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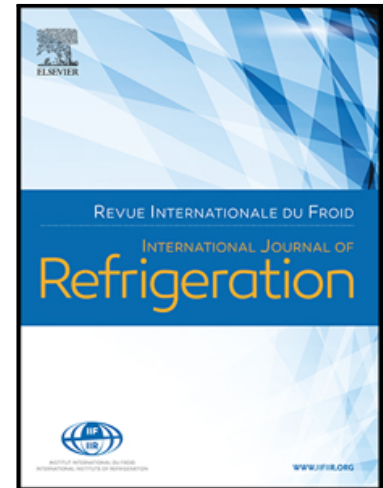
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**Highlights**

- Entropy generation of R134a flow condensation inside helical coils is studied
- The effect of geometrical parameters on entropy generation is investigated
- The effect of flow conditions on entropy generation is assessed
- An entropy generation sensitivity analysis is performed

# Entropy generation and sensitivity analysis of R134a flow condensation inside a helically coiled tube-in-tube heat exchanger

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## Abstract

Entropy generation analysis is a key to evaluating heat transfer augmentation along with pressure drop in enhanced heat exchangers. In the present study, entropy generation during R134a flow condensation inside a helically coiled tube-in-tube heat exchanger is investigated. In that regard, heat transfer and pressure drop contributions to entropy generation are assessed and compared for such condensers. Also, the influence of variation in geometrical parameters and flow conditions on entropy generation is studied. The findings reveal that larger tube diameters and diameter ratios, higher mass and heat fluxes and inlet vapor qualities, and lower saturation temperatures result in higher entropy generation. Additionally, a sensitivity analysis is conducted to analyze such condensers' sensitivity to changes in geometrical and flow conditions from an entropy generation perspective. The analysis results indicate that at high mass velocities ( $G \geq 300 \text{ kg.m}^{-2}.\text{s}^{-1}$ ) and inlet vapor qualities ( $x \geq 0.5$ ) and low saturation temperatures ( $T_{sat} \leq 46 \text{ }^{\circ}\text{C}$ ), coils with larger tube diameters are of higher sensitivity to alterations in aforementioned flow conditions, while coils with different coil diameters are of nearly the same sensitivity at both high and low ranges of flow parameters. On the contrary, when heat flux changes, coils with larger coil diameters are seen to be more sensitive to, while coils with different tube diameters show virtually the same level of sensitivity.

**Keywords:** Flow Condensation; Helically Coiled Tube; Enhanced Heat Transfer; Entropy Generation, Sensitivity Analysis

## Nomenclature

|                            |                                                                                               | Greek symbols |                                                  |
|----------------------------|-----------------------------------------------------------------------------------------------|---------------|--------------------------------------------------|
| A                          | Cross sectional area ( $\text{m}^2$ )                                                         |               |                                                  |
| L                          | Length (m)                                                                                    | $\rho$        | Density ( $\text{kg. m}^3$ )                     |
| $D_i$                      | Tube diameter (mm)                                                                            | $v$           | Specific volume ( $\text{m}^3. \text{kg}^{-1}$ ) |
| $D_c$                      | Coil diameter (mm)                                                                            | $\mu$         | Dynamic viscosity ( $\text{Ns. m}^{-2}$ )        |
| $D_r$                      | Diameter ratio                                                                                |               |                                                  |
| P                          | Perimeter (m)                                                                                 | Subscripts    |                                                  |
| dz                         | Element discretization (mm)                                                                   | ht            | Heat transfer                                    |
| G                          | Mass velocity/Specific momentum ( $\text{kg. m}^{-2}. \text{s}^{-1}$ )                        | pd            | Pressure drop                                    |
| $\dot{m}$                  | Mass flow rate ( $\text{kg. s}^{-1}$ )                                                        | in            | Inlet                                            |
| T                          | Temperature ( $^{\circ}\text{C}$ )                                                            | l             | Liquid                                           |
| x                          | Vapor quality                                                                                 | sat           | Saturation                                       |
| p                          | Pressure (pa)                                                                                 | tp            | Two-phase                                        |
| q                          | Heat flux ( $\text{W. m}^{-2}$ )                                                              | v             | Vapor                                            |
| $\dot{Q}$                  | Heat rate (W)                                                                                 | w             | Wall                                             |
| U                          | Convective heat transfer coefficient ( $\text{W. m}^{-2}. \text{K}^{-1}$ )                    |               |                                                  |
| h                          | Specific enthalpy ( $\text{J. kg}^{-1}$ )                                                     |               |                                                  |
| k                          | Thermal conductivity ( $\text{W. m}^{-1}\text{K}^{-1}$ )                                      |               |                                                  |
| $c_p$                      | Heat capacity ( $\text{kJ. kg}^{-1}\text{K}^{-1}$ )                                           |               |                                                  |
| s                          | Specific entropy ( $\text{J. K}^{-1}$ )                                                       |               |                                                  |
| $\dot{S}'_{\text{gen}}$    | Total entropy generation per unit length ( $\text{W. m}^{-1}. \text{K}^{-1}$ )                |               |                                                  |
| $\dot{S}'_{\text{gen-ht}}$ | Entropy generation per unit length due to heat transfer ( $\text{W. m}^{-1}. \text{K}^{-1}$ ) |               |                                                  |
| $\dot{S}'_{\text{gen-pd}}$ | Entropy generation per unit length due to pressure drop ( $\text{W. m}^{-1}. \text{K}^{-1}$ ) |               |                                                  |

## 1. Introduction

Flow condensation in curved channels such as helical coils, a common passive heat transfer enhancement technique (HTET), has been an engrossing field of study among researchers. In essence, in such channels, flow undergoes some phenomena like radial pressure gradient, centrifugal effects, secondary flow, and higher turbulence, enormously contributing to an augmentation in phase-change heat transfer [1–4]. Hence, these coils are widely applied in condensers, evaporators, refrigerating systems, air-conditioning systems, and food and chemical industries.

Heat transfer and pressure drop characteristics of refrigerants flow condensation in helical coils have been studied by countless researchers both experimentally and numerically [5–7]. The results of these studies show that flow condensation heat transfer coefficient (HTC) is higher for helical coils than plain straight tubes under the same flow conditions. But, such an enhancement in heat transfer for helical coils occurs at the expense of a considerable increase in pressure drop as a penalty. On the other hand, helical coils are widely applied in refrigeration cycles, where condensation commonly takes place. Thus, the significant increase in the hydraulic losses (pressure drop) can cause reductions in the energy and exergy efficiencies of these cycles as well [8,9]. In fact, to compensate the higher pressure drop caused by helical coils, refrigeration cycles utilize higher pumping energy (electrical power). The higher energy consumption, in turn, can increase the cooling production costs of the refrigeration cycles [10,11]. Therefore, identifying favorable flow and geometrical conditions at which helical coils are of lower exergy destruction and higher energy efficiency can significantly contribute to improving the energy, exergy, and economic performances of refrigeration cycles. To determine such conditions, a practical and reliable method like entropy generation analysis by which heat transfer enhancement can be assessed alongside pressure drop must be employed. This study aims to introduce a method based on entropy generation analysis by which the favorable conditions for enhanced condensers and subsequently, refrigeration cycles using such condensers like helical coils, can be identified. The following paragraphs review the literature for entropy generation analysis in two-phase flows (boiling and condensation).

Many researchers have either developed new or employed existing mathematical models to analyze entropy generation in two-phase flow problems (both flow boiling and condensation). A one-dimensional model ignoring the pressure drop term was developed by Collado [12] to investigate entropy generation of sub-cooled flow boiling. Eskin and Deniz [13] used entropy

generation analysis to study the effects of design parameters on entropy generation during R134a flow boiling in straight tubes. Two design parameters, including tube diameter and saturation temperature, were used to calculate irreversibilities due to heat transfer and pressure drop by this study. A mathematical model was developed and introduced by Revellin et al. [14] to analyze local entropy generation during diabatic saturated two-phase flow of a pure fluid in both plain and enhanced tubes. The findings revealed that from an entropy generation viewpoint, the enhanced tube is favorable at low mass velocities. In another study, Revellin and Bonjour [15] utilized entropy generation analysis as a criterion together with the model developed in the previous study to optimize diabatic two-phase flows of R134a pure refrigerant and a refrigerant-oil mixture. It was found that an increase in oil concentration leads to a higher entropy generation. Abdous et al. [9,16,17] and Holagh et al. [18] employed the models developed by Revellin to study entropy generation during R134a flow boiling inside helical coils, micro-fin and twisted-tape tubes as common passive HTETs. They determined the flow and geometrical conditions at which enhanced tubes are of lower entropy generation compared to plain tubes.

Some studies have studied entropy generation during condensation. Adeyinka and Naterer [19] developed a model to analyze and minimize entropy generation for laminar film condensation on an isothermal plate. The optimization results revealed that entropy generation analysis can be considered as a beneficial criterion when thermally designing two-phase flow systems. Two expressions were derived by Li and Yang [20] to calculate and minimize entropy generation during free convection film-wise condensation on an elliptical cylinder. It was discovered that although the local heat transfer coefficient improves with increasing ellipticity, the entropy generation increases. Dung and Yang [21] applied the second law of thermodynamics to optimize free convection film-wise condensation of saturated vapor on a horizontal tube. The results indicated that the local entropy generation rate due to heat transfer changes as the square of Nusselt number, while the local entropy generation due to gravity-driven film flow friction has a direct relation with Brinkman number. The same approach was adopted to optimize saturated vapor film condensation on an isothermal sphere by Tzeng and Yang [22]. They found that entropy generation increases as Brinkman and Rayleigh numbers go up. Esfahani et al. [23] analyzed entropy generation number and the influence of geometrical parameters, and Brinkman and Reynolds numbers on irreversibilities during flow condensation on a horizontally positioned isothermal elliptical tube. It was deduced that the ellipticity significantly affects the entropy generation. Ye and Lee [24] proposed a model to optimally design a refrigerant circuit for a fin-and-tube

condenser based on entropy generation minimization. The resulting circuit was found to be superior in terms of heat transfer performance and entropy generation compared to plain refrigerant circuitries. Entropy generation as well as exergy loss of Nano-refrigerant flow condensation inside a horizontal tube were experimentally studied by Sheikholeslami et al. [25]. They examined the impact of nanoparticles concentration, mass velocity, and vapor quality on entropy generation and discovered that an increase in nanoparticles concentration results in higher frictional pressure drops and entropy generation. Utilizing two-phase Eulerian-Eulerian model together with SST  $k-\omega$  turbulence model, Vatanmakan et al. [26] numerically analyzed the impact of volumetric heating on entropy generation during steam condensation in the cascade of turbine blades. It was concluded that the entropy generation declines as the volumetric heating rises. The same phenomenon was numerically studied by Ding et al. [27] to assess the influence of roughness on entropy generation. The results indicated that entropy generation rises as the blade's roughness increases. In a recent study, Holagh et al. [28] compared entropy generation between helical coils and straight tubes for R134a flow condensation under the same flow conditions. Using entropy generation number, the authors managed to identify geometrical configurations and flow conditions at which helical coils are of superior performance compared to straight tubes.

As the literature review shows, refrigerant flow condensation inside helically coiled tube-in-tube heat exchangers has been inadequately studied from an entropy generation perspective. Thus, there is no clear understanding of thermal-hydraulic losses occurring inside such an enhanced condenser under different flow and geometrical conditions. Hence, to fill this gap, this study aims at determining the influence of flow and geometrical parameters on heat transfer and pressure drop contributions (thermal-hydraulic losses) to entropy generation and total entropy generation during R134a flow condensation inside a helically coiled tube-in-tube heat exchanger. Considered flow and geometrical parameters are tube diameter in the refrigerant side ( $D_i$ ), diameter ratio ( $D_r$ ) which is defined as coil diameter ( $D_c$ ) ratio to tube diameter, mass velocity ( $G$ ), vapor quality ( $x$ ), saturation temperature ( $T_{sat}$ ), and wall heat flux ( $q$ ). In addition, a sensitivity analysis is carried out to analyze and compare the condenser's sensitivity to variations in flow conditions at different geometrical configurations from the entropy generation point of view. Note that unlike the current study, the study conducted by Holagh et al. [28] neither investigates the impact of diameter ratio on entropy generation nor performs a sensitivity analysis.

## 2. Modeling

### 2.1. Mathematical model

Mathematical modeling is the key to the comprehension of a wide variety of thermo-fluid phenomenon [29]. Current section presents a mathematical model upon which entropy generation analysis is conducted. The model was first introduced by Revellin et al. for saturated two-phase flows [14] and then developed for refrigerant and refrigerant-oil mixture flow boiling in enhanced tubes [15]. Abdous et al. [9,16,17] and Holagh et al. [18,28] applied this model to simulate and analyze entropy generation during refrigerant flow boiling and condensation inside helical coils, and micro-fin and twisted-tape tubes. In this study, this model is adjusted to calculate and analyze entropy generation during R134a flow condensation in a helically coiled tube-in-tube heat exchanger. To have simplified and reliable simulations, below mentioned assumptions are considered.

- The system operates under steady-state conditions.
- Entropy generation is computed based on the second law of thermodynamics.
- The R134a two-phase flow is assumed to be saturated, where the temperatures of liquid and vapor phases are the same and equal to the saturation temperature ( $T_{\text{sat}} = T_v = T_l$ ).
- Pressure differences of the liquid and vapor phases are assumed to be the same ( $dp_v = dp_l$ ).
- $dh$  is assumed to be equal to  $Tds + vdp$  for each phase.
- The total pressure drop is divided into three parts, including frictional, accelerational, and gravitational terms.
- Variations in kinetic and potential energy are supposed to be insignificant.
- HTC and pressure drop are calculated using the two-phase flow Nusselt number ( $Nu_{\text{tp}}$ ) and multiplier ( $\phi_l^2$ ) correlations and relations proposed by Wongwises et al. [6].
- All the simulations are conducted at flow and geometrical conditions aligning with those of Wongwises et al. [6] and also other researchers [2,5,7,30–33] who has acknowledged the validity of the employed correlations and equations at wider ranges of flow and geometrical parameters. That is, all the simulations are conducted within the ranges at which the correlations are applicable and valid.

- The grid considered for the helical coil is one-dimensional. Since the simulations are thermodynamic-based and intend to compute entropy generation inside the coil, the application of such a grid is appropriate and justifiable.

When passing through a tube with wall temperature lower than the saturation temperature, the heat, transferred through the wall, leads to the formation of liquid droplets on the tube's inner wall for a saturated two-phase flow. Under such conditions, for a control volume of length  $dz$ , writing entropy generation in unit length ( $\dot{S}'_{gen}$ ) based on the second law of thermodynamics gives Eq. (1a) which can be rewritten as Eq. (1b) utilizing Eq. (2) [15].

$$\dot{S}'_{gen}dz = d[\dot{m}_v s_v + \dot{m}_l s_l] - \delta\dot{Q}/T_w \quad (1a)$$

$$\dot{S}'_{gen}dz = \dot{m}d[xs_v + (1-x)s_l] - \delta\dot{Q}/T_w \quad (1b)$$

$$x = \dot{m}_v/(\dot{m}_v + \dot{m}_l) = \dot{m}_v/\dot{m} \quad (2)$$

where  $\dot{m}$ ,  $x$ ,  $s$ ,  $T_w$ , and  $\delta\dot{Q}$  denote mass flow rate, vapor quality, entropy, wall temperature, and the heat transfer rate to the control volume, respectively. Besides, subscripts v, l, and w represent vapor, liquid, and wall, correspondingly.

Regarding the fact that  $s_{lv} = s_v - s_l$ , Eq. (1b), used for computing entropy generation, is simplified to Eq. (3b).

$$\dot{S}'_{gen}dz = \dot{m}[s_v dx + xds_v - s_l dx + (1-x)ds_l] - \delta\dot{Q}/T_w \quad (3a)$$

$$\dot{S}'_{gen}dz = \dot{m}[s_{lv}dx + xds_v + (1-x)ds_l] - \delta\dot{Q}/T_w \quad (3b)$$

Eq. (4) defines  $s_{lv}$  used in Eq. (3b) as follows [9]:

$$s_{lv} = h_{lv}/T_{sat} \quad (4)$$

On the other hand,

$$dh_v = T_v ds_v + v_v dp_v \quad (5)$$

$$dh_l = T_l ds_l + v_l dp_l \quad (6)$$

Substituting the values of  $s_{lv}$ ,  $ds_v$ , and  $ds_l$  from Eqs. (4)-(6) into Eq. (3b) and assuming  $T_{sat} = T_v = T_l$  and  $dp = dp_v = dp_l$  [14,15], Eq. (3b) can be rewritten as Eq. (7a), for which the simplified form is Eq. (7b).

$$\dot{S}'_{\text{gen}} dz = \dot{m} \left[ \frac{h_{lv}}{T_{\text{sat}}} dx + x \frac{(dh_v - v_v dp)}{T_{\text{sat}}} + (1-x) \frac{(dh_l - v_l dp)}{T_{\text{sat}}} \right] - \frac{\delta \dot{Q}}{T_w} \quad (7a)$$

$$\dot{S}'_{\text{gen}} dz = \frac{\dot{m}}{T_{\text{sat}}} [h_{lv} dx + x dh_v + (1-x) dh_l] - \frac{\delta \dot{Q}}{T_w} - \frac{\dot{m} [x v_v + (1-x) v_l]}{T_{\text{sat}}} dp \quad (7b)$$

Consequently,  $\dot{S}'_{\text{gen}}$  is written in the form of Eq. (8), where  $dh_{tp}$  and  $v_{tp}$  are calculated by Eqs. (9) and (10), individually [15].

$$\dot{S}'_{\text{gen}} dz = \frac{\dot{m}}{T_{\text{sat}}} dh_{tp} - \frac{\delta \dot{Q}}{T_w} - \frac{\dot{m} v_{tp} dp}{T_{\text{sat}}} \quad (8)$$

$$dh_{tp} = h_{lv} dx + x dh_v + (1-x) dh_l \quad (9)$$

$$v_{tp} = x v_v + (1-x) v_l \quad (10)$$

Employing the first law of thermodynamics and considering kinetic energy and gravitational terms negligible,  $\delta \dot{Q}$  can be achieved using Eq. (11) [15].

$$d\dot{Q} = \dot{m} dh_{tp} \quad (11)$$

Eventually, by substituting Eq. (11) into Eq. (8) and performing further simplifications, entropy generation per unit length can be expressed as Eq. (12).

$$\dot{S}'_{\text{gen}} = \dot{m} \left[ \frac{dh_{tp}}{dz} \left( \frac{1}{T_{\text{sat}}} - \frac{1}{T_w} \right) - \frac{v_{tp}}{T_{\text{sat}}} \frac{dp}{dz} \right] \quad (12)$$

This equation can be divided into two parts; the first, written as Eq. (13), indicates entropy generation due to heat transfer, while the second, written as Eq. (14), refers to entropy generation due to pressure drop.

$$\dot{S}'_{\text{gen-ht}} = \dot{m} \left[ \frac{dh_{tp}}{dz} \left( \frac{1}{T_{\text{sat}}} - \frac{1}{T_w} \right) \right] \quad (13)$$

$$\dot{S}'_{\text{gen-pd}} = \frac{\dot{m} v_{tp}}{T_{\text{sat}}} \left( -\frac{dp}{dz} \right) \quad (14)$$

The term,  $\frac{dh_{tp}}{dz}$ , in Eq. (13), can be replaced with Eq. (15) utilizing Eq. (16), where  $q$ ,  $U$ , and  $P$  respectively depict wall heat flux, heat transfer coefficient, and tube perimeter.

$$q = U(T_w - T_{\text{sat}}) = \delta \dot{Q} / P dz = \dot{m} dh_{\text{tp}} / P dz \quad (15)$$

$$\frac{dh_{\text{tp}}}{dz} = \frac{pq}{\dot{m}} \quad (16)$$

Thus, the simplified form of Eq. (13) can be written as Eq. (17), which represents the heat transfer contribution to entropy generation regarding  $T_w - T_{\text{sat}} = \frac{q}{U}$ . In Eq. (17), wall temperature can be computed utilizing heat transfer coefficient and wall heat flux as known values.

$$\dot{S}'_{\text{gen-ht}} = \frac{q^2 P}{UT_w T_{\text{sat}}} \quad (17)$$

On the other hand,  $\frac{dp}{dz}$  in Eq. (14), which presents pressure drop contribution to entropy generation, reveals total pressure drop per unit length during a condensing two-phase flow, which can be written as [6]:

$$\left(\frac{dp}{dz}\right) = \left(\frac{dp_F}{dz}\right)_{\text{tp}} + \left(\frac{dp_G}{dz}\right)_{\text{tp}} + \left(\frac{dp_A}{dz}\right)_{\text{tp}} \quad (18)$$

The terms on the right-hand side of Eq. (18) demonstrate frictional  $\left(\left(\frac{dp_F}{dz}\right)_{\text{tp}}\right)$ , gravitational  $\left(\left(\frac{dp_G}{dz}\right)_{\text{tp}}\right)$ , and accelerational  $\left(\left(\frac{dp_A}{dz}\right)_{\text{tp}}\right)$  components of the total pressure drop.

The gravitational term is calculated using Eq. (19), in which  $\theta$  and  $\alpha$  correspondingly stand for helix angle of the coil and void fraction determined by Eq. (20) [6].

$$\left(\frac{dp_G}{dz}\right)_{\text{tp}} = [\alpha \rho_v + (1 - \alpha) \rho_l] g \sin \theta \quad (19)$$

$$\alpha = \frac{1}{1 + S \left(\frac{1-x}{x}\right) \frac{\rho_v}{\rho_l}} \quad (20)$$

In Eq. (20),  $S$  represents the slip ratio obtained by a correlation proposed by Chisholm [34]

$$S = \left[1 - x \left(1 - \frac{\rho_l}{\rho_v}\right)\right]^{0.5} \quad (21)$$

where,  $\rho_l$  and  $\rho_v$  denote the densities of the liquid and vapor phases.

The accelerational term is also attained from Eq. (22), in which  $G$  is the refrigerant mass velocity.

$$\left(\frac{dp_A}{dz}\right)_{\text{tp}} = G^2 \frac{d}{dz} \left[ \frac{x^2}{\alpha \rho_v} + \frac{(1-x)^2}{(1-\alpha) \rho_l} \right] \quad (22)$$

The frictional pressure drop for a gas-liquid two-phase flow is determined by [6]

$$\phi_1^2 = \left( \frac{dp_F}{dz} \right)_{tp} / \left( \frac{dp_F}{dz} \right)_1 \quad (23)$$

where,  $\phi_1^2$  is the two-phase frictional multiplier.

$\left( \frac{dp_F}{dz} \right)_1$ , representing the single-phase liquid pressure drop, can be calculated from Eq. (24) [6].

$$\left( \frac{dp_F}{dz} \right)_1 = \frac{2f_1 \rho_1 U_1^2}{D_i} \quad (24)$$

In Eq. (24),  $f_1$  stands for friction factor and is determined by Eq. (25) proposed by Ito for a single-phase flow inside a curved tube [35].

$$f_1 \left( \frac{D_c}{D_i} \right)^{0.5} = 0.00725 + 0.076 \left[ Re_1 \left( \frac{D_c}{D_i} \right)^{-2} \right]^{-0.25} \quad (25)$$

Martinelli parameter,  $X^2$  is defined as

$$X^2 = \left( \frac{dp_F}{dz} \right)_1 / \left( \frac{dp_F}{dz} \right)_v \quad (26)$$

In the case that the two abovementioned pressure gradients are based on turbulent flow, then

$$X = X_{tt} \approx \left( \frac{1-x}{x} \right)^{0.9} \left( \frac{\rho_v}{\rho_l} \right)^{0.5} \left( \frac{\mu_l}{\mu_v} \right)^{0.1} \quad (27)$$

On the other hand, the two-phase frictional multiplier is computed by Eq. (28) for smooth circular tubes in the form of the Lockhart-Martinelli correlation

$$\phi_l^2 = 1 + \frac{C}{X_{tt}} + \frac{1}{X_{tt}^2} \quad (28)$$

where,  $C$  is a parameter indicating the two-phase flow condition and its value can be variable between 5 and 20 for different flow conditions [35]. The insertion of Eqs. (24) and (28) into Eq. (23) results in the calculation of frictional two-phase pressure drop.

To analyze entropy generation during R134a flow condensation in a helically coiled tube-in-tube heat exchanger with the help of this model, two experimental correlations are required. One for Nusselt number which is used in the determination of heat transfer coefficient (Eq. (17)) and the other for  $\phi_l^2$  that is utilized in the calculation of frictional pressure drop (Eq. (23)).

Wongwises et al. [6] experimentally studied heat transfer and pressure drop characteristics of R134a flow condensation in a helically-coiled tube-in-tube heat exchanger. According to this study, two-phase Nusselt number,  $Nu_{tp}$ , (Eq. (29)) is correlated between Martinelli parameter

( $X_{tt}$ ), equivalent Dean number ( $De_{Eq}$ ), Prandtl number ( $Pr_l$ ), reduced pressure ( $p_r$ ), and Boiling number ( $Bo$ ) which can be individually calculated using Eqs. (27) and (29)-(33).

$$Nu_{tp} = 0.1352 De_{Eq}^{0.7654} Pr_l^{0.8144} X_{tt}^{0.0432} p_r^{-0.3256} (Bo \times 10^4)^{0.112} \quad (29)$$

$$De_{Eq} = \left[ Re_l + Re_v \left( \frac{\mu_v}{\mu_l} \right) \left( \frac{\rho_l}{\rho_v} \right)^{0.5} \right] \left( \frac{D_i}{D_c} \right)^{0.5} \quad (30)$$

$$Pr_l = \frac{C p_l \mu_l}{k_l} \quad (31)$$

$$p_r = \frac{P_{sat}}{P_{crit}} \quad (32)$$

$$Bo = \frac{q''}{G h_{lv}} \quad (33)$$

where  $Re_l$  and  $Re_v$  indicate liquid and vapor phases Reynolds numbers determined by Eqs. (34) and (35), respectively.

$$Re_l = \frac{G(1-x)D_i}{\mu_l} \quad (34)$$

$$Re_v = \frac{GxD_i}{\mu_v} \quad (35)$$

As for two-phase frictional multiplier ( $\phi_1^2$ ), Wongwises et al. [6] proposed Eq. (36).

$$\phi_1^2 = 1 + \frac{5.569}{X_{tt}^{1.496}} + \frac{1}{X_{tt}^2} \quad (36)$$

Since these correlations have been derived for R134a flow condensation inside a helically coiled tube-in-tube heat exchanger and are in good agreements with not only the experimental results obtained by Wongwises et al. [6], but also findings of other researchers [2,5,7,30–33] for various flow and geometrical conditions, they are perfectly suitable to be applied in the aforementioned model to analyze entropy generation in helical coils.

## 2.2. Physical model

The schematic of a helically coiled tube-in-tube heat exchanger and important dimensions encompassing refrigerant side tube diameter ( $D_i$ ), and coil diameter ( $D_c$ ) is illustrated in Fig. 1. Saturated R134a flows through the inner tube, while cold water passes through the annular part.

## 2.3. Method of solution

Considering the above-expounded mathematical model, a computational code has been developed in MATLAB to simulate entropy generation inside the helical coil. To perform the simulations, first, the coil's geometry is divided into a number of cells to an extent that fulfills the grid independency requirement. Once the number of cells is specified, for each

cell, the mass, momentum, and energy equations are coupled in a loop to attain the local pressure and local wall and liquid temperatures. Having calculated these parameters, the refrigerant's thermo-physical properties are computed and applied in the introduced mathematical model to obtain the amounts of the generated entropy and relevant parameters within each cell. By integrating over the entire geometry, the total entropy generation is calculated and is then compared with the R134a entropy values reported in the thermodynamic charts to verify the simulation results. Once the difference between the total entropy generation calculated by the model and that obtained from the thermodynamic charts becomes lower than  $10^{-5}$ , the calculations are supposed to be sufficiently accurate and therefore, the simulations stop.

## 2.4. Validation

The helical coil is divided into 4000 one-dimensional control volume elements (cells), satisfying grid independency (as shown in Fig. 3). In fact, the grid consists of 4000 cells distributed and placed in 0.625 mm distances along a helical curve passing through the center of the coil. Figs. 2a and 2b reveal a zoomed view of the devised grid and a schematic view of the cells, respectively.

Fig. 3 investigates the validity and grid independency of the entropy generation model expounded in section 2.1. To validate the simulations, the results achieved for the entropy generation via the mathematical model were compared with the values reported in thermodynamic charts for entropy generation that were found based on the flow's inlet and outlet conditions. The inlet conditions for the helical coil are always known. On the other hand, the boundary conditions (wall heat flux) is known; thus, the outlet conditions can be achieved applying the first law of thermodynamics. Using the inlet and outlet conditions and referring to thermodynamic charts provided in [36], the entropies of the input and output flow are attained. The validation has been conducted at three different values of specific momentum (240, 520, and 720  $\text{kg.m}^{-2}\text{s}^{-1}$ ) and saturation temperature, inlet quality, and wall heat flux of 40 °C, 0.8, and 5000  $\text{W.m}^{-2}$ , respectively.

First, the number of cells for the tube was set to be 1000 which resulted in big errors as shown in Fig. 3. Then, to reduce the errors, the number of cells was increased until the errors dropped below  $10^{-5}$ . As can be seen, the entropy generation reaches a fixed value equal to that obtained from thermodynamics charts after the point corresponding to 4000 cells. This suggests that 4000 cells are sufficient for the considered geometry and the results are valid.

Note that the proposed model is limited when some of the considered assumptions are violated. Hence, the model cannot be applied to simulate entropy generation for (a) unsteady operating conditions, (b) unsaturated flows, and (c) ranges of flow and geometrical parameters at which the correlations and relations proposed by Wongwises et al. [6] are not valid. Tables 1 and 2 represents the ranges of flow and geometrical parameters at which the applied correlations are in good agreement with the experimental results of Wongwises et al. [6], respectively. On the other hand, the findings of other researchers [2,5,7,30–33] have corroborated that these correlations are applicable and accurate within wider ranges than what represented in the these tables. In the present study, the ranges of flow and geometrical parameters within which entropy generation is analyzed are shown in Tables 3 and 4, correspondingly. Regarding Tables 1 and 2 (Wongwises et al. [6]), Tables 3 and 4 (current study), and the wide ranges of geometrical and flow parameters at which these correlations have been proven to be accurate [2,5,7,30–33], it is crystal clear that current simulations are conducted within the ranges in which these correlations are valid.

### 3. Results and discussions

#### 3.1. Influence of geometrical parameters at constant flow conditions

##### 3.1.1. Tube diameter ( $D_i$ )

Fig. 4 illustrates the effect of the tube diameter on entropy generation at constant flow conditions provided in Table 5 and constant coil diameter equal to 305 mm. Looking at Fig. 4, one can see that as the tube diameter increases from 5 mm to 25 mm, both heat transfer and pressure drop contributions to entropy generation rise from roughly  $4.2 \times 10^{-3}$  and  $2.8 \times 10^{-2}$   $\text{W.K}^{-1}$  to  $1.5 \times 10^{-2}$  and  $3.7 \times 10^{-1}$   $\text{W.K}^{-1}$ , respectively. In essence, with an increase in  $D_i$ , the tube's cross-sectional area becomes larger. On the other hand, since the mass velocity ( $G = \frac{\dot{m}}{A}$ ) is kept constant, with increasing the cross-sectional area ( $A$ ), the R134a mass flow rate ( $\dot{m}$ ) inside the coil increases. As a result, the difference between the wall temperature and saturation temperature enhances due to the larger cross-sectional area with higher mass flow rate which mean lower HTC. This, in turn, leads to an enhancement in heat transfer contribution to entropy generation. The existing heat transfer phenomenon aims at increasing the mixture density by condensing more vapor, while higher mass flow rate can reduce the mixture density. On the other hand, at a constant wall heat flux, the amount of the heat transferred from the wall to flow is proportional to the tube diameter, but mass flow rate is

proportional to the second power of the tube diameter. Therefore, the influence of the increase in mass flow rate on two-phase mixture density overcomes that of the heat transfer. Thereby, the two-phase mixture density declines. Consequently, with decreasing the mixture density, the two-phase mixture flow velocity increases at constant mass velocity. Such an increase in the two-phase mixture flow velocity results in a higher pressure drop inside the tube, leading to an increase in the pressure drop contribution to entropy generation. Therefore, the total entropy generation is observed to enhance by  $0.35 \text{ W.K}^{-1}$  with an increase in the tube diameter.

### 3.1.2. Diameter ratio ( $D_r$ )

The impact of an increase in diameter ratio (the ratio of coil diameter to tube diameter) on entropy generation at constant flow conditions presented in Table 5 and constant tube diameter equal to 8.3 mm is exhibited in Fig. 5. If the tube diameter remains unchanged and the coil diameter increases, the diameter ratio becomes larger. Under such conditions, as represented in Fig. 5, an increase in diameter ratio (from 20 to 100) leads to an enhancement in the heat transfer contribution by  $0.43 \times 10^{-2} \text{ W.K}^{-1}$  and a reduction in the pressure drop contribution by  $0.32 \times 10^{-2} \text{ W.K}^{-1}$ . In reality, with an increase in the diameter ratio (ensuing from the increase in coil diameter at a constant tube diameter), the coil becomes closer to straight tubes from geometrical standpoint. This closeness, in turn, causes significant reductions in both heat transfer rate and pressure drop amount for R134a flow condensation inside the tube [5]. These are because as the tube becomes straighter, some phenomena occurring inside the curved tubes like radial pressure gradient, centrifugal effects, secondary flow, Dean vortices, etc., that are the root causes behind higher heat transfer rate and pressure drop inside such geometries, diminish [3]. Thus, heat transfer contribution rises, whereas pressure drop contribution reduces by increasing diameter ratio. Nonetheless, since the increase in heat transfer contribution overcomes the reduction in pressure drop contribution, the total entropy generation goes up by  $0.11 \times 10^{-2} \text{ W.K}^{-1}$  as shown in Fig. 5.

## 3.2. Influence of flow conditions at constant geometrical parameters

The behavior of entropy generation versus variations in flow conditions under constant geometrical parameters (Table 6) are studied in this subsection. It is worth noting that in each case, one flow parameter varies, while others are kept constant with Table 5 as the input.

### 3.2.1. Mass velocity

Figs. 6 represents the influence of mass velocity, varying from 200 to 600  $\text{kg.m}^{-2}.\text{s}^{-1}$ , on entropy generation. As can be obviously seen in Fig. 6, total entropy generation and pressure drop contribution remarkably increase from  $1.73 \times 10^{-2}$  and  $6.47 \times 10^{-3} \text{ W.K}^{-1}$  to  $2.23 \times 10^{-1}$  and  $2.18 \times 10^{-1} \text{ W.K}^{-1}$  as the mass velocity goes up, correspondingly. In stark contrast, heat transfer contribution declines from  $1.08 \times 10^{-2}$  to  $4.61 \times 10^{-3} \text{ W.K}^{-1}$ . In fact, with increasing mass velocity under the same geometrical parameters, flow velocity inside the coil rises, bringing about an improvement in heat transfer rate due to higher turbulence and also an enhancement in pressure drop as a result of the increase in frictional losses. Consequently, heat transfer contribution reduces, whereas pressure drop contribution grows. On the other hand, since the increase in pressure drop contribution outweighs the reduction in heat transfer contribution, the total entropy generation remarkably enhances by  $20.5 \times 10^{-2} \text{ W.K}^{-1}$ .

### 3.2.2. Saturation temperature

Alterations in entropy generation against saturation temperature, rising from 30 to 50  $^{\circ}\text{C}$ , is plotted in Fig. 7. From the information gathered in Fig. 7, it is evident that as the saturation temperature goes up, the total entropy generation and pressure drop contribution significantly decline from  $1.12 \times 10^{-1}$  and  $1.06 \times 10^{-1} \text{ W.K}^{-1}$  to  $3.92 \times 10^{-2}$  and  $3.21 \times 10^{-2} \text{ W.K}^{-1}$ , while the heat transfer contribution slightly enhances from  $5.47 \times 10^{-3}$  to  $7.08 \times 10^{-3} \text{ W.K}^{-1}$ . In essence, an increase in saturation temperature means an enhancement in saturation pressure and thus, higher two-phase mixture densities. On the other hand, for a constant mass velocity, the higher the mixture density is, the lower the mixture velocity is. Therefore, with an increase in saturation temperature that entails lower flow velocities, pressure drop, which is directly proportional to flow velocity, and its contribution to entropy generation reduce. Also, unlike flow boiling, heat transfer coefficient drops as saturation temperature rises during flow condensation, yielding to higher heat losses and thereby generating entropy attributable to heat transfer, as depicted in Fig. 7. Besides, since the change in pressure drop contribution is greater than that of the heat transfer, the total entropy generation experiences a decreasing behavior.

### 3.2.3. Inlet vapor quality

The entropy generation behavior against an increase in inlet vapor quality, varying from 0.4 to 0.9, is shown in Fig. 8. Looking at Fig. 8, it is crystal clear that with increasing inlet vapor quality, total entropy generation and pressure drop contribution markedly escalate from  $2.10 \times 10^{-2}$  and  $1.20 \times 10^{-2} \text{ W.K}^{-1}$  to  $8.01 \times 10^{-2}$  and  $7.41 \times 10^{-2} \text{ W.K}^{-1}$ , correspondingly, whereas heat transfer contribution insignificantly reduces from  $0.9 \times 10^{-2}$  to  $0.59 \times 10^{-2} \text{ W.K}^{-1}$ . In reality, as the mass velocity is kept unchanged, the increase in the inlet vapor quality, which brings about lower two-phase mixture densities, contributes to an escalation in flow velocity. On the other hand, the higher the flow velocity is, the higher the heat transfer rate and pressure drop are, leading to a reduction in heat transfer contribution and an enhancement in pressure drop contribution to entropy generation. Also, since the escalation in the pressure drop contribution is greater compared with the reduction in the heat transfer contribution, the total entropy generation enhances by  $5.91 \times 10^{-2} \text{ W.K}^{-1}$ .

### 3.2.4. Heat flux

The influence of changes in wall heat flux from 5000 to 10,000  $\text{W.m}^{-2}$  on entropy generation is manifested in Fig. 9. In accordance with Fig. 9, as the wall heat flux increases, heat transfer contribution grows from  $6.28 \times 10^{-3}$  to  $2.58 \times 10^{-2} \text{ W.K}^{-1}$ , while pressure drop contribution declines from  $5.76 \times 10^{-2}$  to  $4.23 \times 10^{-2} \text{ W.K}^{-1}$ . Thus, the total entropy generation goes up by  $0.5 \text{ W.K}^{-1}$ . In fact, with increasing wall heat flux, condensation process intensifies since more heat is transferred from the two-phase flow to the cooling water flowing in the annular part of the helically coiled tube-in-tube condenser. Therefore, condensation rate enhances, in turn, leading to decreasing vapor quality and increasing two-phase mixture density inside the tube. On the other hand, since the mass velocity is kept constant, the increase in the two-phase mixture density causes a reduction in the flow velocity, resulting in lower heat transfer rate and pressure drop. Consequently, heat transfer contribution rises, while pressure drop contribution declines as the wall heat flux increases.

## 3.3. Sensitivity analysis

In the current section, the sensitivity of helically coiled condensers to variations in geometrical parameters and flow conditions is analyzed from an entropy generation viewpoint by comparing the rate of change in entropy generation (RCEG) for different geometrical parameters at different flow conditions. To this end, the impact of variations in flow conditions on RCEG under different geometrical parameters is evaluated.

This analysis is conducted to measure the sensitivity of helically coiled tube-in-tube condensers to changes in flow conditions at different geometrical configurations from an entropy generation perspective. Such an analysis can give engineers and designers a clear view of the impact of flow conditions on thermal-hydraulic losses at different geometrical conditions. This, in turn, can assist engineers to appropriately select geometrical and flow parameters such that the losses become low. In addition, the results of such a sensitivity analysis can be applied for optimization purposes.

### 3.3.1. Sensitivity to mass velocity

Figs. 10a and 10b respectively reveal entropy generation for different tube and coil diameters when mass velocity increases from 200 to 600  $\text{kg.m}^{-2}.\text{s}^{-1}$ .

As demonstrated in Fig. 10a, up to mass velocity around 300  $\text{kg.m}^{-2}.\text{s}^{-1}$ , RCEG is nearly the same for all considered tube diameters. However, for mass velocities higher than that, it is seen that the coil having the largest tube diameter ( $D_i = 20 \text{ mm}$ ) is of steeper RCEG compared with two others so that for mass velocities between 500-600  $\text{kg.m}^{-2}.\text{s}^{-1}$ , the enhancement in entropy generation for this coil is virtually 3.8 times greater than that of the coil with  $D_i = 8.3 \text{ mm}$ . This means that at mass velocities higher than nearly 300  $\text{kg.m}^{-2}.\text{s}^{-1}$ , coils with larger tube diameters experience greater growth rates of entropy generation as mass velocity augments and thus, they are more sensitive to change in mass velocity. This is because at mass velocities larger than nearly 300  $\text{kg.m}^{-2}.\text{s}^{-1}$ , the increase in pressure drop and subsequently its contribution to entropy generation is sharper for coils with larger tube diameters compared to coils with smaller tube diameters when mass velocity goes up.

Fig. 10b reveals that the RCEG is approximately the same for all considered coil diameters when mass velocity changes. That is, coil diameter has an insignificant effect on the RCEG at both low and high mass velocities. Thus, from an entropy generation standpoint, coils with different coil diameters are the same in terms of sensitivity to variations in mass velocity. Nonetheless, it is observed that for mass velocities lower than 380  $\text{kg.m}^{-2}.\text{s}^{-1}$ , the coil with the largest coil diameter ( $D_c = 800 \text{ mm}$ ) experiences the highest entropy generation, whereas for mass velocities higher than that it is the coil with the lowest coil diameter ( $D_c = 100 \text{ mm}$ ) that witnesses the greatest entropy generation. The cause behind this phenomenon is the larger growth of pressure drop and consequently its contribution to entropy generation that occurs in coils with lower coil diameters at higher mass velocities.

### 3.3.2. Sensitivity to saturation temperature

The behavior of entropy generation for variable tube and coil diameters against saturation temperature, changing from 30 to 50 °C, is represented in Figs. 11a and 11b.

According to Fig. 11a, for saturation temperatures higher than roughly 46 °C, entropy generation in all tube diameters sees nearly an identical rate of change when saturation temperature varies. But, at lower saturation temperatures, the coil with the largest tube diameter has sharper RCEG so that the reduction in entropy generation experienced by this coil when saturation temperature increases from 30 to 35 °C is virtually four times greater than that of the coil with the smallest diameter. Simply put, the coils with larger tube diameters are more sensitive to variations in saturation temperature from an entropy generation perspective, particularly at low ranges of saturation temperature. This is mainly due to the fact that at saturation temperatures lower than approximately 46 °C, compared to coils with smaller tube diameters, the increase in pressure drop and consequently its contribution to entropy generation is severer in coils with larger tube diameters as the saturation temperature declines.

Regarding Fig. 11b, it is evident that at both low and high saturation temperatures, the sensitivities of the three considered coils with different coil diameters to changes in saturation temperature are almost the same from an entropy generation viewpoint since all the coils show nearly identical values of RCEG within the whole range. Nevertheless, it is seen that before the saturation temperature of 40 °C, the amount of the entropy generated in the coil with the smallest coil diameter is greater than those produced in coils with the largest and mediocre coil diameters. After the saturation temperature of 40 °C, this is the coil with the largest coil diameter that has the highest value of entropy generation. This can be attributable to the larger enhancements in pressure drop and the respective contribution to entropy generation experienced by coils with larger coil diameters at higher saturation temperatures.

### 3.3.3. Sensitivity to inlet vapor quality

Entropy generation against variation in inlet vapor quality under different values of tube and coil diameters is correspondingly plotted in Figs. 12a and 12b.

As can be seen in Fig. 12a, for three considered coils with different tube diameters, the RCEG is almost identical at inlet vapor qualities lower than virtually 0.5. Yet, for inlet vapor qualities higher than that, the coil with the largest tube diameter undergoes sharper RCEG

compared with the two others to such an extent that for such a coil, an increase in inlet vapor quality from 0.8 to 0.9 yields to an enhancement in entropy generation which is 4.2 times greater than that of the coil with the smallest diameter. In other words, from entropy generation standpoint, the sensitivity of coils with larger tube diameters to alterations in inlet vapor quality is higher compared to coils with smaller tube diameters. This can be attributable to the fact that at inlet vapor qualities larger than about 0.5, when inlet vapor quality rises, compared to the coils having smaller tube diameters, the coils with larger tube diameters experience sharper enhancements in pressure drop and its contribution to entropy generation.

Fig. 12b reveals that coils having different coil diameters show approximately the same RCEG throughout the considered range for inlet vapor quality. That is, from entropy generation viewpoint, coils with different coil diameters are the same when it comes to sensitivity to variation in inlet vapor quality. In addition, it is clear that up to saturation temperature equal to 80 °C, the coil with the largest coil diameter has the greatest amount of entropy generation since the augmentation in its pressure drop and respective contribution to entropy generation is higher compared to the two other coils when inlet vapor quality rises from 0.4 to 0.8.

#### 3.3.4. Sensitivity to heat flux

Figs. 13a and 13b, exhibit the variation of entropy generation versus wall heat flux varying from 5000 to 10000 W.m<sup>-2</sup> for coils with different tube and coil diameters, individually.

Looking at Fig. 13a, one can see that throughout the whole considered range for wall heat flux, notably at heat fluxes lower than 8000 W.m<sup>-2</sup>, coils with different tube diameters are of approximately identical values of RCEG. To put it differently, it is crystal clear that coils with different tube diameters have nearly the same level of sensitivity to variations in heat flux from entropy generation standpoint. This is because coils with different tube diameters are virtually the same in terms of the rate of change in the reductions experienced by heat transfer rate and respective contribution to entropy generation as wall heat flux varies.

With respect to Fig. 13b, the comparison of RCEG between coils with different coil diameters when wall heat flux increases indicates that coils with larger coil diameters are of higher values of RCEG, in particular at wall heat fluxes higher than nearly 8000 W.m<sup>-2</sup>. That is, the larger the coil diameter is, the more sensitive the coil is to variation in wall heat flux

from entropy generation viewpoint. For instance, an increase in wall heat flux from 9000 to 10000  $\text{W.m}^{-2}$ , leads to an augmentation in entropy generation inside the coil with coil diameter equal to 800 mm (largest diameter) which is roughly 10 times greater than that of the coil with coil diameter equal to 100 mm (smallest diameter). This is because at wall heat fluxes higher than approximately 8000  $\text{W.m}^{-2}$ , when wall heat flux augments, compared to coils with smaller coil diameters, coils with larger coil diameters face sharper reductions in heat transfer rate and consequently more rapidly increases in the respective contribution to entropy generation.

#### 4. Conclusions

In the current study, entropy generation was studied for R134a flow condensation inside a helically coiled tube-in-tube condenser at wide ranges of flow and geometrical conditions through developing a code in MATLAB software based on a mathematical entropy generation model. First, the influence of variations in geometrical and flow parameters on entropy generation and its heat transfer and pressure drop components was assessed. Then, a sensitivity analysis was conducted to measure and compare the sensitivity of the helically coiled tube-in-tube condensers with different geometrical configurations to variations in flow conditions from an entropy generation viewpoint. Future studies can extend the current study and use the entropy generation results to investigate exergy destruction in helically coiled tube-in-tube heat exchangers as well as refrigeration cycles in which such exchangers are utilized. Also, future researchers can combine the introduced mathematical model in the Modeling section (Section 2) with various optimization algorithms like genetic algorithm to optimize the performance of such heat exchangers. The main outcomes of this study are as follows.

- At constant flow conditions, total entropy generation enhances as the tube diameter and diameter ratio become larger.
- At constant geometrical conditions, total entropy generation augments as mass and wall heat fluxes and inlet vapor quality increase and saturation temperature decreases.
- When mass velocity, saturation temperature, and inlet vapor quality vary, this is the pressure drop contribution to entropy generation that plays the role of the dominant component in the determination of the behavior experienced by the total entropy generation. But, for variations in wall heat flux, the dominant component is the heat transfer contribution to entropy generation.

- From an entropy generation perspective, helically coiled tube-in-tube condensers with larger tube diameters in the refrigerant side are more sensitive to variations in mass velocity, saturation temperature, and inlet vapor quality at  $G \geq 300 \text{ kg.m}^{-2}.\text{s}^{-1}$ ,  $T_{sat} \leq 46 \text{ }^{\circ}\text{C}$ , and  $x \geq 0.5$ . Nonetheless, it is seen that coils with different tube diameters are almost of the same sensitivity to changes in wall heat flux within the whole considered range.
- Helically coiled tube-in-tube condensers with different coil diameters are of nearly the same sensitivity to alterations in mass velocity, saturation temperature, and inlet vapor quality at both high and low flow conditions. But, coils with larger coil diameters are found to be of higher sensitivity to variations in heat flux, particularly at  $q \geq 8000 \text{ W.m}^{-2}$ .

### Declaration of interests

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

### References

- [1] M.A. Abdous, S.G. Holagh, M. Shamsaiee, H. Saffari, The prediction of bubble departure and lift-off radii in vertical U-shaped channel under subcooled flow boiling based on forces balance analysis, *Int. J. Therm. Sci.* 142 (2019) 316–331. doi:10.1016/j.ijthermalsci.2019.04.021.
- [2] A. Kumar, R. Kumar, Two-phase flow condensation heat transfer characteristic of R-600a inside the horizontal smooth and dimpled helical coiled tube in shell type heat exchanger, *Int. J. Refrig.* 107 (2019) 155–164. doi:10.1016/j.ijrefrig.2019.07.017.
- [3] S.G. Holagh, M.A. Abdous, M. Shamsaiee, H. Sa, An experimental study on the influence of radial pressure gradient on bubbles dynamic behavior in subcooled flow boiling, *Therm. Sci. Eng. Prog.* 16 (2020). doi:10.1016/j.tsep.2019.100468.
- [4] S.G. Holagh, M.A. Abdous, P. Roy, M. Shamsaiee, M. Shafiee, H. Saffari, L. Vali, R. Andersson, An experimental investigation on bubbles departure characteristics during sub-cooled flow boiling in a vertical U-shaped channel

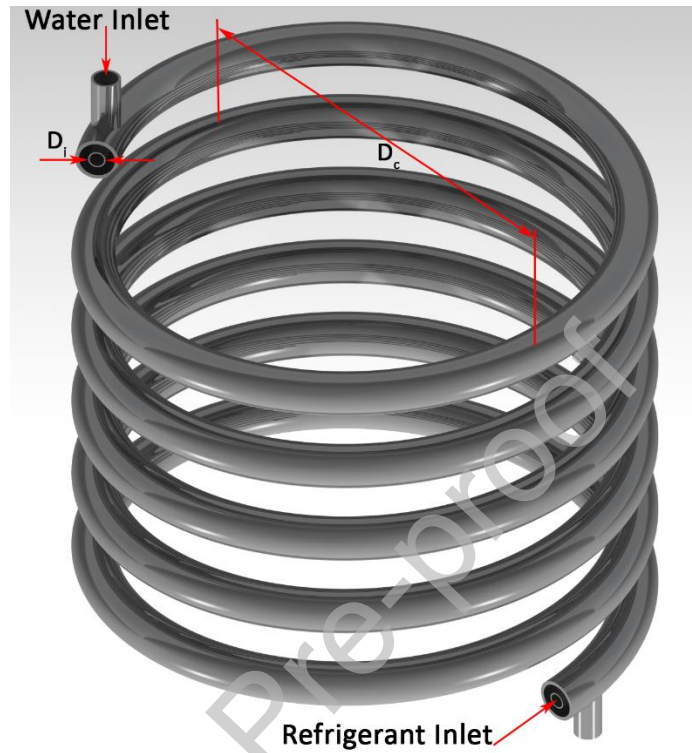
- utilizing high-speed photography, *Therm. Sci. Eng. Prog.* 22 (2021). doi:10.1016/j.tsep.2020.100828.
- [5] S. Zakeralhoseini, B. Sajadi, M. Ali, A. Behabadi, S. Azarhazin, Experimental investigation of the heat transfer coefficient and pressure drop of R1234yf during flow condensation in helically coiled tubes, *Int. J. Therm. Sci.* 157 (2020) 106516. doi:10.1016/j.ijthermalsci.2020.106516.
- [6] S. Wongwises, M. Polsongkram, Condensation heat transfer and pressure drop of HFC-134a in a helically coiled concentric tube-in-tube heat exchanger, *Int. J. Heat Mass Transf.* 49 (2006) 4386–4398. doi:10.1016/j.ijheatmasstransfer.2006.05.010.
- [7] M.E. Mosaad, M. Al-Hajeri, R. Al-Ajmi, A.M. Koliub, Heat transfer and pressure drop of R-134a condensation in a coiled double tube, *Heat Mass Transf.* 45 (2009) 1107–1115. doi:10.1007/s00231-009-0484-y.
- [8] S. Yan, H. Moria, S. Pourhedayat, M. Hashemian, S. Assadi, H. Sadighi, K. Jermisittiparsert, A critique of effectiveness concept for heat exchangers; theoretical-experimental study, *Int. J. Heat Mass Transf.* 159 (2020) 120160. doi:10.1016/j.ijheatmasstransfer.2020.120160.
- [9] M.A. Abdous, H. Saffari, H.B. Avval, M. Khoshzat, Investigation of entropy generation in a helically coiled tube in flow boiling condition under a constant heat flux, *Int. J. Refrig.* 60 (2015) 217–233. doi:10.1016/j.ijrefrig.2015.07.026.
- [10] M. Abdollahi, S. Ghazanfari Holagh, A. Chitsaz, K. Parham, Thermodynamic assessment of a novel multi-generation solid oxide fuel cell- based system for production of electrical power , cooling , fresh water , and hydrogen, *Energy Convers. Manag.* 197 (2019) 111895. doi:10.1016/j.enconman.2019.111895.
- [11] S.G. Holagh, Exergoeconomic and environmental investigation of an innovative poly-generation plant driven by a solid oxide fuel cell for production of electricity , cooling , desalinated water , and hydrogen, *Int. J. Energy Res.* (2020) 1–29. doi:10.1002/er.5626.
- [12] F.J. Collado, The entropy balance for boiling flow, *Fusion Eng. Des.* 56–57

(2001) 199–203. doi:10.1016/S0920-3796(01)00259-9.

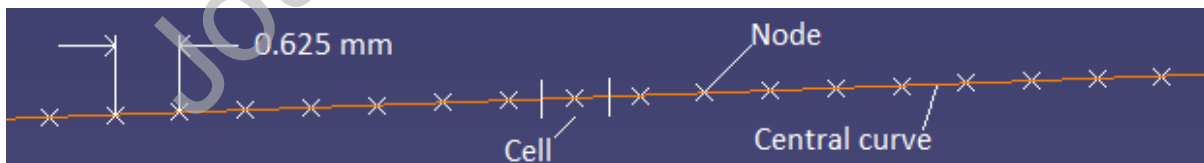
- [13] N. Eskin, E. Deniz, Influence of design parameters on two-phase pressure drop and entropy generation during evaporation of refrigerants, in: 22nd Int. Conf. Effic. Cost, Optim. Simul. Environ. Impact Energy Syst., 2009: pp. 159–168.
- [14] R. Revellin, S. Lips, S. Khandekar, J. Bonjour, Local entropy generation for saturated two-phase flow, *Energy*. 34 (2009) 1113–1121. doi:10.1016/j.energy.2009.03.014.
- [15] R. Revellin, J. Bonjour, Entropy generation during flow boiling of pure refrigerant and refrigerant-oil mixture, *Int. J. Refrig.* 34 (2011) 1040–1047. doi:10.1016/j.ijrefrig.2011.01.010.
- [16] M.A. Abdous, S.G. Holagh, M. Shamsaiee, An evaluation of heat transfer enhancement technique in flow boiling conditions based on entropy generation analysis: micro-fin tube, in: 4th Int. Electron. Conf. Entropy Its Appl., 2017: pp. 1–9. doi:10.3390/ecea-4-05002.
- [17] M.A. Abdous, H. Saffari, H. Barzegar Avval, M. Khoshzat, The study of entropy generation during flow boiling in a micro-fin tube, *Int. J. Refrig.* 68 (2016) 76–93. doi:10.1016/j.ijrefrig.2016.04.008.
- [18] S.G. Holagh, M.A. Abdous, M. Shamsaiee, H. Saffari, Assessment of heat transfer enhancement technique in flow boiling conditions based on entropy generation analysis: twisted-tape tube, *Heat Mass Transf.* 56 (2020) 429–443. doi:https://doi.org/10.1007/s00231-019-02705-y.
- [19] O. Adeyinka, G. Naterer, Optimization correlation for entropy production and energy availability in film condensation, *Int. Commun. Heat Mass Transf.* 31 (2004) 513–524. doi:https://doi.org/10.1016/S0735-1933(04)00032-6.
- [20] G. Li, S. Yang, Entropy generation minimization of free convection film condensation on an elliptical cylinder, *Int. J. Therm. Sci.* 46 (2007) 407–412. doi:10.1016/j.ijthermalsci.2006.06.007.
- [21] S. Dung, S. Yang, Second law based optimization of free convection film-

- wise condensation on a horizontal tube, *Int. Commun. Heat Mass Transf.* 33 (2006) 636–644. doi:10.1016/j.icheatmasstransfer.2006.01.013.
- [22] S.-H. Tzeng, S.-A. Yang, Second law analysis and optimization for film-wise condensation from downward flowing vapors onto a sphere, *Heat Mass Transf.* 43 (2007) 365–369. doi:10.1007/s00231-006-0119-5.
- [23] J.A. Esfahani, M. Modirkhazeni, Entropy generation of forced convection film condensation on a horizontal elliptical tube, *Comptes Rendus Mec.* 340 (2012) 543–551. doi:10.1016/j.crme.2012.03.008.
- [24] H. Ye, K. Lee, Refrigerant circuitry design of fin-and-tube condenser based on entropy generation minimization, *Int. J. Refrig.* 35 (2012) 1430–1438. doi:10.1016/j.ijrefrig.2012.03.013.
- [25] M. Sheikholeslami, M. Darzi, Z. Li, Experimental investigation for entropy generation and exergy loss of nano-refrigerant condensation process, *Int. J. Heat Mass Transf.* 125 (2018) 1087–1095. doi:10.1016/j.ijheatmasstransfer.2018.04.155.
- [26] M. Vatanmakan, E. Lakzian, M. Reza, Investigating the entropy generation in condensing steam flow in turbine blades with volumetric heating, *Energy.* 147 (2018) 701–714. doi:10.1016/j.energy.2018.01.097.
- [27] H. Ding, Y. Li, E. Lakzian, C. Wen, C. Wang, Entropy generation and exergy destruction in condensing steam flow through turbine blade with surface roughness, *Energy Convers. Manag.* 196 (2019) 1089–1104. doi:10.1016/j.enconman.2019.06.066.
- [28] S.G. Holagh, M.A. Abdous, M. Shafiee, M.A. Rosen, Performance evaluation of helical coils as a passive heat transfer enhancement technique under flow condensation by use of entropy generation analysis, *Therm. Sci. Eng. Prog.* 23 (2021) 100914. doi:10.1016/j.tsep.2021.100914.
- [29] S. Ghazanfari, F. Talati, M. Shamsaiee, Analytical solution of steady state heat conduction equations in irregular domains with various BCs by use of Schwarz-Christoffel conformal mapping, *Therm. Sci. Eng. Prog.* 11 (2019) 8–18. doi:10.1016/j.tsep.2019.03.008.

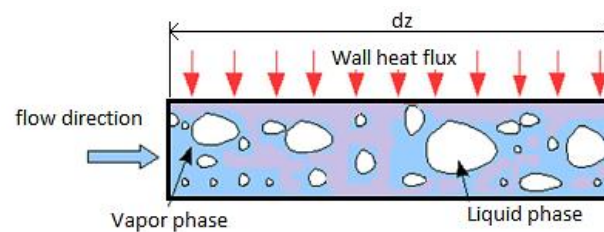
- [30] A.K. Solanki, R. Kumar, Condensation of R-134a inside dimpled helically coiled tube-in-shell type heat exchanger, *Appl. Therm. Eng.* 129 (2018) 535–548. doi:10.1016/j.applthermaleng.2017.10.026.
- [31] A.K. Solanki, R. Kumar, Condensation of R-134a inside micro-fin helical coiled tube-in-shell type heat exchanger, *Exp. Therm. Fluid Sci.* 93 (2018) 344–355. doi:https://doi.org/10.1016/j.expthermflusci.2018.01.021.
- [32] M. Mozafari, M.A. Akhavan-Behabadi, H. Qobadi, M. Fakoor, Condensation and pressure drop characteristics of R600a in a helical tube-in-tube heat exchanger at different inclination angles, *Appl. Therm. Eng.* 90 (2015) 571–578. doi:10.1016/j.applthermaleng.2015.07.044.
- [33] A. Gupta, R. Kumar, A. Gupta, Condensation of R-134a inside a helically coiled tube-in-shell heat exchanger, *Exp. Therm. Fluid Sci.* 54 (2014) 279–289. doi:10.1016/j.expthermflusci.2014.01.003.
- [34] D. Chisholm, Pressure gradients due to friction during the flow of evaporating two-phase mixtures in smooth tubes and channels, *Int. J. Heat Mass Transf.* 16 (1973) 347–358. doi:https://doi.org/10.1016/0017-9310(73)90063-X.
- [35] H. Ito, Friction factors for turbulent flow in curved pipes, *J. Basic Eng.* 81 (1959) 123–132. doi:https://doi.org/10.1115/1.4008390.
- [36] Y.A. Cengel, Michael A. Boles, *Thermodynamics: An Engineering Approach*, 6th ed., The McGraw-Hill Companies, New York, 2007.



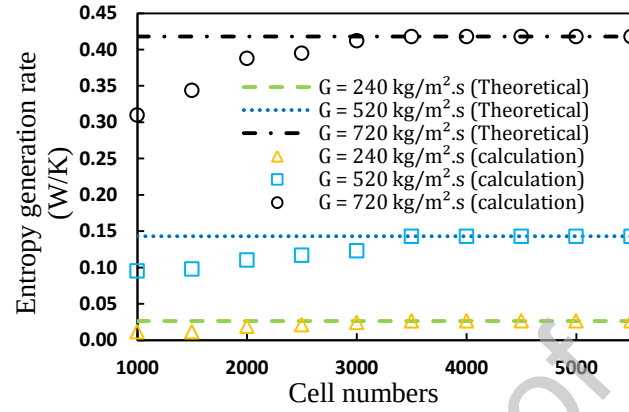
**Fig. 1: Schematic of the helically coiled tube-in-tube condenser and the important geometrical parameters (adopted from [28])**



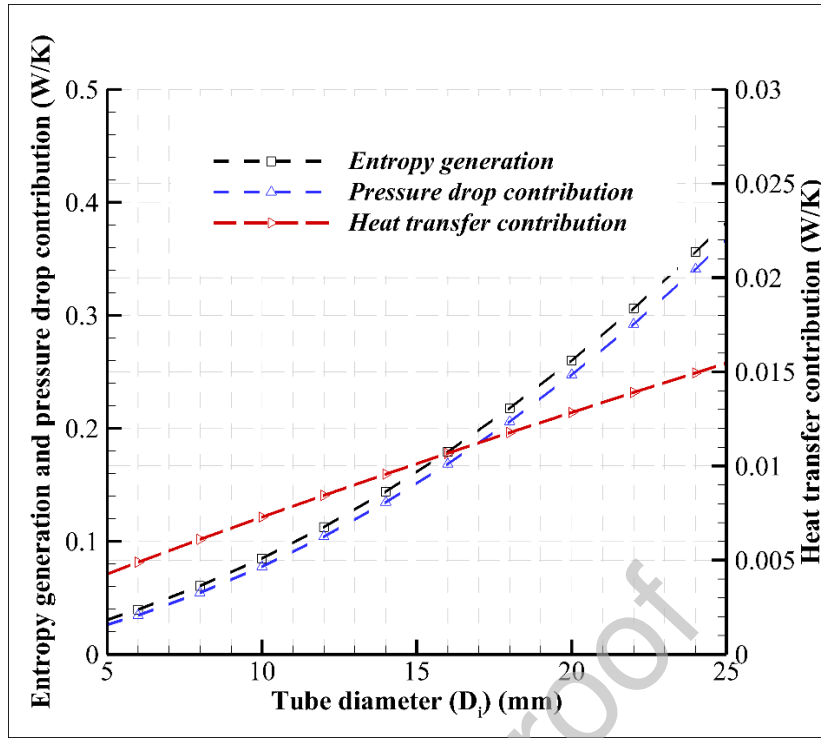
**Fig. 2a: Zoomed view of the grid considered for the helical coil**



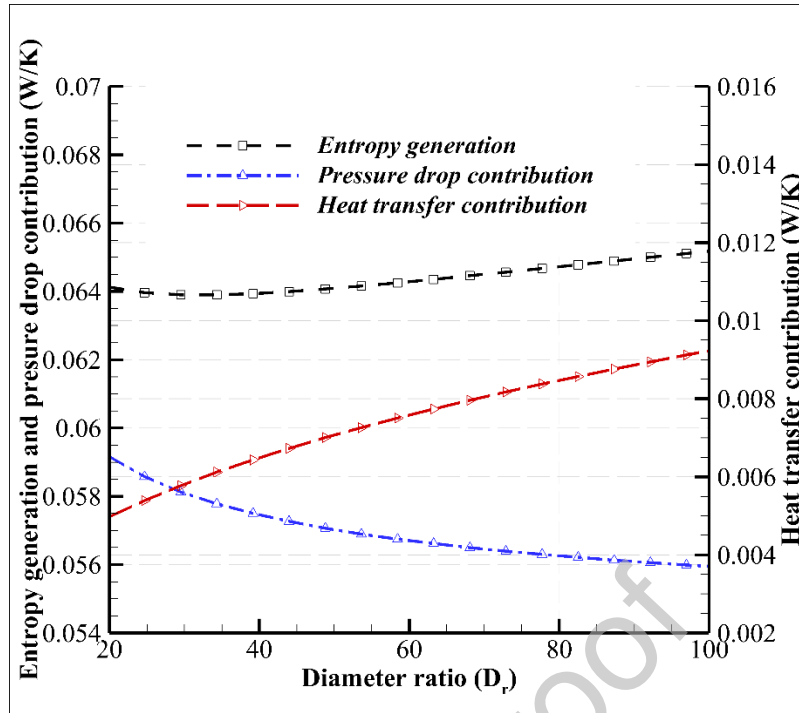
**Fig. 2b: The schematic view of R134a diabatic saturated two-phase flow inside a cell (adopted from Revellin et al. [14,15].**



**Fig. 3: Grid independency and validation for helical coil simulation based on flow conditions in Table 5 and geometrical conditions in Table 6.**



**Fig. 4: Variation of total entropy generation, heat transfer and pressure drop contributions versus the tube diameter regarding flow conditions presented in Table 5 and other constant geometrical parameters shown in Table 6.**



**Fig. 5: Variation of total entropy generation, heat transfer and pressure drop contributions versus the diameter ratio regarding flow conditions presented in Table 5 and other constant geometrical parameters shown in Table 6.**

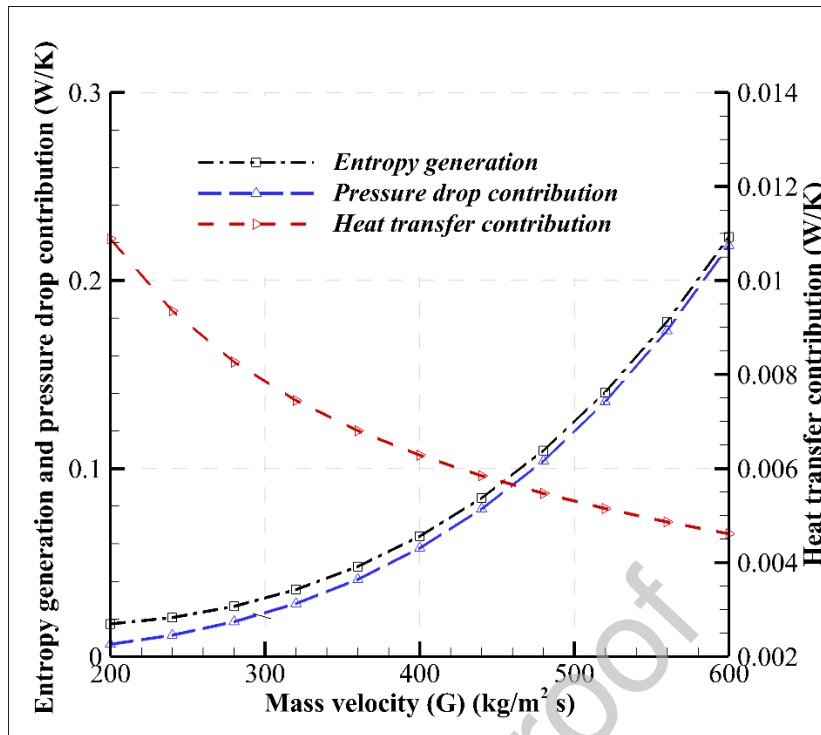
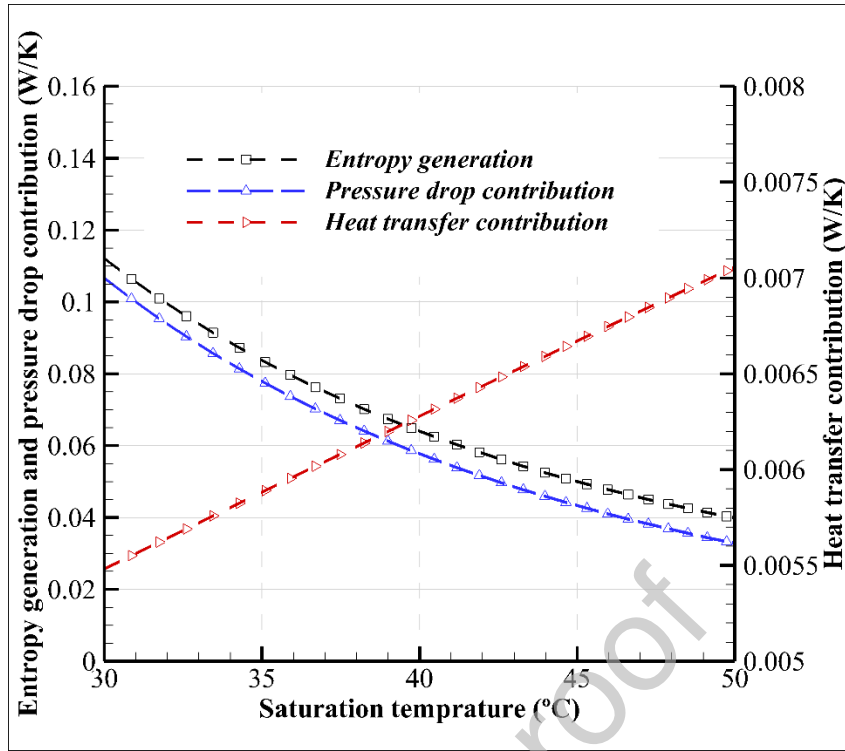


Fig. 6: Variation of total entropy generation, heat transfer and pressure drop contributions versus mass velocity regarding flow and geometrical conditions presented in Tables 5 and 6, respectively.



**Fig. 7:** Variation of total entropy generation, heat transfer and pressure drop contributions versus saturation temperature regarding flow and geometrical conditions presented in Tables 5 and 6, respectively.

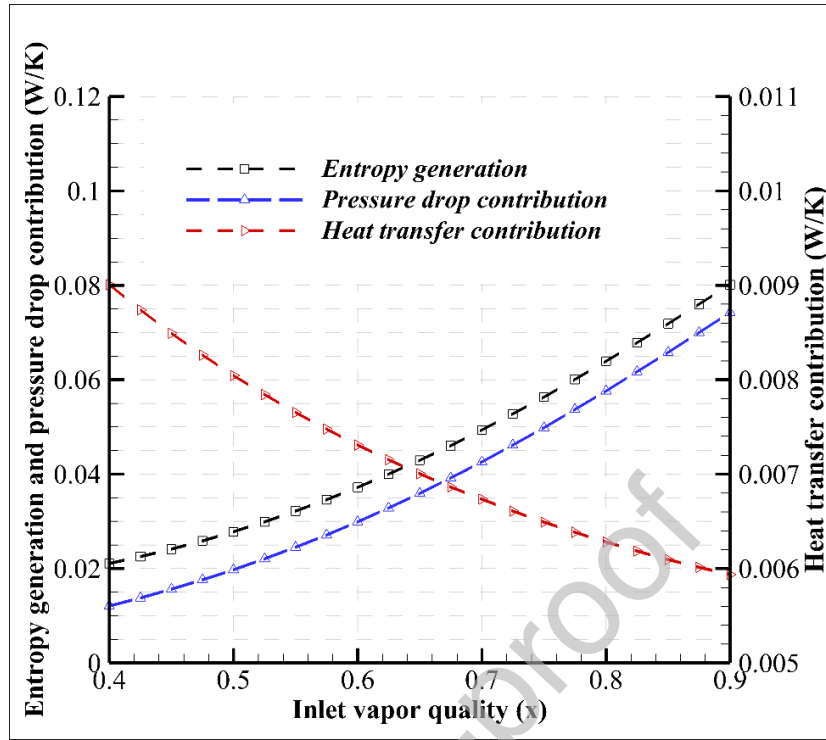
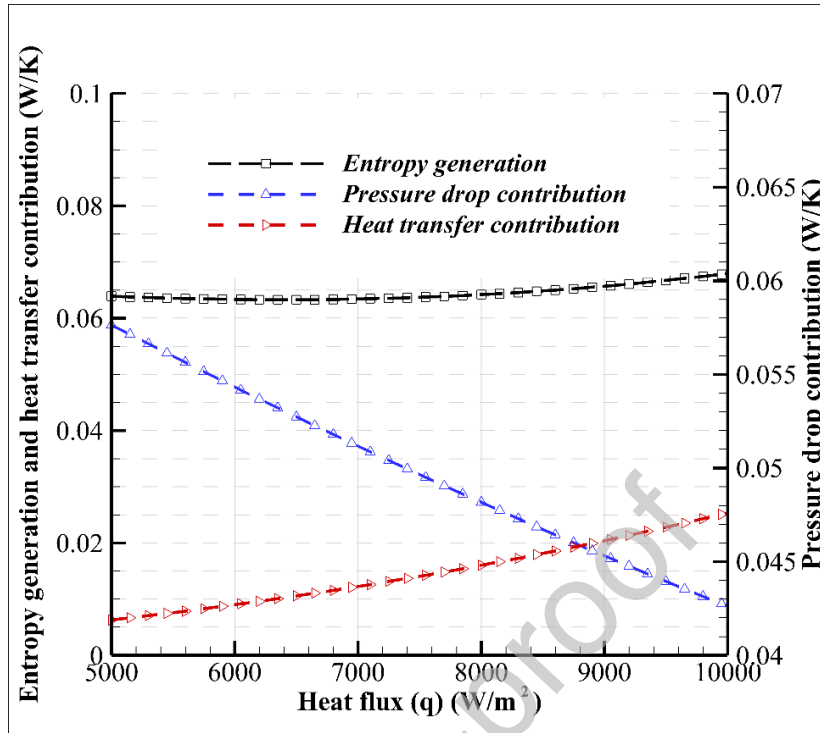
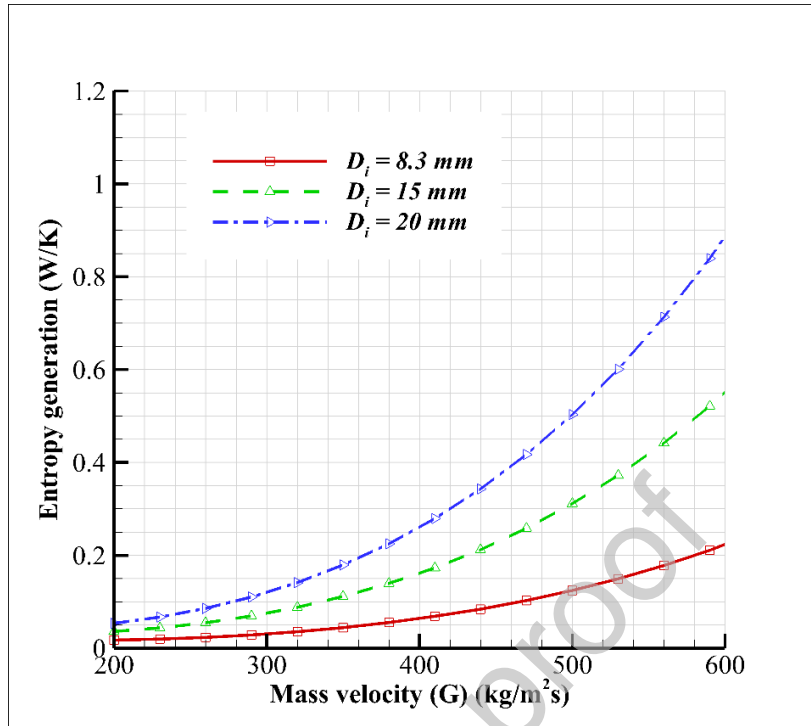


Fig. 8: Variation of total entropy generation, heat transfer and pressure drop contributions versus inlet vapor quality regarding flow and geometrical conditions presented in Tables 5 and 6, respectively.



**Fig. 9:** Variation of total entropy generation, heat transfer and pressure drop contributions versus wall heat flux regarding flow and geometrical conditions presented in Tables 5 and 6, respectively.

a



b

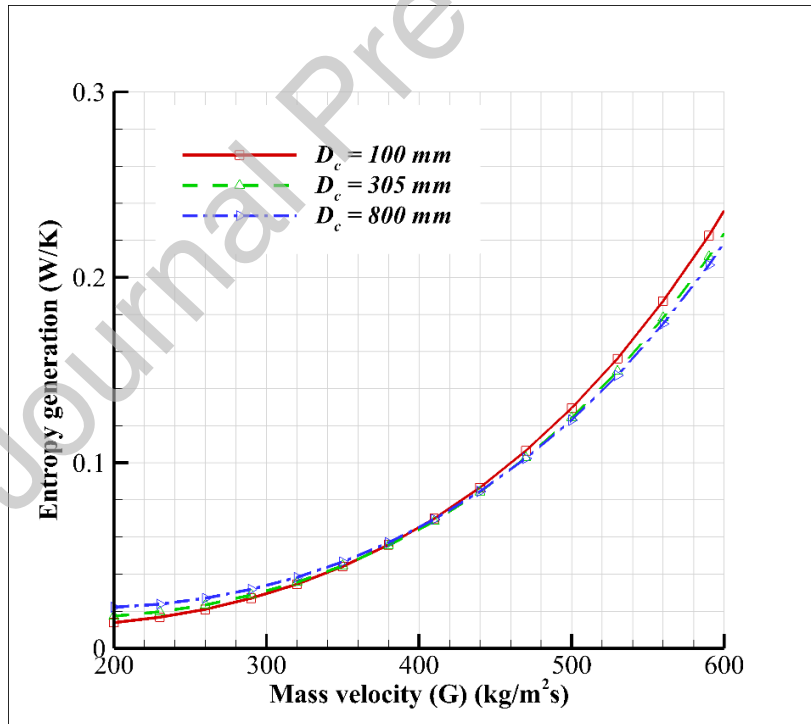
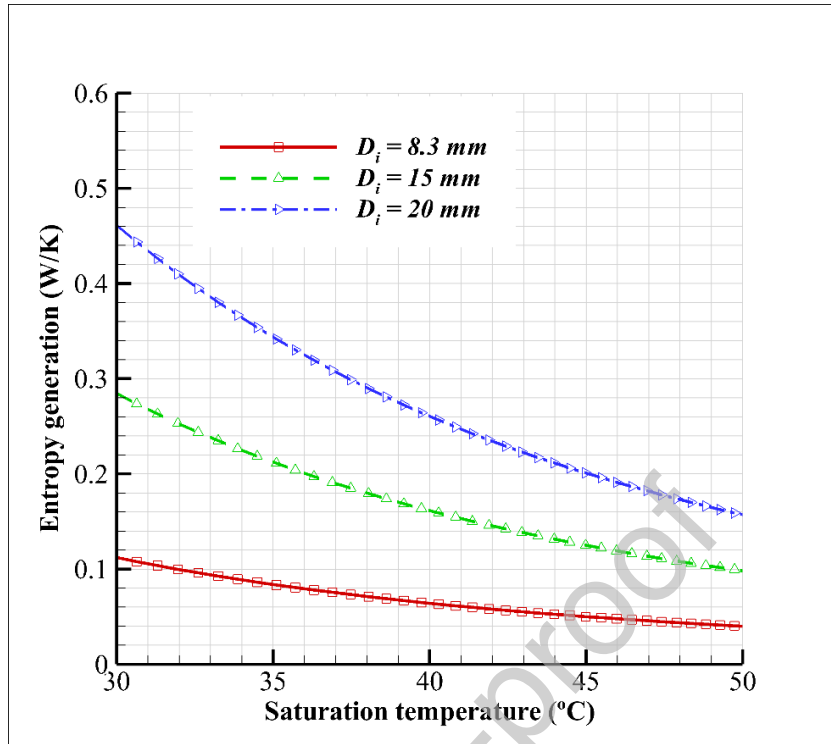


Fig. 10: Comparison of RCEG for helical coils with (a) different tube diameters at constant coil diameter ( $D_c = 305$  mm) and (b) different coil diameters at constant tube diameter ( $D_i = 8.3$  mm) versus variation mass velocity regarding flow conditions presented in Table 5.

a



b

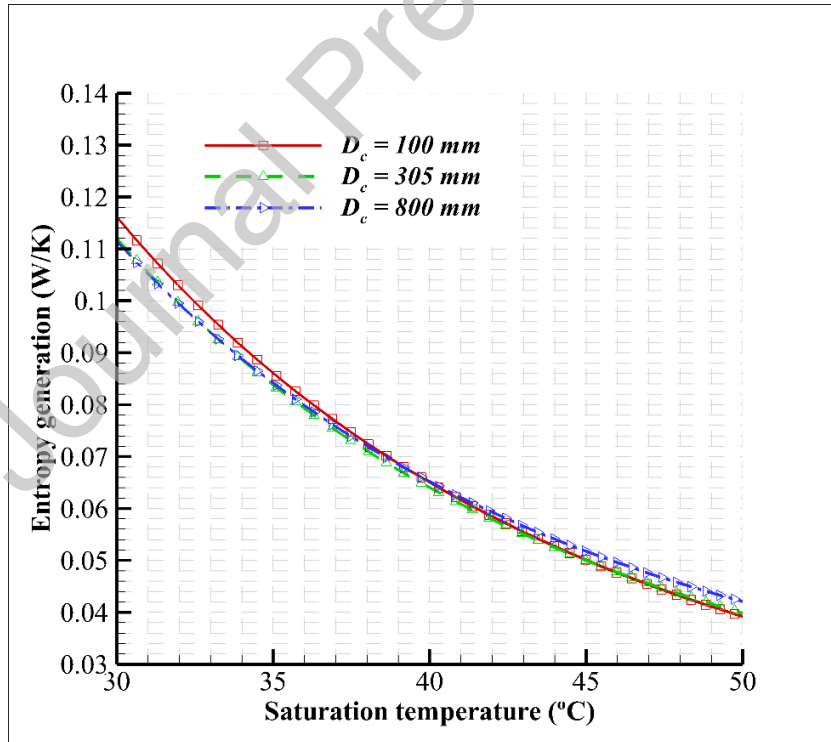
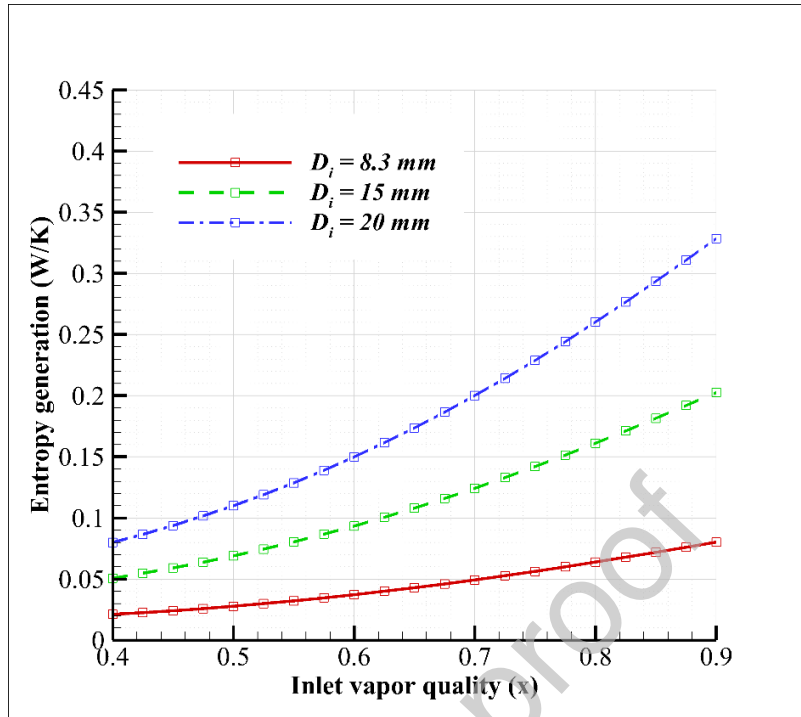


Fig. 11: Comparison of RCEG for helical coils with (a) different tube diameters at constant coil diameter ( $D_c = 305$  mm) and (b) different coil diameters at constant tube diameter ( $D_i = 8.3$  mm) versus variation in saturation temperature regarding flow conditions presented in Table 5.

a



b

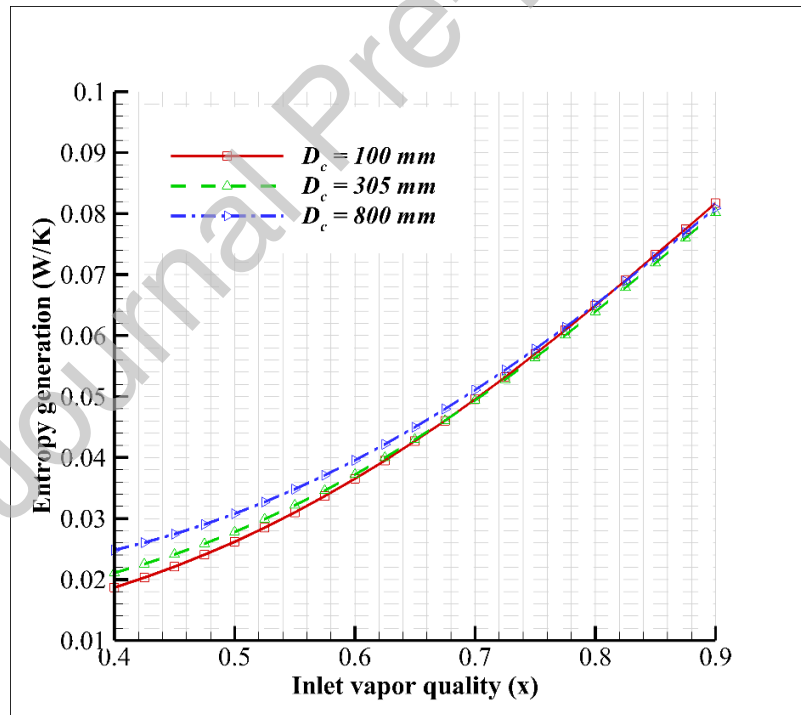
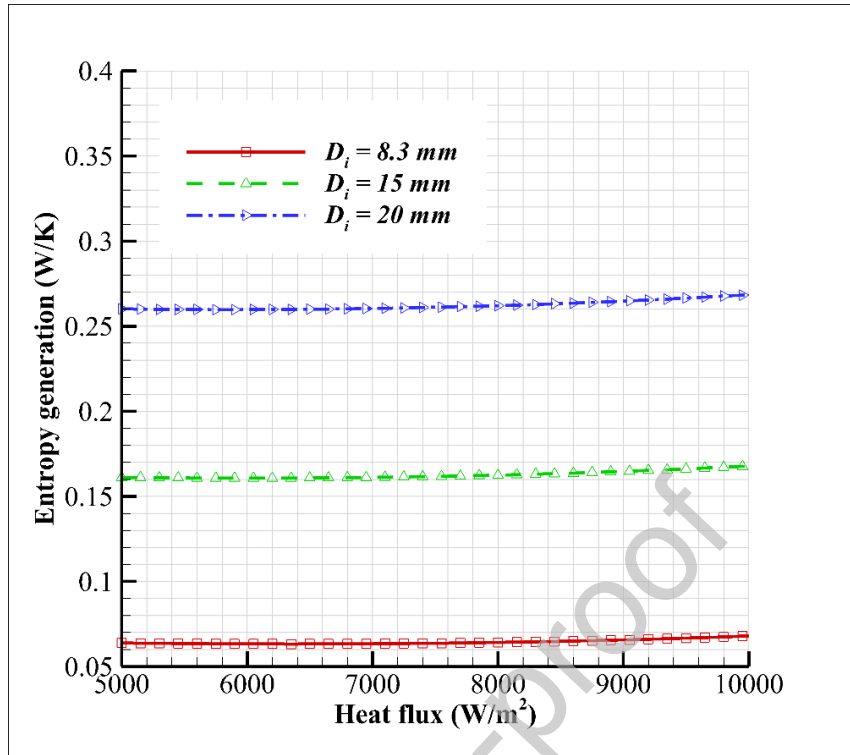


Fig. 12: Comparison of RCEG for helical coils with (a) different tube diameters at constant coil diameter ( $D_c = 305$  mm) and (b) different coil diameters at constant tube diameter ( $D_i = 8.3$  mm) versus variation in inlet vapor quality regarding flow conditions presented in Table 5.

a



b

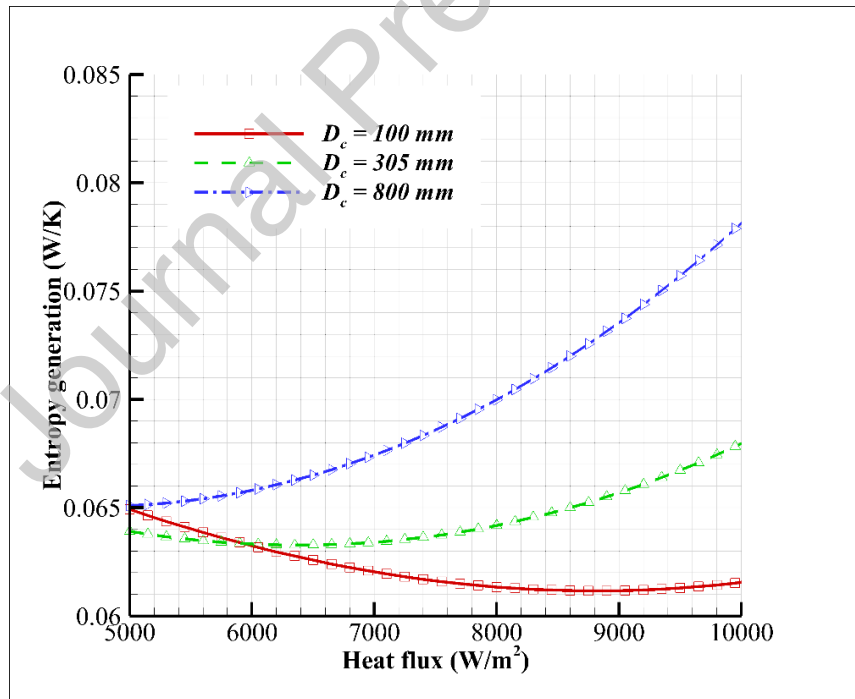


Fig. 13: Comparison of RCEG for helical coils with (a) different tube diameters at constant coil diameter ( $D_c = 305$  mm) and (b) different coil diameters at constant tube diameter ( $D_i = 8.3$  mm) versus variation in heat flux regarding flow conditions presented in Table 5.

**Table 1: Flow conditions studied by Wongwises et al. [6]**

| $x$     | $T_{\text{sat}}$ (°C) | $G$ (kg.m <sup>-2</sup> .s <sup>-1</sup> ) | $q$ (W.m <sup>2</sup> ) |
|---------|-----------------------|--------------------------------------------|-------------------------|
| 0.2-0.9 | 40-50                 | 400-800                                    | 5000-10000              |

**Table 2: Geometrical conditions studied by Wongwises et al. [6]**

| Parameter        | Unit | Inner tube |
|------------------|------|------------|
| Outside diameter | mm   | 9.52       |
| Inside diameter  | mm   | 8.3        |
| Coil diameter    | mm   | 305        |
| Coil pitch       | mm   | 35         |
| Diameter ratio   | mm   | 36.74      |
| Number of turns  | -    | 6          |
| Length           | mm   | 5786.8     |

**Table 3: Ranges of flow parameters at which entropy generation is analyzed in the present study**

| $x$     | $T_{\text{sat}}$ (°C) | $G$ (kg.m <sup>-2</sup> .s <sup>-1</sup> ) | $q$ (W.m <sup>2</sup> ) |
|---------|-----------------------|--------------------------------------------|-------------------------|
| 0.4-0.9 | 30-50                 | 200-600                                    | 5000-10000              |

**Table 4: Ranges of geometrical parameters at which entropy generation is analyzed in the present study**

| Parameter        | Unit | Inner tube |
|------------------|------|------------|
| Outside diameter | mm   | 6.22-26.22 |
| Inside diameter  | mm   | 5-25       |
| Coil diameter    | mm   | 305        |
| Coil pitch       | mm   | 35         |
| Diameter ratio   | mm   | 20-100     |
| Number of turns  | -    | 6          |
| Length           | mm   | 5786.8     |

**Table 5: Constant flow conditions utilized in simulations (Section 3)**

| $x$ | $T_{\text{sat}}$ (°C) | $G$ (kg.m <sup>-2</sup> .s <sup>-1</sup> ) | $q$ (W.m <sup>2</sup> ) |
|-----|-----------------------|--------------------------------------------|-------------------------|
| 0.8 | 40                    | 400                                        | 5000                    |

**Table 6: Constant geometrical conditions utilized in simulations (Section 3)**

| Parameter        | Unit | Inner tube |
|------------------|------|------------|
| Outside diameter | mm   | 9.52       |
| Inside diameter  | mm   | 8.3        |
| Coil diameter    | mm   | 305        |
| Coil pitch       | mm   | 35         |
| Diameter ratio   | mm   | 36.74      |
| Number of turns  | -    | 6          |
| Length           | mm   | 5786.8     |