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THERMODYNAMIC PERFORMANCE EVALUATION FOR HELICAL PLATE HEAT EXCHANGER BASED ON SECOND LAW ANALYSIS

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Abstract. Second-law analysis has affected the design methodology of different heat and mass transfer systems to minimize the entropy generation rate, and so to maximize system available work. In this paper, thermodynamic performance evaluation for helical plate heat exchanger (HPHE) based on second law analysis is studied. The entropy generated per unit amount of heat transferred and by friction are investigated in the entropy generation analysis. A three-dimensional numerical simulation of a whole plate heat exchanger with is carried out by using computational fluid dynamics (CFD) code of Ansys 16.2 for modeling and computational calculations. Helical plate with different pitch ratios and different flow channel cross section aspect ratio were studied for variation of Reynolds numbers. The results showed that the maximum total entropy generation is 0.074 in case of pitch ratio and aspect ratio 1.31 and 0.67 respectively.

Key words: Helical plate, CFD, Heat exchanger, Second law analysis.

1. INTRODUCTION

Heat exchangers performance plays vital role in many industrial applications. Because of high energy costs and low energy sources, there are many efforts to enhance heat exchangers' efficiency. As a result, it is very important to determine the performance of heat exchange devices on both heat transfer and thermodynamic considerations. Heat exchangers are the equipments that provide the flow of thermal energy between two or more fluids at different temperatures. The second law of thermodynamics has proved to be a very powerful tool in the optimization of complex thermodynamic systems such as heat exchangers and is required to establish the difference in quality between mechanical and thermal energy in it [1]. Yilmaz *et al.* [2] presented second-law based performance evaluation criteria in order to evaluate the heat exchangers performance. Firstly, they recalled and discussed the need for the systematic design of heat exchangers using a second law-based procedure. After that, the researchers classified the evaluation techniques for heat exchangers based on the second law of thermodynamics into two categories: the evaluation techniques using exergy as an evaluation parameter, and the evaluation techniques using entropy as an evaluation parameter. They presented and reviewed collectively both categories, and gave their respective characteristics and constraints. It was shown how some of these criteria were related to every other. In addition, emphasis was placed on the importance of second law-based thermoeconomic analysis of heat exchangers, and these methods were discussed briefly. Etghani and Baboli [3] investigated numerical model of shell and helical tube heat exchanger in order to assess heat transfer coefficient and exergy loss. The researchers took to consideration four design parameters including tube diameter, pitch coil, cold and hot flow rate that were more significant for the performance of heat exchanger. Then, they applied Taguchi approach to figure out the optimum levels of the design factors. They modeled and analyzed numerically sixteen cases with diverse design parameters. They found that tube diameter and cold flow rate were the most significant design parameters of heat transfer and exergy loss, respectively. In addition, the highest Nusselt number was achieved by more both cold and hot flow rates and also, heat transfer coefficient was reduced by pitch coil increasing as well as by hot flow rate increasing, the exergy loss increased. The optimum levels for heat transfer coefficient were: tube diameter 12 mm, pitch 13 mm, cold and hot flow rate 4 LPM. Moreover, the optimum level for exergy loss are: tube diameter 12 mm, pitch 13 mm, cold and hot flow rate 1 LPM. İpek *et al.* [4] investigated experimentally exergy loss analysis of newly designed compact heat exchanger (CHE). The researchers designed and constructed experimental system used for experimental analysis of the newly designed CHE and brazed plate heat exchanger (BPHE). Also, they investigated thermodynamic analysis of newly designed CHE and BPHE. They compared the experimental results of the CHE and BPHE. They calculated exergy loss values for every type of heat exchanger. Their experimental results showed that similar exergy loss values were obtained. The least exergy loss value for newly designed CHE has been obtained as about 4.65 kW, while the highest exergy loss value has been obtained as about 7.6 kW for the same heat exchanger. The compared and presented graphically the experiments results. Dizaji *et al.* [5] studied experimentally exergy analysis for shell and tube heat exchanger made of corrugated shell and corrugated tube. The researchers evaluated said parameters for various arrangements of corrugated tubes. They produced corrugated tubes using a special machine that was developed for this purpose. They found that corrugations caused increment of both exergy loss and NTU. If both tube and shell were corrugated, the exergy loss and NTU increased about 17–81% and 34–60% respectively. They observed maximum exergy loss for heat exchanger made of convex corrugated tube and concave corrugated shell. The present paper presents an evaluation of the thermodynamic performance of a promising type of heat exchanger with helical plate (HPHE) based on second law analysis with different helical plate pitch ratios and flow channel cross section aspect ratio.

2. HPHE GEOMETRY AND PROBLEM FORMULATION

The Helical plate heat exchanger with nine helix turns is shown in Figs. 1 and 2. The hot fluid flows in the helical channel with the series arrangement in counter with cold fluid. The heat transfer process occurs through a Helical copper plate with thickness 1 mm. These plates are repeated in the x-direction, with a pitch P, and height h. Here, the dimensionless geometric parameter; pitch ratio = P/h and aspect ratio =w/h were used in the numerical study. The hydraulic diameter D_h was used as the characteristic flow channel diameter.

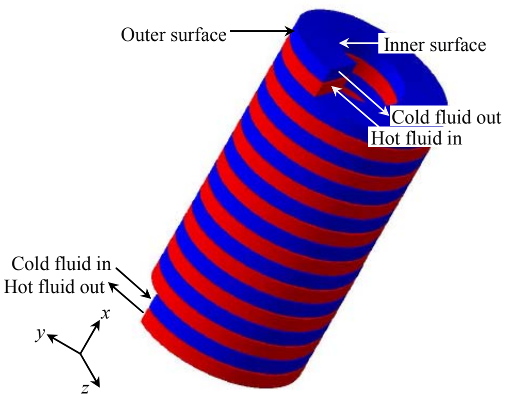

Fig. 1 – Counter flow arrangement .

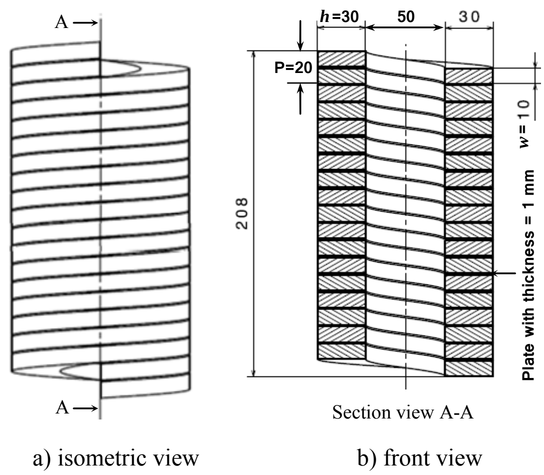

Fig. 2 – Helical plate heat exchanger with pitch ratio = 0.67.

2.1. Numerical domain and grid generation

The whole HPHE with 9 helix turns was modeled as the numerical domain. Commercial software (ANSYS CFX 16.2) was used with a structural hexahedral grid of a total number of nodes in the range of 197,324 to 237,888 using the multi-zone meshing approach. The grid spacing is non-uniform, being concentrated near the interfaces because of the heat transfer and frictions in that region.

2.2. Governing equations and solution assumptions

The problem investigated is a three-dimensional steady state turbulent flow through a helical flow channel fitted with plain tube using the governing equations for the mass, momentum, and energy conservations, and for k and ε turbulence model. The following assumptions were employed: 1. The heat transfer and fluid flow are time-independent (steady-state), three-dimensional, and incompressible, 2. Phase changes and heat transfer by radiation and natural convection are neglected, 3. All the thermo-physical properties of the solid are assumed to be constants.

Mass conservation equation

$$\nabla \cdot (\rho \vec{V}) = 0. \tag{1}$$

Momentum conservation equation

$$\nabla\left[\vec{V}\cdot\left(\rho\vec{V}\right)\right] = -\nabla p - \frac{2}{3}\nabla\left[\mu\left(\nabla\cdot\vec{V}\right)\right] + \nabla\cdot\left[\mu\left(\nabla\cdot\vec{V}\right)^{T}\right] + \nabla\cdot\left[\mu\left(\nabla\cdot\vec{V}\right)\right]. \tag{2}$$

Energy conservation equation

$$\rho c_p \vec{V}\times\nabla T = \nabla\times\left[k\left(\nabla T\right)\right] + \left[\frac{\partial p}{\partial t} + \vec{V}\times\nabla p\right] + \varphi. \tag{3}$$

Turbulence model

$$\nabla \bullet (\rho k \vec{V}) = \nabla\cdot\left[\left(\mu + \frac{u_t}{\sigma_k}\right)(\nabla\cdot k)\right] + G_k - \rho\varepsilon. \tag{4}$$

Entropy generation. In order to evaluate irreversibility loss in heat exchanger, the modified number of entropy generation units (N_s) is defined as [6].

$$N_s = \frac{\dot{S}_g\left(T_{\mathrm{h,i}} - T_{\mathrm{c,i}}\right)}{\dot{Q}}. \tag{5}$$

The entropy balance for an open system such as the heat exchanger is defined as Eq. (6). In a steady-flow process, $\Delta\dot{S}_{sys}$ is zero. In addition, the heat exchanger is often seen as an adiabatic system; therefore, $\Delta\dot{S}_{sys}$ is also zero.

$$\Delta\dot{S}_{\mathrm{sys}} = \dot{S}_i - \dot{S}_0 + \dot{S}_f + \dot{S}_g. \tag{6}$$

Then the entropy balance equation can be reduced to:

$$\dot{S}_g = \dot{S}_o - \dot{S}_i. \tag{7}$$

From Eq. (7), the total rate of entropy production $\dot{S}_{g,total}$ in the heat exchanger can be expressed as follows [7]:

$$\dot{S}_{g,total} = \underbrace{\dot{m}_h c_{p,h} \ln T_{h,o}/T_{h,i} + \dot{m}_c c_{p,c} \ln T_{c,o}/T_{c,i}}_{\text{Entropy gneration due to heat transfer}} + \underbrace{\beta_h \dot{V}_h \Delta p_h + \beta_c \dot{V}_c \Delta p_c}_{\text{Entroy generation due to friction}}. \tag{8}$$

Then, according to Eqs. (5), (7) and (8), the total number of entropy generation units ($N_{s_{total}}$), N_s due to heat transfer ($N_{s_{\Delta T}}$), and Ns due to friction ($N_{s_{\Delta P}}$) are defined as follows:

$$Ns_{total} = Ns_{\Delta T} + Ns_{\Delta P}, \tag{9}$$

$$N_{s_{\Delta T}} = \frac{1}{\varepsilon}\frac{\dot{m}_h c_{p,h}}{\left(\dot{m}\ c_p\right)_{\min}}.\ln\left[1 + \varepsilon\frac{\left(\dot{m}\ c_p\right)_{\min}}{\dot{m}_h c_{p,h}}.\left(\frac{T_{c,i}}{T_{h,i}} - 1\right)\right] + \frac{1}{\varepsilon}\frac{\dot{m}_c c_{p,c}}{\left(\dot{m}c_p\right)_{\min}}.\ln\left[1 + \varepsilon\frac{\left(\dot{m}c_p\right)_{\min}}{\dot{m}_c c_{p,c}}.\left(\frac{T_{h,i}}{T_{c,i}} - 1\right)\right], \tag{10}$$

$$N_{s_{\Delta p}} = \frac{1}{\varepsilon\left(\dot{m}c_p\right)_{\min}}\left(\beta_h \dot{V}_h \Delta P_h + \beta_c \dot{V}_c \Delta p_c\right). \tag{11}$$

2.3. Numerical method

The above mentioned equations were solved with the commercial software ANSYS CFX 16.2. The renormalization group (RNG) *k–ε* model is adopted because it can provide improved predictions of near-wall flows and flows with high streamline curvature [8]. Solution sequential algorithm (segregated solver algorithm) with settings including implicit formulation, steady (time-independent) calculation, SIMPLE as the pressure-velocity coupling method, and first-order upwind scheme for energy and momentum equations was selected for simulation.

2.4. Boundary and initial conditions

The inlet boundary and initial conditions of hot and cold fluid are axial velocity and outlet boundary condition is fixed average static pressure equal to the standard atmohpheric pressure. The inner and outer surface of the HPHE is adiabatic (isolated). All blocks are starting with water. The hot and cold fluids have inlet temperatures of 400 and 300 K for all simulations. The numerical values of velocities and pitch ratios, which were used in a number of the simulations are given in Table 1.

Table 1

Numerical values of the parameters used for simulations

<table>
<tr><th rowspan="2">Pitch ratio</th><th colspan="10">Velocity, m/s</th></tr>
<tr><th colspan="5">Hot fluid</th><th colspan="5">Cold fluid</th></tr>
<tr><td>0.24</td><td rowspan="3">0.1</td><td rowspan="3">0.2</td><td rowspan="3">0.3</td><td rowspan="3">0.4</td><td rowspan="3">0.5</td><td rowspan="3">0.07</td><td rowspan="3">0.13</td><td rowspan="3">0.20</td><td rowspan="3">0.27</td><td rowspan="3">0.33</td></tr>
<tr><td>0.67</td></tr>
<tr><td>1.31</td></tr>
<tr><td>0.24</td><td>0.93</td><td>0.75</td><td>0.56</td><td>0.37</td><td>0.19</td><td>0.65</td><td>0.52</td><td>0.39</td><td>0.26</td><td>0.13</td></tr>
<tr><td>0.67</td><td>0.67</td><td>0.53</td><td>0.40</td><td>0.27</td><td>0.13</td><td>0.47</td><td>0.37</td><td>0.28</td><td>0.19</td><td>0.09</td></tr>
<tr><td>1.31</td><td>0.59</td><td>0.47</td><td>0.35</td><td>0.24</td><td>0.12</td><td>0.41</td><td>0.33</td><td>0.25</td><td>0.17</td><td>0.08</td></tr>
</table>

3. RESULTS AND DISCUSSIONS

Figures 3–5 show the variations in number of entropy generation units due to friction ($Ns_{\Delta p}$), heat transfer ($Ns_{\Delta T}$) and total entropy generation as a function of Reynolds number, pitch ratio and aspect ratio. For constant pitch and aspect ratios, it is shown that the increasing of Reynolds number for both fluids; the number of entropy generation units will decrease. Maximum entropy generation due to friction is 0.003 in case of pitch ratio and aspect ratio 0.24 and 0.12 respectively. Maximum entropy generation due to heat transfer and total entropy generation are 0.073 and 0.074 in case of pitch ratio and aspect ratio 1.31 and 0.67 respectively.

In this section a comparison of helical plate with flat plate heat exchangers is presented according to thermal, hydraulic and thermodynamic parameters. Figures 6–8 show the variations in number of entropy generation units due to friction ($Ns_{\Delta p}$), heat transfer ($Ns_{\Delta T}$) and total entropy generation for helical plate and flat plate heat exchangers as a function of Reynolds number at constant aspect ratio. For constant aspect ratio, it is shown that an increasing in the number of entropy generation units for flat plate heat exchanger over helical plate heat exchanger by about 1.24 to 1.65 on average.

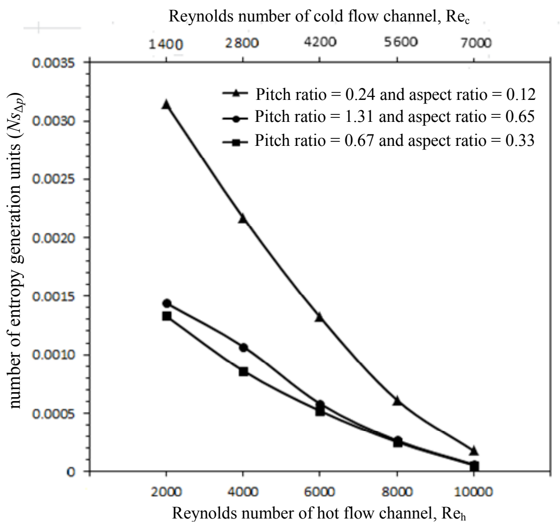

Fig. 3 – Variations in number of entropy generation units due to friction ($Ns_{\Delta p}$) as a function of Reynolds number, pitch ratio and aspect ratio.

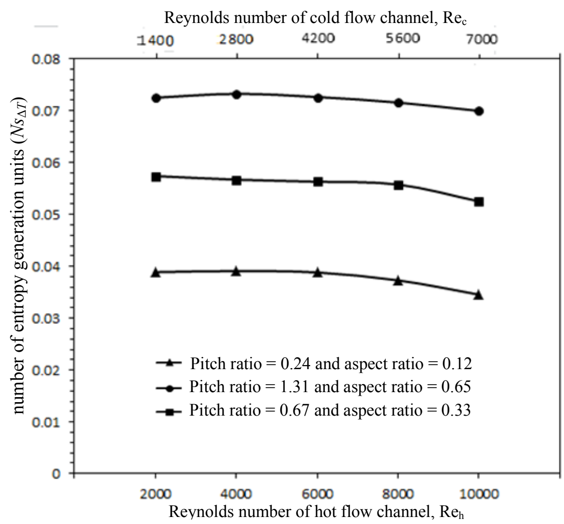

Fig. 4 – Variations in number of entropy generation units due to heat transfer ($Ns_{\Delta T}$) as a function of Reynolds number, pitch ratio and aspect ratio.

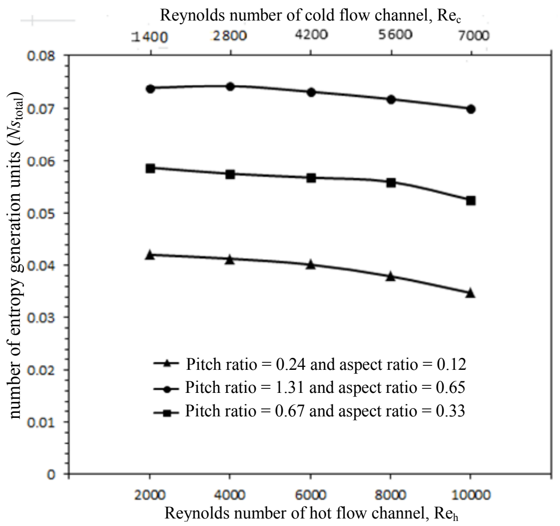

Fig. 5 – Variations in number of total entropy generation units (Ns_{total}) as a function of Reynolds number, pitch ratio and aspect ratio.

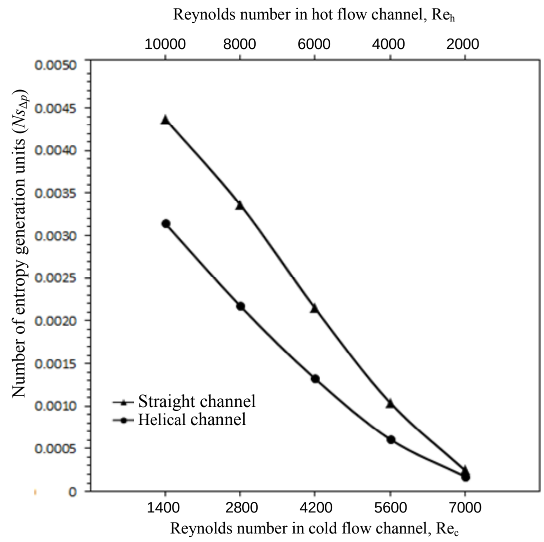

Fig. 6 – Number of entropy generation units due to friction ($Ns_{\Delta p}$) for helical and straight flow channel as a function of Reynolds number at aspect ratio = 0.12.

4. CONCLUSIONS

In this paper a three dimensional simulation model of flow and heat transfer in the fluids channels of a whole HPHE were established numerically to evaluate the thermodynamic performance based on second law analysis with different helical plate pitch ratios and flow channel cross section aspect ratio. The main conclusions are summarized:

1. Maximum total entropy generation is 0.074 in case of pitch ratio and aspect ratio 1.31 and 0.67 respectively.
2. The present numerical simulation had been compared and a good agreement with experimental data trend had been obtained from another published work.

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