INVESTIGATION OF THE EFFECTS OF PLATE PATTERNS ON EFFECTIVENESS AND ENTROPY GENERATION IN PLATE HEAT EXCHANGERS

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ABSTRACT

Compact plate heat exchangers (PHE) are widely used in various industry and power generation fields, especially as condenser and evaporator. Since heat rejection is an essential part of ORC systems, all improvements on heat transfer will lead higher efficiencies. There are some design criteria for such devices. Heat transfer, pressure drop, compactness, entropy generation, and effectiveness are the most essentials. The thermal performance and compactness of these devices can be increased by modifying the surface patterns of plates. However, attention should be paid to pressure drop and entropy generation since any modification on surface can easily led to pressure drop and entropy generation due to viscous effects on the plate surface which increase pumping duty eventually. Thus, there exists an optimization problem that one should take care of. This paper presents an investigation of the effect of the plate surface patterns of a standard PHE and trachea patterned compact PHE in terms of effectiveness and entropy generation. Trachea shaped fins are placed on the surface of the plate to form a trachea patterned PHE designed in accordance with the additive manufacturing method. Compactness which is heat transfer surface area scaled by the heat exchanger volume is an important design parameter for heat exchangers. In order to increase compactness, plate surface is decorated with trachea shaped fins. Consequently, compactness reached to 8150 while it is only 4280 for the standard PHE with three plates. In order to perform the thermodynamic analysis of the trachea pattern compact PHE, different flowrates were taken as 0.01, 0.03 and 0.05 kg/s. As a result of the study, the heat transfer amount was 2950 W, the efficiency was 0.282, and the entropy production was 20.370 W/K.

1 INTRODUCTION

Today, the speed of industrialization and the depletion risk of natural resources cannot be considered separately. Therefore, the sustainability of energy resources is very important. Power plants (PPs) are highly energy-intensive hence their energy recovery potential especially in terms of heat is greater. It is obvious that studies on that area can lead more environmentally friendly production and contribution to the national economy (Kocabas, 2018). Heat exchangers (HEs) are the key equipment of the PP and heat recovery processes. For example, 99% of thermoelectric PPs in the USA use water cooling systems (Acharya *et al.*, 2013). The use of fresh water for PP cooling is becoming increasingly difficult and expensive to obtain, adding to the environmental impact. If cooling is provided by air, it will cause the HE to increase in size. Similarly, these problems are observed for the PP that draws heat from a heat source (geothermal fluid, waste hot water, etc.). In this case, for cooling or heating of a plant, the design of the plant' HEs is important regard of the working fluid. It is necessary to increase the heat transfer performance of the working fluid side without significantly increasing the HE size and the pumping power of the plant. In particular, changing the working fluid from water to refrigerants shows that design and manufacturing should be given more attention. Among many other types, a compact plate heat exchanger (PHE) is a special type in which heat transfer is occurred between two fluids with different temperature levels without mixing versus leakage as a whole. In the literature, there are many studies on HEs. Akyurek *et al.* (2020) have comparatively investigated the thermodynamic analysis and experimental results for double tube HE, shell tube HE and PHE. Savat *et al.* (2020) have simulated the PHE, which is widely used in air-to-air heat recovery devices. They have identified recurring zones in the HE and created various meshes. They have carried out computational fluid dynamics (CFD) analyses using those meshes. As the outcome of study, they have calculated a consistent value for thermal efficiency (51%) against the experimental study which is 52%. Islam *et al.* (2020) have designed a compact-corrugated PHE with a simple modification that significantly improve the thermal performance. General tests have performed for two symmetrical chevron angles 30°/30° and 60°/60° in four compact-corrugated PHEs of which the twos have the basic corrugation and the twos have newly designed corrugations. They have gathered data for steady-state single phase (water-water) counter flow arrangement for Reynolds numbers (Re) varying between 500 and 2500. They have adopted sophisticated mesh techniques for the plates and the fluid between the plates. They have performed a proper system improvement test for the accuracy of numerical results. Finally, they have verified numerical results with the experimental results. For the most accurate prediction of thermal performance of HE, they have use scalable wall function and realizable k-ε turbulence model in numerical analysis. According to the numerical results, Nusselt number (Nu) and efficiency of their compact-corrugated PHEs is higher than the basic PHEs hence they are useful when the extreme heat duty is required. Increase in Nu and efficacy is up to 75% and 42% respectively, and linearly proportional with Re. According to Zhu *et al.* (2020), previous studies on experimental measurements and correlations of the friction factor in PHEs are quite inconsistent. Therefore, to establish a benchmark, they have used Great Vortex Simulation to model fully developed flow in the cross-corrugated channels of the PHE, and then investigated the friction factor for various conditions for the Re ranging from 10 to 6000 with tilt angles ranging from 18° to 72°. They have derived new correlations to predict the critical point for the transition from laminar to turbulence in the PHE. They have also analyzed the average flow properties for the vortex structure and different slope angles. They have stated that a larger angle of inclination leads to more intense eddies and aperture secondary flows in the channel resulting a larger friction factor. İpek *et al.* (2020) have designed and set up a system to experimentally examine the exergy loss analysis of their newly designed compact HE. In their experimental system, they have carried out the experiments of their newly designed compact HE and brazed PHE and also conducted the thermodynamic analysis. They have calculated the exergy loss for each HE type. While the highest exergy loss is obtained as approximately 7.6kW for their newly designed compact HE, they have calculated the lowest exergy loss as approximately 4.65kW for the same HE. In the same study, numerical investigation of various parameters (temperature, velocity, pressure, Re and heat transfer coefficient) has been performed by obtaining the trachea pattern compact PHE design and meshing. Water is the working fluid for all studies. Lee *et al.* (2014) have investigated the flow-boiling heat transfer and pressure drop properties of water under low mass flow condition. They have stated that the evaporation temperature was 102.8–105°C and the mass flux was in the range of 14.5–33.6kg/m^{2}s. According to the results, the flow boiling heat transfer is mainly dominated by the convective boiling mechanism. In addition, the two-phase heat transfer coefficient decreases as the steam quality increases. This phenomenon intensifies when the steam quality rises above 0.3 due to partial drying of the liquid layer. Hence, they have developed two-phase Nusselt number and friction factor correlations based on vapor Reynolds number and liquid Reynolds number values. A typical Organic Rankine Cycle (ORC) should include at least two heat exchangers (a condenser and an evaporator). In ORC systems, overall efficiency is highly related with evaporator performance. Hence exchanger type and design are great concerns. PHEs are used in a broad range of engineering applications because of great heat transfer performance, compactness and relatively simple production process (Ayub, 2003). Furthermore, since it is easy to manipulate surface geometry in compact PHEs designed in accordance with the additive manufacturing, it is possible to obtain a turbulent flow in lower Reynolds ranges comparing to other HE types. In this study, the effects of plate surface shapes on the effectiveness and entropy production of a compact PHE designed for additive manufacturing were investigated under different flow rate and ORC operation conditions. PHE in ORC has been accepted as a water-to-water compact PHE in here due to the reduction in size and complex manufacturing difficulties.

2 MATERIAL AND METHOD

The main purpose of a compact plate heat exchanger (PHE) design is obtained better heat transfer with minimum pressure drop. In this study, in order to improve these parameters, the surface pattern of the plate have been designed with inspiration from the arthropod respiratory system and trachea (spherical) fins have been placed on the plate surface. The geometry of the designed plate is given in Figure 1. To create a 3D trachea pattern, hemispherical fins with a diameter of 3.60mm and a height of 1.60mm were drawn on the plate surface. There are a total of 528 fins on a plate surface, 16 transversely and 33 longitudinally. In order to ease the flow and increase the heat transfer, the pits of the spherical fins are formed at the back of the plate. In order to increase the plate surface area and enhance the heat transfer, extra geometries which helps to maintain a homogenous flow have been added to the hot and cold inlet and outlet ports. The PHE has been designed as three plates (74mm x 192mm) with two hot and one cold flow. It is stated that the first HEs have been produced by pressing have fewer plate contact points and the plates are less resistant to high pressures since it is made of rather thick material. To design a HE with higher heat transfer and lower pressure drop, the HE design must be innovative. However, conventional HE fabrication techniques are limited in their ability to fabricate the complex HE design as a whole. However, additive manufacturing has the ability to produce complex geometries that are difficult or impractical to produce with conventional methods. With this method, the PHE is produced compactly, and there will be benefits such as leakage, formation of micro channels, errors in joining the plates, forming thin plates that withstand the desired pressures of the plates. In PHEs, the surfaces where the actual heat transfer is made are made of thin metal plates and today this plate thickness is up to 0.4mm thin. In addition, these metal surfaces can be smooth or wavy. PHEs provide the most effective heat transfer in liquid-to-liquid applications among all HEs. This is because of the turbulence within the wrinkled plate channels. Heat transfer can be increased by using thinner plates, but as the plate thickness decreases, working pressures become lower (Vestergren, 2003; İpek, 2016). Considering the studies in the literature, the distance between the plates has been taken as 2.25mm and the plate thickness 0.60mm.

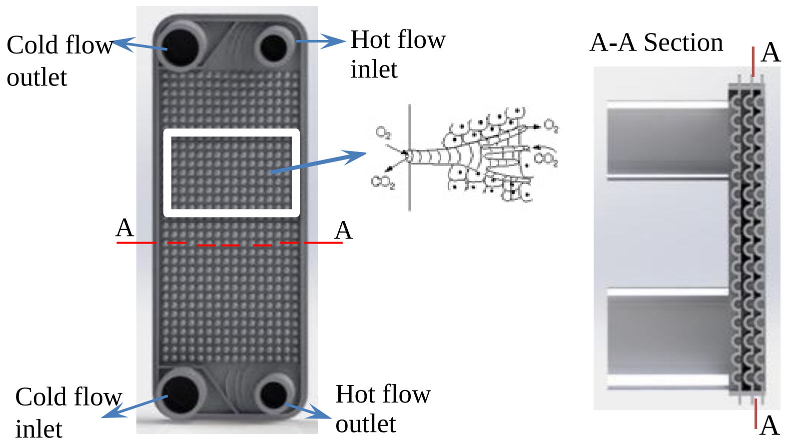

Figure 1: The 3D model and details of design trachea patterned PHE and the location of 17 regions in y-plane on plate surface of hot and cold side.

3 NUMERICAL ANALYSIS

The numerical analysis of 3D trachea patterned compact PHE, is carried out by time independent combined (conduction and convection) heat transfer approach. A finite volume based CFD software, Ansys Fluent ™ is used. It uses a control volume-based technique in which conservation equations are linearized and converted into an algebraic system of equations that can be numerically solved. This method involves obtaining discrete equations that provide the control volume for the variables after integrating the conservation equations for each control volume. Solution iterates until the given convergence parameters such as velocity, pressure and temperature is satisfied. Quality of meshing is an essential parameter for the reliability of solution especially in complex geometries such as spherical structures used in this study. Therefore, HE geometry is meshed with 11 million tetrahedral nodes. Figure 2 shows the meshing structure of the trachea patterned compact PHE designed for this study. The numerical mesh structure of the compact PHE, which was designed by drawing trach-shaped wings on the plate surface, was created with the Ansys-Mesh software. The 18th version of the Ansys Fluent software, based on the finite volume method, was used in the analyzes (Fluent, 2017). In the finite volume method, the analysis of flow and heat transfer through geometry is based on the analysis of partial differential equations obtained from the laws of conservation of energy, mass and momentum regardless of time (Wang, 2009; Fluent, 2017). Boundary conditions given in Table 1 were used for the analyzes. Hot and cold temperatures have been selected, respectively, 90°C and 40°C (Tchanche *et al.*, 2009). Water was used as the working fluid in the PHE. Analyzes were carried out for three flowrates (0.01, 0.03 and 0.05kg/s). Ambient temperature is accepted as 27°C. The parameters used in numerical modeling are given in Table 2. In addition, the material properties for the working fluid water and plate material steel used in the analysis are given in Table 3.

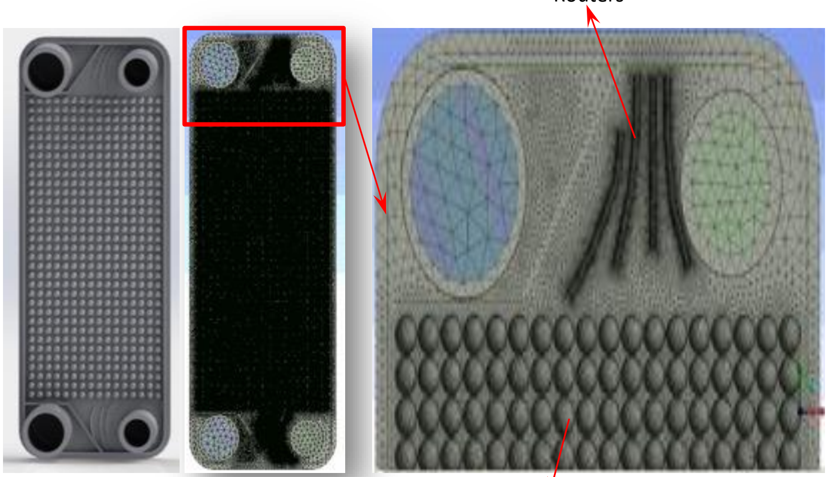

Figure 2: Numerical mesh of designed heat exchanger.

Table 1: Boundary conditions for compact heat exchanger.

parameters	values
Plate thickness (mm)	2.25
Compact heat exchanger height (mm)	74
Number of plates	3
Hot inlet temperature (°C)	90
Cold inlet temperature (°C)	40
Ambient temperature (°C)	27
Heat and cold mass flowrates (kg/s)	0.01, 0.03, 0.05
Compact heat exchanger surface area (mm^2)	139903.33

Inlet-outlet water temperature and hot-cold water velocity are measured values. Heat transfer for plate heat exchanger is as in Eq. (1) (Genceli, 1999; Incropera, 1990).

$$\dot{Q} = \dot{m}_h\, c_{ph}(T_{h,in} - T_{h,out}) = \dot{m}_c\, c_{pc}(T_{c,out} - T_{c,in}) \quad (1)$$

Heat transfer rate can be calculated as follow:

$$\dot{Q} = UA\Delta T_{lmtd} \tag{2}$$

where A is the total heat transfer area, U is the total heat transfer coefficient and ΔT_{lmtd} is the logarithmic temperature difference. Logarithmic temperature difference is:

$$\Delta T_{lmtd} = \frac{\Delta T_1 - \Delta T_2}{\ln\frac{\Delta T_1}{\Delta T_2}}\ ,\ \Delta T_1 = T_{h,in} - T_{c,out}\ ,\ \Delta T_2 = T_{h,out} - T_{c,in} \tag{3}$$

Hot and cold flow heat capacity is calculated from Eq. (4) (Genceli, 1999; Incropera, 1990).

$$\dot{C}_h = \dot{m}_h c_{p,h}\ ,\ \dot{C}_c = \dot{m}_c c_{p,c} \tag{4}$$

Effectiveness of HE is given as (Genceli, 1999; Incropera, 1990):

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} \quad \text{for } \dot{Q}_{max} = C_{min}(T_{h,in} - T_{c,in}) \tag{5}$$

where C_{min} refers to the minimum heat capacity of hot and cold flows.

Table 2: Numerical modelling parameters.

simulation condition	**steady-state**
solver type	pressure based
mesh structure/mesh number	tetrahedral/10 million
turbulence model	standard k-ε turbulence model
wall–turbulence interaction	standard wall-function
pressure–velocity coupling	SIMPLE algorithm
discretization method	second order upwind

Table 3: Thermal properties of the materials.

thermal properties of the materials	**density (ρ) (kg/m^3)**	**specific Heat (Cp) (j/kgK)**	**thermal Conductivity (λ) (W/mK)**
Fluid properties (water)	998.2	4182	0.6
Plate material (AISI 316L, Stainless steel powder)	8030	502.48	381

As the entropy generation amount approaches zero in the HE, the quality of the energy transfer increases. In all real applications, entropy generation is always greater than zero. The maximum available energy that can be taken from a system is known as exergy (also called availability). According to Fettaka *et al.* (2013), exergy is the maximum usable work that can be obtained from the system in a given situation under a certain environmental condition. There is a great deal of temperature distribution on thermodynamic irreversibility in HEs. The amount of the corresponding entropy generation is also defined as irreversibility. Entropy generation for a HE considered as an adiabatic open system is shown as follows (Yilmaz *et al.*, 2013):

$$\dot{S}_{gen} = \Delta\dot{S} = \dot{m}_1\Delta s_1 + \dot{m}_2\Delta s_2 \tag{6}$$

One can write change in entropy in differential form in terms of enthalpy as ds = dh/T. Entropy change for the ideal gas or incompressible fluid in the HE under steady state operating conditions (Yilmaz *et al.*, 2013) are given as follows:

$$\dot{m}_1\Delta s_1 = \int_{in}^{out}\left(\frac{\dot{m}dh}{T}\right)_1 = \int_{in}^{out}\left(\frac{\dot{m}c_p dT}{T}\right)_1 = \left(\dot{m}c_p\right)_1 \ln\frac{T_{1,out}}{T_{1,in}} \quad \text{and} \quad \dot{m}_2\Delta s_2 = \left(\dot{m}c_p\right)_2 \ln\frac{T_{2,out}}{T_{2,in}} \tag{7}$$

Hence the entropy generation is:

$$\dot{S}_{gen} = \sum_{j=1}^{2}\dot{m}_j\Delta s_j = \left(\dot{m}c_p\right)_1 \ln\frac{T_{1,out}}{T_{1,in}} + \left(\dot{m}c_p\right)_2 \ln\frac{T_{2,out}}{T_{2,in}} \tag{8}$$

One of the important design parameters for PHEs is the pressure drop or alternatively the friction factor. Considering the flow velocity and plate surface geometry, the friction factor enables the pressure drop to be calculated. Smaller pressure drop is directly related to lower friction losses in the PHE, which results lower pumping power requirements finally (Kilic, 2017). Reynold (Re), Prandtl

(Pr) and Nusselt numbers (Nu) and friction factor (f) are defined as follows (Kakac, 2012):

$$\mathrm{Re} = \frac{\rho U D_h}{\mu},\ \ \mathrm{Nu} = \frac{U D_h}{k},\ \ \mathrm{Pr} = \frac{\mu C_P}{k},\ \ f = \frac{\rho \Delta P D_h}{2 L_P (\rho U)^2} \tag{9}$$

where L_p is the length of the exchanger between inlet and outlet ports and D_h is the hydraulic diameter which is $2 \times a$. Here, a is the distance between the plates (Genceli, 1999). Mesh size next to the wall should be selected properly for the sake of robustness of simulation. Hence, mesh size was calculated according to the desired y+ value. The Eq. (10) are uses for such task. Average Re number between plates in PHEs is 1470, hence the flow is in transition. According to the literature (Tennekes and Lumley, 1972), the y+ value in the transition flow is between 5-30. In order to be a more sensitive solution, y+ value was taken as 5. When the geometric properties and fluid properties of the PHE given in Table 4 are used in the equations, Δs is calculated as 0.00015 m.

$$\mathrm{Re}_x = \frac{\rho U_\infty L}{\mu},\ \ C_f = \frac{0.026}{\mathrm{Re}_x^{1/7}},\ \ T_{wall} = \frac{C_f \rho U_\infty^2}{2},\ U_{fric} = \sqrt{\frac{T_{wall}}{\rho}},\ \Delta s = \frac{y^+ \mu}{U_{fric}\rho} \tag{10}$$

Table 4: Dimensions of plate heat exchanger and flow properties.

U_∞ (m/s)	ρ (kg/m^3)	μ (kg/ms)	L (m)	y+	Δs (m)	Re_x
0.49	1000	0.001	0.003	5	0.0001506743062897145	1470

4 RESULTS AND DISCUSSION

In this study, CFD simulations were carried out to examine the heat transfer, pressure drop and entropy generation of a compact PHE with a trachea pattern on the plate surface designed for additive manufacturing under ORC operating conditions. Thus, the hot and cold flow inlets were accepted as 90°C and 40°C (Tchanche *et al.*, 2009), and analyzes were carried out at 0.01kg/s, 0.03kg/s and 0.05kg/s. Temperature, pressure, velocity, heat convection and Reynolds number parameter values were calculated for compact PHE.

In Figure 3(a) and (b) mid-plane temperature distributions of hot and cold sides are given for mass flowrates of 0.01, 0.03 and 0.05kg/s respectively. Hot inlet temperature is 90°C for all flowrates. Smooth temperature gradients have been obtained in each case at hot and cold sides. However, gradient becomes steeper as the flowrate increases. Widthwise temperature distribution is nearly uniform.

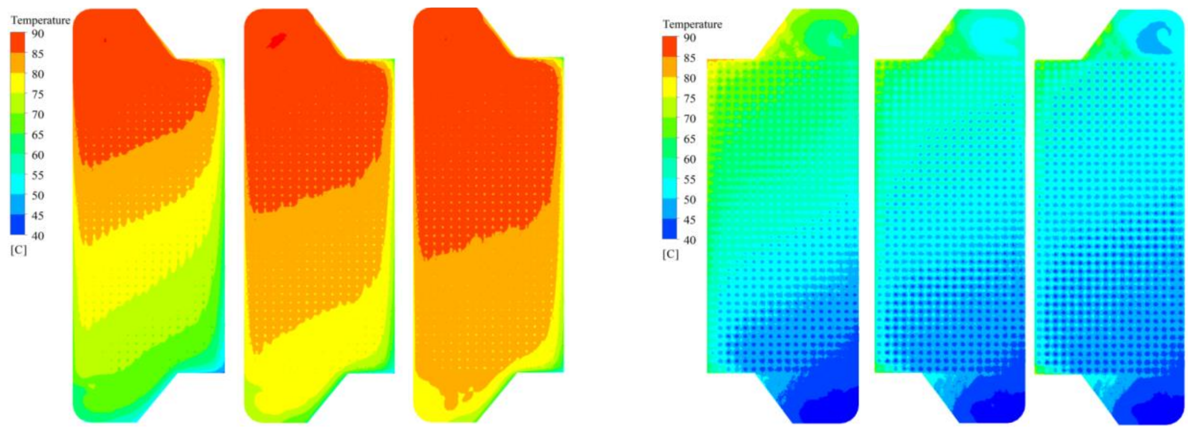

Figure 3: Temperature distribution on mid plane for (a) hot and (b) cold flows.

According to CFD analysis results of the compact plate heat exchanger designed in this study, heat transfer (Q), efficiency value (ε), entropy generation ($\dot{S}_{gen}$) values were calculated. Figure 4 shows the heat transfer and efficiency of the compact heat exchanger at different flowrates. While a heat transfer of 919.6W was realized at a flowrate of 0.01kg/s, it reached its maximum value at a flowrate of 0.03kg/s (2950W). At the highest flow (0.05kg/s), it was observed that the heat transfer was only

2800W. Similarily, the maximum efficiency was calculated at a flowrate of 0.03kg/s with a value of 0.282. Efficiency values at high and low flowrates were obtained as 0.267 and 0.08, respectively. Accordingly, both heat transfer and efficiency of the compact HE are highly dependent on the flow velocity. When the flow velocity is higher or lower (far from optimum), less heat transfer occurs between hot and cold flows. As shown in Figure 4, the values obtained as a result of their work in Sevilgen *et al.* (2020) are given. In their work, the PHE obtained the heat transfer amounts of 1800W and 2800W at 0.02kg/s and 0.04kg/s, respectively. Sevilgen *et al.* (2020) obtained heat transfer amounts as 1800 W and 2800 W, respectively, by using water at a flowrate of 0.02 and 0.04 kg/s in a plate heat exchanger. When the flowrate increases from 0.03 to 0.05 heat transfer decreases. This is because of increasing pressure drop. One of the main factors for pressure drop is friction factor which affects also the amount of transferred heat.

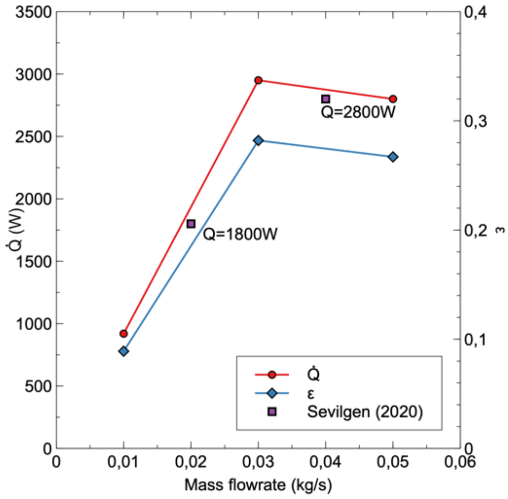

Figure 4: Variation of heat transfer and effectiveness with mass flowrate.

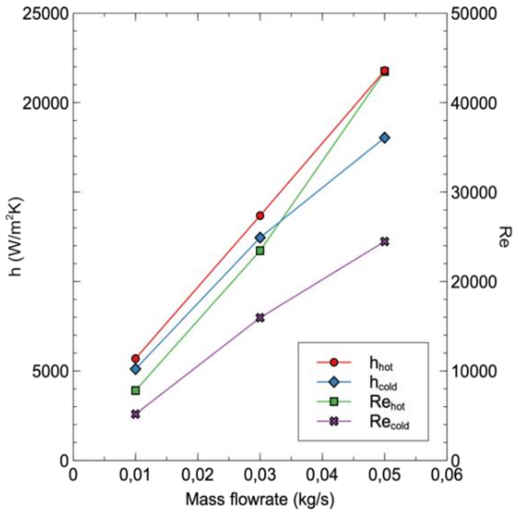

Figure 5: Variation of Reynolds number and convective heat transfer coefficient with mass flowrate.

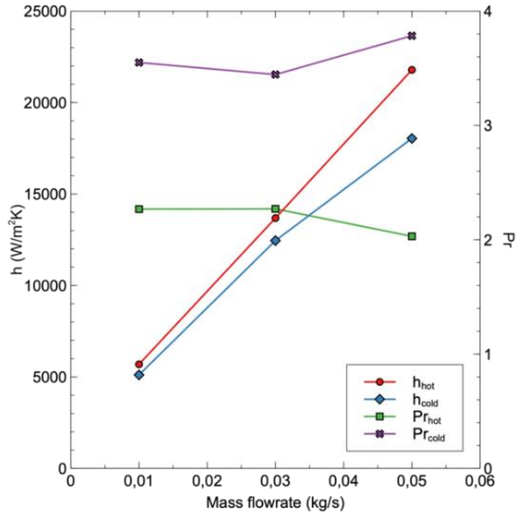

Figure 6: Variation of Prandtl number and convective heat transfer coefficient with mass flowrate.

In Figure 5, the convective heat transfer coefficient (h) and Reynolds numbers (Re) calculated at different flowrates of the compact heat exchanger are given. It is seen that both values increase as the flowrate increases. At 0.05kg/s flow, h and Re for the hot side are maximum with 21785.4W/m^{2}K and 43486.5W/m^{2}K, respectively, while the cold side h and Re values are maximum with 18038W/m^{2}K and 24476W/m^{2}K, respectively, at the same flow. Since the hot water outlet temperature decreases when the mass flow of cold water increases, the Re number increases for the hot flow at low cold water flowrates. Figure 6 shows the variation of the convective heat transfer coefficient and Prandtl number value of the compact heat exchanger at different flowrates. Maximum convective heat transfer coefficient was obtained at a flowrate of 0.05kg/s for the hot and cold sides. For the hot side, the Prandtl number was calculated as 2.030, 2.270 and 2.268 at flowrates of 0.05, 0.03 and 0.01kg/s, respectively, while it was 3.784, 3.445 and 3.55 for the cold side, respectively. The convective heat transfer coefficient was found to be higher on the colder side for each flowrate.

Figure 7 shows the effectiveness and entropy generation in the compact plate heat exchanger at different flowrates. As the flowrate increases, effectiveness and entropy generation increase. At a flow of 0.03kg/s, the effectiveness is maximum (0.282) and the entropy generation is 0.20W/K. The maximum entropy generation (27.071W/K) was realized at 0.05kg/s. Minimum effectiveness and entropy production were achieved at a flowrate of 0.01kg/s. While the effectiveness should be higher at a flow of 0.05kg/s, it is seen that it is lower than at a flow of 0.03kg/s. Entropy generation increases as the flowrate increases. This is because of the increase in entropy generation due to fluid friction with increasing flow (Kotcioğlu *et al.*, 2010).

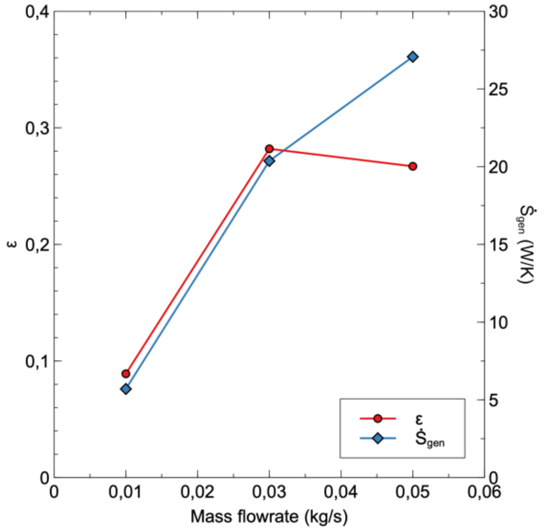

Figure 7: Variation of heat exchanger effectiveness and entropy generation with mass flowrate.

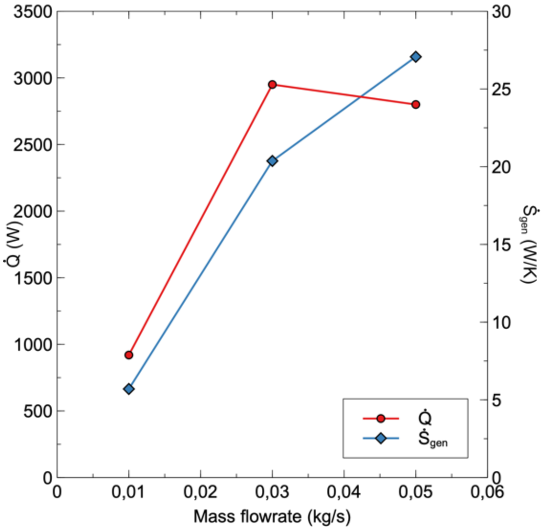

Figure 8: Variation of heat transfer and entropy generation with mass flowrate

The variation of heat transfer and entropy production at different flowrates is given in Figure 8. It was observed that as the mass flowrate of the trachea pattern compact plate heat exchanger increases, the entropy production increases and the heat transfer increases from 0.01kg/s to 0.03kg/s and then decreases. Entropy generation at maximum flowrate is 27.071W/K, but heat transfer is only 2800W. It has been shown in the literature that minimum entropy generation does not always guarantee the highest heat transfer (Qian and Li, 2010). Accordingly in this study, it was seen that heat transfer was the lowest at a flow of 0.01kg/s with minimum entropy generation.

Table 5. Inlet outlet pressures of hot side according to mass flowrate

mass flowrate (kg/s)	inlet pressure, P_{in} (Pa)	pressure drop (ΔP) (Pa)	$\Delta P/P_{in}$
0.01	700	650	0.928
0.167 (Kilic, 2013) (Reference study)	5800	4750	0.815
0.03	3200	2700	0.844
0.05	3500	2600	0.743

Hot side inlet and outlet pressures (gauge) are given for three flowrates and the reference study (Kilic, 2013) in Table 5. Both inlet and outlet pressure values are minimum at the flowrate of 0.01kg/s and incrérase together with increasing flowrate. In the reference study (Kilic, 2013). The inlet and outlet pressures are 5800Pa and 1050Pa respectively. Since mass flowrate is different in each case, in order to make a fair comparison, the ratio of pressure drop to the inlet pressure have been calculated. Although new design have many surface treatment for better heat transfer, no negative effect have been observed in pressure drop.

5 CONCLUSION

In this study, the heat transfer amount, pressure drop and entropy generation of the trachea patterned compact plate heat exchanger (PHE) have been investigated. Analyzes were carried out for three different flowrates. In addition, the tracheal pattern on the plate surface was divided into 17 regions and results were obtained from these regions. Reynolds number, heat transfer coefficient, temperature, pressure and velocity values were taken from each of these regions separately and evaluated. Heat transfer amounts and efficiency values were calculated and the results obtained were presented. According to these results;

- Channels designed as in trachea form have increased heat transfer.
- The highest pressure drop in both hot and cold flow regions has occurred in trachea patterns as expected, but also it was observed that it has gradually decreased from inlet to outlet.
- Heat transferred at 0.03kg/s flowrate is 5.085% and 68% more than at 0.05kg/s and 0.01kg/s respectively.
- While the standard plate heat exchanger used for verification has six plates, the trachea design investigated in this study consists of three plates. For the reference study, the hot and cold inlet temperatures are 32.8°C and 25°C respectively and flowrate is 0.167kg/s. Numerical results of transferred heat is 2143W (Kilic, 2013). For the trachea design, transferred heat has been calculated as 2950W when the hot and cold inlet temperatures are 90°C and 40°C respectively and the flowrate is 0.03kg/s.
- Although new design has more complex surface geometry surface no additional pressure drop have been observed.
- While entropy production was 12.35W/K for the reference standard plate heat exchanger, it was obtained as 20.37W/K for designed heat exchanger when the flowrate is maximum (0.03kg/s).

It was concluded that the designed compact plate heat exchanger with trachea pattern provides better heat transfer and pressure drop in less plates compared to the standard plate heat exchanger. In heat exchanger designs, the effect of entropy generation on heat transfer should be considered. Plate heat exchangers are used in a wide range of areas, especially in thermal power plants, chemical industries, heating, air conditioning, cooling installations, vehicles, electronic devices. In order to increase energy efficiency and thus contribute to energy economy, it is necessary to determine the optimum operating conditions of plate heat exchangers used in a wide range of areas. The results of this study are intended to contribute to the optimum operating conditions, design and production and use of systems using plate heat exchangers.

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