Y., Jeong, J.J., 2013. An implicit code coupling of 1-D system code and 3-D in-house CFD code for multi-scaled simulations of nuclear reactor transients. Ann. Nucl. Energy 59, 80–91

[5] Cho H.K, Yun J. C, Seung-Jun L, Han Y. Y., 2013. Heat structure coupling of CUPID and MARS for the passive auxiliary feedwater system analysis, Transaction of the Korean Nuclear Society Spring Meeting, Gwangju, Korea, May 30-31,

[6] Jeong, J.J., Yoon, H.Y., Park, I.k, Cho H.K., Lee, H.D., 2010. Development and preliminary assessment strategy of a three-dimensional thermal-hydraulics code, CUPID. Nucl. Eng. Tech. 42(3), 279-296

[7] Kang M.G., 2013. Variation of local pool boiling heat transfer coefficient on 3-degree inclined tube surface. Nucl. Eng. Tech. 45(7), 911-920

[8] Jeon, S.S., Hong, S.J, Park, J.Y., Seul, K.W., Park, G.C., 2013. Assessment of horizontal in-tube condensation models

using MARS code, Part 1: Stratified flow condensation. Nucl. Eng. Des. 262, 510-514

[9] N. Minocha, J.B Joshi, A.K Nayak, P.K Vijay, 2016. 3D CFD Simulation of passive decay heat removal system under boiling condition: role of bubble sliding motion on inclined heat tubes, Chemical Engineering Science 145,245-265

[10] N. Minocha, J.B Joshi, A.K Nayak, P.K Vijay, 2016. 3D CFD Simulation to study the effect of inclination of condenser tube on natural convection and thermal stratification in a passive decay heat removal system, Nuclear Engineering and Design 305,582-603

[11] Kang M.G., 2000. Effect of tube inclination on pool boiling heat transfer, ASME J. Heat Transfer 127,88-192

[12] Olivier S.P., Meyer J.P., De Paepe M., De Kerpel K., 2016. The influence of inclination angle on void fraction and heat transfer coefficient during condensation inside a smooth tube, International Journal of Multiphase Flow, 80 1-14

[13] Sateesh, G., Das, S.K., Balakrishnan, A.R., 2009. Experimental studies of tube inclination on nucleate pool boiling. Heat Mass Transfer. 45,1493-1502