A Water Cooled Condenser with Nanostructured Surfaces for Energy Efficient Air Conditioners

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Abstract:

Water cooling has been proven to be more effective than air-fin cooling thanks to the property of water. Water with high specific heat capacity is able to transfer substantial amounts of energy than air. Replacing the air-fin condenser with plate heat exchangers can provide a more efficient way to dissipate heat from the refrigerant. Different factors such as flow arrangements and flow rate also play an important role in determining the performance of the heat exchanger. Therefore, with the aid of Ansys Fluent software, different flow rates and flow arrangements were simulated case by case. It was found out that the counter flow rate and high flow rate did increase the heat transfer coefficient of the plate heat exchanger. To further increase the performance of the air-conditioner, nanoparticles, Ceo2, are also used. At the beginning of the proposal, it was suggested that the nanoparticles will be mixed with water. However, it was not feasible to put nanoparticles into water since the water was only for one-time purposes. After injecting nanoparticles into the refrigerant, the coefficient of performance of the air conditioner was improved.

Keywords: Plate heat exchanger, Cerium(IV) oxide, Ansys, Refrigerant R410A

Introduction:

The aim of this report is to review the progress and interpret results of the project, A Water-Cooled Condenser with Nanostructured Surfaces for Energy Efficient Air Conditioners, which puts emphasis on the mechanism of heat transfer and nanotechnology through the simulation software and experimental research. In the era of sustainability, this report will briefly conduct a practical prediction of the potential energy saving for the implementation of nanotechnology in water-cooled air conditioners. According to the real operational principle, the heat rejection system is involved with thermodynamic principles and heat transfer between fluids. The process of thermal convection can be analyzed with the net difference between the initial and final states by the heat flux, heat transfer coefficient and fluid flow. With an overview of current nanotechnologies, the application of nanofluids is considered the most desired and effective way to enhance the overall efficiency in thermal fluid systems.

Aims and Objectives

The main objective is to achieve the goal of promoting energy efficient air conditioners with the integration of knowledge and skills acquired through our study. To redesign the original air conditioning technologies, our teammates strived to study the working principle of air conditioning mechanism and therefore identify the main function of major parts in air conditioners which includes cooler, heater, condenser and vaporizer. With a better understanding of heat transfer characteristics, it gives rise to hope for the identification of innovative solutions to redefine the future of air conditioning and revolutionize the conventional structure.

In addition, application of nanotechnologies is another important goal which optimizes the energy system in air conditioners. In terms of hydrodynamic performance analysis, nanotechnology plays a prominent role in determining the thermo-physical properties of refrigerants. Consequently, the type of nanoparticles and their volume concentration are our key determining factors to enhance the performance of heat transfer in plate heat exchangers. Under this circumstance, our team has dedicated much effort to summarize the correlation between the heat transfer coefficient ratio with different concentration of nanoparticles at different temperatures and flow rate.

Scope

Methodology

The type of plate heat exchanger used in the project is the brazed plate heat exchanger. The heat exchange plate is combined with two types of plate. Each plate provides the individual channels for the water and refrigerant flow and prevents contamination of the fluid. Copper foil and stainless-steel foil were combined by a butt joint. It allows the two pieces of material joined at the ends without shaping. The heat exchange plate is manufactured by hydraulic press with a complete auto-production line undergoing metal cutting and plate pressing. The nozzles of the channel plate were then generated by punching. The different types of plate are soldering with the copper foil in the vacuum environment under high temperature after assembling in an alternative pattern where soldering temperature used to be under 800°C. The chevron angle suggested in $\beta \sim 60°/60°$ in PHE indicates reduction in irreversible heat transfer due to frictional losses contributing to better qualitative response in cooling performance [1].

PHE used in the project consists of 16 layers of Grade 316 stainless steel channel plates with thickness 2.7mm, 17 pieces of copper foils for soldering, back and front cover plates to allow the inlet and outlet of water and refrigerant inside the plate heat exchanger.The plate was then assembled with an alternative AB pattern and soldered by copper foil. Compression was required before the soldering process to fix the relative position and combine the frame plate. The assembled plates were experienced brazing under a vacuum and high temperature environment. Several tests, including leakage test and pressure test, were conducted to ensure the quality of the product. Thus, the brazed plate heat exchanger was allowed for further installation in the refrigeration system, mentioned in other sessions. The counterflow strategy is suggested as the flow principle in a brazed plate heat exchanger for refrigeration applications.

With the aid of Ansys Fluent simulation software, the fluid flow inside the brazed plate heat exchanger can be easily analyzed. The software integrates with a broad range of physical models that is necessary for modeling heat transfer, turbulent flow and other related physical phenomena. This has always offered insightful ideas for engineers to design and optimize new equipment.

Throughout the simulation, three different models were used. The first two models were not able to give accurate results. The following parts will discuss the problem of the models.

The experimental setup consists of two parts – a test unit and the testing device which referred to the Jockey Club Controlled Environment Test Facility. The test unit is the adjusted air conditioner. Test unit consists of a condenser, expansion valve, evaporator and compressor. In this project the condenser was replaced from a condenser coil to a brazed plate heat exchanger that allows water as the cooling media instead of air cooling. Moreover, the adjustable expansion valve was installed in the system for adjusting the capillary tube. Since there are no external cooling devices for the water, the open water loop system was applied to the plate heat exchanger. These allow the stable cooling water temperature and prevent the temperature change of water affecting the experimental result. Meanwhile, the other side of the plate heat exchanger formed a closed loop with the refrigeration system. The refrigerant flows in the opposite direction from the cooling water. The counterflow principle was represented in Figure 16. Since the parallel flow principle would be checked in the experiment, the position of two plastic pipes was changed and fixed with the saddle clamp

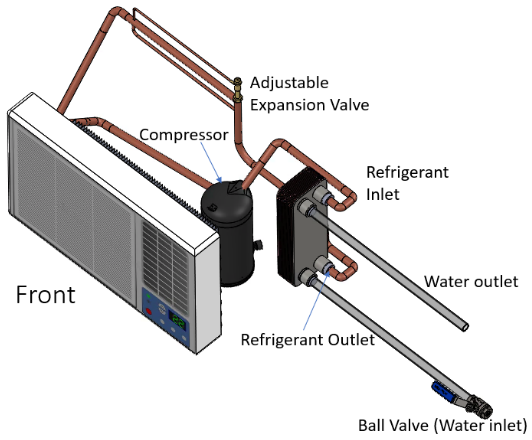

Figure 1 General view of Experimental Test Unit (Air conditioner)

Attempt number 1

The plate heat exchanger in the experiment was made up of 10 corrugated plates. In order to save the substantial amount of computing resources, a simple model as shown in figure 8 was used to simulate the heat exchanger. This model consisted of 10 layers in which the water domain and refrigerant domain were stacked in alternating order.

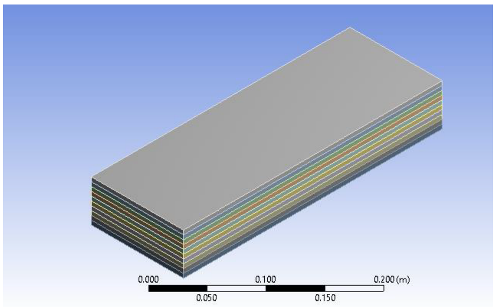

Fig.2 Model of attempt 1

This model was high in computing efficiency, the solution from this model was far from the reality that it failed to depict both inlet and outlet temperature and the streamline of the two fluids. This could be attributed to the absence of obstacles in the channels in which the flow in the two domains is laminar flow.

Attempt number 2

The model for the Ansys simulation therefore was re-designed so that the model could accurately describe the reality. Figure 3 shows a modified version of the plate heat exchanger model which matches the geometry of the exchanger in the experiment.

One thing to note is that for Ansys Fluent simulation a fluid profile is needed to represent a fluid domain as shown in figure 4. To create this fluid profile, 4 planes were firstly created on the 4 openings which are denoted as red colour. 2 fluid profiles were then created by using the 'fill' function in the DesignModeler.

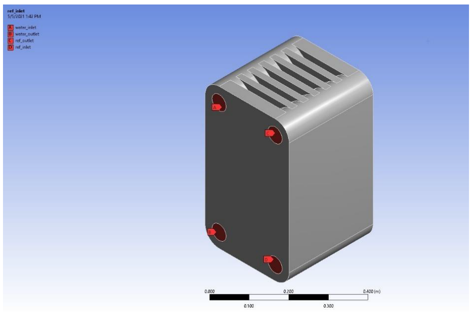

Fig.3 Model of plate heat exchanger

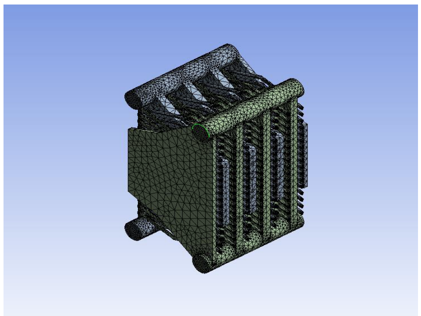

Fig.4 Water domain and refrigerant domain

From Figure 5 to 8, the heat flux and the temperature across the domains were not even. Especially from the temperature distribution graph, the temperature had a clear-cut relative to other regions. This could be due to the water, or the refrigerant directly flow from the inlet to the outlet without covering the whole plate.

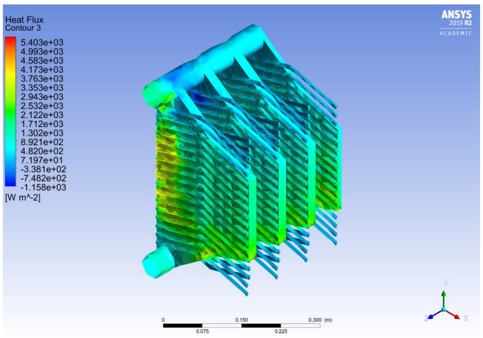

Fig.5 water domain Heat flux

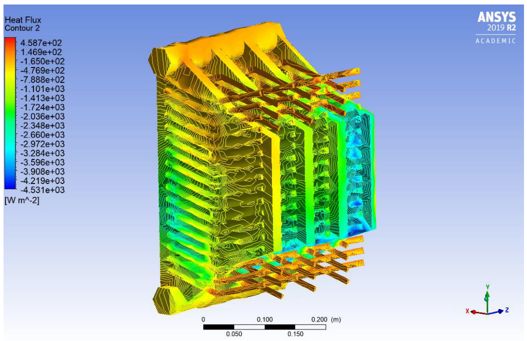

Fig.6 refrigerant domain Heat flux

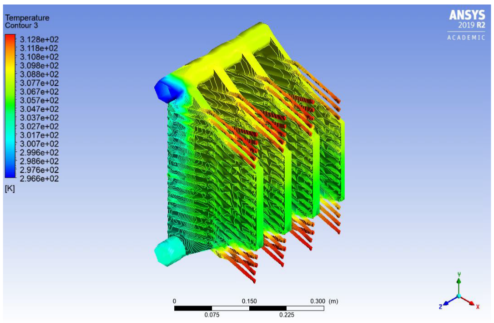

Fig.7 water domain Heat flux

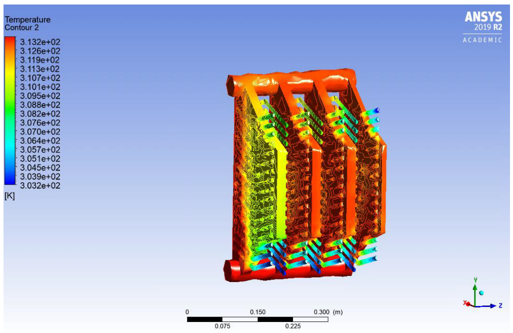

Fig.8 water domain Heat flux

This could be verified by plotting the streamline graph of the two fluids. In figure 14, the water domain only passed through the first plate while the refrigerant did not cover the whole surface area of the plate although it passed through all channels. This also explained why it took 4000 iterations before it finally converged.

Therefore, this model failed. The reason for that is the channels for the fluids were too large which could lead to inaccurate results. This could also be the reason why this model needed to run for 4000 iterations before the result was converged.

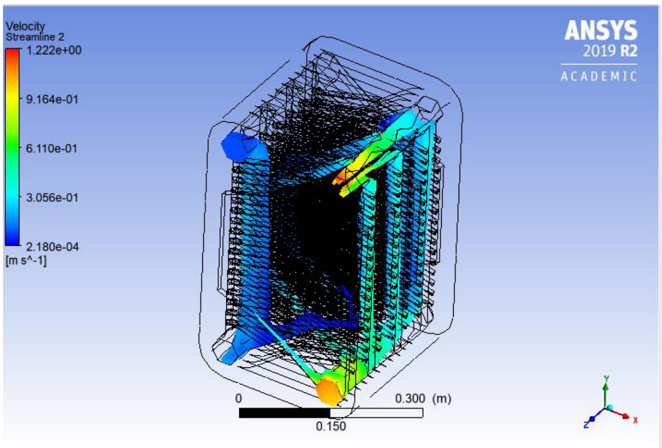

Fig. 9 Streamline of the water and refrigerant.

Data Anlaysis

After doing some modification on the dimension of the plate heat exchanger model, the model could offer accurate results agreeing with the data from the experiment. The dimension of the plate heat exchanger model was strictly following the dimension of the PHE used in the experiment and surprisingly the model only needed to run 160 iterations for the result to be converged.

Plate width (mm)	Plate length (mm)	Plate height (mm)	Corrugated Depth (mm)	Corrugated pitch (mm)	Chevron Angle (°)
50	311	111	0.05	10	60

Dimension of model 3

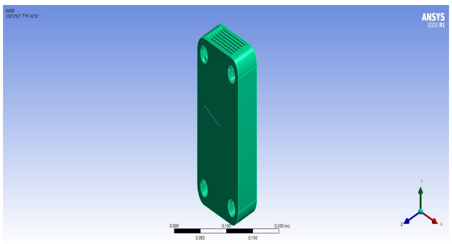

Fig. 10 Design of model 3

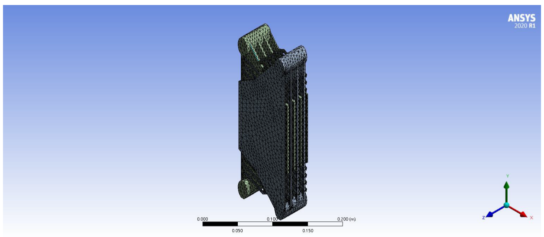

Fig.11 Fluid domains

Boundary Condition

To simulate heat transfer between water and refrigerant, NIST real gas models, which uses the refrigerant mixture database from National Institute of Standards and Technology (NIST), were used. To use this database, one can activate these real gas models following the console prompt. Figure 13 shows the hydrocarbon and refrigerant supported by the library.

Named selection in mesh	Direction	Type	Temperature (K)	Gauge Pressure (Pa)
Water inlet	Normal to boundary	Mass flow inlet	296.8	
Water outlet		Pressure outlet		
Refrigerant inlet		Pressure inlet	313.8	2e6
Refrigerant outlet		Pressure outlet		1.999e6

Case1: 8kg/s Counter flow

Named selection in mesh	Direction	Type	Temperature (K)	Gauge Pressure (Pa)
Water inlet	Normal to boundary	Pressure outlet		
Water outlet		Mass flow inlet	297.4	
Refrigerant inlet		Pressure inlet	315.2	2e6
Refrigerant outlet		Pressure outlet		1.999e6

Case 2: 8kg/s Parallel flow

Named selection in mesh	Direction	Type	Temperature (K)	Gauge Pressure (Pa)
Water inlet	Normal to boundary	Mass flow inlet	296.6	
Water outlet		Pressure outlet		
Refrigerant inlet		Pressure inlet	313.5	2e6
Refrigerant outlet		Pressure outlet		1.999e6

Case 3: 7kg/s Counter flow

Method for obtaining heat transfer coefficient

The bulk temperature needs to be found out in order to calculate the heat transfer coefficient. The bulk temperature can be found out as follows:

$$T_{bulk} = \frac{\iint uT dxdy}{\iint u\, dxdy}$$

As shown in the graph below, 4 planes were constructed. The area integral can therefore be performed on these four planes to find out the bulk temperature.

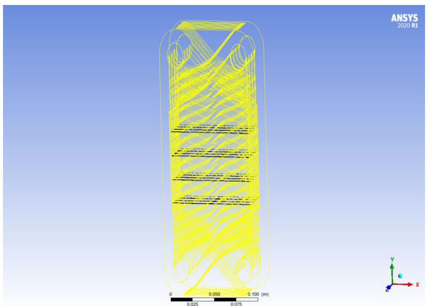

Fig.12 Construction of 3 planes in water domain

Figure below shows the expression for finding the bulk temperature in the post-processing software.

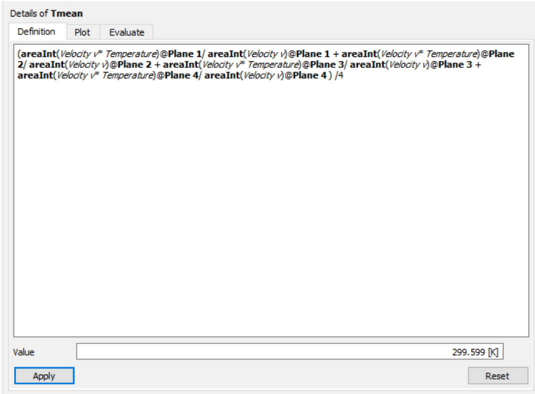

Fig. 13 Expression for finding bulk temperature

For finding the wall temperature, 4 points were constructed, and the maximum value of this point was used as the wall temperature.

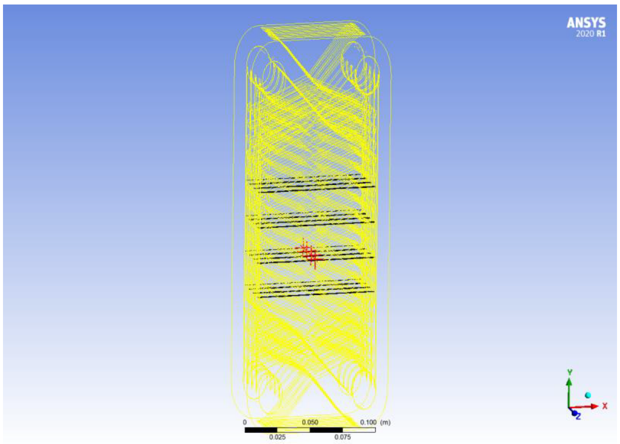

Fig. 14. Construction of 4 points

	Overall heat transfer coefficient (W/m^2 K)
Case 1	6.50E+02
Case 2	6.47E+02
Case 3	6.63E+02

Based on the result from the simulation, the outlet temperature at the refrigerant side and the overall heat transfer rate of PHE were figured out. This proved that higher flow rate and counter flow arrangement indeed improved the performance of plate heat exchanger.

Discussion

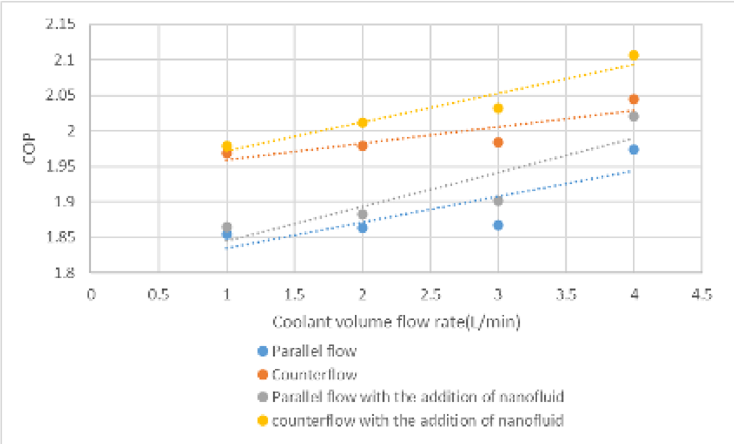

Fig. 15. Comparison between the parallel flow, counterflow with and without the addition of nanofluids

Based on the acquired result from Figure 15, it was predicted that the performance of countercurrent flow would be much better than parallel flow arrangement after the addition of CeO_2 nanofluid because of its higher heat transfer rate. After the collection of data, that was close to the prediction and the improved performance could be attributed to the enormous thermal conductivity provided by nanofluids, as it offered enhanced performance compared with the traditional fluids in the process of heat transfer.

		6 (L/min)	6.5 (L/min)	7 (L/min)	8 (L/min)
COP	Parallel flow	1.854	1.863	1.867	1.974
	Parallel flow with the addition of nanofluid	1.865	1.882	1.901	2.02
Improvement (%)		0.593	1.02	1.82	2.33

Table 1 Improvement before and after the addition of CeO2 in parallel flow

		6 (L/min)	6.5 (L/min)	7 (L/min)	8 (L/min)
COP	Counter flow	1.969	1.979	1.983	2.044
	Counterflow with the addition of nanofluid	1.978	2.012	2.032	2.107
Improvement (%)		0.457	1.64	2.47	3.08

Table 2 Improvement before and after the addition of CeO2 in counterflow

After the addition of CeO2 nanofluid into brazed plate heat exchangers, the coefficient of performance was observed to have an extraordinary enhancement in both counterflow and parallel flow arrangement. As summarized, the gradient of the curve (in the color of yellow and gray) with the addition of CeO2 nanofluid was larger than the original curve (in the color of orange and blue). The underlying reason was revealed for the characteristics of better convective heat transfer coefficient in CeO2 nanofluids. Owing to its smaller particle sizes, CeO2 nanoparticles could enjoy lower particle momentum and higher mobility, and this implied that the goal of faster heat dispersion could be achieved due to the reduction of viscosity and micro convection. Based on the calculation shown in table 1 and table 2, the addition of CeO2 nanofluids could promote performance amelioration of brazed plate heat exchangers. For the arrangement of parallel flow and counterflow, there was near 2.4 and 3.1 percentage improvement after the addition of CeO2 nanofluids, respectively. As the coefficient of performance (COP) is a measurement of heat dissipation and the amount of work required, higher COP value exhibits a high operating performance as well as lower power consumption. Therefore, as opposed to traditional heat exchangers, it was redesigned to be a more environmentally friendly system and resulted in lower operational costs.

Parallel flow		Counterflow	
Q_1	60000 J/s	Q_2	60000 J/s
U_1	150 W/(m^2°C)	U_2	200 W/(m^2°C)
ΔT_{LM1}	15°C	ΔT_{LM2}	20°C
A_1	26.7 m^2	A_2	15 m^2

Table 3

On the other hand, it was surprisingly observed that the normal operation of counterflow was much better than the performance of parallel flow with the addition of nanofluid. Through a profound analysis, the difference could be justified by the leading role of the overall heat transfer coefficient in heat convection between fluids. As evidenced by equation $Q = UA\Delta T_{LM}$, the overall heat transfer coefficient was directly proportional to heat transfer rate, heat transfer surface area and logarithmic mean temperature difference. From the theoretical calculation, since larger value obtained from the multiplication of U and ΔT_{LM} in counterflow, table 3 showed that it took less heat transfer surface area (15 m^2) compared to the value of parallel flow (26.7 m^2) under the same condition of heat transfer rate provided. As a result, it was concluded that the counter flow heat exchanger design served a higher heat transfer rate per unit surface area and the above phenomenon could be attributed to its better heat dissipation.

In conclusion, the goal of improved energy efficiency in the air conditioner was fully achieved with the implementation of CeO2 nanofluid. During the process of the two-step method, concern was raised about the aggregation and suspension stability of CeO_2 nanoparticles to provide non-standard nanofluids. And after the action of flow rate adjustment and the addition of surfactant, the problem of particle coagulation was greatly minimized with the observation. In order to investigate the optimal operating conditions of the system by the correlation of its thermophysical properties, different experimental setups were implemented to evaluate the actual operating performance under different conditions including the adjustment in volume flow rate of coolant, the change of fluid direction and the addition of nanofluid. Under the condition of the same volume flow rate provided, it was found the performance of countercurrent flow was more reliable than parallel flow heat exchange devices. This result could be interpreted by a more uniform heat transfer rate and slower decline in temperature gradient provided by countercurrent flow.

Regarding the use of nanofluids, both design of the counter flow heat exchanger and parallel flow heat exchanger showed significant enhancement in their coefficient of performance. Thanks to its smaller particle sizes, lower particle momentum and higher mobility were provided to disperse the heat more effectively through micro convection. Therefore, as opposed to a traditional brazed-plate heat exchanger, it could provide a higher coefficient of performance and result in lower operational costs. Based on the calculation of experimental results, the improvement percentage reached around 3.1% in the counter flow arrangement, and this exhibited a higher operating performance and lower power consumption compared to the original design. And according to our data analysis, the optimal operating conditions of air conditioners which were under 8kg/s volume flow rate with the addition of CeO2 nanofluid in the counterflow. To sum up, the maximum energy efficiency of air conditioners could be achieved by the adjustment in volume flow rate and the enhancement of thermophysical properties in nanomaterials.

Conclusion

In conclusion, in order to simulate the complex geometries and provide higher accuracy in the engineering simulation, the body of the channel was initially divided into several broken elements, which are solid, water and refrigerant, as a discrete local approximation of the entire domain. After the structural analysis of the deflection at the edge, it is found that grade 316 stainless steel can withstand tremendous pressure created by the refrigerant. Thanks to the aid of the computational fluid dynamics, the outlet temperature at the refrigerant side and the overall heat transfer rate of PHE were fully analyzed and it is proved that the performance of plate heat exchanger can be enhanced by higher flow rate and counter flow arrangement. Referring to the calculation of experimental results, it is shown that the improvement percentage reached around 3.1% in terms of COP, and this demonstrates a higher operating performance and lower power consumption compared to the original design. And for the future design and research of the project, it is suggested that much more focus should be put on the experimental uncertainty analysis such as the preparation of nanofluids. Since the two-step method is highly dependent on the effect of atmospheric pressure, it is impossible to determine how much air is mixed and entered the system. If a large amount of air enters the system during the process, it will hugely reduce the heat performance because it occupies space and decreases the heat exchanging surface in the heat exchanger. In addition, due to the concern of high density for CeO2 nanoparticles, sedimentation problem may be underestimated, and it leads to an additional number of nanoparticles that used to deal with the issue of resultant loss.

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