This paper aims to investigate the cubical fins and the thermal-hydraulic characteristics in a Printed Circuit Heat Exchanger (PCHE). The working fluid is considered a supercritical LNG. The outcomes show that the thermal-hydraulic performance is enhanced using the cubical fins comparing it to the straight channel. The maximum and minimum difference of $Nu/Eu$ between the cubical fins channel and the straight channel were 71.7% and 64.8% respectively. It is noticed that the pressure drop and Heat Transfer coefficient are increased simultaneously with increasing the mass flux. Using Ansys Fluent 15.0, numerical optimization is performed to examine and analyze the influence of the sparser staggered arrangement. The computed results show that this type of arrangement improves the thermal-hydraulic efficiency of the cubical fins in a PCHE. A comparison between the straight fins arrangement and the staggered fins arrangement is reported. It has been noticed that the velocity of the LNG flow in the cubical fins channel is increased along the length of the PCHE, and decreased by increasing the vertical separation $LV$. The side effect of the vertical separation $LV$ on the PCHE's efficiency was more evident than that of the staggered arrangement $LS$. Finally, feasibility study is performed to inspect the power consumption of the new design.

Cite this article as: El-Hakim N, Assaf J, Nehme B, Zeghondy B, Said W, Jelwan J. CFD analysis and heat transfer characteristics of printed circuit heat exchanger. J Ther Eng 2022;8(3):335–348.
heat exchangers using computational fluid dynamics (CFD) and modeling a series of topology enhancements. His model is expressed by a user-defined function to optimize the porous medium having a sequence of high and low porosity. Consequently, the 3D straight fin structure has been diverted and tested for a quick enhancement with constant heat flux. It has been observed that this approach resulted in a drop in fin weight and pressure drop of 23% and 13% respectively while conserving the same heat enrichment. Embracing this technique for the FSRU heat exchanger is ultimate, and constructive features could be exploited to improve the LNG process.

The most important component of the LNG process is the heat exchanger which vaporizes the liquefied natural gas using a specific heating medium such as seawater or other means. This transfer procedure implicates in power production as a direct and/or an indirect contact, where the mechanization of the Heat Transfer follows the geometry of the fins. For an FSRU heat exchanger, LNG is normally heated up using seawater. In such an application, heat exchangers are subject to corrosion due to high salinity water, imposing the plant to a finite life cycle. Relatively, a reduction of rate energy transfer is expected, resulting in an increase in fluid mixing and velocities and initiating a decline of the fluid flow properties where the thermal conductivity and density are radially disoriented.

To conquer the corrosion problem, yet, acquire high energy efficiency, robust and more thermally-conductive materials are desired. Cohen et al. [2], conduct a parametric analysis of a highly thermal conductivity polymer. His application was based on a doubly finned heat exchanger. Cohen observed that heat exchangers made from polymers offer a promising alternative material in designing heat exchangers in seawater cooled comparing it to aluminum heat exchangers. Such heat exchangers offer better corrosion resistance. This observation is affirmed by Zaheed et al. [3], underpinning the prominent use of the polymer film heat exchanger and addressing the disadvantages of the metallic heat exchangers.

Although polymers are recognized to be an effective physical material for thermal conductivity, most of the research which has been compiled from the literature was about the flexibility of the PCHE channels [4, 5]. For instance, Lee et al. [4] conducted a CFD analysis for a PCHE channel made of a thin plate for various thermal performances. Lee's et al. [5] simulation was optimized with a variable Reynolds number of 67 × 10^3 to 280 × 10^3. He found out that reducing the space of Heat Conduction and expanding the overlapping area sandwiched between the hot and cold channels will maximize the Heat Transfer rate. Kwon et al. [6] studied the thermal efficiency of the Printed Circuit Heat Exchanger using mini channels in various cryogenic heat transfer conditions. His experimental results show a percentage of error of 5% for Reynolds numbers ranging from 8500 to 17 × 10^3. This percentage of error is significantly increased by 20% once the Reynolds number is setup between the range of 2100 and 2500. Even though Kwon's work is distinctive and very substantial, the cryogenic application on the PCHE has been shown to obtain axial conduction causing a low thermal performance [7]. Most of the analyses and experiments which have been channeled on PCHE are engaged using helium and water [8, 9, 10]. For instance, Figley et al. [8], tested a PCHE in a high-temperature helium facility and developed a mathematical model. The numerical model was associated with the geometric factors and material properties. The flow conditions were checked through a comparative along with a prognostic investigation to assess the thermal efficiency and the Heat Transfer Performance of the Printed Circuit Heat Exchanger using the cross, parallel, and counter-flow PCHEs. Kim et al. [9] examined the thermal-hydraulic performance of Alloy 800H Printed Circuit Heat Exchanger using the helium test loop under variable mass flow rates ranging from 40 – 100 kg/h, operating under a pressure of 1.5 MPa and 1.9 MPa and an inlet temperature ranging between 25°C – 550°C. Pra et al. [10] conduct a steady-state assessment of a PCHE and examined its thermal-hydraulic performance in helium test ring at a temperature of 510°C. It has been observed that the effectiveness of the PCHE was about 95%. Moreover, a transient assessment was realized to induce the total life cycles in the heat exchanger. The process was repeated 100 times under different temperatures and unconventional channel geometries. Praet al. [10] reported that no trace of fatigue and failure was observed, and the reduction of the high-pressure drop was investigated numerically by performing a direct shape optimization procedure.

While the above sections highlights some of the work done using helium and water, many other researchers yield different approaches in their investigation. For instance, Nikitin et al. [11] investigated experimentally the heat transfer characteristics and the pressure drop features of the PCHE using supercritical water and supercritical CO₂. The experiment was conducted at an inlet temperature of 280°C to 300°C and a pressure of 2.2 to 3.2 MPa respectively. An empirical correlation was achieved to calculate the local heat transfer coefficient and pressure drop factor as a function of the Reynolds number. Moreover, supercritical CO₂ (S-CO₂) was investigated by Jeon et al. [12] using a heterogeneous type of PCHE. The thermal performance was optimized using different channel sizes and spacing between the channels. Jeonet al. [12] reported that the thermal performance is very sensitive to the adjustment of spacing, and the results are affected significantly. However, it has been observed that the hydraulic diameter and the thermal performance remain constant by modifying the channel cross-section. Kim et al. [13] examined the heat transfer and pressure drop characteristics of supercritical CO₂ flow in a PCHE model that incorporates airfoil-shaped fins. A comparative study between airfoil and zigzag PCHE channel configurations showed a significant decrease in pressure...
In a recent study by Zhao et al. [14], a new sort of microchannel PCHE heat exchanger is proposed. Sparser staggered airfoil fin arrangement has been investigated using the supercritical LNG as a working fluid. It has been reported that the thermal-hydraulic performance in airfoil fin PCHE is considerably improved.

In light of the above discussion, most of the proposed PCHE fin-type models are very complex to manufacture and require lots of optimization and predictive analysis to select the NACA number of the airfoil, each of which has its characteristics in terms of quantifying its mass and its energy saving. Hence, the objective of this paper is to thermally conduct a customary type of fin system within a heat exchanger made of steel material and compare it to the results obtained by Zhao et al. [15], the details of which are represented in the following sections.

### System Configuration, Modelling, and Simulation Environment

**Physical Model and Cube Fin Arrangement Parameters**

In this paper, a cube-fin PCHE model was considered. Figures 1 and 2 show the complete configuration of the PCHE model and the fluid passage configuration with a semicircular crossflow at entry and exit regions with a diameter of 1.5 mm. The heat exchanger is made of steel with thermal conductivity of 16.27 W/(m.K) as reported in [15]. The PCHE has a full length of 260 mm, 1 cold and 2 hot plates diffusion bonded in a single banking pattern with 18 cubical fins. Figure 3 shows the cube fins arrangements. For simplicity, 157 periodic cubic fin structures along one channel were selected with a total length of 260 mm. The staggered pitch $L_s$ and the vertical pitch $L_v$ were 0 mm and 1.67 mm respectively. CFD analysis has been carried out using 9 different configurations (refer to table 1 for more details) to acquire a comparative study between different fins arrangements. The flow characteristics and heat transfer of different cubical fins configurations were investigated using six periodic fins (see figure 4 for more details) all along the streampath, and three in the oriented transverse direction. The same length and the same hydraulic diameter were employed for both states. From figure 3, $L_h$ represents the split gap sandwiched between a cube top and adjacent cube top of the same string, and $L_v$ represents the distance separating two vertical rows in the vertical direction. Accordingly, the width ($W$) varied from 3.9 to 6 mm (refer to figure 5 for more details). All configurations are tabulated in table 1.

**Materials Properties and Boundary Conditions**

The performance of any heat exchanger can be optimized by relating the total heat transfer rate $q$, heat transfer surface area $A$, the heat capacity of each fluid $C_p$, overall heat transfer coefficient $U$, and fluid terminal temperatures. The fluid considered in these simulations is supercritical LNG, which thermo-physical properties are reported in [14, 15].

The applied boundary conditions in this paper were 10 MPa and 121 K at the inlet, exceeding its critical pressure of $P_{cr} = 4.59$ MPa, with a mass flux of 325 kg/(m².s). The thermo-physical properties of the supercritical LNG are reported

### Table 1: Different configurations for the cubical fins position

| Configuration(s) | 1   | 2   | 3   | 4   | 5   | 6   | 7   | 8   | 9   |
|------------------|-----|-----|-----|-----|-----|-----|-----|-----|-----|
| $L_v$ (mm)       | 1.3 | 1.3 | 1.3 | 1.67| 1.67| 1.67| 2   | 2   | 2   |
| $L_s$ (mm)       | 0   | 0.4 | 0.8 | 0   | 0.4 | 0.8 | 0   | 0.4 | 0.8 |
| Width (mm)       | 3.9 | 3.9 | 3.9 | 5.01| 5.01| 5.01| 6   | 6   | 6   |
| $L_h$ (mm)       | 1.2 | 1.2 | 1.2 | 1.2 | 1.2 | 1.2 | 1.2 | 1.2 | 1.2 |
| Thickness (mm)   | 0.75| 0.75| 0.75| 0.75| 0.75| 0.75| 0.75| 0.75| 0.75|

**Figure 1:** Complete configuration of the PCHE model.

**Figure 2:** Fluid passage configuration with a semicircular cross flow.
For simplicity, the flow channel is prearranged as periodic in both transverse and longitudinal directions as it is shown in Figure 4.

Supercritical LNG is used as the working fluid and steel is chosen as the material of construction for the plates and fins as well. The outlet boundary condition was defined by a pressure of 0 Pa. The heating medium used was seawater maintained at 298 K with the aid of exhaust fumes and BOG if it is lower than the desired temperature and a mass flow rate of 30 kg/s. Subsequently, the heat provided by the medium over the heat exchanger of 1 m² surface area is:

\[ Q = hA_s(T_s - T_{\infty}) \]  

(1)

\( T_s \) is the surface temperature and it is considered 210 K. The properties of the steel plate are:

\( k = 0.60603 \quad \text{W/m.K}, \quad L = 1 \quad \text{m}, \quad \text{whereas the properties of the seawater at a mean temperature of 298 K are} \quad \rho = 1035 \quad \text{Kg/m}^3 \quad \text{for a salinity concentration of} \quad 0.035 \quad \text{g salt/g water.} \)

This leads to dimensionless numbers \( Pr = 6.13, \quad \theta = 8.927 \times 10^{-6} \quad \text{m}^2/\text{s}. \) The thickness of the water channel is 1 mm, and the width of 1 m. Therefore, the cross-sectional area is equal to 0.001 m². Accordingly, the velocity is calculated as:

\[ V = \frac{m}{\rho \times A_c} \]  

(2)

Hence, the Reynolds number is calculated as:

\[ Re_L = \frac{V \times L}{\rho} \]  

(3)

Consequently, the convective heat transfer \( h \) could be calculated using the following relation:

The Nusselt number is calculated as follows:

\[ Nu = \frac{h \times L}{k} = 0.3387 \times Re_L^\frac{1}{2} \times Pr^\frac{1}{4} = 1116.97 \]  

(4)

Using equations (2), (3), and (4), the velocity is computed as 28.99 m/s, \( Re_L = 3246948.27, \quad Nu = 1116.97, \quad h = 676.91 \quad \text{W/m}^2\text{K}. \) Thus, the rate of heat transfer per unit area is 58891 W. While the left and right walls are considered to be adiabatic, the top and bottom walls are predetermined with a constant heat flux of 58891 W/m² conducted throughout a thickness of 0.5 mm, and an outlet pressure was determined as 0 Pa.

Numerical Approach

ANSYS FLUENT R15.0[17] is used to solve the governing equations for 3D steady flow and heat transfer in the PCHE channel. The governing equations are represented as the following:

\[ \frac{\partial}{\partial t} \rho u + \frac{\partial \rho u^2}{\partial x} + \frac{\partial \rho u v}{\partial y} + \frac{\partial \rho u w}{\partial z} + \nabla = 0 \]  

(5)

\[ \frac{\partial}{\partial t} \rho v + \frac{\partial \rho v^2}{\partial x} + \frac{\partial \rho u v}{\partial y} + \frac{\partial \rho v w}{\partial z} + \nabla = 0 \]  

(6)

\[ \frac{\partial}{\partial t} \rho w + \frac{\partial \rho w^2}{\partial x} + \frac{\partial \rho u w}{\partial y} + \frac{\partial \rho w v}{\partial z} + \nabla = 0 \]  

(7)

Conservation of Mass equation:

\[ \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = \frac{dP}{dt} \]  

(8)
\( \rho \) is the density; \( u, v, w \) are the vector velocities; and \( P \) is the operating pressure. The \( k-\omega \) model is substantially more used than \( k-\varepsilon \)[18]. This model is good to describe the pressure gradient for any application [19]. The mathematical form of this model is:

\[
\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_i} \left( \frac{k}{\sigma_t} \right) + G_k - Y_k + S_k \quad (9)
\]

\[
\frac{\partial (\rho \omega)}{\partial t} + \frac{\partial (\rho \omega u_i)}{\partial x_i} = \frac{\partial}{\partial x_i} \left( \frac{\omega \rho u_i}{\sigma_w} \right) + T_k - \frac{\partial \omega}{\partial x_i} + G_\omega - Y_\omega + S_\omega \quad (10)
\]

The effective diffusivities for the \( k-\omega \) model are given by:

\[
T_k = \mu \frac{\rho u_i}{\sigma_t} \quad (11)
\]

\[
T_\omega = \mu \frac{\rho u_i}{\sigma_w} \quad (12)
\]

From equation 12\( \mu_\omega \) is computed by combining \( k \) and \( \omega \) as follows:

\[
\mu_\omega = \alpha * \frac{\rho u_i}{\omega} \quad (13)
\]

The coefficient \( \alpha^* \) damps the turbulent viscosity causing a low-Reynolds number correction. The SST model is formulated as the following:

\[
\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_i} \left( \frac{k}{\sigma_t} \right) \left( \frac{k}{\sigma_t} \right) + \frac{\partial}{\partial x_i} \left( \frac{\sigma_t}{\sigma_t} \right) \left( \frac{k}{\sigma_t} \right) + G_k - Y_k + S_k \quad (14)
\]

Equation (14) could be re-written in terms of a blending function denoted as:

\[
\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_i} \left( \frac{k}{\sigma_t} \right) + \frac{\partial}{\partial x_i} \left( \frac{\sigma_t}{\sigma_t} \right) + \frac{\partial}{\partial x_i} \left( \frac{\sigma_t}{\sigma_t} \right) + \frac{\partial}{\partial x_i} \left( \frac{\sigma_t}{\sigma_t} \right) + G_k - Y_k + S_k \quad (15)
\]

\( S \) is the invariant strain rate and \( F_i \) is expressed as:

\[
F_i = \tanh \left\{ \min \left[ \max \left( \frac{\sqrt{k}}{\beta* \omega} \frac{500 \nu}{\gamma^2} \right) \right] \right\} \left( \frac{4 \rho \sigma_{\omega \omega}}{CD_{\omega \omega}} \right)^\frac{1}{4} \quad (16)
\]

\[
CD_{\omega \omega} = \max \left( 2 \rho \sigma_{\omega \omega} \frac{1}{\omega} \frac{\partial \omega}{\partial x_i} \frac{\partial \omega}{\partial x_i} \right) \quad (17)
\]

\[
\mu = \frac{a_i k}{\max(a_i, SF_i)} \quad (18)
\]

\[
F_i = \tanh \left\{ \min \left[ \max \left( \frac{2 \sqrt{k}}{\beta* \omega} \frac{500 \nu}{\gamma^2} \right) \right] \right\} \left( \frac{4 \rho \sigma_{\omega \omega}}{CD_{\omega \omega}} \right)^\frac{1}{2} \quad (19)
\]

To prevent the build-up of turbulence in stagnation regions, a production limiter is introduced as:

\[
P_k = \mu \frac{\partial U_i}{\partial x_i} \left( \frac{\partial U_i}{\partial x_i} + \frac{\partial U_j}{\partial x_j} \right) P_k = \min(P_k, 10, \beta* \rho k \omega) \quad (20)
\]

The constants of this model are computed by a blend function given as [18]:

\[
\alpha = \alpha F + \alpha (1 - F) \quad (21)
\]

All constants of the \( k-\omega \), SST are tabulated in table 2. The convective heat transfer coefficient is calculated using the following equation:

\[
h = \frac{q_u}{T_u - T_{bulk}} = \frac{q_u}{(T_u - (T_{in} + T_{out})/2) \quad (22)}
\]

Where \( q_u \) is the heat flux applied on the top and bottom wall, \( T_u \) is the area-averaged wall temperature. \( T_{in}, T_{out} \) and \( T_{bulk} \) are the inlet, outlet, and bulk temperature of LNG, respectively. The Nusselt number is calculated using the following equation:

\[
Nu = \frac{hD_t}{\lambda} \quad (23)
\]

With \( \lambda \) is the LNG thermal conductivity. The Reynolds number is calculated by:

\[
Re = \frac{\nu D_t}{\mu} \quad (24)
\]

**Table 2:** Shear Stress Transport Constants (Menter, 1991).

| Function | \( a_{x_1} \) | \( a_{x_2} \) | \( \beta^* \) | \( \beta_1 \) | \( \beta_2 \) | \( a_1 \) | \( a_2 \) | \( \sigma_{x_1} \) | \( \sigma_{x_2} \) |
|----------|----------------|----------------|--------------|------------|------------|--------|------|-------------|-------------|
| SST      | 0.5            | 0.852          | 0.09         | 3          | 0          | 0.0828 | 5    | 0.44        | 0.85        | 1          |
Where \( \rho \) and \( \mu \) are the density and the dynamic viscosity of LNG, respectively. The total pressure drop between inlet and outlet is the sum of both friction and acceleration effect resulting from density augmentation.

\[
\Delta P = \Delta P_{ac} + \Delta P_{fric} \tag{25}
\]

\[
\Delta P_{ac} = \frac{1}{G} \left( \frac{1}{\rho_n} - \frac{1}{\rho} \right) \tag{26}
\]

\[
\Delta P_{fric} = \frac{2f \ell_n \rho \nu^2}{D_h} \tag{27}
\]

Where \( \Delta P \) is the total pressure drop between inlet and outlet. \( \Delta P_{ac}, \Delta P_{fric} \) are the acceleration and friction pressure drop, respectively. The wall shear stress is described as:

\[
\tau_w = \mu \frac{\partial w}{\partial x} \tag{28}
\]

This could be represented in terms of the mean velocity along the axial direction and the hydraulic diameter along the transverse direction as:

\[
\tau_w \sim \frac{\mu n}{D_h} \tag{29}
\]

Therefore, a relationship could be obtained between the Fanning friction factor and the wall shear stress as:

\[
f = \frac{\tau_w}{2 \rho \nu^2} \tag{30}
\]

In this dimensionless study, the pressure loss coefficient is expressed in terms of the Euler number. It shows the relationship between dynamic velocity head, pressure drop, also the relative momentum loss rate, as follows:

\[
E_u = \frac{\Delta P}{\rho \nu^2} \tag{31}
\]

In a dimensionless study, it is necessary to determine the hydraulic diameter “\( D_h \)” which means the characteristics length of the flow channel. Thus, the cube fin placement is considered periodic, so the hydraulic diameter is calculated using the equations below:

\[
V = (LW - S_t) t \tag{32}
\]

\[
S = 2 \left( \frac{P_t}{2} \right) + 2(L - L_c) t + 2(WL - S_t) \tag{33}
\]

\[
D_h = 4V / S \tag{34}
\]

Where \( S_t \) and \( P_t \) are the top area and the wet perimeter of the fin respectively. \( V \) and \( S \) represent the volume and side surface area of the flow channel, respectively. Table 3 shows the calculated hydraulic diameter \( D_h \) for each configuration.

The accuracy of the results is interrelated to the mesh approach. The inflation method was predetermined by

**Table 3: Hydraulic diameter calculation**

| \( L_v = 1.3 \) mm | \( L_v = 1.67 \) mm | \( L_v = 2 \) mm |
|----------------|----------------|----------------|
| \( L (\text{mm}) \) | 0.4 | 0.8 | 0 | 0.4 | 0.8 | 0 | 0.4 | 0.8 |
| \( W (\text{mm}) \) | 1.3 | 1.3 | 1.3 | 1.67 | 1.67 | 1.67 | 2 | 2 | 2 |
| \( t (\text{mm}) \) | 0.75 | 0.75 | 0.75 | 0.75 | 0.75 | 0.75 | 0.75 | 0.75 | 0.75 |
| \( L_c (\text{mm}) \) | 1.175 | 1.575 | 1.975 | 1.175 | 1.575 | 1.975 | 1.175 | 1.575 | 1.975 |
| \( S_f (\text{mm}^2) \) | 0.32 | 0.32 | 0.32 | 0.32 | 0.32 | 0.32 | 0.32 | 0.32 | 0.32 |
| \( P_t (\text{mm}) \) | 2.263 | 2.263 | 2.263 | 2.263 | 2.263 | 2.263 | 2.263 | 2.263 | 2.263 |
| \( V (\text{mm}^3) \) | 0.54 | 0.93 | 1.32 | 0.762 | 1.263 | 1.764 | 0.96 | 1.56 | 2.16 |
| \( S (\text{mm}^2) \) | 2.575 | 3.614 | 4.655 | 3.167 | 4.503 | 5.839 | 3.695 | 5.295 | 6.895 |
| \( D_h (\text{mm}) \) | 0.839 | 1.029 | 1.134 | 0.963 | 1.122 | 1.209 | 1.039 | 1.179 | 1.253 |

**Figure 6:** Mesh strategy.
increases rapidly in the cube fins channel and reaches three times the velocity of the inlet at the contracted region of the cube fin. This could be explained that the cubical fins play a significant role in the LNG distribution causing a rise in the flow resistance and the heat transfer simultaneously, enhancing the distribution of the LNG from the outset of the cubical fins.

To investigate the effect of the mass flux, six different values were selected for a straight and cube fins PCHE respectively. Figures 9 and 10 display the heat transfer coefficient and the Nusselt number at various mass fluxes respectively. From figure 9, it is clearly shown that the turbulence intensity of the flow increases, this is due to a rise in the mass flux; thus, the heat transfer coefficient and Nusselt number are increased in both cases.

Referring to figure 10, the Nusselt number of the cube fins is higher than that of the straight channel. This could be explained by the fact that the flow and the heat transfer are highly affected by the fins in two different ways: the flow disturbance will increase and the heat transfer area will enlarge. Accordingly, the cube fin PCHE shows an improved thermal and hydraulic performance than that of

setting the thickness of the first boundary layer to 0.01 mm and setting six boundary layers near the bottom and top walls and fin surface described as sensitive areas for flow and heat transfer. The accuracy of the results is highly dependent on the grid density. Thus, an extremely fine grid of 3659194 elements was obtained with a mesh sizing of 0.09 mm (see figure 6). The residual for each parameter was set as $10^{-6}$, and the second-order upwind was used.

RESULTS AND DISCUSSION

Comparison of Straight Channel Pche Vs Cube Fin PCHE

In this study, the flow characteristics of LNG and the heat transfer in the straight channel and cube fin PCHE were numerically investigated with the same hydraulic diameter (0.917 mm). Figure 7 shows the velocity contour in a straight channel and cube fin channel with a mass flux of 325 kg/m²s. When the flow was continuously heated along the channel, the bulk velocity increased due to a drop in its density. However, due to the continuous contraction and expansion in the sectional area of the channel, the velocity

Figure 7: Straight channel velocity contour.

Figure 8: Cube fins channel velocity volume rendering.

Figure 9: Heat transfer coefficient in the function of mass flux.
minimum differences of the ratio between cube fin PCHE and straight channel PCHE were 51.07% and 46.2% subsequently. Noticeably, the cube fin PCHE represents an enriched thermal-hydraulic performance, and the ratio difference between cube fin and straight channel PCHEs increased with increasing mass flux. Consequently, fin arrangement is favored for cube fin PCHE.

**Staggered Pitch L Effect**

Generally, the flow and heat characteristics were affected by the shape of the fins and their arrangement. In this study, the cube fins were first placed in parallel ($L = 0$ mm), and then in a staggered position ($L = 0.4$ and 0.8 mm) along the transverse direction. In cube fin PCHE, the fins have a narrow head and tail suitable for forming a smoother channel when the fins are staggered; also, it prevents the vortex separation due to the streamlined fins, leading to less flow resistance in a staggered position compared to that in a parallel arrangement. Figures 14, 15, and 16 describe the velocity contour of the flow in a cubefin PCHE positioned in parallel and staggered manners for a vertical pitch ($L_v$) of 1.3 mm. At the inlet of the channels, the velocities of flow were relatively small and uniform.
Figure 17 displays the Nusselt number of a cube fin PCHE in parallel ($L_s = 0\ mm$) and staggered positions ($L_s = 0.4\ mm$ and $0.8\ mm$). As expected, better thermal performance is revealed in the case of staggered arrangements. For instance, for $L_v = 1.67\ mm$, the Nusselt number at $L_s = 0\ mm$ varies with that at $L_s = 0.8\ mm$ by 21.9%. Figure 20 shows the $N_u/E_u$ ratio in function of $L_v$. While increasing

in both parallel and staggered arrangements. However, as a result of the continuous contraction and expansion of the sectional area of the flow channel, the velocity becomes uniform and it increased in the flow direction. Thus, the staggered arrangement promoted the formation design by creating a smooth flow channel and keeping the flow field more uniform.
the staggering number \( L_s \), the ratio increased as well. As shown in figure 20, \( N/Eu \) has the lowest value at \( L_s = 0 \) mm, about 38% than that at \( L_s = 0.8 \) mm, indicating that \( L_s = 0.8 \) mm has better pressure drop and heat transfer. In general, staggered arrangement leads to less flow resistance and enhanced thermal-hydraulic performance. However, the pressure drop was more affected to \( L_s \) than heat transfer.

**Vertical Pitch (Lv) Effect**

For constant \( L_h \) (1.2 mm), \( L_v \) directly specifies the width of the flow channel and the density of the fins. So, the consequence of changing \( L_v \) on the thermal and hydraulic performance of a cube, fin was analyzed.

**Figure 17**: Plot of Nusselt number vs. \( L_v \) for different \( L_s \).

**Figure 18**: Plot of pressure drop vs. \( L_v \) at different \( L_s \).

**Figure 19**: Plot of Euler number vs. \( L_v \) at different \( L_s \).

**Figure 20**: Plot of Nu/Eu vs. \( L_v \) at different \( L_s \).

**Figure 21**: Velocity contour for \( L_v = 1.3 \) mm and \( L_s = 0.8 \) mm.
Figures 21-23 show the velocity contour of the flow in the cube fin PCHE disposed at different \( L_v \) while staggered at \( L_s = 0.8 \text{ mm} \). For a small vertical pitch \( L_v \), the flow resistance is depicted to be higher, causing the flow velocity to increase. This is due to the narrow vertical separation \( L_v \) which is controlled by the thin flow channel sectional area. Accordingly, for the same inlet mass flux, the flow velocity increases with decreasing \( L_v \). As an example, the maximum velocities of the flow in the adjoining sectional of the channel were about 0.91 m/s and 2.17 m/s with \( L_v = 2 \text{ mm} \) and \( L_v = 1.3 \text{ mm} \) respectively. Hence, the flow resistance and the turbulence intensity were both locally enhanced.

Figures 24 and 25 describe the variation of the heat transfer coefficient and Nusselt number in the function of \( L_v \) and at different \( L_s \). Setting \( L_s \) unchanged, the convective heat transfer coefficient increased with decreasing \( L_v \), which results in increased flow velocity. Figure 26 shows the effect of \( L_v \) on Euler number. The pressure drop decreased while increasing \( L_v \). So for the same \( L_s \), the Euler number dropped significantly with increasing \( L_v \). For example, Euler number was about 54.8% at \( L_v = 2 \text{ mm} \) comparing it to \( L_v = 1.3 \text{ mm} \) for \( L_s = 0.8 \text{ mm} \). Therefore, reducing \( L_v \) improved the heat transfer but also resulted in an increased pressure drop.

As discussed previously, the ratio \( N_u/E_u \) was adopted to assess the efficiency of the cube fin PCHE (please refer to figure 27 for more details), \( N_u/E_u \) significantly increases as \( L_v \) rises. \( N_u/E_u \) at \( L_v = 2 \text{ mm} \) is about 2 times and 1.16 times comparing it to \( L_v = 1.3 \text{ mm} \) and \( L_v = 1.67 \text{ mm} \) respectively, indicating that the condensed fin arrangement is optimal to enhance the heat transfer rate. At the same time, it is difficult to surpass flow resistance. Accordingly, the fins should be moderately arranged in the cube fin PCHE.

**FEASIBILITY STUDY**

A feasibility study is performed to investigate the power consumption of the new design. Accordingly, the pump power is emphasized and correlated to the pressure drop. It should be noted that for an incompressible fluid with a mass flow rate of \( m \), the power required by an adiabatic pump is defined as:

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**Figure 22:** Velocity contour plot for \( L_v = 1.67 \text{ mm} \) and \( L_s = 0.8 \text{ mm} \).

**Figure 23:** Velocity contour plot for \( L_v = 2 \text{ mm} \) and \( L_s = 0.8 \text{ mm} \).
Table 4 represents the parameters used to compute the power consumption in terms of USD.

Accordingly, table 5 represents a summary of the obtained results. It is clearly shown that once $L_s$ increases the pressure drop decrease. This observation has been predicted for $L_v$ as well, where the pressure drop decrease as $L_v$ increases. Thus, this trend is strictly proportional to the power consumed by the pump, this has been observed in figure 28.

From figure 28, one can predict the cost-effectiveness of the fin arrangements. For $L_v = 1.3 \text{ mm}$, for a variable $L_s$, it can be seen that the cost is extremely high with an optimum result for the heat transfer coefficient rates which enhance a better heat transfer rate. This value starts to decrease gradually for $L_v = 1.67 \text{ mm}$, depicting a value of $h = 5277 \frac{W}{m^2K}$, indicating a lower cost for comparing it to $L_v = 1.3 \text{ mm}$. Thus, for $L_v = 1.3 \text{ mm}$, it is clearly shown that the $h$ is most of the time constant between $L_s = 0$ and $L_s = 0.4 \text{ mm}$, corresponding to the lowest cost which is equivalent to 0.0018 USD at $L_v = 0.4 \text{ mm}$. It should be noted that this value could be marginal since the lowest point is represented at $L_v = 1.67 \text{ mm}$ and $L_s = 0.8 \text{ mm}$. However, for this particular case $h$ has been perceived to decline, which is not the case of an efficient and well-enriched system performance. Finally, the predicted values for $L_v = 2 \text{ mm}$ is depicted to have the lowest heat transfer coefficient in all cases. Hence, this value is not considered any design point. It should be noted that this cost study, has been performed with a singlechannel within one plate. This study has been performed on 18 fins (6 in the longitudinal direction and 3

Table 4: Parameters to simulate the cost

| Parameter            | Value   |
|----------------------|---------|
| KWh ($)              | 0.091   |
| Mass flux (kg/m².s)  | 325     |
| Density (kg/m³)      | 447     |
| Pump efficiency      | 0.9     |
| Time (hr)            | 24      |

\[
P = \frac{1}{\eta_p} \frac{\dot{m}}{\rho} \Delta p
\] (35)
in the transverse direction), with periodic boundary conditions on the left and right hand of the channel as well as for the top and bottom.

To sum up, the relatively highest heat transfer coefficient that corresponds to the lowest pumping cost is achieved for $L_v = 1.67$ mm and $L_s = 0.4$ mm. This configuration corresponds to the optimal design of the PCHE.

**CONCLUSION AND DESIGN OUTCOMES**

This paper investigated the cubical fins in a PCHE and the thermal-hydraulic characteristics considering the working fluid as a supercritical LNG. It has been shown that the turbulence model SSTK-ω predicts very well the temperatures and the pressure drop. The obtained results show that the cube fins improved the thermal-hydraulic performance comparing it to the straight channel using LNG as a working medium. The maximum and minimum difference of $Nu/Eu$ between the cube fin channel and the straight channel was 71.7 % and 64.8 % respectively. The pressure drop and heat transfer coefficient were raised in both PCHEs with increasing mass flux. Comparing the straight fin arrangements to the staggered fin arrangement, it is clearly shown that better thermal-hydraulic performance is acquired than the parallel arrangement. This has been observed for the same $L_h$ and $L_v$, the cube fins arranged at $L_s = 0.8$mm presented a better thermal-hydraulic performance than those situated at other $L_s$. The velocity of LNG flow in the cube fin channel increased along the length and decreased with increasing $L_v$. The effect of vertical separation $L_v$ on the PCHE’s performance was more evident than that of staggered arrangement $L_s$. Regarding the analysis of the pressure drop and the heat transfer coefficient, the sparser staggered arrangement can enhance the thermal-hydraulic performance of the cube fin PCHE.

**AUTHORSHIP CONTRIBUTIONS**

Authors equally contributed to this work.

**DATA AVAILABILITY STATEMENT**

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

**CONFLICT OF INTEREST**

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

**ETHICS**

There are no ethical issues with the publication of this manuscript.
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