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STUDY OF PLATE HEAT EXCHANGER PERFORMANCE WORKING WITH THREE TYPES OF REFRIGERANTS EXPOSED TO HOT AIR FLOW

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ABSTRACT: The thermal effectiveness of plate heat exchanger exposed to uniform hot air flow working with three Refrigerants (R134a, R290 and R513a) that are heat exchanged with cold water assuming constant flow rate have been calculated via the AUTODESK CFD 2019 package using the standard K-ε turbulence model, the variation effect of the Refrigerants inlet temperature range (35-50)°C on the effectiveness is investigated considering the hot air flow inlet temperature range between (45-50)°C. The results showed that the PHE effectiveness is maximumly decreased by (14%) for the R513a and by (17%) and (19%) for R134a and R290, respectively.

KEYWORDS

Plate heat exchanger, Hot air flow, Effectiveness evaluation

Introduction

One of the most efficient heat exchangers that are the best alternative ones to the shell and tube type in the mobile and the compact air-conditioning units is the plate heat exchanger (PHE) [4] due to its small size, and efficient heat exchange rate consists of several pressed metal chevron or pillow plates with or without adjusting hard rubber gaskets between them. J.F. Seara et al. [1] implemented an experimental evaluation of the heat transfer rate and the overall heat transfer coefficient of the water-water ethylene mixture in the titanium brazed offset strip fins PHE. They developed a general correlation for the heat transfer coefficient in PHE channels in terms of Reynolds number for isothermal outbound conditions, and the overall heat transfer coefficient increased with increased fluids mass flow rate.

The Hydro Fluro Olefins (HFO’s) refrigerants attains high heat transfer rate, non-flammability and lower pressure drop are the R1234yf-ze Refrigerants where several researches have been performed to evaluate their performance and formulate convection heat transfer correlations. J. Zhang et al. [2] performed experimental tests on the brazed PHE working as an evaporator in an organic Rankin cycle using the HFO’s in addition to the HFC R134a, they measured the boiling heat transfer coefficient and the pressure decrease for variable inlet saturation conditions, mass flux and attained outlet vapor qualities. They developed an evaporation correlation for the heat transfer coefficient.

The PHE thermal effectiveness is calculated numerically using the NTU method in parallel flow process by H. Dardour et al. [3] using water liquid as the hot and the cold fluid. The numerical solution is performed using the Runge-Kutta method involving Newton Raphson convergent criteria. The results showed that the effectiveness is decreased with the increased thermal flow rate ratio of the heat-exchanged fluids, while it remains nearly constant for NTU greater than [4].

The main objective of the present research is to evaluate the typical PHE thermal effectiveness (using the NTU method) subjected to forced convection by means of uniform hot air flow from both sides of the plates with variable inlet temperature range from (45-50°C), considering three types of Refrigerants in a superheated state; R134a, R290 and the (R1234yf/R134a) mixture R513a, respectively. The heat exchange of the Refrigerant passage flow through the HE will be with cold water by adopting the counter-flow process, the CFD numerical simulation solution is performed using the AUTODESK CFD 2019 student version considering the K-ε as the turbulence model considering the refrigerants inlet temperature range between (35-50)°C.

Heat exchanger geometry

The chevron non-corrugated pattern shape of seven plates typical heat exchanger with stainless steel 316 material selection and inserted nitrile hard rubber gaskets between each plate is sketched using the Solidworks CAD 2016 shown in Fig. (1). Table (1) denotes the geometry main selected dimensions for the PHE.

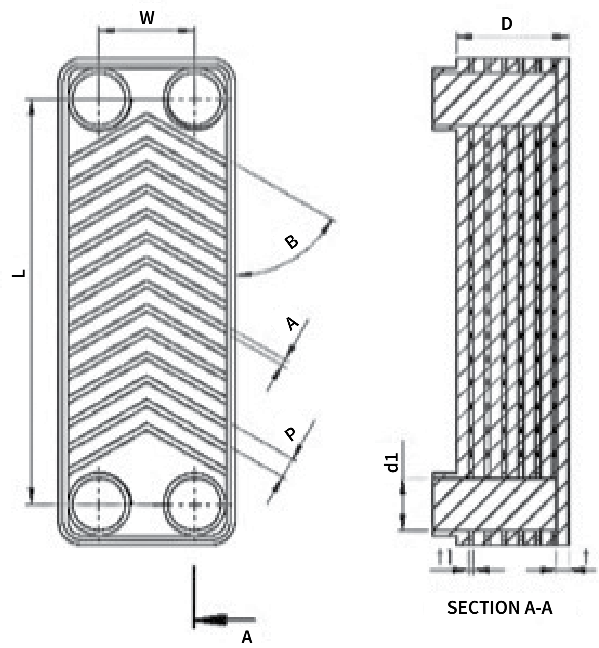


FIGURE 1. PHE geometry

Dimension	Value (m)
Chevron thickness (A)	1.227m
Chevron angle (B)	60°
PHE depth (D)	0.05
Flow inlet diameter (d1)	0.02
Inlet/outlet height (L)	0.17
Chevron pitch (P)	0.01
Plate thickness (t)	0.01
Gasket thickness (t1)	0.002
Refrigerant/water inlet width (W)	0.05

TABLE 1. PHE main dimensions

Solution methodology

Prior to evaluate the PHE thermal effectiveness, it is necessary to simulate the heat exchange process transitionally until reaching the steady-state condition. The continuity, momentum, the energy in addition to the K-ε turbulence model is utilized to calculate the flow domains, where the counter flow process has been adopted as shown in Fig. (2). The external air flows normal to the both sides of the heat exchanger plates and exits upwards the PHE.

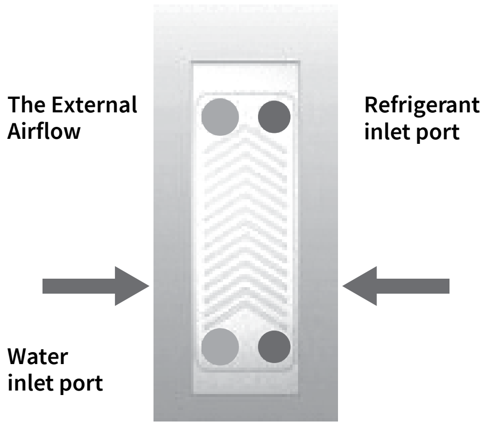


FIGURE 2. The fluids flow adopted directions

Numerical Simulation Procedure

Geometry Mesh Generation

The PHE with the three fluids flow volumes have been meshed using the tetrahedrons method with automatic sizing and activation of surfaces refinement option, the total mesh statistics were more (410K) nodes and (1280K) elements, respectively.

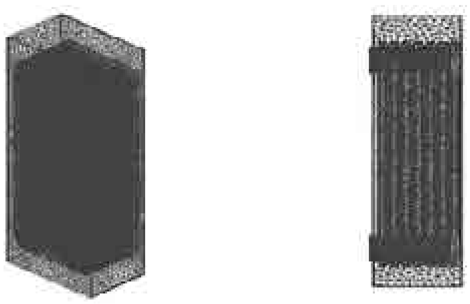


FIGURE 3. The generated mesh PHE

Applying Turbulence Model

The standard K-ε turbulence model differential equations is utilized to calculate the flow turbulence using the finite element method.

$$\frac{\partial(\rho K)}{\partial t}+\frac{\partial}{\partial x}(\rho UK)=\frac{\partial}{\partial x}\left[\left(\mu+\frac{\mu_c}{\sigma_k}\right)\frac{\partial K}{\partial x}\right]+P_k-\rho\varepsilon \quad (1)$$

$$\frac{\partial(\rho\varepsilon)}{\partial t}+\frac{\partial}{\partial x}(\rho U\varepsilon)=\frac{\partial}{\partial x}\left[\left(\mu+\frac{\mu_c}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial x}\right]+\frac{\varepsilon}{K}[C_{\varepsilon1}P_\varepsilon-C_{\varepsilon2}\rho\varepsilon] \quad (2)$$

Assigning Boundary Conditions

It is necessary to assign the inlet conditions of the internal (water/Refrigerant) flow and the out-bonding air in addition to the convection heat transfer coefficient with the PHE surfaces as well as the initial PHE volume temperature, then the transient solution is adopted until reaching the steady state condition. Table (2) denotes the estimated boundary conditions for the simulated case.

Flow domain	Inlet condition	Temperature (°C)	Pressure (KPa)
Refrigerant	Velocity (5) m/sec	(35-50)	300
Water	Velocity (5) m/sec	(10)	atm
Air	Velocity (2) m/sec	(45-50)	atm

TABLE 2. Boundary conditions for the flow domains

To compute the film convection heat transfer for the flow field during the iteration run execution, the Wu-Little correlation is utilized in the CFD package.

$$Nu = C \times Re^a \times Pr^b \quad (3)$$

Where the values of the constants are:
C=0.0022, a=1.04 and b=0.4.

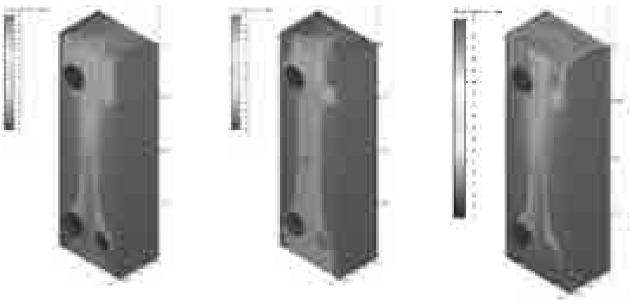
Solution scheme

The modified Petrov-Galerkin solution method is selected as the turbulence modeling advection scheme in the iteration procedure setting the residual target for the numerical solution to (1e-3) in the governing equations.

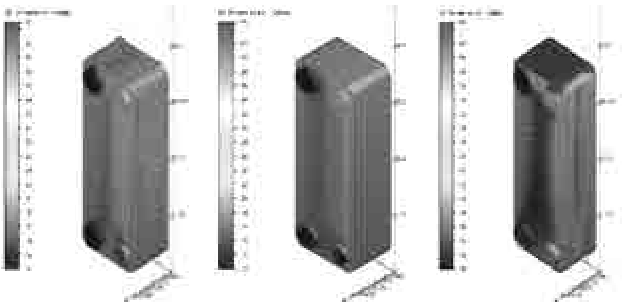
Results and discussion

PHE transient response:

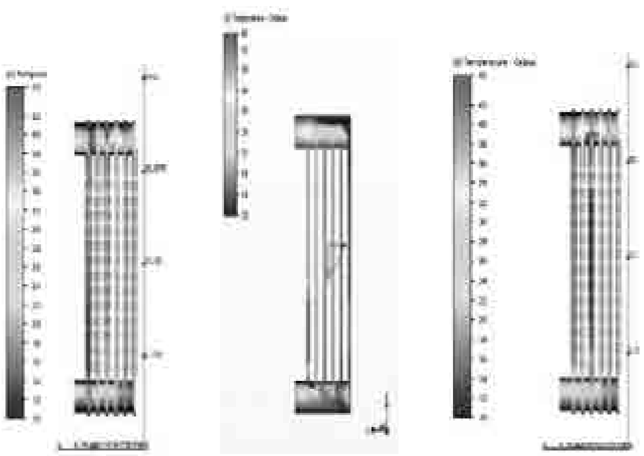
Figs. (4-9) demonstrates the transient heat exchange for the air flow at (45°C) with the PHE body and the cooling water effect for 12 seconds of simulation using the R134a noticing increased gradual heating on the PHE side and upper surfaces, while the water cooling becomes more significant with the passage of time. Figures (10-12) represent the comparison between the Refrigerants subcooling (R134a, R290 and R513a) with water after reaching the steady-state condition. The R513a refrigerant gives relatively better subcooling degree (12)°C to the traditional R134a and the propane gas R290 (10-9)°C.



FIGURES 4-6. Transient response for the air flow with time



FIGURES 7-9. PHE transient response with time



FIGURES 10-12. R134a, R290 and R513a T-contours

Thermal effectiveness calculation:

The PHE effectiveness (η) is defined as the percentage ratio of the Realistic to maximum permissible heat transfer rate between the cold and the hot flow streams, [3] is represented by:

$$\eta = \frac{m_H \times C_{pH}\,(\Delta T_H)}{m_{min} \times C_{pmin}\,(\Delta T_{in})} \tag{4}$$

Table.3 shows the calculated PHE effectiveness subjected for the air temperature range (40-50)°C for corresponding Rerigerants inlet temperature (35-50)°C taking into consideration constant velocity for the three incorporated fluids flow. From the tabulated results, it is obvious that the effectiveness is lowest afftected with utilizing R513a as the ambient temperature rises up, while the two other refrigerants gives nearly equivalent result. Also, the increase in inlet refrigerant temperature had caused a gradual decline in the PHE effectiveness that's for the dual heating effect from both sides of the PHE.

Tair (°C)	R134a		R290		R513a	
	Tin	η	Tin	η	Tin	η
40	35	0.84	35	0.81	35	0.86
	42	0.75	42	0.72	42	0.79
	50	0.70	50	0.68	50	0.74
45	35	0.78	35	0.76	35	0.81
	42	0.71	42	0.69	42	0.75
	50	0.68	50	0.66	50	0.71
50	35	0.71	35	0.69	35	0.75
	42	0.63	42	0.59	42	0.68
	50	0.58	50	0.55	50	0.67

TABLE 3. Effectiveness calculation results

Solution validation:

The comparative evaluation of the Refrigerant wall film heat transfer coefficient using the utilized empirical correlation Eq.(3). Table. 4 shows a percentage error range between (12-17)% for R134a inlet temperaure of 35°C when compared with the energy equation residual calculation method.

Flow Velocity	h (Wh&Little)	h (Thermal residual)
5	4623	5306
10	6237	7538

TABLE 4. Wall film heat transfer coefficient

Conclusions

1. The bounding hot air flow effect on the PHE effectiveness have a maximum of (14%) when the R513a is the working fluid and around (17&19)% for R134a and R290, respectively.
2. Both of the increased air and the refrigerants inlet temperature causes decline in the PHE effectiveness.

Acknowledgment

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References

Seara J.F, Diz R. and Uhia F.J. (2013) Pressure drop and heat transfer characteristics of a titanium brazed plate-fin heat exchanger with offset strip fins. Applied Thermal Engineering; 51:502-511.

Zhang J., Desideri A., Kaern M.R, Ommen T.S, Wronski J. and Haglind F. (2017). Flow boiling heat transfer and pressure drop characteristics of R134a, R1234yf and R1234ze in a plate heat exchanger for organic Rankine cycle units. International Journal of Heat and Mass Transfer;18:1787-1801.

Dardour H., Mazouz S. and Bellagi A. (2009). Numerical Analysis of Plate Heat Exchanger Performance in Co-Current Fluid Flow Configuration. International Journal of Physical and Mathematical Sciences; 3:213-217.

Hesselgreaves, J.E, Law R. and Reay D.A. (2017). Compact Heat Exchangers, Selection, Design and Operation: Elsevier Ltd.

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Nomenclature

a, b	Nusselt number correlation
C	Coefficients
Cp	Specific heat (KJ/kg.°C)
h	Film heat transfer coeff. (W/m^2K).
K	Kinetic energy (Kg/m^2sec^2)
m	Mass flow rate (Kg/sec)
T	Temperature (°C)
t	Time (sec)

Subscripts

H	Hot fluid
min	Minimum value
in	Inlet flow ports

Greek Symbols

K-ε	Turbulent viscosity, kg/m.sec
ε	Dissipation of kinetic energy
σ_k, σ_ε	Turbulent model constants

Non-dimensional Numbers

Nu	Nusselt number
Pr	Prandtl number, [Cpμ/k]
Re	Reynolds number, [ρUD/μ]