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Effect of flame spray coating on falling film evaporation for multi effect distillation system

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ABSTRACT

Horizontal tubefalling film evaporators find various applications like multi effect distillation for sea water desalination, power and process applications, refrigeration applications, etc. In this system, latent heat released inside the tube due to condensation is transferred to the falling film on the tube surface resulting in convective evaporation. Among many heat transfer enhancement techniques, thermal spray coatings enjoy diverse applications with economic advantage for commercial applications. This paper focuses on Computational Fluid Dynamic (CFD) analysis of falling film evaporation of water on horizontal tubes under the influence of gravity at vacuum. ANSYS Fluent with VOF two-phase model is used for the falling film studies on the horizontal tube. Half section of the tube is modeled with gravitational flow of sea water from the top. This model is used to predict the enhancement in convective evaporation on thermal spray coated tube surface with varying roughness. These CFD results are validated with published experimental data available in the literature. From this study, it is observed that heat transfer coefficient is increased by 10–15% due to increased roughness for laminar convective film boiling under vacuum. The film coefficient enhancement is higher for the larger diameter tubes. The tubes in the tube bank located beneath the top tube exhibit reduction in performance, which may be due to partial drying out of film on tube surface or due to evaporation and maldistribution of flow. At high heat flux and low feed rate, the decrease in film coefficient is more predominant. The effect of feed rate, tube diameter, wall temperature, heat flux, tube arrays, etc. on the heat transfer characteristics is studied and presented in this paper.

Keywords: Multi effect distillation; Falling film evaporation; Thermal spray coating; Surface roughness

1. Introduction

Sea water desalination using multi effect distillation (MED) principle has been increasingly predominant during the last two decades due to higher thermal efficiency and low initial and maintenance expenses. Large capacity MED plants are being installed and presently about 14% of the water production from renewable energy in the world is by MED technology as reported by Mathioulakis et al. [1]. The

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MED system makes use of thin falling film evaporation outside a horizontal tube by utilizing the latent heat from the condensate inside the tube. Falling film evaporation finds applications in refrigeration, chemical, ocean thermal energy conversion systems, process industries, etc. Due to simultaneous condensation and evaporation across the tube wall, high film coefficient is observed in MED systems. Absence of hydrostatic head, as in pool boiling, makes the falling film evaporation more effective and pressure drop for the liquid flow is negligible. However, the film coefficient on evaporation side (on the outside of the tube) is considerably lower than the condensation side.

There are many heat transfer enhancement technologies developed for the applications in aerospace, nuclear technology, refrigeration, power generation, chemical industries, etc. Passive enhancement techniques are more attractive due to low investment and energy consumption compared to active techniques which are more complicated and nonproductive. Structured surfaces (porous metallic coatings), rough surfaces (grooved surfaces), and extended surfaces (fins) are the common passive enhancement techniques. Metallic spray coating on heat transfer surface has been widely reported in recent years for heat transfer enhancement. Plasma spray, electric arc spray, and flame spray are the three widely used thermal spray coating methods. Porous layer in the coated surface acts as nucleation sites in addition to increasing the turbulence level in the falling film. Thus, the enhancement by coating is achieved by increasing the effective surface area, active nucleation sites, and changing the mode of heat transfer. Heat and mass transfer mechanism in a falling film is complex due to combined effects of convective film evaporation at the liquid-vapor interface and nucleate boiling in the liquid film. Nucleate boiling occurs at high heat flux or at large temperature difference. Falling film evaporation provides higher heat transfer coefficients than pool boiling even at low heat flux conditions.

Various analytical and experimental studies were carried out in the past on the estimation of heat transfer coefficients for the falling film evaporation on horizontal tubes. Plain tubes of Cu–Ni (90/10), stainless steel, etc. were studied for horizontal tubes of various diameters. Fluids like ammonia, R134a, etc. were studied in addition to water or sea water for the application in ocean thermal energy conversion (OTEC) systems. Single tube as well as bundle tube tests were reported. Also, experimental investigations were carried out for boiling and film evaporation and combination of these modes.

An overall review of falling film evaporation on horizontal tubes with a focus on refrigeration and air conditioning is published by Ribatski and Jacobi [5]. Previous works on the influence of heat flux, flow rate, film temperature, tube diameter, feeder height, flow patterns, bundle effect, etc. are compared. It is reported that the structured surfaces can give heat transfer coefficients approximately 10 times higher than those in plane surfaces. Parken et al. [2] conducted experimental studies on electrically heated brass tubes of 25.4 and 50.8 mm diameters for measuring film coefficients under boiling and evaporation for transient flow over the plain tube. Heat transfer measurements were done with thermocouples installed at different locations on the tube circumference. It was also observed large fluctuation in film coefficient under local boiling conditions. Alam et al. [3] carried out studies on heat transfer enhancement by copper coating of various thicknesses on a 32 mm outside diameter mild steel tube. Plasma spray technique was used and the maximum coating thickness was 60 μm with porosity 13.7%. Experimental studies with water at atmospheric and subatmospheric conditions showed that nucleate pool film coefficient increased to certain value of coating thickness and thereafter, it was adversely affected. Lakhera et al. [4] carried out a study on the effect of porous thermal spray coating of stainless steel 316 on plain AISI 304 tube having 19.05 mm outer diameter and 130 mm length under flow boiling conditions for distillate water at atmospheric pressure. Surface roughness was varied from 0.33 to 4.73 μm with porosity <2% and varied from 12 to 45 kW/m^2 with heat flux. A proportionate increase in heat transfer coefficient with coating thickness was observed and maximum enhancement was about 153% corresponding to the maximum heat flux. Experimental studies with porous coating of different materials carried out by Ceislinski [7] showed that coating thickness and porosity were key parameters that influence the enhancement of heat transfer. Among the materials tested such as aluminum, copper, brass, molybdenum, stainless steel, etc. aluminum coated surfaces proved the superiority. Under pool boiling conditions, boiling had commenced at even 0.1 K temperature difference.

Evaporation outside the tube is influenced by film Reynolds number (Re), Prandtl number, dimensionless tube spacing, and tube diameter [8]. Heat transfer correlations were developed for constant wall temperature and constant heat flux conditions. Also for single tube and tube banks, it is shown that the film coefficient is reduced from first tube onwards. Some of the known correlations in the literature used for the present studies for comparison are listed below in Eqs. (1) [9] and (2) [10]:

$$\mathrm{Nu_D} = 0.2071\mathrm{Re}^{0.24}\mathrm{Ar}^{-0.111}\mathrm{Pr}^{0.53} \tag{1}$$

$$\mathrm{Nu_D} = 0.0137\mathrm{Re}^{0.349}\mathrm{Pr}\left[\frac{\left(\frac{s}{d_o}\right)^{0.158}}{\left(1+e^{\left(-0.0032\mathrm{Re}^{1.32}\right)}\right)}\right] \quad (2)$$

Heat is transferred from the tube wall across the thin liquid film and heat transfer to the evaporating liquid is governed by the following equation.

$$\frac{k\Delta T_{ef}}{\delta} = -h\Delta T_e \quad (3)$$

Temperature difference between the tube wall and saturation temperature is taken as the local evaporation temperature difference (ΔT_e) and ΔT_{ef} is the temperature difference across the film.

In the present study, falling film evaporation of seawater on horizontal tubes under vacuum conditions for MED system is carried out for the operating conditions as shown in Fig. 1. Heat transfer takes place from tube wall to the flowing thin film outside. Evaporation of the water takes place at the liquid vapor interface. The porous metallic coating on the tube surface is shown in schematic diagram in Fig. 2. A sample of coated aluminum tubes for the experimental studies is shown in Fig. 3. The flame spray coating enhances the surface roughness and porosity which accelerates the film evaporation.

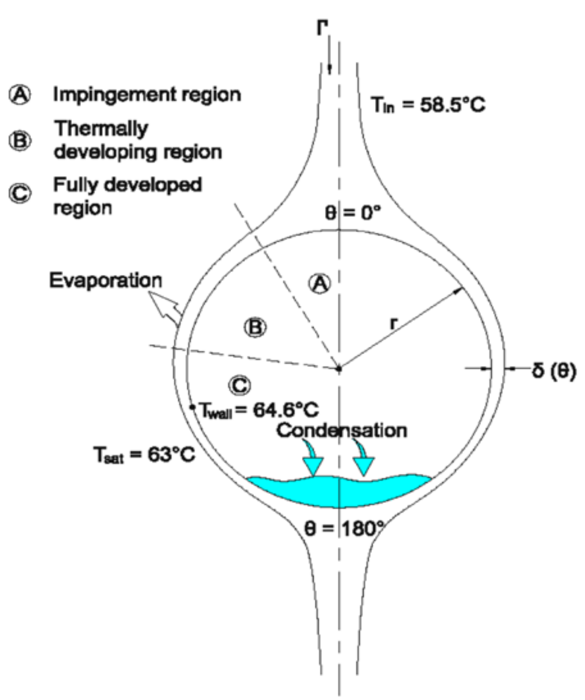

Fig. 1. Falling film over the horizontal tube.

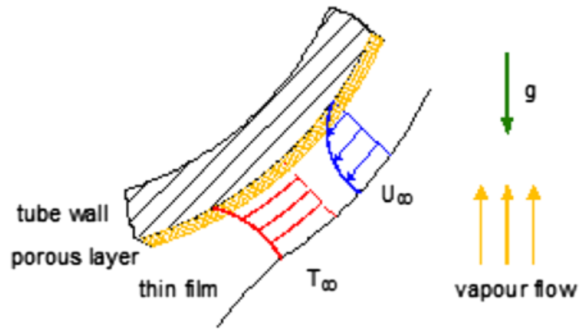

Fig. 2. Spray coating over the tube wall.

Fig. 3. Al coated Al tube of 25.4 mm dia.

The present work is important as:

- Earlier experimental studies were using an intrusive measuring device which considerably affects the results for falling film flow over the tube.
- Influence of porous metallic coating on surface with falling film flow is considerably different from nucleate or pool boiling in submerged conditions.
- There is an optimum thickness of coating which gives the better performance.

2. Computational fluid dynamic analysis

ANSYS Fluent® was used to carry outflow modeling of a horizontal tube falling film [14]. Half section of the film flow outside the tube is modeled with two-phase VOF model. The salient features of the modeling are listed in Table 1. Quad-Pave scheme with 10,138 meshes were used for studies. A grid dependence analysis was carried out from grid sizes from 2,000 with an increment of about 1,000. Beyond 10,138 meshes, the accuracy was not improved considerably further, however computing time was increased. The film coefficient increase was within 2%. Fig. 4 shows a typical contour of volume of fraction and temperature distribution.

The effect of heat flux, feed flow rate, etc. is studied. Surface tension and viscosity are two important parameters that influence the flowing film characteris-

Table 1
Parameters for fluent analysis

Simulation	2D
Solver	Pressure based, laminar, unsteady
Two-phase model	VOF
Phases	
Primary	Air
Secondary	Water vapor Liquid-sea water
Gravitation acceleration	−9.81 in Y
Tube wall	Constant temperature/heat flux
Volume of fraction	1 for water 0 for vapor
Pressure velocity coupling	PISO
Discretization pressure	PRESTO
Discretization momentum	1st order Upwind
Mass transfer	udf based
Viscosity, density	udf based
Tube wall temperature	64.6°C
Saturation temperature	62°C
Operating pressure	21,742 Pa
Feed water temperature	58.5°C

tics. Temperature dependence on the variation of film thickness, temperature gradient, heat transfer coefficient, mass generation, etc. is studied. This analysis is repeated for plain tube and coated tube of 19.04, 25.4, and 50.8 mm diameters. Falling film under both laminar and turbulent flows is compared. Heat transfer under both constant heat flux and constant temperature conditions is compared. Computational Fluid Dynamic (CFD) study is helpful to arrive at the optimum conditions for the operation of the MED plant. The effect of tube bundle is also evaluated and compared with a single tube to understand the interference effects. Details on CFD studies on bundles are reported by Raju Abraham and Prakash Kumar [13].

3. Results and discussion

In falling film evaporation, thickness of the film is an important factor for heat transfer. While the uniformity of the film to be assured for effective heat exchange, the local heat transfer coefficient for a developed laminar flow is inversely proportional to the film thickness. The simplified expression of film thickness for laminar horizontal tube falling film without inertial force term as developed by Sarma et al. [11] is

$$\delta = \frac{d_o}{2} \cdot \left[\frac{0.75 \cdot \mathrm{Re}}{\mathrm{Ar} \cdot \sin\theta}\right]^{1/3} \tag{4}$$

Highest film thickness is predicted at the top and bottom of the tube. Thickness of the film is decreased for increased diameter for the same feed rate. It is observed that film thickness is constant until the first half of the tube and after that surface wave appears for the Re greater than 700.

Several intrusive and nonintrusive attempts were made in the past to measure the film thickness. Intrusive methods like micrometer perturb the flow and thickness measurements. Nonintrusive optical methods are more popular recently. Thickness measurement on plain as well as Turbo CII tube using laser induced fluorescence is reported by Wang et al. [6]. The minimum value of thickness tends to locate at angular position between 95° and 120° under different conditions. In Fig. 5, the CFD results are compared with theoretical predictions and experimental results.

Average heat transfer coefficient of the falling film is influenced by feed water flow rate, fluid properties, tube wall temperature, heat flux, tube diameter, water distribution method, etc. Heat transfer is highly influenced by convective and boiling conditions. The boiling evaporation dominates under high heat flux or wall super heat conditions. Under convective evaporation, an increase in heat transfer coefficient with feed rate is reported by many authors. It is also reported that heat transfer coefficient first reduces and then increases due the transition from laminar to turbulent at about film Re of 1,500. Nondependence on feed rate for heat transfer coefficient is also observed. Similar inconsistency exists for nucleate boiling condition in falling film evaporation. Change in operating pressure, flow pattern such as droplet mode, jet mode, or sheet mode and change in laminar to turbulent and wave conditions on film surface, local dry out, etc. influence the results to a great extent. Fig. 6 shows the comparison between the fluent and experimental studies.

Surface roughness plays a major role in convective evaporation and nucleate boiling. Surface roughness and porosity play the vital role in heat transfer enhancement. Heat transfer enhancement is much more than the increase in area due to coated surface. Thickness of coating, porosity, and roughness height are the key parameters that influence the heat transfer as reported by Ceislinski [7]. The enhancement in film coefficient can be observed as shown in Fig. 7 and it is more significant for higher diameter tubes. However, the heat transfer coefficient decreases beyond certain limits.

An increase in film coefficient is reported for the decrease in tube diameter under convective evapora-

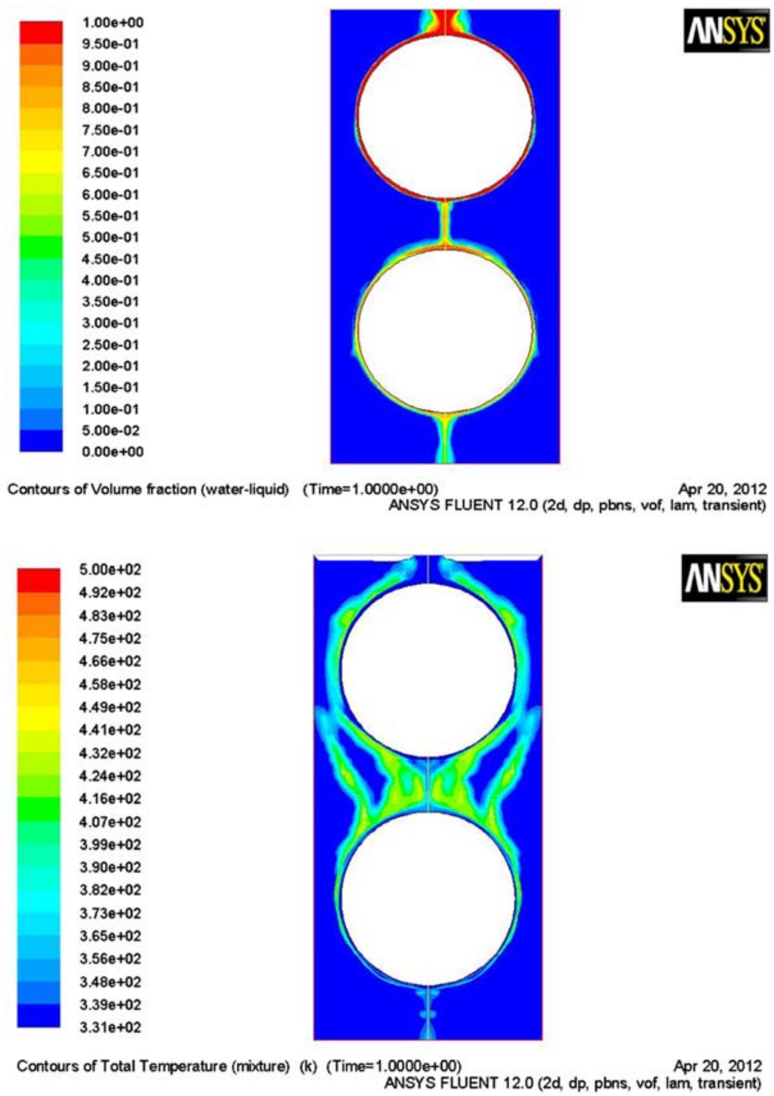

Fig. 4. Contours of falling liquid film and temperature.

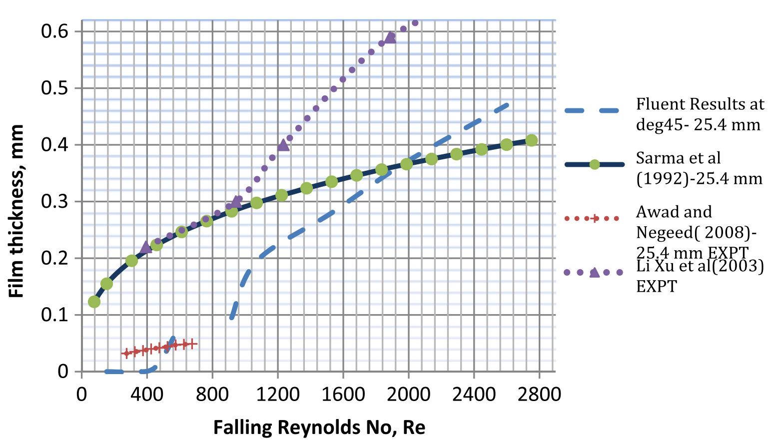

Fig. 5. Influence of feed flow on film thickness.

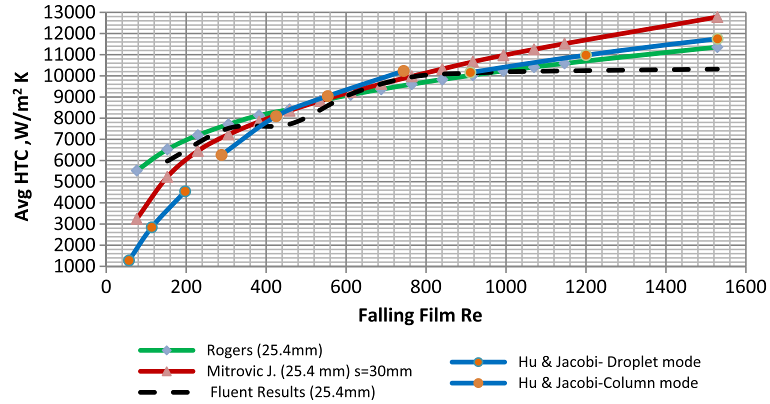

Fig. 6. Effect of feed rate on average heat transfer coefficient.

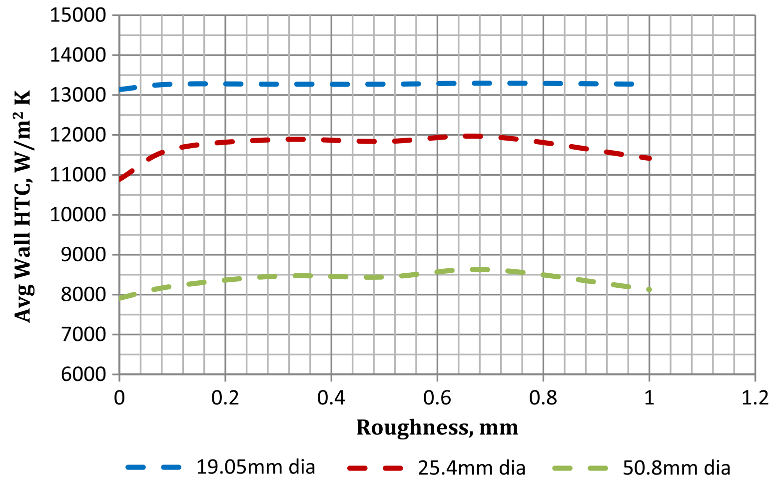

Fig. 7. Influence of surface roughness on convective film coefficient.

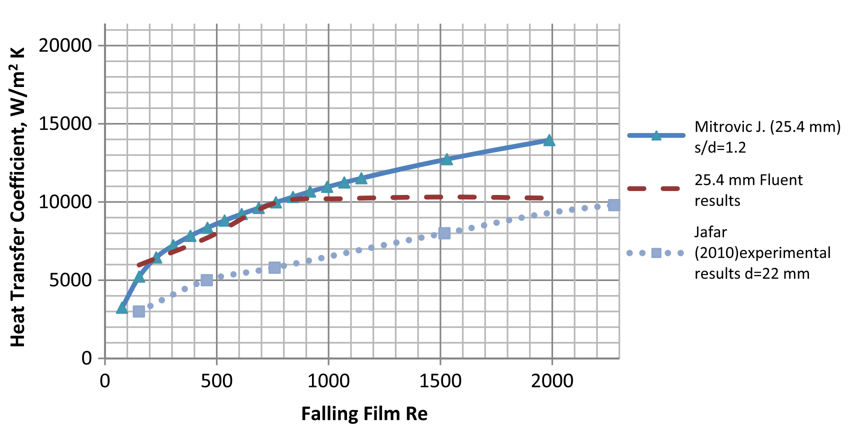

Fig. 8. Effect of Re on heat transfer coefficient.

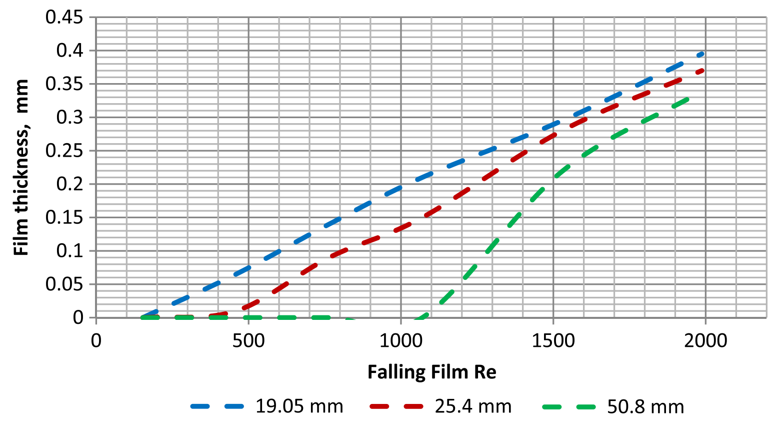

Fig. 9. Effect of Re on film thickness.

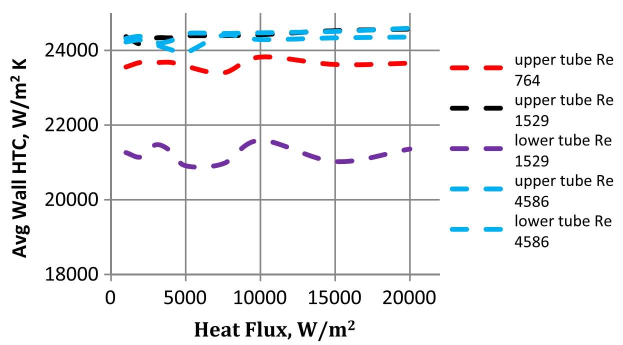

Fig. 10. Effect of heat flux on average heat transfer coefficient for $d_o = 25.04$ mm.

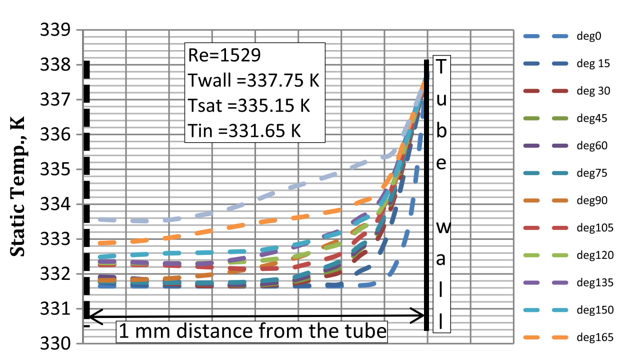

Fig. 11. Contour of temperature around the single tube.

tion. This can be due to the high impingement and flow development for large fraction of the tube surface as reported by Ribatski and Jacobi [5]. However, under nucleate boiling conditions differing trends were reported. Local film coefficient varies due to the bubble formation as well during the bubble growth. Variation of film coefficient with Re is plotted as shown in Fig. 8. CFD results are compared with the predictions of Mitrovic [10]. It is noted that the results are matching only with tube diameter of 25.4 mm. At higher tube size the film coefficients fall well below the predictions probably due to the film dry out. At smaller tube size the CFD results are not matching for the Re 1,000 and above probably due to the film thickening.

Under convective evaporation, in completely wetted surface, heat flux does not affect the heat transfer coefficient. Under boiling conditions, it is observed that porous coating provides high heat transfer enhancements even at low heat flux. Heat transfer enhancements become weak at high heat flux due to the partial dry out at cavities in the porous region. Data for understanding the influence of heat flux on film coefficient at reduced pressure are scanty. Fig. 10 shows the influence of heat flux at three different flow conditions for the upper and lower tubes of tube bundle. At low Re, the film coefficient degrades in the lower tubes due to the maldistribution of flow and local dry out conditions.

As the temperature increases the viscosity is reduced which results in reduction in film thickness and increase in film coefficient for convective evaporation. This also enhances active nucleation due to the reduction in surface tension in case of nucleate boiling. Temperature gradient is reduced as film flows over the tube and hence there is reduction in film coefficient. Local detachment or dry out at 180˚ causes an abnormal gradient at that location.

4. Conclusions

CFD studies are carried out on thin film evaporation of seawater under vacuum conditions. The strong dependence of film coefficient on film thickness and feed rate is observed. Heat transfer increases with film Re, but stabilizes as film thickness increases. Increase in tube diameter adversely affects heat transfer. Upper half of the tube exhibits higher local film coefficient than the lower half. Temperature at different angular positions increases as the film is falling down. Wall roughness increases for the thermal spray coated tube surface and it is observed that the film coefficient increases and then decreases beyond certain limits of wall roughness. Heat transfer enhancement rate is higher for higher diameter tubes.

Nomenclature

Ar	—	Archimedes number, $gd_o^3v^{-3}$
C_p	—	specific heat, $J\,kg^{-1}\,K^{-1}$
d_o	—	tube outside diameter, m
g	—	gravitational acceleration, $m\,s^{-2}$
h	—	heat transfer coefficient, $Wm^{-2}K^{-1}$
k	—	thermal conductivity, $W\,m^{-1}\,K^{-1}$
Nu_D	—	Nusselt number, $hv^{2/3}k^{-1}g^{-1/3}$
Pr	—	Prandtl number, $\mu\,Cp\,k^{-1}$
Re	—	film Reynolds number, $4\Gamma\,\mu^{-1}$
s	—	tube pitch, m
ΔT_e	—	Local temperature difference for evaporation, K

Greek symbols

Γ	—	half liquid mass flow rate per length of tube, $kg\,m^{-1}\,s^{-1}$
δ	—	film thickness, m
θ	—	angle along the tube perimeter measured from the apex, °
μ	—	dynamic viscosity, $kg\,m^{-1}\,s^{-1}$
v	—	kinematic viscosity, $m^2\,s^{-1}$

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