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Investigation of Thermal-Hydraulic Characteristics of Pillow Plate Heat Exchangers Using CFD

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ABSTRACT

The compactness and desirable thermal characteristics of plate heat exchangers (PHXs) have made them a strong competing heat exchanger technology in the heating, ventilating, air conditioning and refrigeration (HVACR) industry. The miniaturization of plate heat exchangers has become a focal point of attention in recent research. It is desirable to utilize less material and refrigerant charge to obtain the same heat transfer performance. Pillow plate heat exchangers (PPHXs) consist of wavy plates that are welded together with a certain pattern using spot welding, sealed at the edges, and then inflated in a hydroforming process. The complex wavy structure of the pillow plates creates an excellent heat transfer medium with a fully developed turbulent flow between the plates. Thus, PPHXs are used in various single-phase as well as two-phase applications in the industry. This paper presents an investigation of the effect of critical geometrical parameters and flow conditions, on the thermal-hydraulic performance of PPHXs. The pillow surface is created using ANSYS structure simulation resembling the actual manufacturing process. The flow between two adjacent pillow plates is then investigated using Computational Fluid Dynamics (CFD) in ANSYS Fluent. The post-processed data from the CFD simulations is used to run an optimization study to maximize the heat transfer coefficient and minimize the pressure drop. The preliminary results show that the heat transfer coefficient can be up to 3 times higher than the selected baseline while the pressure drop can be reduced by 30%.

1. INTRODUCTION

PHXs positioned very well in the HVACR industry due to their favorable characteristics such as compactness, and high heat transfer effectiveness. However, the computational modeling of PHXs is highly challenging due to the complexity of its geometry. Furthermore, the availability of heat transfer coefficient and pressure drop correlations for PHXs are limited especially for two-phase flow (Eldeeb et al., 2016). In order to design more efficient PHXs, reliable design and modeling methods are required. The development of reliable design and modeling methods for PHXs is a continuous effort.

PPHX is a type of PHXs that has great potential to outperform existing PHXs types such as chevron PHXs. This is due to their compactness, easier manufacturing process, lower capital cost, and high approach temperature. The miniaturization of PHXs is highly desired in order to reduce the material used and the refrigerant charge to reduce the negative environmental impact while achieving the same performance. This can be achieved by optimizing the performance of PPHXs.

PPHXs are manufactured by a hydroforming process. The process consists of two thin metal sheets that are first spot welded together and then sealed at the edges by seam welding. The pattern of the spot weld highly affects the thermal-hydraulic performance of the PPHX. The pattern can be inline or staggered. The plates are then inflated using a hydroforming process creating the pillow shape. The inflated plates are then stacked together to form the channels of the PPHX. The thermal-hydraulic performance of the PPHX is affected by the longitudinal and transverse pitches between the spot welds, the shape of the weld, the diameter of the spot weld, the thickness of the plate, and the height of the pillow. Since the structure of the PPHX is fully welded, it has high structure stability and a sealed construction which is very favorable. There is also a great flexibility in the design of the plate geometry since the weld pattern and of the plate. The geometry of the plate can be varied easily unlike chevron plate, for example, where every new design might need a special die to manufacture.

Although PPHXs are used by the HVACR industry, very limited research has been done on this type of heat exchangers. PPHXs were initially developed for the dairy industry about two decades ago and then appeared in the HVACR industry more recently. Mitrovic and Peterson (2007) claim to be the first in literature to study the single phase and condensation heat transfer and pressure drop in what they called thermoplate, which has the same manufacturing process and geometric characteristics as pillow plates. They used isopropanol as the working fluid in their study. They developed heat transfer and pressure drop correlations based on their experimental results. The correlations developed are only valid for isopropanol and for the range of parameters specific to their experiments as they noted. Mitrovic and Maletic (2011) performed numerical simulations using CFD in the Reynolds number range of 50-3800 for which they developed and proposed heat transfer correlation. The CFD simulations used a laminar flow model although the Reynolds range covered part of the turbulent region which led to the underestimation of the heat transfer rate and pressure drop compared to their experimental results. Later Maletic (2009) improved this assumption by using a $k-\varepsilon$ turbulence model. However, the approximation of the pillow plate surface by trigonometric function in this study resulted in inaccurate areas as mentioned by Piper et al. (2015).

More recently, Piper et al. (2015) developed an alternative approach to determine the geometric characteristics of PPHXs based on numerical forming simulations. Piper et al. (2015) claim that their approach is flexible, and well predicts the actual hydroforming process during the manufacture of the PPHXs. They developed correlations to calculate the pillow plate channel volumetric mean hydraulic diameter, wetted heat transfer area, channel cross-section area, and channel volume. However, these parameters were employed only for few geometries in their studies. Although, as an initial attempt to calculate PPHXs geometric parameters, the correlations developed by Piper et al. (2015) can be useful, but as mentioned in their work and also as will be shown in this study, more accurate design methods are required for a detailed design of a PPHX in order to reduce the design uncertainty.

Piper et al. (2016) performed a comprehensive CFD study using forming simulations to obtain the geometry and using a turbulent single-phase flow in PPHXs. The Reynolds number in their study ranged from 1000-8000. They defined an efficiency of the PPHX based on the total heat transfer divided by the total pumping power required by the PPHX. They concluded that this efficiency is higher with lower Reynolds number, higher pillow height, and with the transverse pattern in which the transverse pitch of the welding spot pattern is greater than the longitudinal pitch. They also showed that the shape of the welding can greatly affect the performance of the PPHX such that using a smaller diameter for the spot weld, or using a more streamlined shape such as an oval shape can greatly reduce the pumping power required and thus enhancing the PPHX efficiency. However, since the oval weld has a greater surface area than the circular weld, the total heat transfer area was reduced.

Since PPHXs are in a very early stage of research and development, this study aims at exploring the potential of further optimization of their thermal-hydraulic performance. In this study, a comprehensive CFD study is performed using ANSYS structure simulations linked with ANSYS Fluent. The geometry of the resulting pillow plates are compared against the correlations developed by Piper et al. (2015). CFD simulations are performed on single-phase flow using water. One set of CFD simulations is used to develop a metamodel which is used to solve an optimization problem to demonstrate the size of potential improvement possessed in PPHXs. The results show up to 3 times improvement in the heat transfer coefficient and 30% reduction in pressure drop per unit length relative to the selected baseline PPHX.

2. PPHX GEOMETRY

The geometry of PPHX is a complex 3D geometry that requires accurate prediction. One way to predict the PPHX geometry is to perform forming simulations as denoted by Piper et al. (2015). The pillow surface in this study is attained by simulating two thin metal plates made of stainless steel of material 1.4541 (AISI 321), bonded together at the welding spots, and undergoing a hydroforming process using ANSYS 16.2 structure simulation. Figure 1 shows the result of the forming simulation of half of the smallest cell of a pillow plate of one of the cases studied. The geometric parameters are defined similar to Piper el at. (2015) as shown in Figure 1. The diameter of the spot weld in given by d_{sp} , the height of the pillow is given by h_i , and s_T , and $2s_L$ are the transverse and longitudinal pitches of the smallest cell, respectively.

An accurate reliable design method is required for PPHXs. However, numerical simulations can have several advantages. First, the resemblance of the actual manufacturing process gives an acceptable initial prediction of the geometry. Second, it allows for stress analysis and shows if the maximum stress or even failure is reached at any part of the pillow plate especially at the welding spots. The area surrounding the welding spot is the most vulnerable to failure even before attaining the maximum stress due to the necking of the metal sheet as shown by Piper et al. (2015). Thus, numerical simulation can give a very informative initial insight of the design of the PPHX. However, it cannot be an accurate reliable design method due to the many uncertainties embedded.

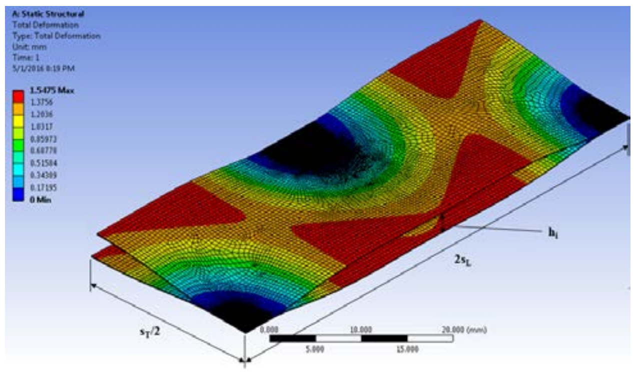


Figure 1: Half of a cell of a “conventional” PPHX geometry simulated using forming simulation.

Figure 2 shows the heat transfer area of the PPHXs simulated in this study by structure simulation plotted against the area calculated using correlations from Piper et al. (2015) which was developed using forming simulations using another commercial finite element tool.

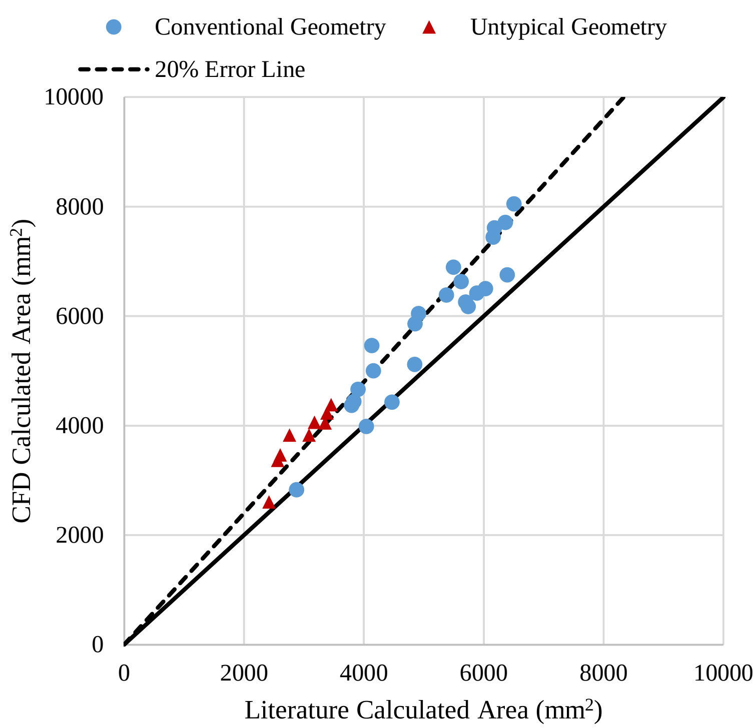


Figure 2: Heat transfer area calculated using forming simulations compared to heat transfer area calculated using correlations from Piper et al. (2015).

Two sets of data are plotted in Figure 2. The first set is for a conventional PPHX geometry as that shown in Figure 1, while the second geometry is what was called "untypical" geometry by Piper et al. (2015) and an example is shown in Figure 3. Although some of the predicted heat transfer areas using the correlations are very well predicted within less than 5%, as shown in Figure 2, the correlations greatly over predict the heat transfer area for both types of geometry up to about 40%.

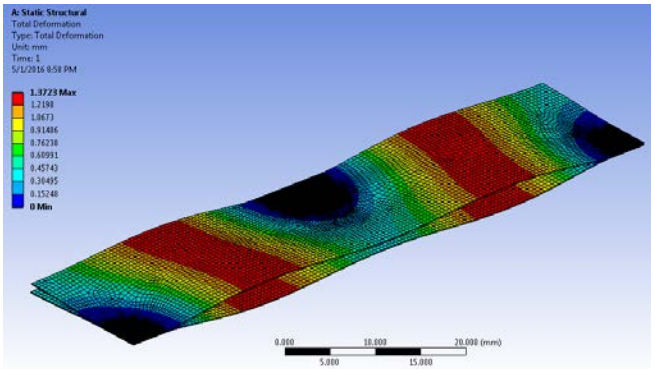


Figure 3: Half of a cell of an "untypical" PPHX geometry simulated using forming simulation.

Figure 4 shows the volume of the channel calculated using forming simulation against that calculated using correlations from Piper et al. (2015). Unlike the heat transfer area, the volume is very well predicted within 5% but only for the conventional geometry. For the untypical geometry, the volume is greatly under predicted by up to 40%.

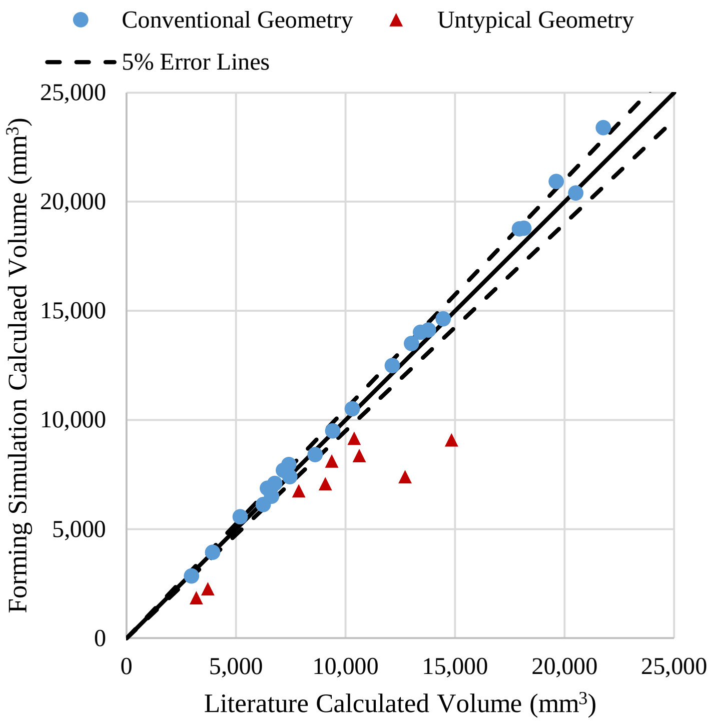


Figure 4: Volume calculated using forming simulations compared to volume calculated using correlations from Piper et al. (2015).

3. CFD SIMULATIONS

The CFD simulations are performed using ANSYS 16.2 Fluent, which is based on finite volume method. An example of the computational domain obtained using forming simulation is shown in Figure 5. A no-slip boundary condition (u=0) is applied and constant wall temperature. Single phase, incompressible, turbulent, steady-state water flow is studied. The turbulence is studied using the realizable $k-\varepsilon$ model (Shih et al., 1995) available in ANSYS Fluent 16.2. The set of PPHXs used in this study were generated using Latin Hypercube Sampling (LHS) with three design variables, namely, diameter of spot weld, d_{sp} $(3.0-10.0\ \text{mm})$, pillow height, h_i $(3.0\text{-}12.0\ mm)$, and the ratio of weld pitches, $s_T/2s_L$ $(0.4\text{-}1.0)$ The generated parameters are shown in Table 1, except for case 32 which is the baseline case corresponding to one of the geometries studied in Piper et al. (2016).

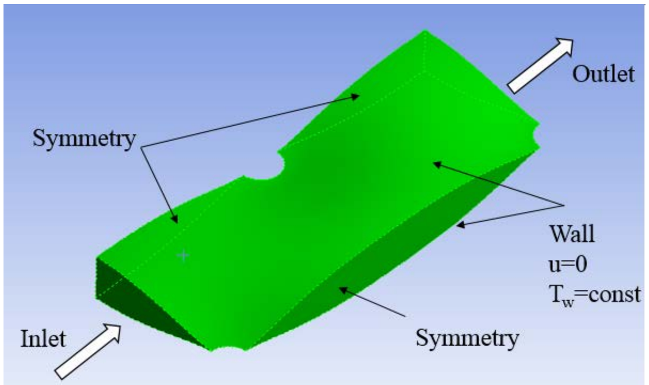


Figure 5: Computational domain of the cell of a pillow plate.

The Reynolds number in this study is defined by the following equation

$$\mathrm{Re} = \frac{ud_h}{\nu},\ where\ \mathrm{d}_h = \frac{4V_i}{A_{w,i}} \tag{1}$$

The heat transfer coefficient is calculated using the logarithmic mean temperature difference

$$HTC = \frac{Q}{A_{w,i}LMTD} \tag{2}$$

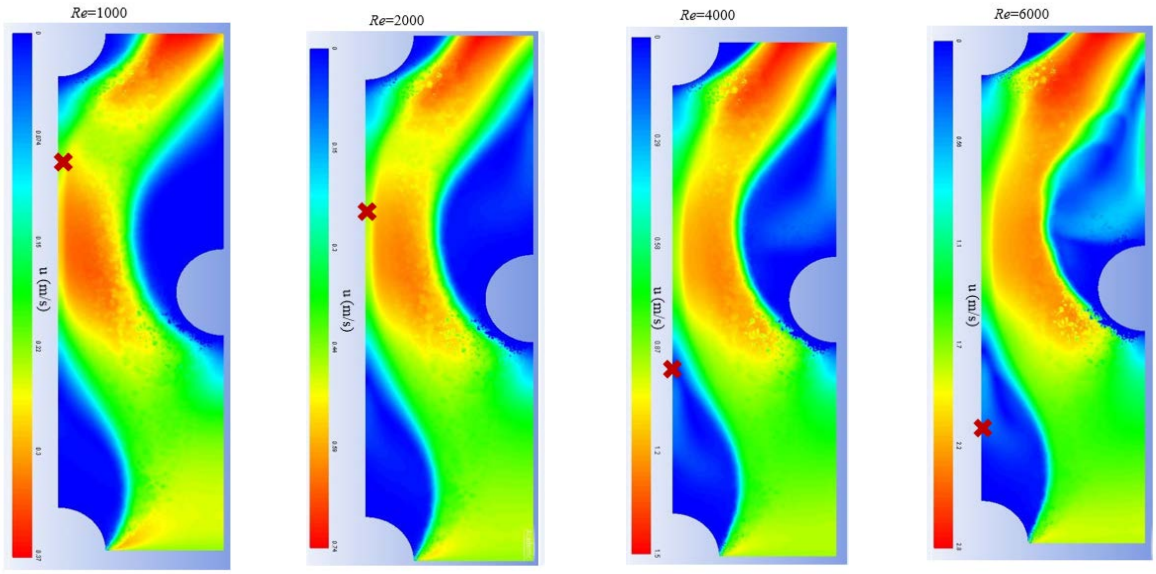
The friction factor is calculated using the following equation

$$f = \frac{d_h}{2\rho u^2}\left(\frac{\Delta P}{L}\right) \tag{3}$$

Table 1: Investigated pillow plate geometries.

Case	$s_T / 2s_L$	h_i (mm)	d_{sp} (mm)	d_h
1	0.7	7.5	6.5	9.5
2	0.4	3.0	3.0	2.7
3	0.4	3.0	10	2.9
4	0.4	12	3.0	9.5
5	0.4	12	10	8.6
6	1	3.0	3.0	3.3
7	0.65	5.6	10	7.5
8	1	12	3.0	11.6
9	1	12	10	10.6
10	0.63	11.2	3.9	10.7
11	0.77	9.7	6.6	9.4
12	0.90	4.0	3.5	4.5
13	0.54	8.9	8.9	8.4
14	0.49	8.3	5.4	8.3
15	0.84	8.5	6.1	8.5
16	0.65	7.0	6.3	7.6
17	0.78	3.1	9.0	3.1
18	0.62	6.9	8.9	7.2
19	0.92	4.3	3.6	4.8
20	0.89	4.5	3.5	5.1
21	0.76	8.4	6.1	8.5
22	0.59	4.9	3.1	5.6
23	0.52	6.2	4.6	6.7
24	0.52	7.2	4.0	7.7
25	0.88	8.5	5.8	8.5
26	0.85	11.7	6.5	10.9
27	0.95	11.1	4.0	11.0
28	0.49	7.8	9.7	7.4
29	0.60	5.1	4.0	5.9
30	0.96	10.3	5.8	10.1
31	0.95	3.9	5.6	4.4
32	0.58	3.0	10	4.1

Figure 6 shows how the flow pattern changes with increasing Reynolds number. As it was shown by Piper et al. (2016), as the Reynolds number increases, recirculation takes place with a larger effect around the welding spots. At Re=6000, the recirculation zone around the welding spot is more significant and extends further to the next welding spot as shown in Figure 6 by the red cross mark. Figure 7 shows the flow through a cell from within the PPHX showing the wake region behind the welding spot.



(a) (b) (c) (d)

Figure 6: The developed velocity field for (a) Re=1000, (b) Re=2000, (c) Re=4000, and (d) Re=6000, for case 32, the red cross mark is the extension of the recirculation zone along the length.

The flow through the PPHX is thus highly dependent on the diameter and shape of the weld, as well as the pillow height, and the spot weld pitches. These parameters highly affect the thermal-hydraulic characteristics of PPHX and are where the improvement of PPHX performance can start. For example, Piper et al. (2016) studied oval shaped welds and showed that the thermal-hydraulic characteristics becomes more favorable than circular welds.

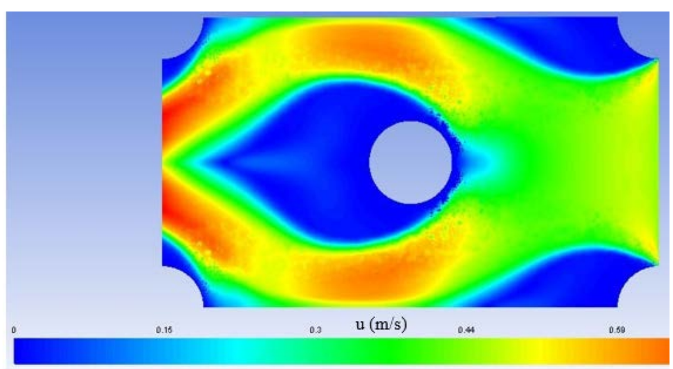


Figure 7: The flow pattern in a full cell of the PPHX.

4. OPTIMIZATION

The CFD simulations in this study are utilized to evaluate the turbulent flow thermal-hydraulic characteristic in PPHXs with different geometry parameters. It is desired to show the potential improvement in PPHX design in order to maximize heat transfer. A Design of Experiments (DOE) consisting of the cases presented in Table 1 with the three design variables previously mentioned is computed using LHS algorithm. Using the post-processed results from the CFD simulations, a metamodel is developed as an approximation to the CFD simulations to predict the PPHX performance. This saves computational time with an acceptable accuracy. The computational time of this Approximation Assisted Optimization (AAO) technique has a fraction of computational cost compared to CFD.

The baseline PPHX used for comparison in this study is picked as one of the designs in Piper et al. (2016). However, it should be noted that this design is not the optimum design developed in their work, their optimum design involved an oval shaped weld which is out of the scope of the current work. The geometry of the baseline is given in Table 1 as case 32. The optimization problem is given by

$$\begin{aligned} &\max\ HTC \\ &\min\ \Delta P / L \\ &s.t. \begin{cases} 0.4 \le \dfrac{s_T}{2 s_L} \le 1.0 \\ 3.0 \le h_i \le 12.0 \\ 3.0 \le d_{sp} \le 10.0 \end{cases} \end{aligned} \tag{4}$$

Two sets of results are presented in Figure 8 and Figure 9. In Figure 8, the Pareto Front of the heat transfer coefficient is plotted against the pressure drop per unit length of the PPHX. Figure 8 shows a maximum improvement of about 3 times the baseline heat transfer coefficient with 30% less pressure drop per unit length (0.7 of the baseline pressure drop). It also shows a maximum improvement of 70% in pressure drop per unit length (0.3 of the baseline pressure drop) with the heat transfer coefficient about 1.6 times better than the baseline.

In Figure 9, Nusselt number is plotted against the friction factor. The maximum improvement in Nu is about ten times the baseline with 2.2 times the friction factor, while the maximum improvement in the friction factor is 0.6 the baseline (40% reduction) with 1.5 improvement in the heat transfer coefficient. Generally, the results suggest that there is great potential for improving the geometric characteristics of PPHXs for maximum performance. More design variables must be considered in the optimization problem as it could be more constrained or otherwise shows more favorable performance. An example is the shape of the weld, and the velocity of the fluid studied.

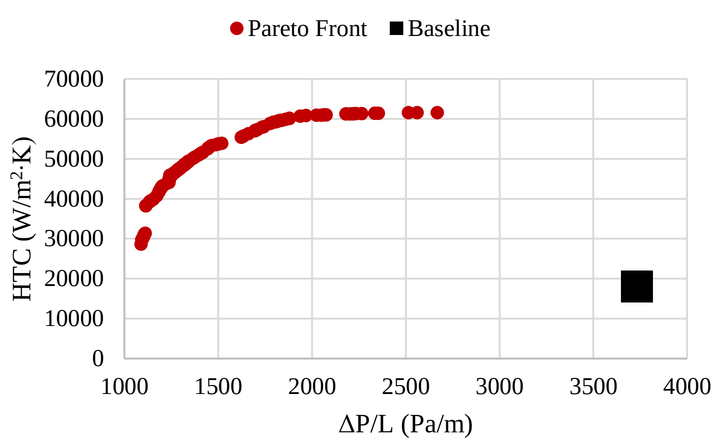


Figure 8: Optimization results of heat transfer coefficient with pressure drop per unit length.

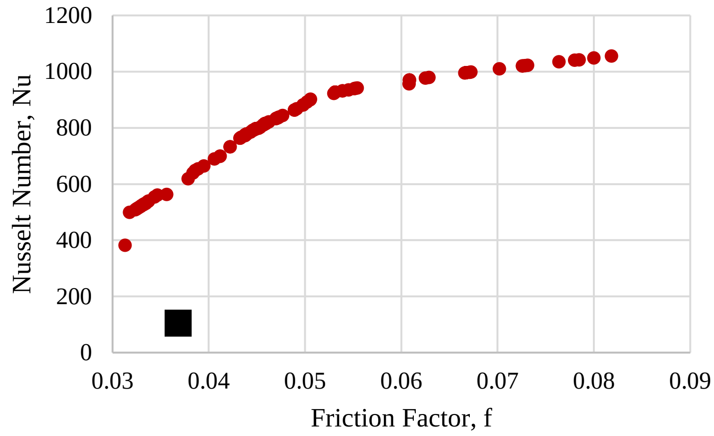


Figure 9: Optimization results of Nusselt number with friction factor.

5. CONCLUSIONS

This study is a continuous effort to study PPHXs which is very limited in literature. The growing favorability of PHXs in general, and PPHXs specifically in the HVACR industry due to their compactness and high heat transfer effectiveness makes them excellent candidates for different processes. PPHXs show great potential to be used due to their easy manufacturing, low capital cost, and more favorable heat transfer characteristics. The geometry of the PPHX is generated using ANSYS structure analysis. The generated geometry is then utilized in ANSYS Fluent to perform CFD simulations on a set of different geometries generated using LHS algorithm.

The CFD simulations show that the PPHX thermal-hydraulic performance is greatly dependent on the geometric characteristics of the PPHX in addition to the thermal characteristics of the materials and fluid used. The welding shape and size, the height of the pillow, and the pitches between the welds are crucial parameters that highly control the performance of PPHXs. The post-processed results from the CFD simulations are utilized to create a metamodel which is used for a simple AAO with three design variables.

The preliminary results presented in this study show that PPHX have a great potential for improvement and requires further investigation. This is the first study to perform geometrical optimization using AAO technique to further expand the improvement of PPHXs thermal-hydraulic performance. The results show up to 3 times the heat transfer coefficient with 30% less pressure drop per unit length, and up to 10 times increase in Nusselt number with 2.2 times increase in the friction factor can be achieved. Extended studies will be done to investigate the effect of flow characteristic and further geometrical parameters such as the weld shape.

NOMENCLATURE

A	area	(m^2)
d	diameter	(m)
f	friction factor	(–)
h	height	(m)
HTC	heat transfer coefficient	($W/m^2{\cdot}K$)
L	length in the direction of flow	(m)
LMTD	logarithmic mean temperature difference	(K)
Nu	Nusselt number	(–)
P	pressure	(Pa)
Q	total heat transfer	(W)
Re	Reynolds number	(–)
u	velocity	(m/s)

s	welding spot pitch	(m)
V	volume	(m^3)

Greek Symbols

Δ	difference	(–)
ρ	density	(kg/m^3)
ν	kinematic viscosity	(m^2/s)

Subscript

i	inner
L	longitudinal
h	hydraulic
sp	spot weld
T	transverse
w	wetted

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