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An Evaluation of Heat Transfer Enhancement Technique in Flow Boiling Conditions Based on Entropy Generation Analysis: Micro-Fin Tube

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Abstract: The flow boiling heat transfer is one of the common phenomenon happening in the industries. The micro-fin tubes are one of the geometries widely used to enhance heat transfer rate in boiling condition. The entropy generation analysis is presented with its formulation to find precisely the best operating conditions in micro-fin tubes in terms of geometrical parameters and flow conditions. This analysis shows important aspects of losses in fluid systems undergoing boiling. The losses include thermal loss related to the heat transfer and hydraulic one related to the pressure drop. The relevant terms are described for both of these losses. The optimum tube diameter under specified conditions is found. The effect of different flow conditions such as mass velocity, inlet vapor quality on contribution of pressure drop and heat transfer in entropy generation is discussed. It is discovered that there is a desirable set of conditions of fluid flow and micro-fin geometrical shape for which the minimum entropy generation is reached.

Keywords: entropy generation analysis (EGA); flow boiling; enhanced heat transfer; micro-fined tube; pressure drop

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1. Introduction

Various researchers have conducted studies on the entropy generation in heat exchangers. Manjunath and Kaushik (2014) performed a review on the importance of first and second law of thermodynamics in entropy generation analysis. Also, the application of the entropy generation analysis in design and optimization of engineering systems was reviewed by Sciacovelli et al. (2015).

Naphon (2011) presented experimental results as well as theoretical formulations for entropy generation and exergy destruction in a tubular heat exchanger. They included the effect of fluid temperature variations along the length of the heat exchanger. Their results indicated that for a constant hot water mass flux, the entropy generation number increases for higher inlet temperatures. Ye and Lee (2012) investigated a numerical model for fin-and-tube condensers with complex refrigerant circuit based on entropy generation. Compared to the condensers with simple refrigerant path, there was an appreciable enhancement in heat transfer performance. Manjunath and Kaushik (2015) conducted an analytical analysis on an unbalanced heat exchanger based on the second law of thermodynamics. They varied the length-to-diameter ratio for both counter flow and parallel flow configurations. Their results showed that the dimensions for optimum heat exchanger design, namely length-to-diameter ratio, can be found.

Some studies have been published with the focus on the entropy generation analysis in two-phase flows in heat exchanger. Kaushik and Manjunath (2011) analyzed a single pass double pipe two phase flow heat exchanger. Based on the second law, they mainly considered three two-phase flow regimes. They reported their results of entropy generation number by varying parameters such as diameter, mass velocity, wall superheat, and quality.

Few efforts have been made to investigate entropy generation in two-phase flows in tubes and channels. The local entropy generation for diabatic two-phase flow was studied by Revellin et al. (2009). They performed a study on local entropy generation of two-phase diabatic flows. In order to analyze the entropy generation in saturated flow boiling in pipes, they developed two models, namely separated flow model and mixture model. Entropy generation in the evaporator of a vapor compression refrigeration cycle was inspected by Türkakar and Okutucu-Özyurt (2015) by studying the variations of channel height and width, the heat flux, the mass flow rate and several other parameters. They concluded that increasing heat flux increases the entropy generation. Abdous et al. (2015) conducted a comprehensive study on using of entropy generation analysis in a helically coiled tube in flow boiling condition under a constant heat flux. They investigated changing of important flow parameters such as saturation temperature, heat flux and vapor quality on entropy generation. The optimum coil diameter was found based on their analysis.

The main goal of this study is to find the optimum diameter of the micro-fin tube by means of entropy generation analysis. Then, variation of flow conditions such as vapor quality and mass velocity is studied on entropy generation and contribution of pressure drop and heat transfer one.

2. Mathematical Modeling

Suppose a mixture of liquid and vapor in saturated conditions enters a tube under constant wall heat flux. The mathematical model of saturated two-phase flow is based on the second law of thermodynamics. For a finite control volume of length $\mathrm{d}l$, the total entropy generation per unit length $\dot{S}'_{\mathrm{gen}}$ can be written as:

$$\dot{S}'_{\mathrm{gen}} = \frac{\dot{m}\mathrm{d}(xs_{\mathrm{v}} + (1-x)s_{\mathrm{l}}) - \dfrac{\dot{Q}}{T_{\mathrm{w}}}}{\mathrm{d}l} \tag{1}$$

where $\dot{m}$ is the total mass flow rate, T_{w} denotes the channel wall temperature, $\dot{Q}$ is the rate of heat transfer into the control volume and *s* is the entropy. The vapor quality can be defined as Equation (2):

$$x = \frac{\dot{m}_{\mathrm{v}}}{\dot{m}_{\mathrm{v}} + \dot{m}_{\mathrm{l}}} \tag{2}$$

The entropy generation per unit length ($\dot{S}'_{\mathrm{gen}}$) becomes:

$$\dot{S}'_{\mathrm{gen}}\mathrm{d}z = \dot{m}(s_{\mathrm{lv}}\mathrm{d}x + x\mathrm{d}s_{\mathrm{v}} + (1-x)\mathrm{d}s_{\mathrm{l}}) - \frac{\dot{Q}}{T_{\mathrm{w}}} \tag{3}$$

And, the values of $\mathrm{d}h_{\mathrm{v}}$ and $\mathrm{d}h_{\mathrm{l}}$ are:

$$\mathrm{d}h_{\mathrm{v}} = T_{\mathrm{v}}\mathrm{d}s_{\mathrm{v}} + v_{\mathrm{v}}\mathrm{d}p_{\mathrm{v}} \tag{4}$$

$$\mathrm{d}h_{\mathrm{l}} = T_{\mathrm{l}}\mathrm{d}s_{\mathrm{l}} + v_{\mathrm{l}}\mathrm{d}p_{\mathrm{l}} \tag{5}$$

Therefore, in Equation (3), the values of $\mathrm{d}s_{\mathrm{v}}$ and $\mathrm{d}s_{\mathrm{l}}$ are obtained from Equations (4) and (5). Assuming $T_{\mathrm{v}} = T_{\mathrm{l}} = T_{\mathrm{sat}}$, $\mathrm{d}p_{\mathrm{v}} = \mathrm{d}p_{\mathrm{l}}$ and $s_{\mathrm{lv}} = \frac{h_{\mathrm{lv}}}{T_{\mathrm{sat}}}$ in saturated flow boiling, the entropy generation per unit length ($\dot{S}'_{\mathrm{gen}}$) can be rewritten as:

$$\dot{S}'_{\mathrm{gen}}\mathrm{d}z = \frac{\dot{m}}{T_{\mathrm{sat}}}(h_{\mathrm{lv}}\mathrm{d}x + x\mathrm{d}h_{\mathrm{v}} + (1-x)\mathrm{d}h_{\mathrm{l}}) - \frac{\dot{Q}}{T_{\mathrm{w}}} - \frac{\dot{m}(xv_{\mathrm{v}} + (1-x)v_{\mathrm{l}})}{T_{\mathrm{sat}}}\mathrm{d}p \tag{6}$$

$$\dot{S}'_{\mathrm{gen}}\mathrm{d}z = \frac{\dot{m}}{T_{\mathrm{sat}}}\mathrm{d}h_{\mathrm{tp}} - \frac{\dot{Q}}{T_{\mathrm{w}}} - \frac{\dot{m}v_{\mathrm{tp}}\mathrm{d}p}{T_{\mathrm{sat}}} \tag{7}$$

Using the first law of thermodynamics $\dot{Q}$ is obtained as:

$$\dot{Q} = \dot{m}\mathrm{d}h_{\mathrm{tp}} \tag{8}$$

Simplifying Equation (7) by using Equation (8), the final term for the entropy generation per unit length is

$$\dot{S}'_{\mathrm{gen}} = \dot{m}\left[\frac{\mathrm{d}h_{\mathrm{tp}}}{\mathrm{d}z}\left(\frac{1}{T_{\mathrm{sat}}} - \frac{1}{T_{\mathrm{w}}}\right) - \frac{v_{\mathrm{tp}}}{T_{\mathrm{sat}}}\frac{\mathrm{d}p}{\mathrm{d}z}\right] \tag{9}$$

Two contributions can be understood of the entropy generation per unit length ($\dot{S}'_{\mathrm{gen}}$) in Equation (9). The heat transfer contribution [(Equation (10)], and the pressure drop contribution [Equation (11)];

$$\dot{S}'_{\mathrm{gen,ht}} = \dot{m}\frac{\mathrm{d}h_{\mathrm{tp}}}{\mathrm{d}z}\left(\frac{1}{T_{\mathrm{sat}}} - \frac{1}{T_{\mathrm{w}}}\right) \tag{10}$$

$$\dot{S}'_{\mathrm{gen,pd}} = \frac{\dot{m}v_{\mathrm{tp}}}{T_{\mathrm{sat}}}\left(-\frac{\mathrm{d}p}{\mathrm{d}z}\right) \tag{11}$$

$\frac{\mathrm{d}p}{\mathrm{d}z}$ being the two-phase pressure drop of tube covering gravitational and accelerative pressure drops. The relationship between vapor quality and void fraction is obtained from (Woldesemayat and A. J. Ghajar, 2003);

$$\varepsilon = \frac{U_{\mathrm{sv}}}{U_{\mathrm{sv}}\left(1 + \left(\frac{U_{\mathrm{sl}}}{U_{\mathrm{sv}}}\right)^{\left(\frac{\rho_{\mathrm{v}}}{\rho_{\mathrm{l}}}\right)^{0.1}}\right) + 2.9\left[\frac{gD\sigma(1+\cos\theta)(\rho_{\mathrm{l}} - \rho_{\mathrm{v}})}{\rho_{\mathrm{l}}^2}\right]^{0.25}(1.22 + 1.22\sin\theta)^{\frac{P_{\mathrm{atm}}}{P_{\mathrm{system}}}}} \tag{12}$$

where, U_{sl} and U_{sv} are the superficial liquid and gas velocity, respectively, g is the gravitational acceleration and the θ is tube inclination angle. ρ_{v} and ρ_{l} are respective densities of vapor and liquid. σ is the fluid surface tension. In the case of a micro-fin tube, D is the tube hydraulic diameter. P_{atm} is the atmospheric pressure and the P_{system} is the system pressure.

3. Results and Discussions

Figure 1 shows a schematic of a tube-in-tube straight heat exchanger designed by Wongsangam et al. of a micro-fin tubes for obtaining correlations of heat transfer coefficient and frictional pressure drop. HFC-134a flows in the inner tube, while heating water flows in the annulus in counter direction.

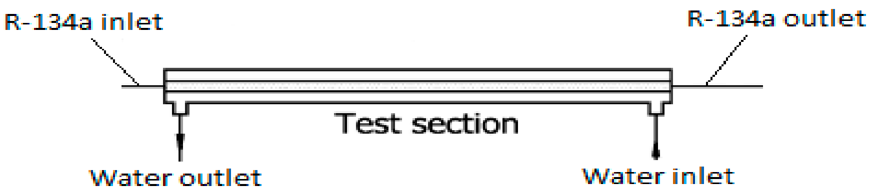

Figure 1. The schematic figure of the micro-fin tube-in-tube heat exchangers [Wongsa-ngam et al., 2002].

3.1. The Geometrical Parameters of the Micro-Fin Tube

First, the geometry of micro-fin must be demonstrated here by some details. Important Parameters height (e_f), the bottom width (B_w), the bottom thickness (B_T) and the angle α are shown in Figure 2. Symbol (N) denotes the number of fins for the micro-fin tube.

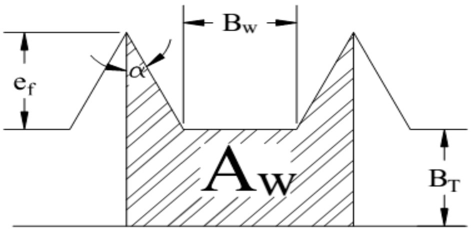

Figure 2. The cross sectional view of the micro-fin tube (Wongsa-ngam et al., 2002).

S_p is the distance between two tips of fins (Wongsa-ngam et al., 2002);

$$S_p = B_w + \frac{2e_f}{\cos(\alpha)} \tag{137}$$

The cross sectional tube wall area per fin A_w and cross sectional flow area A_c are defined as follow;

$$A_w = e_f^2 \tan(\alpha) + (2e_f \tan(\alpha) + B_w)B_T \tag{148}$$

$$A_c = \frac{\pi D_o^2}{4} - NA_w \tag{19}$$

The hydraulic diameter is;

$$D_h = \frac{4A_c \cos(\beta)}{NS_p} \tag{20}$$

The alpha angle (α) in Figure 2 is:

$$\alpha = \tan^{-1}\left(\frac{\frac{\pi D_i}{N} - B_W}{2e_f}\right) \tag{21}$$

The flow conditions are constant and presented in Table 1.

Table 1. The assumed flow parameters.

G ($kgm^{-2}s^{-1}$)	q (kWm^{-2})	T_{sat} (°C)	x_{in}
400	10	15	0.2

The increase in the value of the mass flow rate is directly proportional to the square of tube outer diameter (D_o^2). In the micro-fin tube, the mass flow rate ($\dot{m} = GA_c$) varies with A_c (Equation (19)). In addition, at constant heat flux, the value of total heat rate entering the micro-fin tube is proportional to the tube outer diameter (D_o). Therefore, the vapor quality decreases which leads to increase in the value of mixture density. Consequently, at constant mass velocity, the mixture velocity and pressure drop decrease. At low values of tube outer diameter (D_o), the decrease in pressure drop causes a reduction in pressure drop contribution. Gradually, the increase in the value of tube outer diameter (D_o), leads to an increase in pressure drop contribution.

Therefore, at this region the increase in the value of mass flow rate becomes the dominant parameter in increasing the pressure drop contribution as shown in Figure 3. Consequently, the minimum point appears in the pressure drop contribution for the micro-fin one. Finally, the tube with 5 mm outer diameter is recognized as an optimum for mentioned conditions.

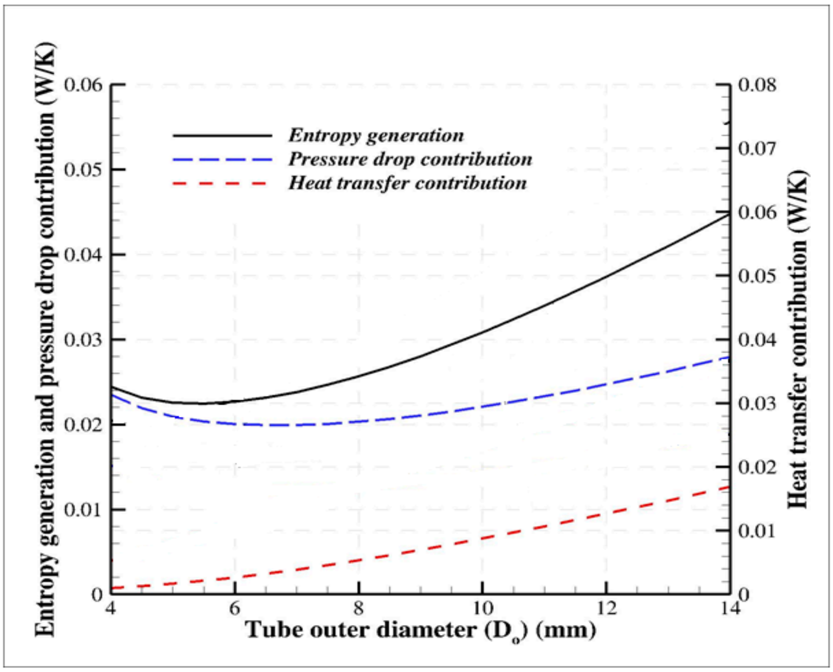

Figure 3. The entropy generation, pressure drop contribution and the heat transfer one for the micro-fin tube with flow conditions (Table 1) and geometrical parameters (Table 2).

Table 2. The assumed geometrical parameters when the tube outer diameter ($\boldsymbol{D_o}$) changes from 4 mm to 14 mm.

e_f (mm)	B_w (mm)	N	B_T (mm)	L (m)
0.2	0.27	60	0.3	2.5

3.2. Change in Flow Conditions

In this section, the geometrical parameters for the micro-fin tube are assumed to match with the study of Wonsagam et al. as mentioned in Table 3.

Table 3. The geometrical parameters for the micro-fin tube.

D_o (mm)	e_f (mm)	B_w (mm)	N	B_T (mm)	L (m)
9.52	0.2	0.27	60	0.3	2.5

Assuming the geometrical parameters in Table 4, the effect of changing in the flow conditions such as mass velocity (G), inlet vapor quality (x_{in}) is studied.

The variation of entropy generation, and the respective pressure drop and heat transfer contributions are plotted versus mass velocity (G) in Figure 4 for micro-fin tube. At constant heat flux, as the value of mass velocity (G) increases, the heat transfer coefficient rises (Wongsangam et al., 2002). Therefore, the heat transfer contribution in entropy generation decreases (Equation (10)). Moreover, the increase in G results in an increase in the frictional losses in both tubes. Thus, the pressure drop contribution in entropy generation increases. Figure 5 illustrates the effect of variation of inlet vapor qualities (x_{in}) on total entropy generation and its components for the micro-fin tube (Table 3).

Table 4. The assumed flow conditions when the value of mass velocity ($\boldsymbol{G}$) changes from $\mathbf{200\ kgm^{-2}s^{-1}}$ – $\mathbf{600\ kgm^{-2}s^{-1}}$.

q ($\mathrm{kWm^{-2}}$)	$\mathrm{T_{sat}}$ (°C)	$\mathrm{x_{in}}$
10	15	0.2

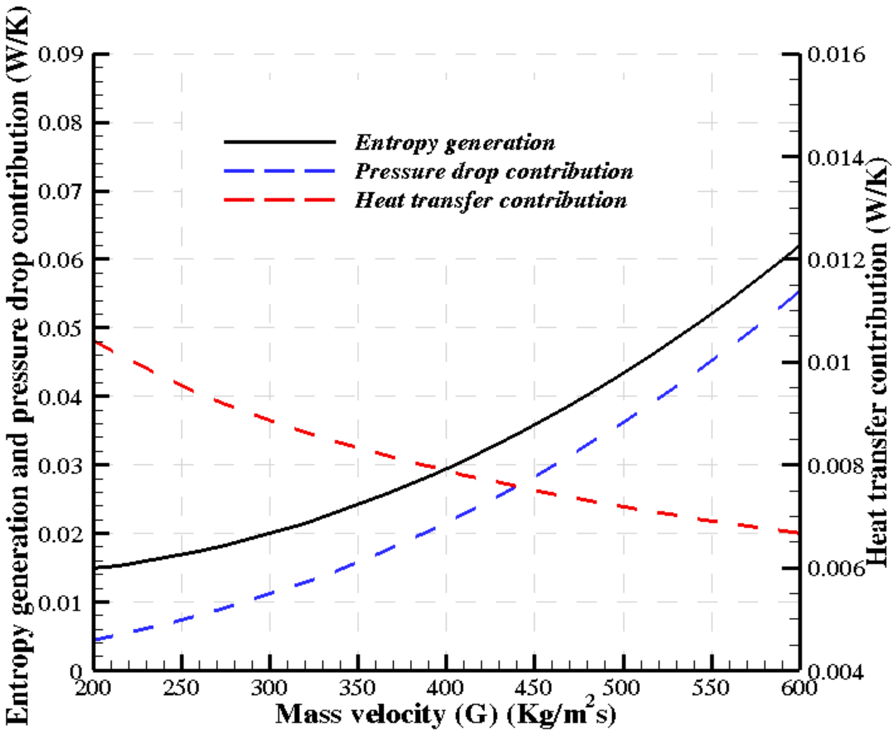

Figure 4. The variation of entropy generation, pressure drop contribution and heat transfer one versus mass velocity ($\boldsymbol{G}$) for the micro-fin tube geometrical parameters (Table 3) with assumed flow conditions (Table 4)

The flow conditions are presented in Table 5. The reduction in two-phase mixture density at constant mass velocity (450 $\mathrm{kgm^{-2}s^{-1}}$), is the result of increasing the inlet vapor quality from 0.1 to 0.5. Therefore, the mixture velocity and pressure drop contribution increasing gradually. The rising the value of inlet vapor quality results in higher heat transfer coefficients (Wongsangam et al., 2002). This means that for higher x_{in} the heat transfer contribution is lower.

Table 5. The flow conditions when inlet vapor quality (x_{in}) changes from 0.1–0.5.

q (kWm^{-2})	T_{sat} (°C)	G ($kgm^{-2}s^{-1}$)
10	15	450

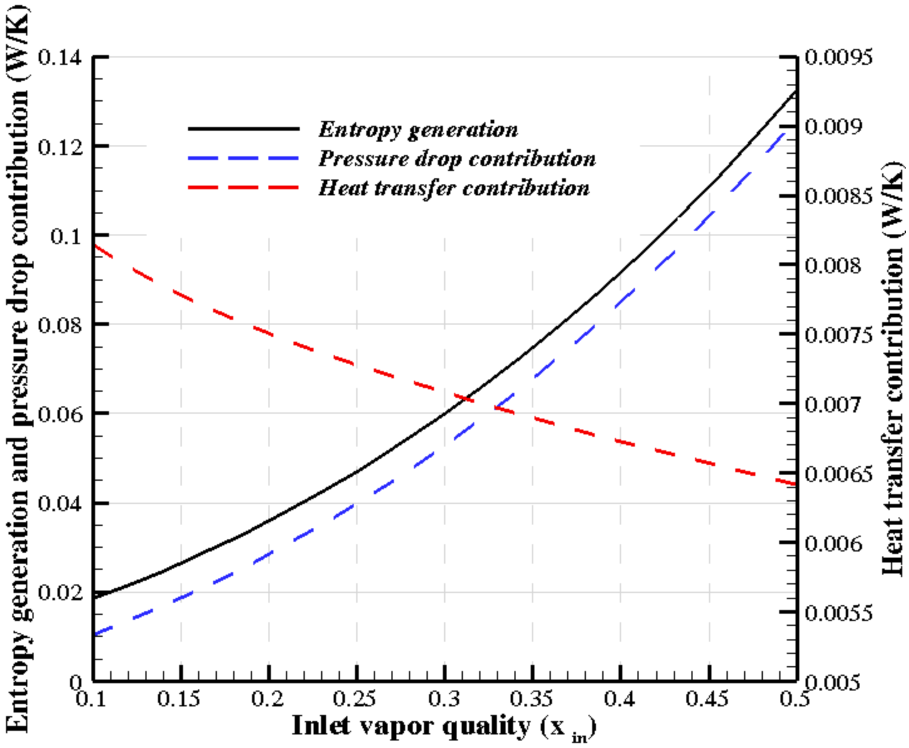

Figure 5. The variation of entropy generation, pressure drop contribution and heat transfer one versus inlet vapor quality (x_{in}) for the micro-fin tube geometrical parameters (Table 3) with assumed flow conditions (Table 5).

4. Conclusions

Enhancement of heat transfer rate is possible using various passive methods. Creating micro-fins in the tube is one of the passive methods to augment the heat transfer coefficient. To evaluate this method, the entropy generation analysis is used in flow boiling. By using entropy generation analysis, the tube optimum diameter is found for specified flow conditions. The variation of flow parameters such as mass velocity and vapor quality is considered. It is showed that, by increasing both of these factors, the contribution of pressure drop increases while the heat transfer one decreases.

Conflicts of Interest: The authors declare no conflict of interest.

Nomenclature

A	Cross section (m^2)
A_c	Cross sectional flow area (m^2)
A_W	Cross sectional tube wall area per fin (m^2)
Be	Bejan number (−)
B_T	Bottom thickness (m)
B_W	Bottom width (m)
dl	Element discretization (m)
D_h	Hydraulic diameter (m)
D_o	Tube outer diameter (m)
e_f	Fin height (m)

G	Mass velocity ($\mathrm{kgm^{-2}s^{-1}}$)
H	Specific enthalpy ($\mathrm{Jkg^{-1}}$)
L	Length (m)
$\dot{m}$	Mass flow rate ($\mathrm{kgs^{-1}}$)
N	Number of fins(−)
N_s	Entropy generation number (−)
P	Pressure (Pa)
P	Perimeter (m)
Q	Heat flux ($\mathrm{Wm^{-2}}$)
$\dot{Q}$	Heat rate (W)
S	Specific entropy ($\mathrm{JK^{-1}}$)
$\dot{S}'_{\mathrm{gen}}$	Entropy generation per unit length ($\mathrm{Wm^{-1}K^{-1}}$)
T	Temperature (°C)
U	Convective heat transfer coefficient ($\mathrm{Wm^{-2}K^{-1}}$)
U_{sl}	Liquid superfacial velocity (ms^{-1})
U_{sv}	Gas superfacial velocity (ms^{-1})
Greek symbols	
$\boldsymbol{\varepsilon}$	Void fraction (−)
α	Fin angle (deg.)
ρ	Density ($\mathrm{kgm^{-3}}$)
β	Fin spiral angle (deg.)
v	Specific volume ($\mathrm{m^3\,kg^{-1}}$)
Subscripts	
ht	Heat transfer
in	Inlet
l	Liquid
pd	Pressure drop
sat	Saturation
tp	Two-phase
v	Vapor
w	Wall
x	Vapor quality

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