

## Subcooled Flow Boiling of Water enhancement by Using Internal Surface Coating

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### Abstract

The main objective of the present work, is to study the effect of coating on boiling heat transfer of compressed water entering a vertical tube and leaving at near saturated vapor conditions, while the tube is heated with stream of cross flow hot gases. This is carried out by employing different types of internal tube coatings with different surface roughnesses and thicknesses. Numerical investigation is carried out by using software called EES (Engineering Equation Solver). It is concluded that, the heat transfer enhancement over the smooth tube case reaches 34.063% for the heat transfer coefficient while the pressure drop is higher than that in the case of smooth tube by 64.64%.

**Keywords:** *Flow Boiling, Pressure Drop, Internal Surface Coating.*

### 1. Introduction

Boiling is the process of rapidly converting a liquid to its gaseous (vapor) state, typically by heating the liquid to a temperature called its boiling point. The boiling of a substance is known as a phase change or phase transition. Chemically, the substance remains the same, but its physical state (or "phase") changes. The energy for phase change is generally supplied by the surface on which boiling occurs. Boiling differs from evaporation at predetermined vapor/gas-liquid interfaces because it also involves creation of these interfaces at discrete sites on the heated surface. Boiling is an extremely efficient process for heat removal and is utilized in various energy-conversion and heat-exchange systems and in the cooling of high-energy density components, i.e. Boiler, Steam generators, water tube boiler, heat exchangers, heat recovery steam generators and nuclear power plant steam generators.

Heat transfer phenomena associated with liquid vapour phase change play an important role in boiling process as in water tube boiler, heat exchangers, heat recovery steam generators and nuclear power plant steam generators. The objective of this work is to enhance the boiling heat transfer of compressed water enters a vertical tube and leaves near saturation vapor conditions while the tube is heated with stream of cross tube hot gases. This is carried out by employing different types of internal tube coatings with different surface roughnesses and different internal coating thicknesses.

The model also aims at obtaining the non-uniform heat flux and wall temperature along the tube height. The boiling heat transfer enhancement is carried out by using different materials with different surface roughness as an internal coating inside

the inner surface of the vertical tube. Moreover, this work is used to define the most adequate heat transfer models form used in the different flow regions and also define the surface roughness and coating thickness which give more enhancements for the heat transfer mechanism.

The scope of the present work can be summarized:

- The effect of using different surface roughness conditions ( $\epsilon/D$ ) to enhance the flow boiling will be investigated.
- The effect of using different internal coating thicknesses ( $\delta$ ) to enhance the flow boiling will be analyzed.
- The required heated vertical tube length for predetermined inlet, outlet flow and boundary conditions.

### 2. Mathematical Model

#### 2.1. Physical Model

The model describes the situation where water enters a vertical tube as compressed liquid with certain temperature and pressure and leaves at low vapor quality values whereas an external hot gases stream perpendicular to the tube supplies the heat flux required to the evaporation process. Temperature and velocity of hot gases is considered uniform which are thus leading to non uniform wall temperature and heat flux along the tube length. Figure 1 illustrates the test case which is a tube with inlet diameter 2.6 (cm) having inlet water with mass flux (G) equal to 47.2 ( $\text{kg/s.m}^2$ ) and the hot flow stream outside the tube and perpendicular to the tube has a uniform temperature

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DOI:10.5383/ijtee.02.02.004

$T_g$  is equal to 140 (°C) and uniform equivalent heat transfer coefficient  $h_{ge}$  equal to 600 (W/m<sup>2</sup>.K) which will be explained later in the heat transfer equation.

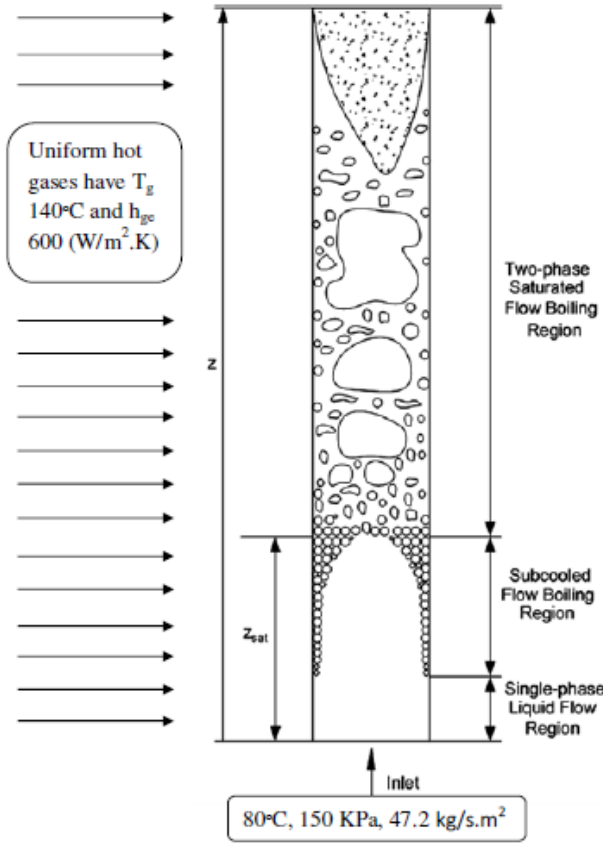


Fig. 1. Single-phase liquid flow, subcooled flow boiling and two-phase saturated flow boiling [1]

The boiling enhancement internal surface paint used for the present surface treatment was a mixture of 1.5 (gm) of silver flakes (3-10) (µm), 0.1 (ml) of omega bond 101 thermal epoxy and 10 (ml) of isopropyl alcohol. This paint composition was determined by conducting a study that systematically varied the silver flakes content in the mixture and was found to provide the highest level of enhancement throughout the boiling curve by O'Connor [2]. The paint was applied to the reference surface using a dripping technique and then the paint was cured under a lamp at 80 (°C) for 100 (minutes), during cure, the alcohol evaporated and the mixture of flakes and epoxy adheres to the surface.

2.1. Simplifying Assumptions

- Conservation of mass.
- Conservation of momentum.
- Conservation of energy.
- Heat transfer correlation.

2.2. General Governing Equations

Applying the well-known equations of conservation mass, momentum and energy as following:

$$G = m \square_{\text{flow}} - \rho \cdot u = \text{const.} \tag{1}$$

Where G is the water mass flux or mass velocity, m' is the water mass flow rate, A is the cross sectional area of the

vertical tube,  $\rho$  is the water density and u is the water vertical velocity in the z direction.

$$-\frac{dp}{dz} = \rho \cdot g + 2 \cdot f \cdot \frac{G^2}{\rho \cdot D_h} + G \cdot \frac{du}{dz} \tag{2}$$

Where (dp/dz) is the pressure drop of the water flow along the vertical direction, g is the acceleration of gravity, f is the flow friction factor,  $D_h$  is the hydraulic diameter of the tube and (du/dz) is the velocity variation along the z direction.

$$A \cdot G \cdot \frac{d(i + u^2/2 + g \cdot z)}{dz} = \pi \cdot D_h \cdot U \cdot (T_g - T_b) \tag{3}$$

Where i is the flow enthalpy, U is the overall heat transfer coefficient and  $T_b$  is the bulk temperature of the water flow. While the overall heat transfer coefficient can be expressed by:

$$\frac{1}{U} = \frac{1}{h_f} + \Sigma R_{\text{fouling}} + \Sigma R_{\text{cond.}} + \frac{1}{(A_o/A_h) \cdot h_g} \tag{4}$$

Where  $h_f$  is the heat transfer coefficient of the internal flow,  $D_o$  and  $D_h$  are the inner outer tube diameters respectively,  $A_o$  and  $A_h$  are the vertical tube surface area corresponding to it inner and outer radiuses and  $h_g$  is the heat transfer coefficient of the hot stream outside gases. The term  $(A_o/A_h) \cdot h_g$ , which is considered as an equivalent heat transfer coefficient, is simplified by assigning it a fixed value ( $h_{ge}$ ), estimated in base to experimental data which is equal to 600 (W/m<sup>2</sup>.K) [1].

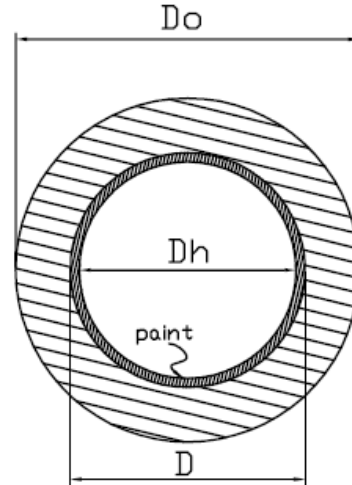


Fig. 2. Tube cross sectional area

According to the conduction resistances it was due to the tube material and due to coating which has a thermal conductivity ( $k_{\text{coating}}$ ) of 0.95 (W/m.K) as defined by O'Connor [2] and according to the fouling resistances it were because of the outside hot flue gases ( $R_{\text{flue, fouling}}$ ) which was equal to 0.002 (m<sup>2</sup>.K/W) which is tabulated in Lienhard [3] and the internal flow boiling ( $R_{\text{flow, fouling}}$ ) which was equal to 0.0002 (m<sup>2</sup>.K/W) which is tabulated in Lienhard [3]. While the heat flux represented by the following equation along the tube length:

$$q = h_{ge} (T_g - T_w) \tag{5}$$

Where (q) is heat flux which is not uniform but both  $h_{ge}$  and  $T_g$  are remaining uniform along the tube. Dividing the vertical tube length into three regimes which are liquid single-phase, subcooled boiling and two-phase saturated boiling. While the subcooled flow boiling regime is divided into three sub regimes which are partial flow boiling, fully developed boiling and significant void flow.

**2.2.1. Liquid Single-Phase Regime**

Because the water enters the tube below its boiling point as a compressed liquid, so the variations of the liquid density with the z-axis can be neglected so the velocity can be taken as constant, then the conservation of mass will be Eq.1 and the conservation of momentum will be.

$$-\frac{dp}{dz} = \rho_L \cdot g + 2 \cdot f_L \cdot \frac{G^2}{\rho_L \cdot D_h} \tag{6}$$

While the friction factor  $f_L$  for fully developed turbulent flow is calculated by the Romeo et al. correlation [4]. And the energy conservation become as follow:

$$A \cdot G \cdot \left[ \frac{d(c_{pL} \cdot T_b)}{dz} + \frac{1}{\rho_L} \cdot \frac{dp}{dz} + g \right] = \pi \cdot D_h \cdot U \cdot (T_g - T_b) \tag{7}$$

Where  $cpL$  is the liquid water specific heat. To get the overall heat transfer coefficient (U) use the Dittus-Boelter correlation [5] to get the pure single-phase liquid heat transfer coefficient of water as follow:

$$h_f = h_L = 0.023 Re^{0.8} Pr_L^{0.4} \left( \frac{k_L}{D_h} \right) \tag{8}$$

Where  $Pr_L$  is the Prandtl number and  $k_L$  is the liquid water thermal conductivity.

**2.2.2. Two-Phase Saturated Flow Boiling Regime**

Eq.1 which represent the conservation of mass for this regime while the conservation of momentum and using the separated flow model will be as follow:

$$-\frac{dp}{dz} = [\alpha \cdot \rho_v + (1-\alpha) \cdot \rho_L] \cdot g + \phi_L^2 \cdot \left[ 2 \cdot f_L \cdot \frac{G^2(1-x)^2}{\rho_L \cdot D_h} \right] + G^2 \cdot \frac{d}{dz} \left( \frac{x^2}{\alpha \cdot \rho_v} + \frac{(1-x)^2}{(1-\alpha) \cdot \rho_L} \right) \tag{9}$$

Where  $\alpha$  is the void fraction or volumetric vapor content,  $x$  is the steam quality and to determine the pressure variation along the tube, the Lockhart & Martinelli generalized correlation method has been followed [6, 7], in the case which is under study each phase is flowing alone in the tube would be turbulent so the relation for the two-phase multiplier ( $\phi_L$ ) will be:

$$\phi_L^2 = 1 + 20/X_{tt} + 1/X_{tt}^2 \tag{10}$$

$$X_{tt} = \left( \frac{\rho_v}{\rho_L} \right)^{0.5} \left( \frac{\mu_L}{\mu_v} \right)^{0.125} \left( \frac{1-x}{x} \right)^{0.875} \tag{11}$$

$$\alpha = [1 + 0.28 X_{tt}^{0.71}]^{-1} \tag{12}$$

Where  $X_{tt}$  is Martinelli parameter. While the energy Eq.2 for the two-phase saturated boiling regime will be:

$$A \cdot G \cdot \frac{\partial}{\partial z} \left\{ x \left( \frac{G^2 x^2}{2\rho_v \alpha^2} \right) + i_L + (1-x) \left[ \frac{G^2(1-x)^2}{2\rho_L(1-\alpha^2)} \right] + x(i_v - i_L) \right\} + A \cdot G \cdot \frac{\partial}{\partial z} (z \cdot g) = \pi \cdot D_h \cdot U \cdot (T_g - T_{sat}) \tag{13}$$

Where  $T_{sat}$  is the water flow saturation temperature. For simplicity, the kinetic term ( $u^2/2$ ) can be neglecting compared to the enthalpy terms [ $i_L, x(i_v - i_L)$ ]. To get the overall heat transfer coefficient (U) using the Kandlikar correlation [8] to get the water heat transfer coefficient which is two-phase for water inside smooth vertical tubes as follow:

$$\frac{h_f}{h_L} = \frac{h_{tp}}{h_L} = \begin{cases} 1.136Co^{-0.9} + 667.2Bo^{0.7} & \text{for } Co < \\ 0.6683Co^{-0.2} + 1058Bo^{0.7} & \text{for } Co > \end{cases} \tag{14}$$

$$Bo = \frac{q}{G \cdot i_{LV}} \tag{15}$$

$$Co = \left( \frac{1-x}{x} \right)^{0.8} \left( \frac{\rho_v}{\rho_L} \right)^{0.5} \tag{16}$$

Where  $h_{tp}$  is the two-phase heat transfer coefficient,  $Co$  is the convection number,  $Bo$  is the boiling number,  $q$  is the heat flux and  $i_{LV}$  is the latent heat of vaporization. To calculate  $h_L$  from Eq.8, put Reynolds number in the form:

$$Re = \frac{G \cdot (1-x) \cdot D_h}{\mu_L} \tag{17}$$

**2.2.3. Subcooled Flow Boiling Regime**

Subcooled flow boiling zone are divided into three different sub-regimes which are as shown in figure 3:

- Partial boiling from ONB to FDB.
- Fully developed boiling from FDB to OSV.
- Significant void flow or net vapor generation from OSV to saturation.

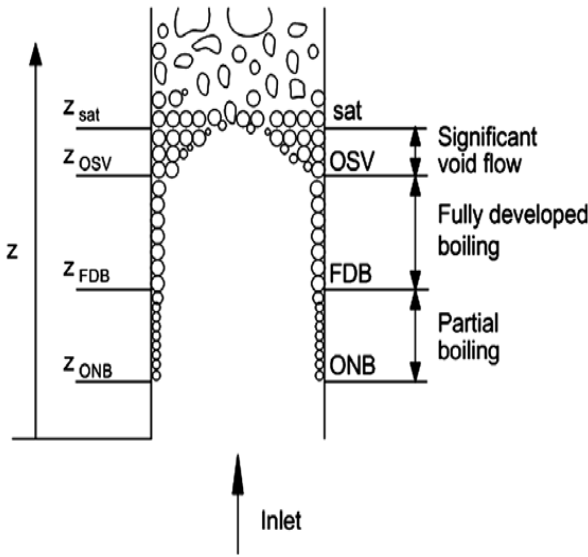


Fig. 3. Subcooled flow boiling in details [1]

In summary partial boiling and fully developed boiling (from ONB to OSV) are highly subcooled so the continuity, momentum and energy equations would be the same of the liquid single-phase regime Eqs.1, 6 and 7 and the significant void flow or net vapor generation (from OSV to saturation) is low subcooled so the continuity, momentum equations would be the same of the two-phase saturated flow boiling regime Eqs.1 and 9 and energy equation will be the same of the liquid single-phase regime Eq.7, all significant void flow or net vapor generation region equations work with the apparent quality ( $x_a$ ) which was used by Saha and Zuber correlations [9] as function of the non-equilibrium for the NVG or OSV point.

To get the overall heat transfer coefficient (U) using the Zambrana et al. correlation [1] to get the water heat transfer coefficient which is subcooled water inside smooth vertical tubes, it is an elliptical expression as function of the two ends which are pure liquid and saturated flow, the relation is as follow:

$$h_f = h_L = h_{L_{so}} - \sqrt{\frac{(B+Z)(Z_{Sat.} - Z)}{(B+Z_o)(Z_{Sat.} - Z_o)}} (h_{L_{so}} - h_{L_o}) \quad (18)$$

$$B = -Z_o + \frac{Z_{Sat.} - Z_o}{1 - \frac{2a(Z_{Sat.} - Z_o)}{(h_{L_{so}} - h_{L_o})}} \quad (19)$$

$$a = \left( \frac{dh_L}{dz} \right)_{z=Z_o} \quad (20)$$

Where  $h_{L_{so}}$  is the heat transfer coefficient at the flow saturation point which is  $Z_{Sat.}$  and  $h_{L_o}$  is the heat transfer coefficient at onset nucleate boiling which is  $Z_o$ .

The heat transfer coefficient ( $h_{L_{so}}$ ) was predicted by Kandlikar correlation [8] at the point of vapor quality zero ( $Z_{Sat.}$ ) using Eqs.14 and 15, and The heat transfer coefficient ( $h_{L_o}$ ) was predicted by Dittus-Boelter correlation [5] at the onset nucleate boiling point ( $Z_o$ ) using Eq.8.

### 2.3. Performance Analysis

To determine if the coating improve the heat transfer of the system or not, Colburn factor ( $j$ ) will be calculated and then plotted versus Reynolds number (Sadik Kakaç [10]), where Colburn factor equal to:

$$j = St \cdot Pr^{2/3} \quad (21)$$

$$St = \frac{h}{G \cdot c_p} \quad (22)$$

$$Pr = \frac{c_p \cdot \mu}{k} \quad (23)$$

Where St is the Stanton number and Pr is the Prandtl number. As the colburn factor increase this mean improvement of heat transferring system.

According to the previous equations for the four different water flow regimes which are liquid single phase (from inlet till onset nucleate boiling), partial and fully developed boiling (from onset nucleate boiling till OSV), significant void flow or net vapor generation (from OSV till saturation) and two-phase saturated flow boiling (from saturation till required steam quality).

To solve these equations professional computerized program used called Engineering Equation Solver (EES) [11]. For further details reference can be made to the work of Abdel Azim [12].

### 2.4. Boundary, Inlet and Physical Conditions

Table 1 summarized the boundary, inlet and physical conditions for the case under study which is consists of vertical tube heated by hot gases with uniform temperature and heat transfer coefficient while subcooled water enter the vertical tube from its bottom.

Table 1. Boundary, inlet and physical conditions

Item	Type	Symbol (unit)	Value
Water temperature	inlet	$T_i$ (°C)	80
Water pressure	Inlet	$P_i$ (KPa)	150
Water mass flux	Inlet	$G$ (kg/m <sup>2</sup> .s)	47.2
Final steam quality	Inlet	$x$ (kg <sub>st</sub> /kg <sub>total</sub> )	20%
Hot gases temperature	Boundary	$T_g$ (°C)	140
Equivalent heat transfer coefficient	Boundary	$h_{eg}$ (W/m <sup>2</sup> .K)	600
Tube diameter	Physical	$D$ (cm)	2.6

## 3. Results and Discussions

This study has been developed for water pressure near to ambient and final low steam quality value it can be near to operating conditions of the low pressure level of a heat recovery steam generator (HRSG), but with enough subcooling to detect the ONB.

Results of water subcooled flow boiling are obtained for cases of no coating is used and for five different coating thicknesses ( $\delta$ ) which are (25, 45, 61, 93 and 124  $\mu\text{m}$ ) in each case four different surface roughness dimensionless groups ( $\epsilon/D$ ) which are (0.05, 0.01, 0.001 and 0.0001) and for the inlet and physical boundary conditions as illustrated in table 1.

### 3.1. Comparison of All Boiling Regimes for Four Different Surface Roughness Results

This samples of the results for this section were taken for internal surface coating thickness of 25 ( $\mu\text{m}$ ) and the surface roughness dimensionless groups ( $\epsilon/D$ ) which were (0.05, 0.01, 0.001 and 0.0001).

Figure 4 illustrate the difference between the four surface roughnesses coefficients on the flow friction factor estimation.

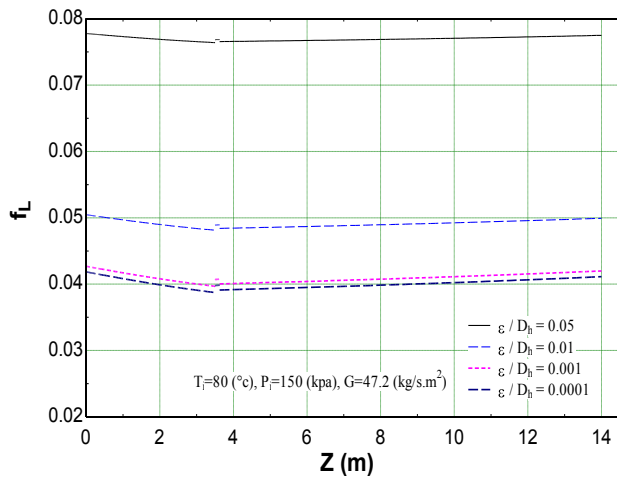


Fig. 4. Flow friction factor versus tube length

Using internal surface coating with different surface roughnesses coefficients make enhancement for the boiling process as Investigation shows in Figure 5 that the heat transfer coefficient is nearly a straight line in the regime of two-phase saturated flow boiling.

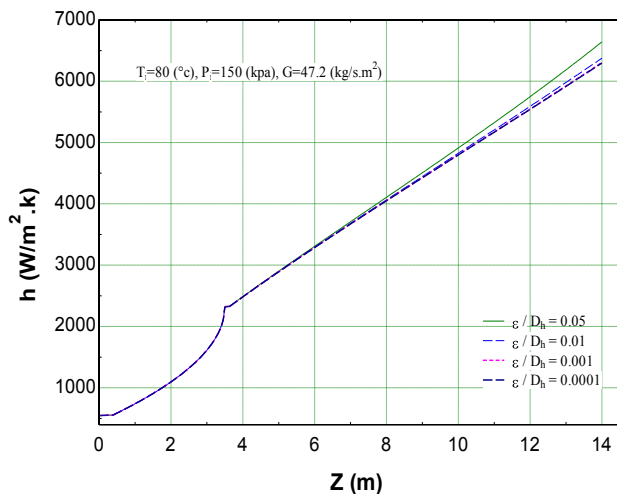


Fig. 5. Heat transfer coefficient versus tube length

The four different cases did not differ for the single-phase and subcooled flow boiling, the noticeable difference occurred in the saturated two-phase flow boiling where the surface roughness ( $\epsilon/D$ ) of 0.05 resulted in the largest value of the heat transfer coefficient. Because of for higher surface roughness,

more cavity places for initiating the steam bubbles over the tube surface, which will give more steam, which has higher velocity than liquid so the heat transfer coefficient will be increased. While the overall heat transfer coefficient doesn't show in figure 6 any noticeable difference for the four surface roughness cases for all flow boiling regimes.

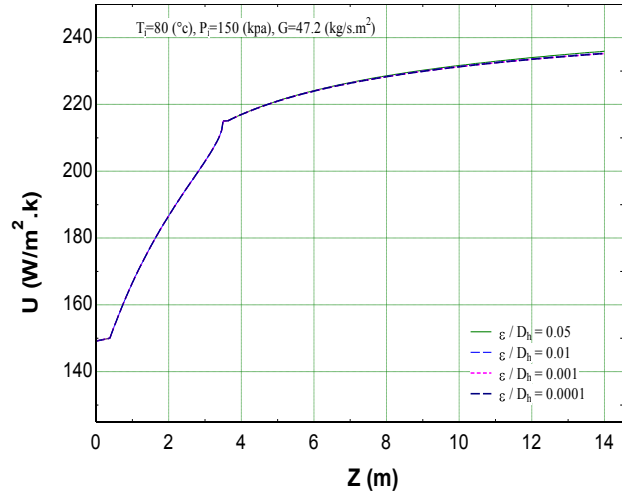


Fig. 6. Overall heat transfer coefficient versus tube length

Because of the hot gases have uniform temperature and heat transfer coefficient along the vertical heated tube Figure 7 shows that the heat flux in this case is not uniform along the vertical tube length; the heat flux is increased with the flow direction. The noticeable different occurred in the saturated two-phase flow boiling where the surface roughness ( $\epsilon/D$ ) of 0.05 resulted in the largest value of the heat transfer coefficient which give more turbulence because of there are more cavities to enhance boiling process. As the value of the surface roughness decreased, this difference diminished.

For judgment on the improvement occurred on the boiling heat transfer, Colburn factor is an indication if the boiling heat transfer was improved or not due to the usage of the rough surface, figure 8 shows Colburn factor increases due to the rough coated surface which mean improvement in the boiling heat transfer process.

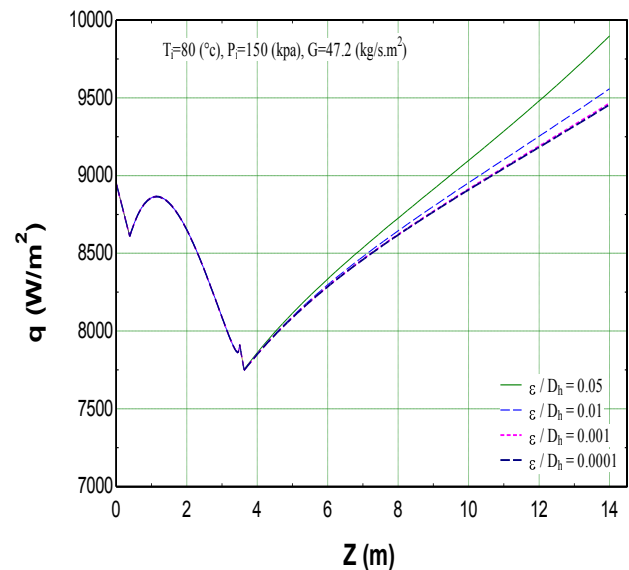


Fig. 7. Heat flux versus tube length

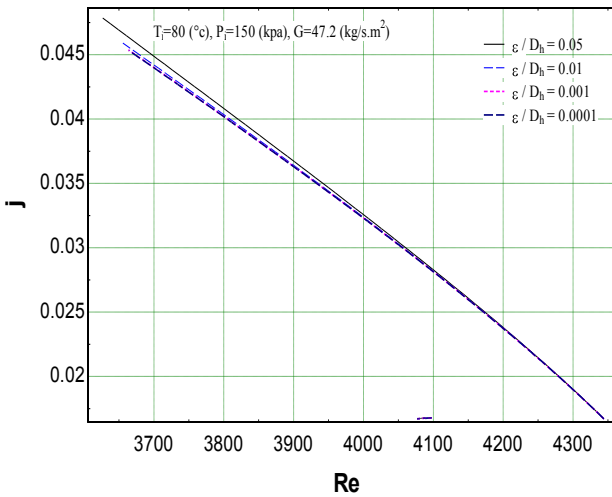


Fig. 8. Colburn factor (j) versus Reynolds (Re)

### 3.2. Comparison of All Boiling Regimes for Four Different Coating Thicknesses Results

The samples of the results for this section were taken for no coating and different coating thicknesses of (25, 45, 61, 93 and 124) ( $\mu\text{m}$ ) and the surface roughness dimensionless group ( $\epsilon/D$ ) = 0.05.

Figures 9 and 10 show that by increasing the internal coating thickness heat transfer coefficient, heat flux and overall heat transfer coefficient will be decreased because of the increase of the conduction thermal resistance of the internal coated material.

It was indicated in figures 12 to 14 that the boiling heat transfer coefficient, overall heat transfer coefficient and heat flux decrease as increasing the internal coating to due add more thermal conduction resistance facing the heat flux crossing the heated tube thickness.

### 3.3. Parameters Variations of All Boiling Regimes

This samples of the results for this section were taken for tube thickness of 25 ( $\mu\text{m}$ ) and the surface roughness dimensionless group ( $\epsilon/D$ ) = 0.05

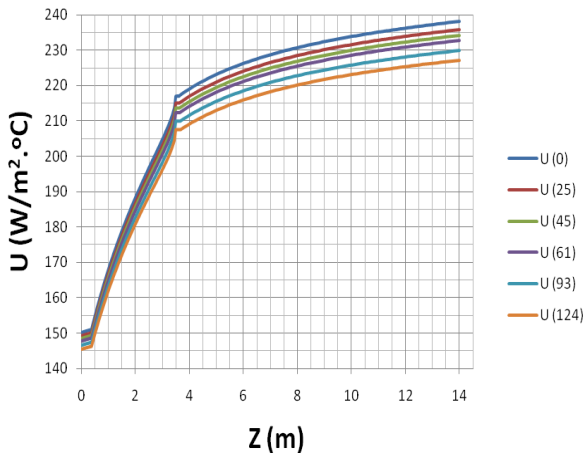


Fig. 10. Overall heat transfer coefficient versus tube length

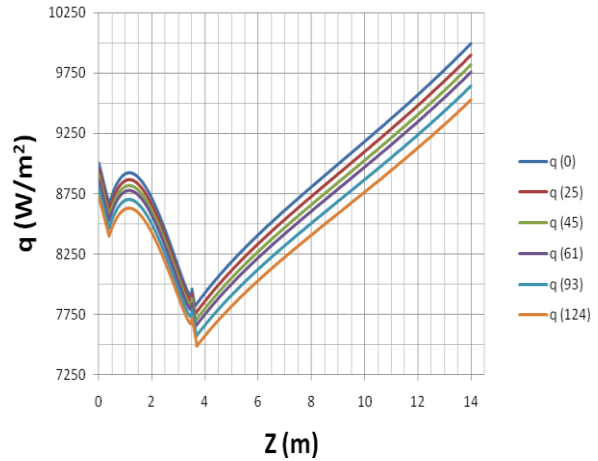


Fig. 11. Heat flux versus tube length

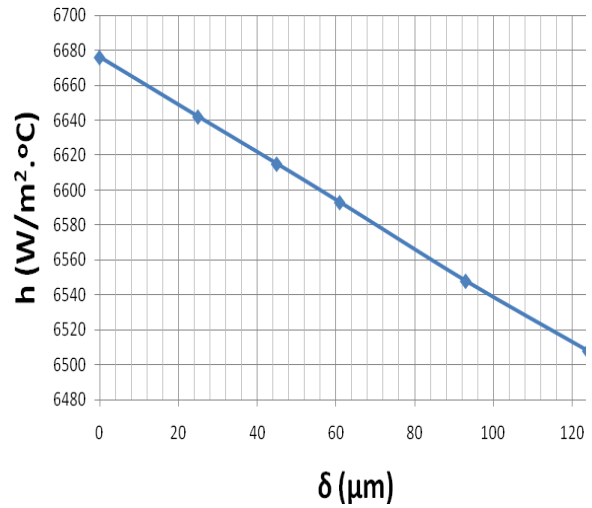


Fig. 12. Heat transfer coefficient versus coating thickness

Figure 15 illustrate the heat transfer coefficient of the water flow inside the heated tube and also the heat flux passes through the tube wall. It is clearly in this figure that the heat flux distribution along the vertical heated tube is non-uniform and varies depending on the boiling heat transfer mechanisms

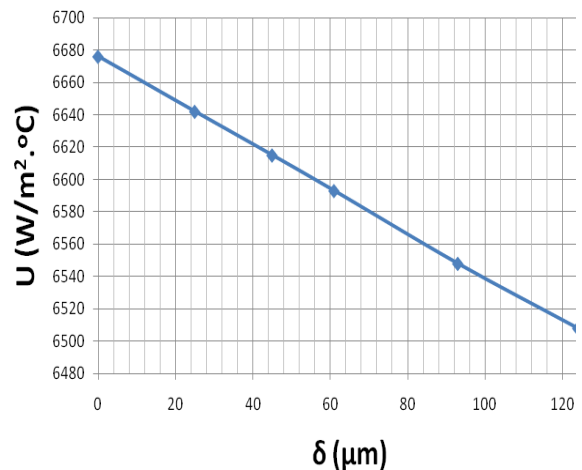


Fig. 13. Overall heat transfer coefficient versus coating thickness

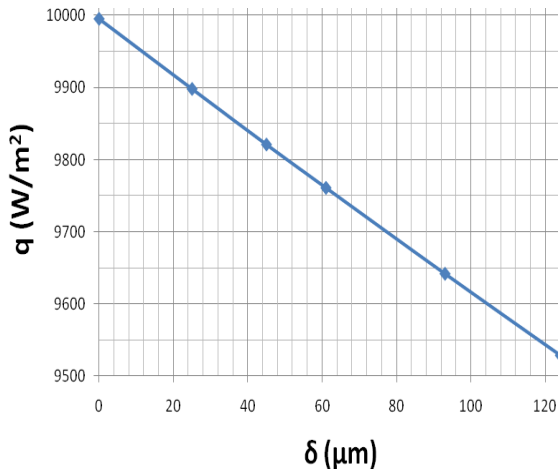


Fig. 14. Heat flux versus coating thickness

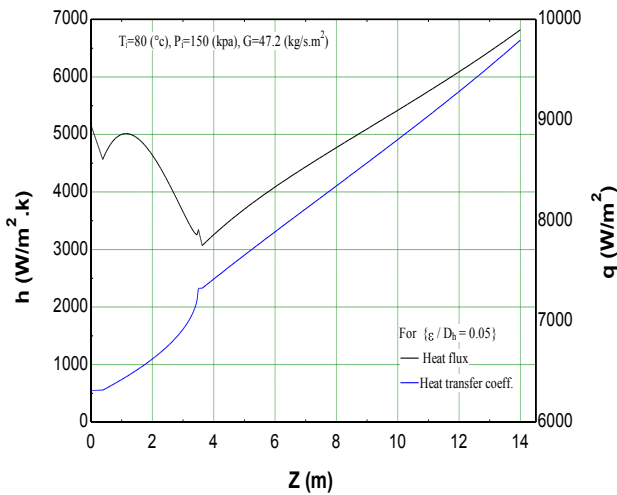


Fig. 15. Heat transfer coefficient and heat flux versus tube length

#### 4. Conclusions

It was concluded that the required tube length was 14 (m) to get the prescribed outlet conditions with the predetermined inlet and boundary conditions.

By changing the surface roughness coefficient for certain internal surface coating thickness it was noticed that most of the results investigated has heat transfer enhancement for the surface roughness value of 0.05, noticed in the figures 5 to 8.

While when internal coating thickness changed for constant value of the surface roughness shows that increasing the internal coating has a bad effect on the boiling heat transfer process which results on decreasing the amount of produced steam at the exit section of the heated tube because of heat flux and heat transfer coefficient decreased. So not using internal surface coating will give highest value for heat flux and amount of produced steam, but it is necessary to use coating for boiling initiating and for chemical corrosion so it was found that using coating thickness of 25 ( $\mu\text{m}$ ) will be efficient.

Finally it was concluded from this results of this work that heat transfer process is enhanced for the internal coating thickness of 25 ( $\mu\text{m}$ ) and surface coating coefficient of 0.05.

#### Nomenclature

A	Cross sectional area, $\text{m}^2$
$c_p$	Specific heat, $\text{J}/\text{kg} \cdot ^\circ\text{C}$
D	Diameter, m
EES	Engineering Equation Solver
f	Friction Factor
G	Mass flux, $\text{kg}/\text{m}^2 \cdot \text{s}$
g	Gravity, $\text{m}/\text{s}^2$
h	Heat transfer coefficient, $\text{W}/\text{m}^2 \cdot ^\circ\text{C}$
i	Flow enthalpy, $\text{J}/\text{kg}$
k	Conductivity, $\text{W}/\text{m} \cdot ^\circ\text{C}$
$\dot{m}$	Mass flow rate, $\text{kg}/\text{s}$
NVG	Net vapor generation
ONB	Onset of nucleate boiling
OSV	Onset of significant void
p	Flow pressure, Pa
PB	Partial boiling
q	Heat flux, $\text{W}/\text{m}^2$
R	Thermal resistance, $\text{m}^2 \cdot ^\circ\text{C}/\text{W}$
SVF	Significant void flow
T	Temperature, $^\circ\text{C}$
U	Overall heat transfer coefficient, $\text{W}/\text{m}^2 \cdot ^\circ\text{C}$
u	Flow velocity, $\text{m}/\text{s}$
x	Steam quality, $\text{kg}_{\text{steam}}/\text{kg}_{\text{total}}$
z	Tube height, m

#### Greek Symbols

$\Delta$	Change interval of any property
$\rho$	Mass density, $\text{kg}/\text{m}^3$
$\delta$	Coating thickness, $\mu\text{m}$
$\eta$	Fin efficiency
$\mu$	Dynamic viscosity, $\text{kg} \cdot \text{m}/\text{s}$
$\alpha$	Void fraction
$\phi$	Two-phase multiplier factor
$\epsilon$	Surface roughness, m

#### Subscripts

b	Flow bulk
Cond.	Conduction resistance
f	Convective
fL	Full liquid
fouling	Fouling resistance
g	Gas
ge	Gas equivalent
h	Hydraulic
L	Liquid
Lo	All liquid
Lso	Liquid saturated
o	Outer
Sat.	Saturation
Sub.	Subcooled
TP	Two phase
w	Wall

#### Non-dimensional Numbers

Co	Convective number
j	Colburn factor
Pr	Prandtl number
Re	Reynolds number
St	Stanton number
$X_{tt}$	Lockhart-Martinelli parameter

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