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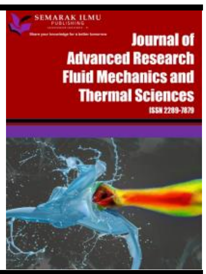

A Review on Heat Transfer Enhancement in a Heat Exchanger

Sashwin Nair Gobinathan[1], Azizuddin Abd Aziz[1,*], Ahmed Nurye Oumer[2], Mohd Yusof Taib[1], Ahmad Basirul Subha Alias[1]

[1] Faculty of Mechanical and Automotive Engineering Technology, Universiti Malaysia Pahang Al-Sultan Abdullah, 26600 Pekan, Pahang, Malaysia
[2] School of Engineering and Physical Sciences, Heriot-Watt University, Dubai, United Arab Emirates

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ABSTRACT

Compact heat exchanger is commonly used in car radiators which is normally small with low flowrate. To produce a large surface area, thin plates or corrugated fins are attached closely separating the two fluids. Compact heat exchanger is normally used in applications where weight and volume are the significant factors for the enhancement of heat transfer. This paper reviews the research related to the characteristics, advantages and findings on various type of compact heat exchanger. It is found that the geometric parameters such as fin shape, tube shape, fin pitch, tube pitch, tube arrangement, tube angle and vortex generators affect the heat transfer performance in general. The air inlet angle of 45° and common flow upstream gives the optimum heat enhancement in most studies while the most suitable air flow is between 1.8 – 3.8 m/s.

1. Introduction

Compact heat exchanger (CHE) is normally used in applications where weight and volume are a matter of interest. The main feature of CHE is efficient heat transfer for a given small volume. Besides, CHEs are energy efficient, they have excellent heat transfer rate, small, lightweight and cheap. As mentioned, CHEs can achieve high heat transfer rates between two fluids in a small volume where they are used in systems that require small dimensions and light weight. This is possible because a large surface area in CHE is obtained by attaching thin plates or corrugated fins closely to the walls separating the two fluids. In CHEs, both fluids are normally in a crossflow configuration whereby they move perpendicular to each other. This produces high turbulence thus generates high heat transfer coefficients. Depending on the flow structure, crossflow may be categorized as unmixed or mixed. CHEs are the most common heat exchanger used in various industries due to their advantages mentioned earlier. CHEs are selected based on their applications in the industry. There are several types of CHE being investigated numerically and experimentally by researchers to improve its heat

* *Corresponding author.*
E-mail address: azizuddin@umpsa.edu.my

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transfer performance. Table 1 shows a summary of different types of CHEs that have been investigated by researchers.

Table 1
Past research on compact heat exchanger

Type of CHE	Methodology	Main Findings	Reference
Wavy duct type plate HE	Numerical	(i) At *Re* ≤ 1000, high heat transfer regions show on both pressure and suction side walls due to the secondary vortex flows which are perpendicular to the flow direction. (ii) At *Re* >1000, these secondary flow cells are not visible and boundary layer type flow is formed on the pressure-side wall.	Hwang *et al.,* [1]
Plate HE	Experimental	(i) Result shows that the inclined discrete rib plate gives an enhancement of heat transfer 20–25% at the constant pumping power compared with the inclined continuous rib plates.	Li *et al.,* [2]
Welded type plate HE	Experimental and Numerical	(i) The results show that the plate with the elliptical shape gives better performance of heat transfer and pressure drop than the plate with the chevron shape.	Jeong *et al.,* [3]
Chevron-type plate HE	Numerical	(i) As the channel's aspect ratio increases and the chevron angle decreases, the tortuosity coefficient and the coefficient K (Kozeny's coefficient in granular beds) increases. (ii) As the chevron angle decreases, the shape factor from plate heat exchanges passages increases.	Fernandes *et al.,* [4]
Corrugated type plate HE	Numerical	(i) Corrugated plate shows an improvement in both flow distribution and heat transfer compared to the smooth plate. (ii) As in pressure drop, the corrugated plate shows a higher result compared to the smooth plate.	Kanaris *et al.,* [5]
Printed circuit heat exchanger with S-shaped fins	Numerical	(i) The new PCHE gives 3.3 times less volume and 37% lower pressure drop in the CO_2 side and by 10 times in H_2O.	Ngo *et al.,* [6]
Airfoil fin printed circuit heat exchanger	Experimental and Numerical	(i) As the pressure increases, the outlet temperature also increases and the average outlet temperature at 8MPa was 7.81 K which is higher than that at 5 MPa for the same mass flow rate. (ii) The pressure drop increased as the mass flow rate increased.	Zhao *et al.,* [7]
Fin-and-tube CHE with wavy and elliptical winglet-type vortex generators	Numerical	(i) Heat transfer performance for up-configuration is higher than the down configuration. (ii) Wavy-up configuration gives the highest pressure difference with the increase of Reynold's number. (iii) Wavy-up configuration gives the lowest value of *(j/f)* compared to others.	Modi and Rathod [8]
Fin-and-tube CHE with wavy rectangular winglet-type vortex generators	Numerical	(i) Heat transfer is enhanced highest in wavy-up configuration. (ii) The friction factor for wavy-down configuration is lower compared to others.	Gholami *et al.,* [9]
Louvered and wavy fin-and-tube CHE	Experimental	(i) Louvered fin heat exchanger has higher pressure drop penalty than wavy fin. (ii) The highest Colburn factor is when the fin pitch is at 2.1mm due to the increase of Reynolds number and ratio of louver pitch to fin pitch.	Okbaz *et al.,* [10]

2. Flow and Heat Transfer Properties

As mentioned earlier, this paper focuses on the heat transfer enhancement of fin-and-tube heat exchangers using vortex generators. Due to the thermo-physical properties of the air which makes the heat transfer coefficient low, the performance of this type of HE is limited by the air side. This results in high thermal resistance at the air side. Therefore, improvements should be focused on the air side of the CHE whereby flow and heat transfer characteristics must be considered. Those significant variables are Nusselt number (Nu), heat transfer coefficient (h), Reynolds number (Re), pressure drop (ΔP), friction factor coefficient (f) and Colburn factor (j).

$$Re = \frac{V_{avg} D_h}{v} = \frac{\rho V_{avg} D_h}{\mu} \tag{1}$$

where,
V_{avg} = average flow velocity (m/s)
D_h = hydraulic diameter (m)
v = kinematic viscosity (m^2/s)
μ = dynamic viscosity (kg/m.s)

Re is a dimensionless ratio that portrays the type of flows experienced by flowing fluid. Types of flow can be indicated as shown

Laminar	$Re \leq 2300$
Transition	$2300 < Re < 4000$
Turbulent	$Re \geq 4000$

Pressure drop is an important parameter that must be considered as it is directly related to the power requirements of the fan or pump that controls the flow of fluid. ΔP represents the pressure difference between 2 points (inlet and outlet) which is due to the viscous effect where it is irreversible. Thus, if there were no friction, the pressure drop would be zero. Since ΔP depends on friction, ΔP for all types of flow in all types of pipes can be represented as follows [11].

$$\Delta P = f \frac{L}{D_h} \frac{\rho V_{avg}^2}{2} \tag{2}$$

where,
V_{avg} = average flow velocity (m/s)
D_h = hydraulic diameter (m)
L = Length (m)
f = friction factor depending on the type of flow as shown below.

Laminar flow in a square duct

$$f = \frac{56.92}{Re} \tag{3}$$

Turbulent flow

$$f = \left(-1.8 log\left[\frac{6.9}{Re} + \left(\frac{\varepsilon/D}{3.7}\right)^{1.11}\right]\right)^{-2} \tag{4}$$

Researchers had proposed several friction factor correlations accompanied by considering the geometric parameters which would benefit the heat exchanger designers. Table 2 shows various friction factor correlations proposed by researchers with various working conditions.

Table 2
Friction factor correlations

Formula	Working conditions	Ref
$f_{RTW}=$ $0.118Re^{-0.385}\left(\frac{e}{2.1}\right)^{0.247}\left(\frac{H}{0.3}\right)^{0.182}(\frac{\alpha}{10})^{-0.011}$ $f_{ARW}=$ $0.172Re^{-0.411}\left(\frac{e}{2.1}\right)^{0.245}\left(\frac{H}{0.3}\right)^{0.182}(\frac{\alpha}{10})^{-0.011}$ $f_{CARW}=$ $0.126Re^{-0.358}\left(\frac{e}{2.1}\right)^{0.247}\left(\frac{H}{0.3}\right)^{0.182}(\frac{\alpha}{10})^{-0.011}$	$500 \leq Re \leq 3000$ $15° \leq \alpha \leq 75°$; α = attack angle of winglet $0.65 \leq e \leq 1.0$; e = tube ellipticity ratio $0.8 \leq H \leq 1.6$; H = wavy fin height	Lotfi *et al.,* [12]
$f = (0.790Re - 1.64)^{-2}$	$Re = 3000$	Ammar and Park [13]
$\frac{1}{\sqrt{f}} = -2log\,(\frac{\in/D}{3.7} + \frac{2.51}{Re\sqrt{f}})$ $\in/D$ = relative roughness, dimensionless	$Re > 4000$	Barker [14]
$\frac{1}{\sqrt{f}} = 2log(Re\sqrt{f}) - 8$	$4000 \leq Re \leq 10^6$	Assefa and Kaushal [15]
$\frac{1}{\sqrt{f}} = 2log\,(\frac{2.51}{Re\sqrt{f}})$ $\frac{1}{\sqrt{f}} = 1.8lo\,g\left(\frac{Re}{6.9}\right)$	$4000 \leq Re \leq 10^8$	Menon [16]
$f = \frac{1}{4}(1.8\,log\,Re - 1.64)^{-2}$	$4000 \leq Re \leq 10^3$	Mortean and Mantelli [17]

Convection is the mode of energy transfer between a solid surface and the adjacent moving fluid. Convection involves a combined effect of conduction along with the fluid motion. The rate of convection heat transfer from Newton's law of cooling is shown below.

$$\dot{Q} = hA(T_s - T_f) \tag{5}$$

where,
$\dot{Q}$ = rate of heat convection (W)
h = convection heat transfer coefficient (W/(m^2°C))
A = surface area (m^2)
T_s = surface temperature (°C)
T_f = bulk fluid temperature (°C)

The h is not a property of a fluid rather it is an experimentally determined parameter that depends on variables such as geometry, fluid motion, properties of the fluid and the bulk fluid velocity [18]. Besides, h also depends on the design of heat exchangers like surface area and material used. In the analysis of convection heat transfer, the number of variables can be reduced if h is non-dimensionalised with thermal conductivity (k) to Nu as shown in Eq. (6).

$$Nu = \frac{hL_c}{k} \tag{6}$$

where,
h = convection heat transfer coefficient (W/(m²°C))
k = thermal conductivity of fluid (W/m.K)
L_c = characteristics length

Nusselt number represents the enhancement of heat transfer through a fluid layer due to convection relative to conduction across a fluid layer. The larger the Nusselt number, the higher the effect of convection [19]. Researchers had proposed several Nusselt number correlations accompanied by considering the geometric parameters which would benefit the heat exchanger designers. Table 3 shows various Nusselt number correlations proposed by researchers with various working conditions.

Table 3
Nusselt number correlations

Formula	Working conditions	Reference
$Nu_{RTW} = 0.236Re^{0.643}\left(1.2+\frac{e}{2.25}\right)^{0.235}\left(\frac{H}{2.25}\right)^{0.146}(1+\frac{\alpha}{45})^{0.021}$ $Nu_{ARW} = 0.214Re^{0.648}\left(1.8+\frac{e}{2.35}\right)^{0.262}\left(\frac{H}{2.35}\right)^{0.161}(1+\frac{\alpha}{45})^{0.014}$ $Nu_{CARW} = 0.247Re^{0.632}\left(1.7+\frac{e}{2.3}\right)^{0.247}\left(\frac{H}{2.3}\right)^{0.150}(1+\frac{\alpha}{45})^{0.017}$	$500 \leq Re \leq 3000$ $15° \leq \alpha \leq 75°$; α = attack angle of winglet $0.65 \leq e \leq 1.0$; e = tube ellipticity ratio $0.8 \leq H \leq 1.6$; H = wavy fin height	Lotfi *et al.,* [12]
$Nu = \frac{\left(\frac{f}{8}\right)(Re-1000)Pr}{1+12.7(f/8)^{\frac{1}{2}}(Pr^{\frac{2}{3}}-1)}$	$Re = 3000$	Ammar and Park [13]
$Nu = 1.953\left(RePr\frac{D}{L}\right)^{1/3}$	$Re < 2100$ $RePr\frac{D}{x} \geq 3.33$	Bai and Bai [20]
$Nu = 3.66+\frac{0.0668\left(\frac{D}{L}\right)Re.Pr}{1+0.04[\left(\frac{D}{L}\right)Re.Pr]^{2/3}}$	$Re < 2100$	Nourafkan *et al.,* [21]
$Nu = 4.364+0.086\,\frac{(\frac{1}{L^*})^{1.33}}{0.1Pr(D_hRe/L)^{0.83}}$ $L^* = \frac{L/D_h}{Re\,Pr}$	$Re < 2100$ $0.7 \leq Pr \leq 7$ $L^* > 0.03$	Mortean and Mantelli [17]
$Nu = 4.364+0.086\,\frac{(\frac{1}{L^*})^{1.33}}{0.1Pr(D_hRe/L)^{0.83}}$ $L^* = \frac{L/D_h}{Re\,Pr}$; L^* = *dimensionless thermal length*	$Re < 2100$ $0.7 \leq Pr \leq 7$ $L^* > 0.03$	Mortean and Mantelli [17]
$Nu = \frac{\left(\frac{f}{2}\right)(Re-1000)Pr}{1+12.7(f/2)^{\frac{1}{2}}(Pr^{\frac{2}{3}}-1)}[1+(D/L)^{\frac{2}{3}}]K$ $K = (Pr/Pr_w)^{0.11}$; *for liquid* $K = (T_b/T_w)^n$; *for gas* $n=0.45$; *for air*	$2300 < Re < 10^4$ $0.5 < Pr < 2000$	Mortean and Mantelli [17]

The Colburn factor is a relation that can be used to evaluate the performance of heat exchangers. It relates heat transfer and fluid flow properties as shown in Eq. (7) [19].

$$j = \frac{Nu}{Re\,Pr^{1/3}} \tag{7}$$

3. Factors Affecting the Heat Transfer

The type of CHE focused on this research is fin and tube heat exchanger. The selection of this HE designs is due to many reasons. The following subtopics portray the factors affecting heat transfer in terms of fin shape, arrangement of tubes, tube inclination angle, usage of vortex generators, air inlet velocity and temperature of liquid for both experimentally and numerically. Fin shapes have an adverse effect on the heat transfer properties which is because different shape affects the fluid flow and as well as the distortion of the boundary layer. The fact that fin shapes are important in heat transfer can be seen in the works of the most recent journal done by Okbaz *et al.,* [10] whereby they had investigated louvered and wavy fin type heat exchanger. The results obtained by Okbaz *et al.,* [10] stated that the louvered fin heat exchanger has a higher Colburn factor and friction factor compared to the wavy fin heat exchanger with the increase of Reynolds Number and number of tube row. Thus, the louvered fin heat exchanger has higher pressure drop penalty than the wavy fin which can be due to more blockage of air flow with the increase of the number of tube row [10].

The influence of various air inlet angles on heat transfer and flow friction of finned oval tube was experimentally and numerically investigated by Tang *et al.,* [22]. The air inlet angle varies from 30° to 90°. Based on the experimental results, heat transfer coefficient for air inlet angle 45° is the highest followed by 60°, 90° and 30° whereas, for friction factor, 30° air inlet angle is the highest followed by 45°, 60° and 90°. The simulation results show the same as the experimental results stated above. Based on the theory of field synergy, a reduction in intersection angle between velocity and temperature gradient enhances the heat transfer. This is the reason why 45° and 60° air inlet angles show a higher heat transfer coefficient than 90° [22]. In 2017, Adam *et al.,* [23] had investigated the effect of tube inclination (0° - 30°) of CHE for automotive applications on thermal and fluid dynamic analysis. The result shows that the Nusselt number increases with the tube inclination angle and Reynolds number. This can be due to the increase in swirling strength behind the tube when the tube inclination increases. Whereas for pressure drop, it increases with the increase of tube inclination angle due to more are blocked in the flow [23].

In a more recent tests, Phu and Hap [24] investigated the influence of inlet velocity on heat transfer performance of plate finned tube HE. The inlet velocity for this study varies from 1-4 m/s. Based on their results, when the inlet velocity increases, the heat transfer rate also increases because the air absorbs more heat at higher velocities. Apart from that, the pressure drops increases with the increase of inlet velocity which due to more air is blocked at higher velocity. Subhedar *et al.,* [25] have performed an experimental investigation on the effect of coolant temperature on car radiator heat transfer performance. Al_2O_3/water-mono ethylene glycol nanofluids were investigated as a coolant for car radiators and the temperature of coolant varies from 65°C – 85°C. The flow rate and air velocity were set constant throughout the experiment which is 4.06 l/min and 1.05 m/s When the inlet temperature increases from 70°C to 85°C, the Nusselt number enhanced by 26% which is due to the variation of thermophysical properties with the increase of temperature of coolant [25].

Heat transfer coefficient and pressure drop of an automobile radiator with graphene nanoplatelets nanofluid were experimentally investigated by Selvam *et al.,* [26]. The inlet temperature of nanofluid, 35°C and 45°C were varied whereas the inlet air velocity and air temperature were kept constant at 3 m/s and 30°C. It is observed that the heat transfer coefficient increases with respect to inlet nanofluid temperature. This is due to the effect of temperature on the thermophysical properties of nanofluids. Moreover, the increase in temperature of nanofluids increases the thermal conductivity and decreases the viscosity [26]. Furthermore, Rai *et al.,* [27] had done an experimental investigation of heat transfer enhancement of automobile radiators using magnesium oxide/distilled water-ethylene glycol nanofluid. The different inlet temperature of 40°C, 45°C, 50°C and 55°C whereas the inlet air velocity and nanofluid flow rate were kept constant at 2.8 m/s and 100 l/min for this research. It is observed that the heat transfer rate increases rapidly from 40°C till 50°C whereas from 50°C till 55°C the rise was not rapid [27]. The increment of heat transfer rate can be due to the thermophysical properties of the nanofluid as mentioned by Selvam *et al.,* [26]. Table 4 lists out the experiment specifications that have been studied in the past.

Table 4
Summary of experiment specifications

Author	Parameters	Working Condition	Remarks
Fin Shape			
Okbaz *et al.,* [10]	Louvered and wavy	$1200 \geq Re \geq 3000$ P_t = 0.0381m P_l = 0.033m δ = 0.00012m W_λ / W_h = 2.5 L_p / L_H = 4.92	i. *j* in the louvered fin is higher than wavy fin with the increase of tube row. ii. *f* in the louvered fin is higher than wavy fin with the increase of tube row.
Tube Inclination Angle			
Tang *et al.,* [22]	$30° \leq \theta \leq 90°$	1.4 m/s≥ v_{air} ≥ 4.5m/s δ = 0.35mm F_s = 2.5mm P_t = 26.7mm P_l = 55mm	i. *h* for air inlet angle 45° is the highest followed by 60°, 90° and 30°. ii. *f* for 30° air inlet angle is the highest followed by 45°, 60° and 90°.
Adam *et al.,* [23]	$0° \leq \theta \leq 30°$	1.8 m/s≥ v_{air} ≥ 3.8m/s δ = 0.6mm D_h = 15.05mm P_t = 40mm P_l = 40mm	i. The Nusselt number increases with the tube inclination angle and Reynolds number. ii. Whereas for pressure drop, it increases with the increase of tube inclination angle.
Air Velocity			
Temperature of Liquid			
Subhedar *et al.,* [25]	65°C – 85°C	δ = 0.00254cm F_s = 0.15875cm H = 0.9cm v_{air} = 1.05 m/s $\dot{v}$ = 4.06 *l*/min	i. When the inlet temperature increases from 70°C to 85°C, the *Nu* enhanced by 26%.
Selvam *et al.,* [26]	35°C – 45°C	δ = 0.0001 m H = 0.00047 m v_{air} = 3 m/s $T_{in,air}$ = 30°C	ii. The heat transfer coefficient increases with respect to inlet nanofluid temperature.

Ke *et al.,* [28] had numerically investigated the thermal-hydraulic performance of plain and wavy fin shape finned tube HE. The wavy fin configuration interrupts the flow and creates a swirling flow which influences the flow at the wake of the tube. Moreover, the results show that the wavy fin configuration gives a higher Nusselt number compared to plain fin with respect to an increase in Reynold's number. Apart from that, flow and thermal characteristics of rectangular and circular fin had been numerically investigated by Sakib and Al-Faruk [29]. The outlet temperature reduces as the inlet air velocity for both types of fins which causes the overall heat transfer of the HE to improve. Based on the temperature distribution of both fins, it is observed that rectangular fins provide higher heat transfer than circular fins due to the higher heat transfer surface area of the rectangular fin.

Adjustment of tube inclination angle can produce vortices which can enhance the heat transfer performance in a HE. Wang *et al.,* [30] had researched the effects of various tube rotation angle ($0° \leq \theta \leq 90°$) on flow and heat transfer performance of CHE. The numerical results show that the Colburn factor increases gradually from 0° to 30° and decreases from 30° to 90° which is due to more air is blocked as the tube angle increases. Furthermore, the pressure drop increases as the tube rotation angle increases. This is acceptable as more air is obstructed in the fluid domain which causes uneven vortices. The heat transfer coefficient is highest when the tube rotated at 30° which is mainly due to the air stagnation at the rear of the tube [30].

Adam *et al.,* [23] performed a numerical investigation of the thermal-hydraulic performance of flat tube HE at various tube inclination angle which are 0°, 30°, 60°, 90°, 120° and 150°. As the inclination angle increases from 0° to 90°, the swirling strength behind the tubes increases which enhances the degree of turbulence. Nevertheless, for tube inclination 120° and 150°, the swirling intensity decreases [23]. Besides that, 90° inclination angle gives the highest Nusselt number with an enhancement of 41.2%. This might be due to the uneven circulation around the tube wall. Even though the 90° inclination angle gives the highest heat transfer enhancement, it also gives the highest pressure drop which can be due to the extra drag formed and results in tube blockage.

Inline and staggered tube arrays can produce variation in flow separation and recirculation. This is proven as Zeeshan *et al.,* [31] investigated the heat transfer performance of an inline and staggered arrangement of flat tubes. The inline arrangement shows that the heat transfer coefficient increases and pressure drop decrease as the flatness of the tube increases. This is due to the flow separation delay in the flat tube. In contrast with inline, heat transfer coefficient and pressure drop increase as the flatness of the tube decreases. This phenomenon occurs because more air is blocked throughout the flow. The author claims that the F1 dimension gives the best thermal-hydraulic performance from the j/f factor [31]. Inline and staggered tube arrangement of rectangular and circular fin-and-tube HE had been numerically investigated by Sakib and Al-Faruk. Their results show that staggered arrangement shows a higher heat transfer characteristic than inline arrangement. Observation of the velocity contour of both inline and staggered arrangements revealed that the vortex formation is obvious in each row which is very important in heat transfer enhancement [29].

Unger *et al.,* [32] had done a numerical analysis of finned tube HE for passive spent fuel pool cooling to ambient air. Both inline and staggered tube arrangements were analysed in this study. This study proves that staggered arrangement gives a higher Nusselt number by 88.5% than inline arrangement. The heat transfer increases because of improved flow mixing, due to flow deflection at staggered arrangement [32]. There are several methods to enhance heat transfer such as increasing the heat transfer coefficient (h) and employing extended surfaces (fins). Fins enhance heat transfer by increasing the surface area of the heated surface to increase the duration of contact with the working fluid which makes more heat energy is transferred to the working fluid. One way to extend a surface is by using winglets. Winglets come in many shapes and forms.

Winglets functions as vortex generators where it increases fluid mixing (turbulence intensity) by inducing vortices. There are longitudinal vortices, where their axis is parallel to the flow; and transverse vortices, where their axis is perpendicular to the flow. When fluid flows, vortices are generated due to friction and separation on the edge of the vortex generator. Longitudinal vortex generators are better than transverse vortex generators in terms of heat transfer enhancement. This is because longitudinal vortex can generate all three mechanisms for heat transfer namely developing boundary layer, swirls and flow destabilization [33]. One of the most recent studies conducted by Modi and Rathod whereby their analysis was focused on heat transfer enhancement and pressure drop for fin-and-circular tube compact heat exchangers with sinusoidal wavy and elliptical curved rectangular winglet vortex generator (RWVG) drop [8]. Based on a previous study by Gholami *et al.,* [9], the author decided to use a 30° attack angle as it gives higher heat transfer performance and moderate pressure drop. Table 5 lists out the numerical studies in the past.

Table 5
Summary of numerical studies

Author	Parameter	Working Conditions	Results
Fin Shape			
Ke *et al.,* [28]	Plain and wavy	$500 \geq Re \geq 1300$ $15° \geq \alpha \geq 75°$ F_p = 3.1mm D_t = 13.75mm P_t = 31.75m P_l = 27.5mmm T_w = 293K	(i) Wavy fin gives a swirling flow compared to plain fin. (ii) Wavy fin gives a higher Nu than plain fin.
Sakib and Al-Faruk [29]	Annular and Rectangular	$4.5 \geq v_{air} \geq 9.5$ D_t = 1.5cm δ = 0.1cm P_t = 3cm P_l = 3cm and 1.5cm (inline and staggered respectively) $T_{in,air}$ = 293.15K T_w= 393.15K	(i) Outlet temperature for both types of fins decrease with the increase in inlet velocity. (ii) The rectangular fin provides higher heat transfer than the annular fin.
Tube Inclination Angle			
Wang *et al.,* [30]	$0° \leq \theta \leq 90°$	F_p = 3.73mm δ = 0.2mm T_w = 373K $T_{in,air}$ = 298K	(i) The j increases gradually from 0° to 30° and decreases from 30° to 90° which is due to more air is blocked as the tube angle increases. (ii) The pressure drop increases as the tube rotation angle increases.
Tang *et al.,* [22]	$30° \leq \theta \leq 90°$	1.4 m/s≥ v_{air} ≥ 4.5m/s δ = 0.35mm Fs = 2.5mm Pt = 26.7mm Pl = 55mm	(i) h for air inlet angle 45° is the highest followed by 60°, 90° and 30°. (ii) f for 30° air inlet angle is the highest followed by 45°, 60° and 90°.
Adam *et al.,* [23]	0°, 30°, 60°, 90°, 120° and 150°.	1.4 m/s≥ v_{air} ≥ 4.5m/s δ = 0.6mm F_s = 20mm P_t = 30mm P_l = 30mm T_w = 100°C $T_{in,air}$ = 27°C	(i) The inclination angle from 0° to 90° and 120° to 150° swirling strength behind the tubes increases and decreases respectively. (ii) 90° inclination angle gives the highest Nusselt number with an enhancement of 41.2%.
Tube Arrangement			
Zeeshan *et al.,* [31]	Inline and Staggered	$400 \geq Re \geq 900$ P_t = 25.4mm P_l = 25.4mm and 22.48mm (for inline and staggered) $T_{in,air}$ = 310.6K T_w = 291.77K δ = 0.18mm	(i) The inline arrangement shows that the heat transfer coefficient increases and pressure drop decrease as the flatness of the tube increases. (ii) For staggered arrangement, heat transfer coefficient and pressure drop increase as the flatness of the tube decreases.
Sakib and Al-Faruk [29]	Inline and Staggered	4.5m/s≥ vair ≥ 9.5m/s P_t = 3cm P_l = 3cm and 1.5cm (for inline and staggered) δ = 0.1cm $T_{in,air}$ = 293.15 K T_w = 393.15K	(i) The staggered arrangement shows a higher heat transfer characteristic than the inline arrangement.

Unger *et al.,* [32]	Inline and staggered	$Re \leq 2300$ P_t = 50mm P_l = 50mm D_t = 27mm	(i) The staggered arrangement gives a higher Nusselt number by 88.5% than the inline arrangement.
Addition of Vortex Generators			
Modi and Rathod [8]	RWVG	400 ≥ Re ≥ 1000 P_t = 25.4mm P_l = 22mm α =30° D_t =10.55mm l_{VG} = 6mm $T_{in,air}$ = 303K T_w = 350K δ_{VG} = 0.2mm	(i) Up configuration gives a higher heat transfer enhancement compared to down configuration. (ii) Wavy-up configuration gives the highest pressure difference with the increase of Reynold's number. (iii) Wavy-up configuration gives the lowest value of (j/f) compared to other configurations. (iv) The curved-down configuration gives the highest value of (j/f).
Xie and Lee [34]	Curved RWVG	P_t = 20.4mm P_l = 20.4mm α =95° F_p= 2.9mm δ = 0.1mm $T_{in,air}$ = 303K T_w = 333K	(i) As the VG's radius increases, the fiction factor also increases significantly. (ii) As the VG's height increases, the friction factor increases significantly. (iii) The author concluded that the optimum winglet parameters are radius ratio = 1.55 and height ratio = 0.8.
Ke *et al.,* [28]	Delta WVG	2m/s ≥ v_{air} ≥ 2.75m/s P_t = 31.75mm P_l = 27.5mm D_t =13.75mm F_p= 3.1mm l_{VG} = 4mm h_{VG} = 2mm	(i) The f increases with the attack angle. (ii) The Nu increases from 15° to 45° attack angle and reduces from 45° to 75°.
Välikangas *et al.,* [35]	Angle RWVG	1354 ≤ Re ≤ 6157 P_t = 25.4mm P_l = 22mm α =30° F_p= 1.81mm δ = 0.115mm	(i) Results for s=0.5D and s=0.375D shows that initially at low velocities, a higher VG design gives a high value of f but as the velocity increases, the f decreases. (ii) In comparison with both geometries, s=0.5D with H=0.6Fp gives the highest volume goodness factor which is 5.23%.
Lotfi *et al.,* [12]	RTW, ARW and CARW	1415 ≥ Re ≥ 7075 P_t = 31.75mm P_l = 27.5mm F_p= 4mm Tw = 293K	(i) Due to the fiction and pressure difference, vortices are formed when the fluid passes the VG. (ii) The Nu of the HE with winglets is higher than the baseline case. (iii) The Nu of HE with RTW, ARW and CARW shows a small difference in their results.
Naik and Tiwari [36]	RWVG	2000 ≥ Re ≥ 10000	(i) The Nu increases as the β increases from 15° to 45° and decreases from 45° to 60°. (ii) The maximum value of Nusselt number obtained for adjacent located RWP which is ΔX = 0 and ΔY = ± 1.5, whereas the maximum value of

			friction factor was observed at $\Delta X = -0.5$ and $\Delta Y = \pm 1.0$.
Hu *et al.,* [37]	Delta WVG, Trapezoid WVG and RWVG	P_t = 20mm P_l = 15.5mm D_t = 10mm α = 35°	(i) The intensity of heat transfer enhancement in the delta winglet is larger than rectangular and trapezoid winglet. (ii) The intensity of the secondary flow produced by the delta winglet is stronger than the other two configurations.
Gupta *et al.,* [38]	RWVG	$1500 \leq Re \leq 9000$ α =45° F_p= 12mm δ = 3mm D_t = 52mm	(i) The effect of orientation for non-punched winglets shows that CFU orientation shows a higher pressure drop compared to CFD orientation. (ii) CFD downstream gives a higher pressure drop than CFD upstream. (iii) CFD orientation shows a better result compared to CFU as it gives a higher *j.* (iv) CFD upstream gives an increase of 17.27% compared to CFD downstream.
Sarangi and Mishra [39]	RWVG	$2\text{m/s} \geq v_{air} \geq 2.75\text{m/s}$ P_t = 25.4mm P_l = 25.4mm α =10° D_t =10.67mm	(i) Winglet at the upstream location for circular tube gives a higher temperature of wake region. (ii) Upstream position winglet gives a higher pressure drop. (iii) The nearer the position of the winglet towards the upstream, the heat transfer performance (*Nu*) and the pressure drop (*f*) decreases.
Air Velocity			
Subhedar *et al.,* [25]	4.06, 5.25, 6.44, 7.63 and 8.82 l/min	δ = 0.00254cm F_s = 0.15875cm H = 0.9cm $T_{in,\,nano}$ = 80°C v_{air} = 1.05 m/s	(i) The *Nu* increases as the flow rate of nanofluid increases.

Based on their results, RWVG for all cases show a steady and stable reduction in the wake region which is behind the tube compared to non-winglet due to delaying of flow separation. Apart from that, the change in temperature for all RWVG are higher compared to non-winglet fin because of the reduction in thermal boundary thickness. Up configuration gives a higher heat transfer enhancement compared to down configuration. In the comparison of pressure drop penalty, all RWVG configuration gives higher pressure drop than non-winglet baseline as Reynold's number increases. Wavy-up configuration gives the highest pressure difference with the increase of Reynold's number. As discussed, the addition of the winglet gives more obstruction which the frictional resistance also increases. Next, their numerical results show that as Reynold's number increases, the Nusselt number also increases for all cases. However, the heat transfer performance for up-configuration is higher than the down configuration. This is because down configuration gives high obstruction and resistance. Their last analysis will be the London area goodness factor (j/f) which the increase in Reynold number gives a lower value of (j/f) for all RWVG cases compared to the non-winglet baseline. Wavy-up configuration gives the lowest value of (j/f) compared to other configurations even though it gives a higher heat transfer rate which being discussed previously and curved down configuration gives the highest value of (j/f). Thus, the author concluded that a curved down configuration is suitable since it gives moderate Nusselt number enhancement and low-pressure drop compared to other cases.

In 2020, research done by Xie and Lee which is flow and heat transfer performance of curved-rectangular vortex generator in a compact fin-tube heat exchanger was investigated. In this research, the author focuses on the thermo-hydraulic performance of a compact fin-tube heat exchanger with different VG height ratio:0.2, 0.4, 0.6,0.8 and radius ratio:1.35, 1.55, 1.75. Based on their results, as the VG's radius increases, the fiction factor also increases significantly. This phenomenon occurs because a larger drag is formed as the recirculation developed. Apart from radius, the height of VG plays a major role in a CHE. It is the same as VG's radius whereas the VG's height increases, the friction factor increases significantly which is due to the flow resistance developed. Based on the investigation, the author concluded that the optimum winglet parameters are radius ratio = 1.55 and height ratio = 0.8. With these parameters, heat transfer performance was enhanced by 1.3-1.5 times better than the baseline [34].

In 2020, Gupta *et al.,* [38] presented both numerical and experimental analysis of punched winglet as vortex generator for performance improvement of a fin-and-tube heat exchanger. For this analysis, a rectangular winglet vortex generator being used with punched and non-punched holes. The Reynolds number varies from 1500-9000 and the angle of attack is set at 45° for optimum heat transfer enhancement [38]. Based on their results, the effect of orientation for non-punched winglet shows that CFU orientation shows a higher pressure drop compared to CFD orientation which it shows an increase by 10.28% while CFD downstream gives a higher pressure drop than CFD upstream. Whereas in comparison with thermal performance, CFD orientation shows a better result compared to CFU as it gives a higher Colburn factor while CFD upstream gives an increase of 17.27% compared to CFD downstream. The air begins to slow down due to the stagnation zone formed caused by the tube and winglet. Besides that, the wake region in CFD is larger than in CFU due to the recirculation formed which enhances the heat transfer.

Kumar and Prasad had investigated the position of RWVG placed on heat transfer performance for a circular and elliptical tube. In this case, common flow up orientation being applied to improve the high-velocity flow and thermal mixing which was also stated by Gupta *et al.,* [38]. The author stated that the winglet at the upstream location for the circular tube gives a higher temperature of the wake region due to its improvement in thermal mixing. To conclude this analysis, the nearer the position of the winglet towards the upstream, the heat transfer performance (Nusselt number) and the pressure drop (friction factor) decreases [39]. In 2019, Ke *et al.,* [28] numerically investigated the effect of delta winglet attack angle (15°, 30°, 45°, 60° and 75°) on the thermal-hydraulic performance of CHE. The addition of the delta winglet increases the blocking area of the flow with the increase of attack angle. Thus, the friction factor increases with the attack angle. It is observed that the Nusselt number increases from 15° to 45° attack angle and reduces from 45° to 75°. This phenomenon is due to the stagnation region formed behind the winglets at a higher attack angle and secondary flow formed with the addition of delta winglets [28].

Välikangas *et al.,* [35] had investigated the dimensions of delta winglet on heat transfer enhancement of fin-and-tube HE. Various of length (s=0.5D and 0.375D) and height (H=0.3Fp, 0.45Fp and 0.6Fp) being used in this study. D and Fp indicate the diameter of tube and fin pitch respectively. Results for s=0.5D show that initially at low velocities, a higher VG design gives a high value of friction factor but as the velocity increases, the friction factor decreases. Whereas for s=0.375D, the results are the same as s=0.5D which are mentioned above. In comparison with both geometries, s=0.5D with H=0.6Fp gives the highest volume goodness factor which is 5.23% [35]. Furthermore, Lotfi *et al.,* [12] had numerically investigated the thermal-hydraulic performance of smooth wavy fin-and-elliptical tube HE with rectangular trapezoidal winglet (RTW), angle rectangular winglet (ARW) and curved angle rectangular winglet (CARW). Due to the fiction and pressure difference, vortices are formed when the fluid passes the VG. Apart from that, the Nusselt number of the HE with winglets is higher than the baseline case. The Nusselt number of HE with RTW, ARW and CARW shows a small difference in their results. This is due to the vortices that create a secondary flow which enhances the heat transfer [12].

The paper done by Naik and Tiwari investigated the effect of attack angle (β) and location of rectangular winglet pair (RWP), which varies from 15° to 60° on fin-tube HE with staggered tube arrangement. Their results show that the Nusselt number increases as the β increases from 15° to 45° and decreases from 45° to 60°. The variance in the trend is due to the dominance of longitudinal and transverse vortices. At lower β, longitudinal vortices are dominant while at higher β, transverse vortices are dominant. Besides that, the maximum value of Nusselt number obtained for adjacent located RWP which is $\Delta X = 0$ and $\Delta Y = \pm 1.5$, whereas the maximum value of friction factor was observed at $\Delta X = -0.5$ and $\Delta Y = \pm 1.0$ [36]. Nevertheless, the upstream position of the winglet gives a higher temperature of the wake region due to its enhancement in thermal mixing which was also claimed by Kumar Sarangi and Prasad Mishra.

The most common VG used as the extended surface of the fin are delta and rectangular winglets. Hu *et al.,* [37] had investigated the effect of delta, trapezoid and rectangular winglet VG on the thermal-hydrodynamic performance of a circular tube bank fin heat exchanger. The intensity of heat transfer enhancement in the delta winglet is larger than rectangular and trapezoid winglet. Furthermore, the author also stated that the intensity of the secondary flow produced by the delta winglet is stronger than the other two configurations. The effect of Reynold's number on the thermal-hydraulic performance of fin-tube HE has been numerically investigated by Naik and Tiwari. Reynold's number varies between 2000-10000. It shows an increment in Nusselt number as Reynold's number increases. The higher value of Reynold's number produces high strength of vortices which gives more disturbance in the thermal and hydraulic boundary layer [40].

4. Conclusion

A review of past research on heat transfer performance of compact heat exchanger has been presented. The studies were mainly focused on the geometric and process parameter namely fin shape, tube angle, arrangement of tube, addition of vortex generator and air flow velocity. The findings shows that air inlet angle of 45° and common flow upstream gives the optimum heat enhancement. Process parameters is one of the key factors that affect the heat transfer performance of compact heat exchanger. The range of airflow is varied between from approximately 1.8 – 3.8 m/s in many cases to investigate the optimum heat transfer. It is interesting that auxiliary equipment was also being considered in many investigations. Vortex generator in the shape of delta winglet was chosen due to stronger intensity of the secondary flow produced compared to the trapezoid and rectangular winglet. For better outcomes, most research recommended the analysis of the attack angle. The location and orientation of the winglet also need to be taken into consideration.

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References

[1] Hwang, Sang Dong, In Hyuk Jang, and Hyung Hee Cho. "Experimental study on flow and local heat/mass transfer characteristics inside corrugated duct." *International Journal of Heat and Fluid Flow* 27, no. 1 (2006): 21-32. https://doi.org/10.1016/j.ijheatfluidflow.2005.07.001

[2] Li, Xiao-wei, Ji-an Meng, and Zhi-xin Li. "An experimental study of the flow and heat transfer between enhanced heat transfer plates for PHEs." *Experimental Thermal and Fluid Science* 34, no. 8 (2010): 1194-1204. https://doi.org/10.1016/j.expthermflusci.2010.04.008

[3] Jeong, Jong Yun, Hi ki Hong, Sun Kuk Kim, and Yong Tae Kang. "Impact of plate design on the performance of welded type plate heat exchangers for sorption cycles." *International Journal of Refrigeration* 32, no. 4 (2009): 705-711. https://doi.org/10.1016/j.ijrefrig.2009.01.028

[4] Fernandes, Carla S., Ricardo P. Dias, João M. Nóbrega, and João M. Maia. "Laminar flow in chevron-type plate heat exchangers: CFD analysis of tortuosity, shape factor and friction factor." *Chemical Engineering and Processing: Process Intensification* 46, no. 9 (2007): 825-833. https://doi.org/10.1016/j.cep.2007.05.011

[5] Kanaris, A. G., A. A. Mouza, and S. V. Paras. "Flow and heat transfer in narrow channels with corrugated walls: a CFD code application." *Chemical Engineering Research and Design* 83, no. 5 (2005): 460-468. https://doi.org/10.1205/cherd.04162

[6] Ngo, Tri Lam, Yasuyoshi Kato, Konstantin Nikitin, and Nobuyoshi Tsuzuki. "New printed circuit heat exchanger with S-shaped fins for hot water supplier." *Experimental Thermal and Fluid Science* 30, no. 8 (2006): 811-819. https://doi.org/10.1016/j.expthermflusci.2006.03.010

[7] Zhao, Zhongchao, Yong Zhang, Xudong Chen, Xiaolong Ma, Shan Yang, and Shilin Li. "Experimental and numerical investigation of thermal-hydraulic performance of supercritical nitrogen in airfoil fin printed circuit heat exchanger." *Applied Thermal Engineering* 168 (2020): 114829. https://doi.org/10.1016/j.applthermaleng.2019.114829

[8] Modi, Ashish J., and Manish K. Rathod. "Comparative study of heat transfer enhancement and pressure drop for fin-and-circular tube compact heat exchangers with sinusoidal wavy and elliptical curved rectangular winglet vortex generator." *International Journal of Heat and Mass Transfer* 141 (2019): 310-326. https://doi.org/10.1016/j.ijheatmasstransfer.2019.06.088

[9] Gholami, A. A., Mazlan A. Wahid, and H. A. Mohammed. "Heat transfer enhancement and pressure drop for fin-and-tube compact heat exchangers with wavy rectangular winglet-type vortex generators." *International Communications in Heat and Mass Transfer* 54 (2014): 132-140. https://doi.org/10.1016/j.icheatmasstransfer.2014.02.016

[10] Okbaz, Abdulkerim, Ali Pınarbaşı, and Ali Bahadır Olcay. "Experimental investigation of effect of different tube row-numbers, fin pitches and operating conditions on thermal and hydraulic performances of louvered and wavy finned heat exchangers." *International Journal of Thermal Sciences* 151 (2020): 106256. https://doi.org/10.1016/j.ijthermalsci.2019.106256

[11] Cengel, Yunus, and John Cimbala. *Fluid mechanics fundamentals and applications*. McGraw Hill, 2013.

[12] Lotfi, Babak, Bengt Sundén, and Qiuwang Wang. "An investigation of the thermo-hydraulic performance of the smooth wavy fin-and-elliptical tube heat exchangers utilizing new type vortex generators." *Applied Energy* 162 (2016): 1282-1302. https://doi.org/10.1016/j.apenergy.2015.07.065

[13] Ammar, Syed Muhammad, and Chan Woo Park. "Validation of the Gnielinski correlation for evaluation of heat transfer coefficient of enhanced tubes by non-linear regression model: An experimental study of absorption refrigeration system." *International Communications in Heat and Mass Transfer* 118 (2020): 104819. https://doi.org/10.1016/j.icheatmasstransfer.2020.104819

[14] Barker, Geoff. "Chapter 18 - Pipe sizing and pressure drop calculations." *The Engineer's Guide to Plant Layout and Piping Design for the Oil and Gas Industries* (2018): 411-472. https://doi.org/10.1016/B978-0-12-814653-8.00018-7

[15] Assefa, K. M., and D. R. Kaushal. "A comparative study of friction factor correlations for high concentrate slurry flow in smooth pipes." *Journal of Hydrology and Hydromechanics* 63, no. 1 (2015): 13-20. https://doi.org/10.1515/johh-2015-0008

[16] Menon, E. Shashi. "Fluid flow in pipes." *Transmission Pipeline Calculations and Simulations Manual* (2015): 149-234. https://doi.org/10.1016/B978-1-85617-830-3.00005-5

[17] Mortean, M. V. V., and M. B. H. Mantelli. "Nusselt number correlation for compact heat exchangers in transition regimes." *Applied Thermal Engineering* 151 (2019): 514-522. https://doi.org/10.1016/j.applthermaleng.2019.02.017

[18] Çengel, Yunus A., Michael A. Boles, and Mehmet Kanoğlu. *Thermodynamics: an engineering approach*. McGraw-Hill Education, 2019.

[19] Çengel, Yunus A., and Afshin Jahanshahi Ghajar. *Heat and mass transfer: fundamentals and applications*. McGraw-Hill Education, 2014.
[20] Bai, Yong, and Qiang Bai. "Chapter 14 - Heat Transfer and Thermal Insulation." *Subsea Engineering Handbook* (2010): 401-450. https://doi.org/10.1016/B978-1-85617-689-7.10014-7
[21] Nourafkan, Ehsan, G. Karimi, and J. Moradgholi. "Experimental study of laminar convective heat transfer and pressure drop of cuprous oxide/water nanofluid inside a circular tube." *Experimental Heat Transfer* 28, no. 1 (2015): 58-68. https://doi.org/10.1080/08916152.2013.803178
[22] Tang, Linghong, Xueping Du, Jie Pan, and Bengt Sundén. "Air inlet angle influence on the air-side heat transfer and flow friction characteristics of a finned oval tube heat exchanger." *International Journal of Heat and Mass Transfer* 145 (2019): 118702. https://doi.org/10.1016/j.ijheatmasstransfer.2019.118702
[23] Adam, A. Y., A. N. Oumer, Azri Alias, M. Ishak, and M. M. Noor. "Investigation of thermal-hydraulic performance in flat tube heat exchangers at various tube inclination angles." *International Journal of Automotive and Mechanical Engineering* 14, no. 3 (2017): 4542-4560. https://doi.org/10.15282/ijame.14.3.2017.12.0359
[24] Phu, Nguyen Minh, and Nguyen Van Hap. "Influence of inlet water temperature on heat transfer and pressure drop of dehumidifying air coil using analytical and experimental methods." *Case Studies in Thermal Engineering* 18 (2020): 100581. https://doi.org/10.1016/j.csite.2019.100581
[25] Subhedar, Dattatraya G., Bharat M. Ramani, and Akhilesh Gupta. "Experimental investigation of heat transfer potential of Al_2O_3/Water-Mono Ethylene Glycol nanofluids as a car radiator coolant." *Case Studies in Thermal Engineering* 11 (2018): 26-34. https://doi.org/10.1016/j.csite.2017.11.009
[26] Selvam, C., D. Mohan Lal, and Sivasankaran Harish. "Enhanced heat transfer performance of an automobile radiator with graphene based suspensions." *Applied Thermal Engineering* 123 (2017): 50-60. https://doi.org/10.1016/j.applthermaleng.2017.05.076
[27] Rai, Praveen Kumar, Anil Kumar, and Anshul Yadav. "Experimental investigation of heat transfer augmentation in automobile radiators using magnesium oxide/distilled water-ethylene glycol based nanofluid." *Materials Today: Proceedings* 24 (2020): 1525-1532. https://doi.org/10.1016/j.matpr.2020.04.472
[28] Ke, Hanbing, Tariq Amin Khan, Wei Li, Yusheng Lin, Zhiwu Ke, Hua Zhu, and Zhengjiang Zhang. "Thermal-hydraulic performance and optimization of attack angle of delta winglets in plain and wavy finned-tube heat exchangers." *Applied Thermal Engineering* 150 (2019): 1054-1065. https://doi.org/10.1016/j.applthermaleng.2019.01.083
[29] Sakib, Shadman, and Abdullah Al-Faruk. "Flow and Thermal Characteristics Analysis of Plate-Finned Tube and Annular-Finned Tube Heat Exchangers fo In-Line and Staggered Configurations." *Mechanics and Mechanical Engineering* 22, no. 4 (2018): 1407-1417. https://doi.org/10.2478/mme-2018-0110
[30] Wang, Pengfei, Jin Jiang, Shunyang Li, Xiangyu Luo, Shaojie Wang, and Wensheng Zhao. "An investigation of influence factor including different tube bundles on inclined elliptical fin-tube heat exchanger." *International Journal of Heat and Mass Transfer* 142 (2019): 118448. https://doi.org/10.1016/j.ijheatmasstransfer.2019.118448
[31] Zeeshan, Mohd, Sujit Nath, and Dipankar Bhanja. "Numerical study to predict optimal configuration of fin and tube compact heat exchanger with various tube shapes and spatial arrangements." *Energy Conversion and Management* 148 (2017): 737-752. https://doi.org/10.1016/j.enconman.2017.06.011
[32] Unger, Sebastian, Eckhard Krepper, Matthias Beyer, and Uwe Hampel. "Numerical optimization of a finned tube bundle heat exchanger arrangement for passive spent fuel pool cooling to ambient air." *Nuclear Engineering and Design* 361 (2020): 110549. https://doi.org/10.1016/j.nucengdes.2020.110549
[33] Song, KeWei, ZhiPeng Xi, Mei Su, LiangChen Wang, Xiang Wu, and LiangBi Wang. "Effect of geometric size of curved delta winglet vortex generators and tube pitch on heat transfer characteristics of fin-tube heat exchanger." *Experimental Thermal and Fluid Science* 82 (2017): 8-18. https://doi.org/10.1016/j.expthermflusci.2016.11.002
[34] Xie, Jinlong, and Hsiao Mun Lee. "Flow and heat transfer performances of directly printed curved-rectangular vortex generators in a compact fin-tube heat exchanger." *Applied Thermal Engineering* 180 (2020): 115830. https://doi.org/10.1016/j.applthermaleng.2020.115830
[35] Välikangas, Turo, Shobhana Singh, Kim Sørensen, and Thomas Condra. "Fin-and-tube heat exchanger enhancement with a combined herringbone and vortex generator design." *International Journal of Heat and Mass Transfer* 118 (2018): 602-616. https://doi.org/10.1016/j.ijheatmasstransfer.2017.11.006
[36] Naik, Hemant, and Shaligram Tiwari. "Numerical investigations on fluid flow and heat transfer characteristics of different locations of winglets mounted in fin-tube heat exchangers." *Thermal Science and Engineering Progress* 22 (2021): 100795. https://doi.org/10.1016/j.tsep.2020.100795
[37] Hu, Wanling, Liangbi Wang, Yong Guan, and Wenju Hu. "The effect of shape of winglet vortex generator on the thermal-hydrodynamic performance of a circular tube bank fin heat exchanger." *Heat and Mass Transfer* 53 (2017): 2961-2973. https://doi.org/10.1007/s00231-017-2042-3
[38] Gupta, Arvind, Aditya Roy, Sachin Gupta, and Munish Gupta. "Numerical investigation towards implementation of punched winglet as vortex generator for performance improvement of a fin-and-tube heat exchanger."

International Journal of Heat and Mass Transfer 149 (2020): 119171. https://doi.org/10.1016/j.ijheatmasstransfer.2019.119171

[39] Sarangi, Shailesh Kumar, and Dipti Prasad Mishra. "Effect of tube shape on thermo-fluid performance of a winglet supported fin-and-tube heat exchanger having staggered tubes." *Materials Today: Proceedings* 41 (2021): 228-232. https://doi.org/10.1016/j.matpr.2020.08.745

[40] Naik, Hemant, and Shaligram Tiwari. "Thermal performance analysis of fin-tube heat exchanger with staggered tube arrangement in presence of rectangular winglet pairs." *International Journal of Thermal Sciences* 161 (2021): 106723. https://doi.org/10.1016/j.ijthermalsci.2020.106723