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# Investigation of heat transfer and pressure drop in plate heat exchangers having different surface profiles

Aydın Durmuş [a,*], Hüseyin Benli [b], İrfan Kurtbaş [a], Hasan Gül [b]

[a] Fırat University, Mechanical Education Department, 23119 Elazığ, Turkey
[b] Fırat University, Vocational High School, 23119 Elazığ, Turkey

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ABSTRACT

It would be misleading to consider only cost aspect of the design of a heat exchanger. High maintenance costs increase total cost during the services life of heat exchanger. Therefore exergy analysis and energy saving are very important parameters in the heat exchanger design. In this study, the effects of surface geometries of three different type heat exchangers called as $PHE_{flat}$ (Flat plate heat exchanger), $PHE_{corrugated}$ (Corrugated plate heat exchanger) and $PHE_{asteriks}$ (Asterisk plate heat exchanger) on heat transfer, friction factor and exergy loss were investigated experimentally. The experiments were carried out for a heat exchanger with single pass under condition of parallel and counter flow. In this study, experiments were conducted for laminar flow conditions. Reynolds number and Prandtl number were in the range of $50 \leqslant Re \leqslant 1000$ and $3 \leqslant Pr \leqslant 7$, respectively. Heat transfer, friction factor and exergy loss correlations were obtained according to the experimental results.



## 1. Introduction

The plate heat exchangers are widely used in warming, heating, cooling applications, food, and cosmetic and chemistry industry. The plate type heat exchangers are initially developed for the pasteurized liquid food domain which mostly requires hygienic application. But, these heat exchangers have a large application area in chemistry and food sector because of being compact and having the quality to be easily cleaned [1–5].

The augmentation techniques of heat exchanger efficiency can be classified as active and passive methods. In active method, heat transfer can be improved by giving extra energy to system. In passive method, however, the improvement can be performed without giving extra energy.

Some examples to active method include the use of mechanical auxiliary elements, turning of surface, mixing of fluid with mechanical parts, constituting of electro-static areas in flow area, vibration of system, etc., Some examples to passive method include covering of surface, changing of surface, forming of the same projection parts of the rough surface, locating of the tabulators in flow area, etc., [6,7].

In the passive methods, usually a turbulence effect is given to the flow by the shapes having different geometries for this purpose; rugged surfaces are used, in a study made to improve the heat transfer [8,9]. Sparrow and his friends [10] examined the heat transfer on twisted surfaces, in Reynolds number, $2000 \leqslant Re \leqslant 35000$ and $Pr$ $4 \leqslant Pr \leqslant 11$ interval. In this study water is used as experimental fluid. These researchers suggest that;

$$Nu = 0.491 \cdot Re^{0.63} Pr^{0.3} \tag{1}$$

The equation can be used for sharp surface geometries. Consequently to the experimental study, in $Re > 250$ condition, they suggested the following correlation.

$$Nu = 4 + 29.2.\ln\left[\frac{(\mathrm{RePr}^{0.4} + 100)}{1200}\right] \tag{2}$$

Fabbri [11] made an interesting optimization study about heat exchangers in wavy canals by using "genetic algorithm" technique. The researcher indicated that the wavy surfaces improve the heat transfer in the situations that the Reynolds and Prandtl numbers are not too low.

In the most of the studies made on the plate surface heat exchangers, it is observed that the configuration is symmetrical; the cold and hot fluid pass by the same number of canals and the favourite flow shape is based on "parallel-counter flow". Kandlikar and Shah [12] investigated multi pass plate heat exchanger effectiveness.

In this study, the influence of the geometrical qualifications of the plate heat exchangers on heat transfer, in "parallel flow" and "counter flow" arrangements, in $Nu$, $f$, $E$ numbers base, is experimentally examined.

* Corresponding author. Tel.: +90 424 2370000/4217; fax: +90 424 2367064.
E-mail address: adurmus1@firat.edu.tr (A. Durmuş).

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**Nomenclature**

| | |
|---|---|
| $A$ | Heat transfer area of the plate, ($m^2$) |
| $A_c$ | Cross-section area, ($m^2$) |
| $C_p$ | Specific heat capacity, ($Jkg^{-1}K^{-1}$) |
| $C$ | Heat capacity rate, ($= \dot{m}C_p$), ($WK^{-1}$) |
| $D_h$ | Hydraulic diameter, ($= 4A_cP_w^{-1}$), ($m$) |
| $E$ | Exergy loss, ($W$) |
| $H$ | Average heat transfer coefficient, ($Wm^{-2}K^{-1}$) |
| $h$ | Specific enthalpy, ($Jkg^{-1}$) |
| $P_w$ | The total wetted perimeter, ($m$) |
| $k$ | The thermal conductivity of the fluid, ($Wm^{-1}K^{-1}$) |
| $T$ | Temperature, ($K$) |
| $V$ | Average velocity of fluid, ($ms^{-1}$) |
| $\mu$ | Dynamic viscosity, ($Pa.s$) |
| $\rho$ | Density of fluid, ($kgm^{-3}$) |
| $\Delta T$ | Logarithmic temperature difference between wall and fluid, ($K$) |

*Dimensionless numbers*

| | |
|---|---|
| $C_r$ | Capacity rate ratio, ($= C_{min}/C_{max}$) |
| $Nu$ | Average Nusselt number, ($= HD_h/k$) |
| $Pr$ | Prandtl number, ($= \mu C_p/k$) |
| $Re$ | Reynolds number, ($= \rho V D_h/\mu$) |
| $f$ | Friction coefficient |

*Subscripts*

| | |
|---|---|
| c | Cold |
| e | Environment |
| h | Hot |
| i | Inlet |
| o | Outlet |
| w | Wall |

*Abbreviations*

| | |
|---|---|
| PHE | Plate heat exchanger |
| SSE | Sum Square Error |

## 2. Materials and methods

The experimental apparatus given in Fig. 1 was designed in order to carry out the experimental study. The plates (Fig. 2) were manufactured by being pressed non-corrodible steel plates, which were shaped by molds. The molds were manufactured by CNC workbench [13]. The plate heat exchanger was formed by means of joining side by side of 15 plates. The heat exchanger is single pass, because hot and cold fluid passes once on the plate surface (Fig. 3). The plate surface profiles were set to three different geometries as flat, corrugated and asterisk. The same shape flow channels on the plates were formed, because of distributing homogenously and easily of the flow in the channel. Because of turning the flow from parallel flow to counter flow or just contrary a by-pass system was added to the experimental apparatus.

Two different water tanks were used to get hot and cold water. Mass flow rate was measured by rotameter, and kept constant in both parallel and counter flow. In the experiments the hot water flow is made between 0.03 kg/s and 0.16 kg/s, and the hot water entries are made among 45–80 °C. The plate surfaces are cleaned before every experiment to avoid the negative influences of the pollution on the surface of the plates on the heat transfer.

In the experiments performed, Reynolds number changes in the range of 50 and 1000, and Prandtl number changes in the range of 3 and 7. Also, heat capacity ($C_r = C_{min}/C_{max}$) 0.2, 0.5, 0.7 and 1.

## 3. Theoretical analysis for heat transfer

The heat load, $Q$, of a PHE in steady-state operation with no heat losses ($Q_{loss}$ neglected or $Q_{loss} = 0$), and no phase changes of hot and cold liquids at constant pressure, can be represented as follows;

$$Q = (\dot{m}C_p)_h\,(T_{h,i} - T_{h,o}) = (\dot{m}C_p)_c(T_{c,o} - T_{c,i}) \tag{3}$$

In the experimental study, the wall temperatures are measured by means of thermocouples (Fig. 3). Logarithmic mean temperature difference between wall and liquids;

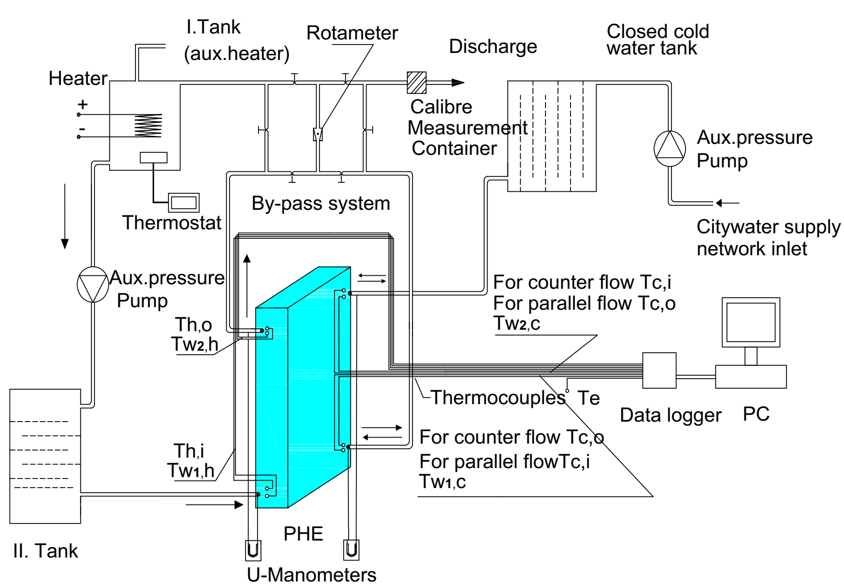


**Fig. 1.** Experimental setup.

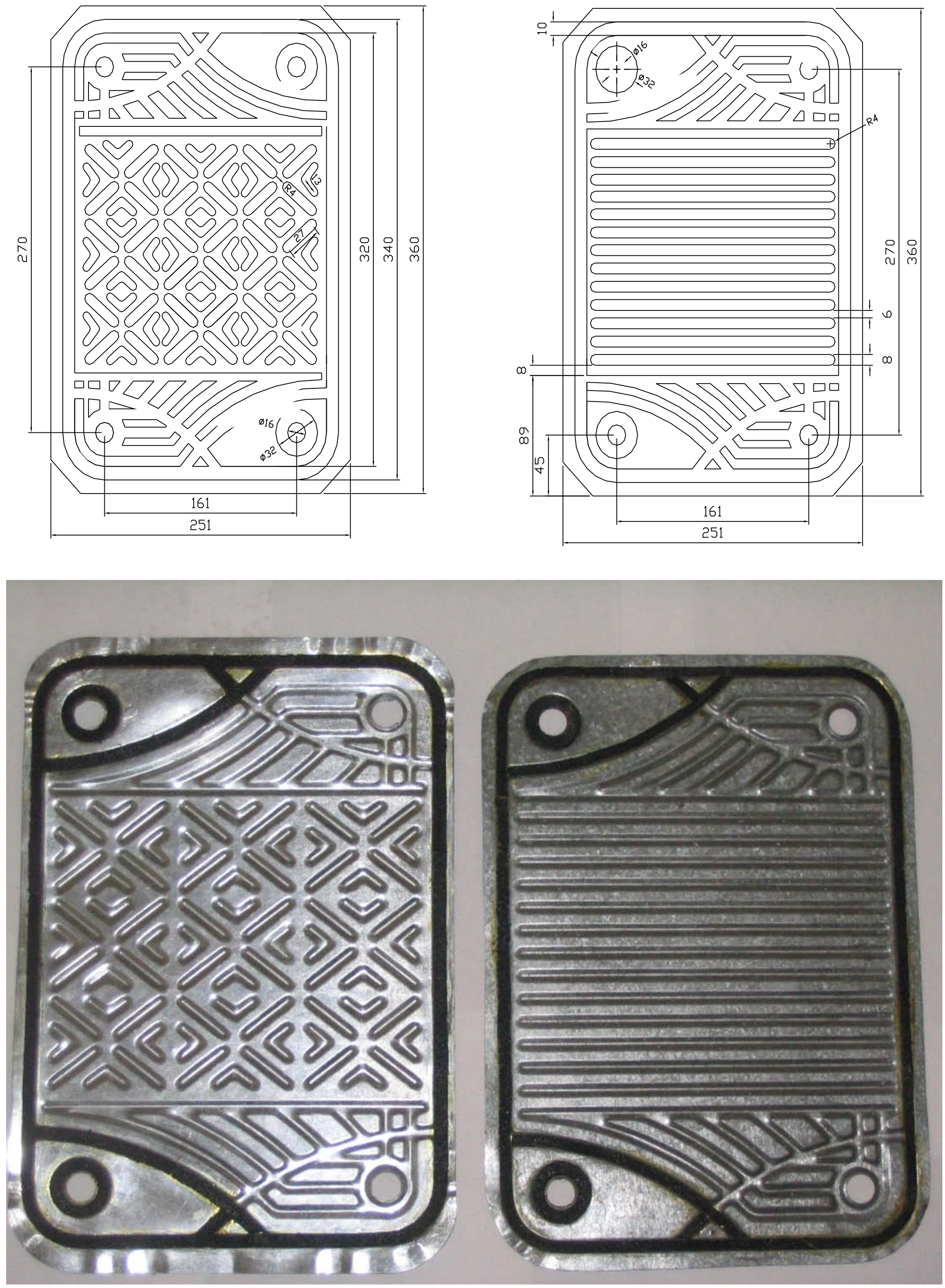


**Fig. 2.** The heat exchanger having the PHE corrugated and PHE asterisk surface structure.

$$\Delta T_{h,c,lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{4}$$

For counter flow (Fig. 3a)

hot water side;

$$\Delta T_{1,h} = T_{h,i} - T_{w1,h}; \quad \Delta T_{2,h} = T_{h,o} - T_{w2,h} \tag{5}$$

cold water side;

$$\Delta T_{1,c} = T_{w1,c} - T_{c,o}\,; \qquad \Delta T_{2,c} = T_{w2,c} - T_{c,i} \tag{6}$$

For parallel flow (Fig. 3b);

hot water side;

$$\Delta T_{1,h} = T_{h,i} - T_{w1,h}; \quad \Delta T_{2,h} = T_{h,o} - T_{w2,h} \tag{7}$$

cold water side;

$$\Delta T_{1,c} = T_{w1,c} - T_{c,i}; \quad \Delta T_{2,c} = T_{w2,c} - T_{c,o} \tag{8}$$

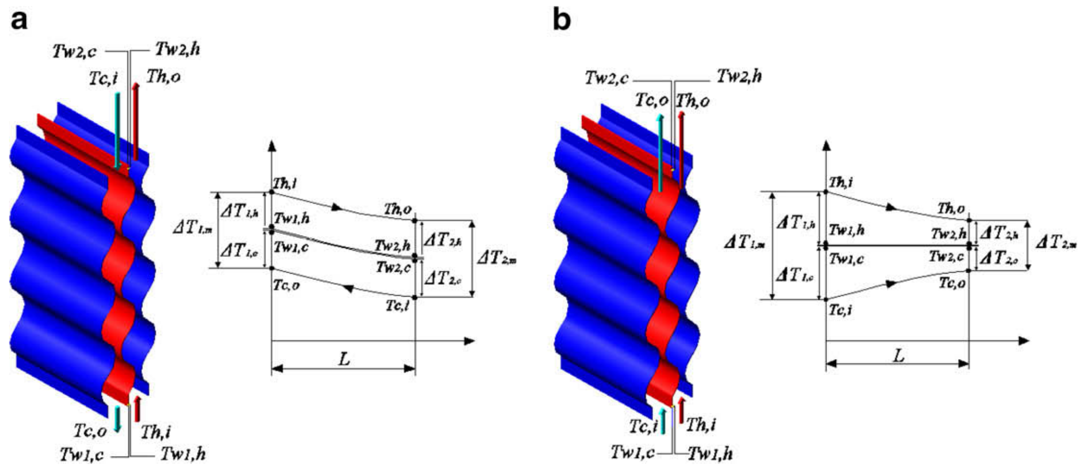


**Fig. 3.** Temperature distributions for (a) counter flow, (b) parallel flow.

Reynolds number is calculated based on hydraulic diameter for cold or hot water side

$$Re = \rho \cdot V \cdot D_h / \mu \tag{9}$$

Fluids properties are evaluated at mean temperature given by

$$T_{m,c} = \frac{T_{c,i} + T_{c,o}}{2} \quad \text{or} \quad T_{m,h} = \frac{T_{h,i} + T_{h,o}}{2} \tag{10}$$

For adiabatic heat exchanger;

$$Q = H_h \cdot A \cdot \Delta T_h = H_c \cdot A \cdot \Delta T_c \tag{11}$$

The Nusselt numbers, in terms of the cold or hot water side are;

$$Nu = H \cdot D_h / k \tag{12}$$

## 4. Theoretical analysis for exergy loss

Exergy is called the amount of maximum work obtained theoretically at the end of a reversible process in which equilibrium with environment should be obtained. According to this definition, exergy balance in a steady-state open system can be written as follows [14]:

$$E = T_e S_{irr} = T_e[\dot{m}_c(S_{c,o} - S_{c,i}) + \dot{m}_h(S_{h,o} - S_{h,i})] \tag{13}$$

$$\mathrm{d}S = \int_{v=\mathrm{const}} \frac{\mathrm{d}Q}{T} = \int_{v=const} \frac{\mathrm{d}u + p\mathrm{d}v}{T} = \int_{v=\mathrm{const}} \frac{C\mathrm{d}T}{T} \tag{14}$$

For liquids ($v$ = const.);

Entropy production of adiabatic heat exchanger;

$$S_{irr} = C_h \ln \frac{T_{h,o}}{T_{h,i}} + C_c \ln \frac{T_{c,o}}{T_{c,i}} \tag{15}$$

Exergy loss;

$$E = T_e \left[ C_h \ln \frac{T_{h,o}}{T_{h,i}} + C_c \ln \frac{T_{c,o}}{T_{c,i}} \right] \tag{16}$$

$T_{h,i}$, $T_{h,o}$, $T_{c,i}$, $T_{c,o}$ are measured and $E$ can directly be determined. For liquids $E$ contains also the exergy loss caused by pressure drop.

## 5. Results and discussion

The experimental results obtained are compared with empirical Nusselt number given as Eq. (17) which had been derived by Gut [15,16].

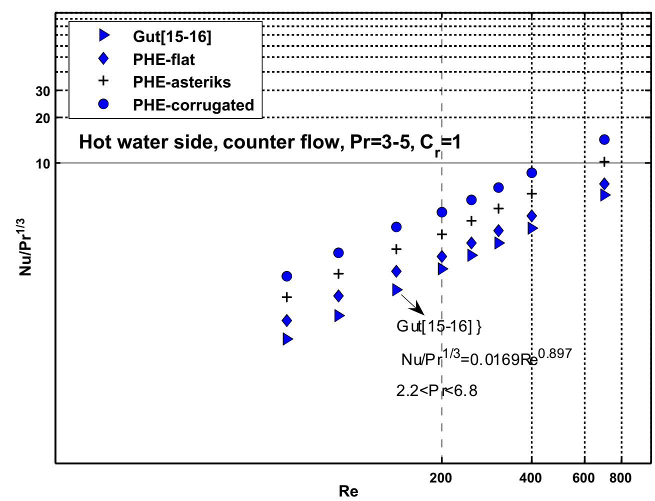


**Fig. 4.** Nusselt number the counter flow as functions of Reynolds number for hot water side.

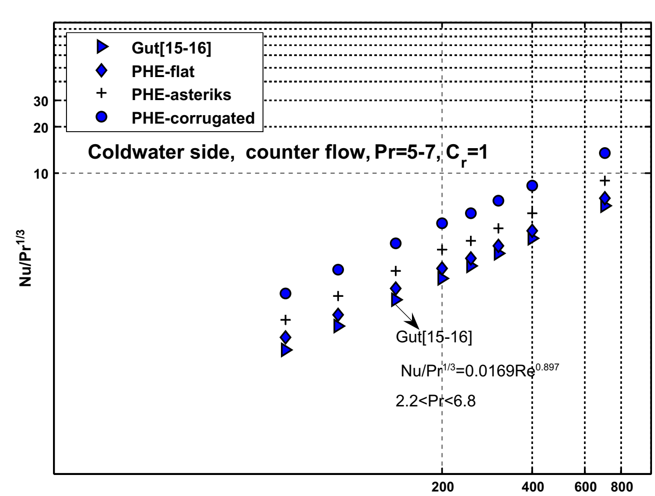


**Fig. 5.** Nusselt number in the counter flow as a function of Reynolds number for cold water side.

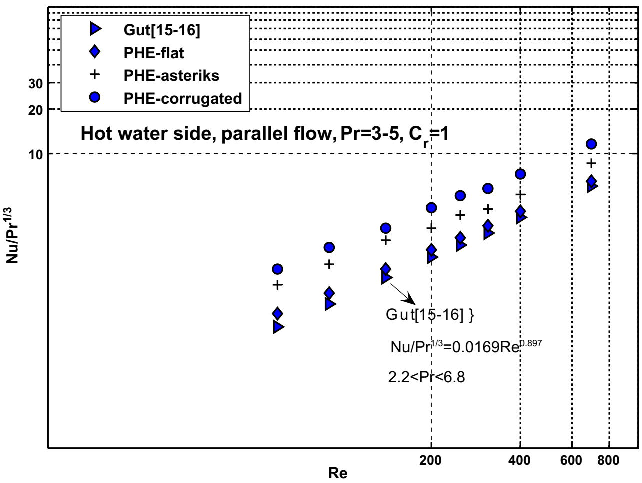


**Fig. 6.** Nusselt number in the parallel flow as a function of Reynolds number for hot water side.

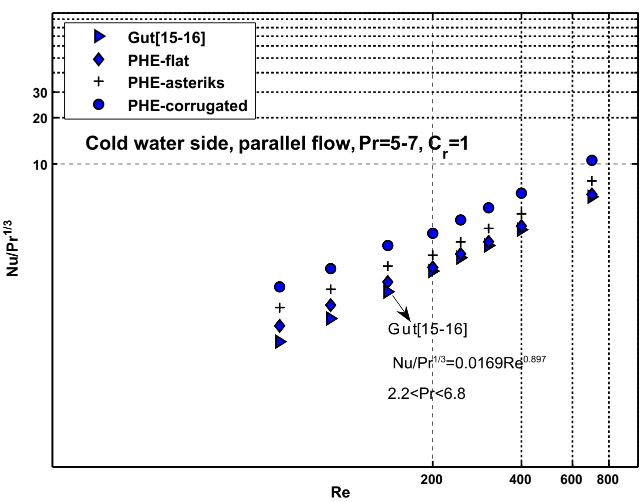


**Fig. 7.** Nusselt number in the parallel flow as a function of Reynolds number for cold water side.

$$\frac{Nu}{Pr^{1/3}} = 0.0169Re^{0.897} \quad 10 < Re < 1000 \quad \text{and} \quad 2.2 < Pr < 6.8 \tag{17}$$

**Table 2**
Typical uncertainties for relevant variable

| Variable | Uncertainty (%) |
|---|---|
| Hot fluid inlet temperature, $T_{h,i}$ | ±0.1 |
| Hot fluid outlet temperature, $T_{h,o}$ | ±0.1 |
| Cold fluid inlet temperature, $T_{c,i}$ | ±0.1 |
| Cold fluid outlet temperature, $T_{c,o}$ | ±0.1 |
| Inner wall temperature, $T_{w1}$ | ±0.1 |
| Outlet wall temperature, $T_{w2}$ | ±0.1 |
| Ambient temperature, $T_e$ | ±0.1 |
| Hydraulic diameter, $D_h$ | ±1.2 |
| Pressure difference, $\Delta P$ | ±2.5 |
| Water mass flow rate, $\dot{m}$ | ±2.5 |
| Uncertainty in reading values of tables, ($\rho$, $C_p$, $k$, $\mu$, ...) | ±0.1 −0.2 |

The variation of Nusselt number in the hot side with respect to Reynolds number for flat, corrugated and asterisk type heat exchanger is clearly depicted in Fig. 4, which is valid for $C_r = 1$ and counter flow conditions. According to Fig. 4 corrugated plate has the highest values. The same variations are also seen for the cold side. Nusselt numbers in the cold side were found to be less than that in the hot side (Fig. 5). The same gradient is also seen for parallel flow (Figs. 6 and 7). On the other hand, the variations in the Nusselt number in both cold and hot side for parallel flow were found to be less than that for counter flow.

The empirical correlations obtained from the experimental results are given in Table 1. The experimental results revealed that asterisk type heat exchanger is the most advantage in terms of heat transfer.

The uncertainties arising in calculating a result ($W_R$) due to several independent variables are given as follows;

$$W_{R^+} = \left( \left( \frac{\delta R^+}{\delta X1} w1 \right)^2 + \left( \frac{\delta R^+}{\delta X_2} w_2 \right)^2 + .. + \left( \frac{\delta R^+}{\delta X_n} w_n \right)^2 \right)^{1/2} \tag{18}$$

where the result $R^+$ is a given function of the independent variables $x_1, x_2, \ldots x_n$ and $w_1, w_2, \ldots w_n$ are uncertainties in the independent variables [17]. Uncertainty calculations showed maximum value of 2.8% for Reynolds number, 5.3% Nusselt number, 4% Prandtl number, 6.4% for friction factor and 6.6% for the exergy losses. The individual contributions to the uncertainties of the non dimensional parameters for each of the measured physical properties are summarized in Table 2.

The changing of friction coefficient with Reynolds number according to the empirical correlation and heat exchanger types is given in Fig. 8. As can be seen from Fig. 8, pressure drop in the

**Table 1**
The empirical correlations obtained from the experimental results

| | | | Nusselt number $\frac{Nu}{Pr^{1/3}} = aRe^b$ | | | |
|---|---|---|---|---|---|---|
| | | | $a$ | $b$ | $R$-square | SSE |
| Corrugated | Parallel flow | Hot side | 0.05774 | 0.8091 | 0.9983 | 0.1222 |
| | | Cold side | 0.04319 | 0.8368 | 0.9961 | 0.2288 |
| | Counter flow | Hot side | 0.0488 | 0.8640 | 0.9989 | 0.1164 |
| | | Cold side | 0.0443 | 0.8709 | 0.9996 | 0.3828 |
| Asterisk | Parallel flow | Hot side | 0.04988 | 0.7830 | 0.9962 | 0.1407 |
| | | Cold side | 0.03131 | 0.8368 | 0.9961 | 0.1202 |
| | Counter flow | Hot side | 0.03516 | 0.8637 | 0.9989 | 0.06026 |
| | | Cold side | 0.02928 | 0.8713 | 0.9996 | 0.01667 |
| Flat | Parallel flow | Hot side | 0.02545 | 0.8508 | 0.9991 | 0.00176 |
| | | Cold side | 0.02372 | 0.8508 | 0.9978 | 0.04017 |
| | Counter flow | Hot side | 0.02503 | 0.8633 | 0.9989 | 0.03041 |
| | | Cold side | 0.02228 | 0.8717 | 0.9962 | 0.00990 |

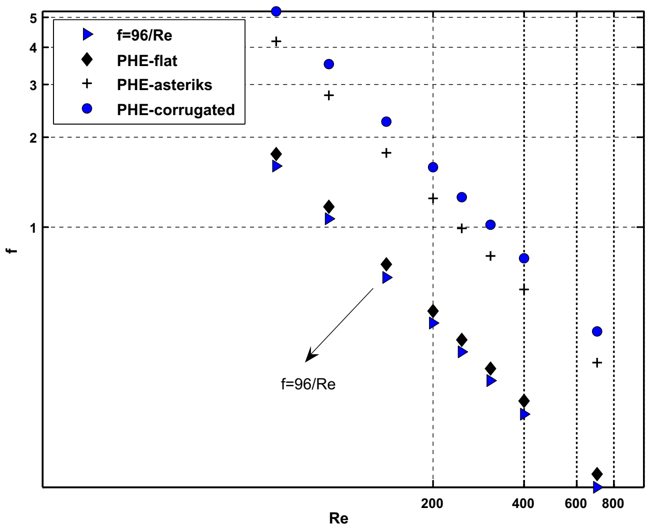


**Fig. 8.** Friction factor as a function of Reynolds.

corrugated and asterisk type heat exchanger increased 2.5 and 3.5 times according to the flat type heat exchanger, respectively.

There are three basic types of exergy losses that occurs in a typical heat exchanger; losses due to the exchange of heat across a finite temperature difference, the fluid friction and heat exchange with the environment which are usually neglected because the heat exchanger surface is insulated to reduce such an exchange of heat [18].

Figs. 9 and 10 show the variation of the exergy loss with hot water mass flow rate for different inlet cold water mass flow rates and counter flow, parallel flow respectively. Figs. 11 and 12 also show the variations exergy loss for the different inlet hot water temperature and counter, parallel flow. Fig. 13 shows the variations of the exergy loss with hot water mass flow rate for different PHE. The experimental results show that heat transfer has great effect on exergy loss. It is clear seen that exergy loss of corrugated type heat exchanger is less than another types.

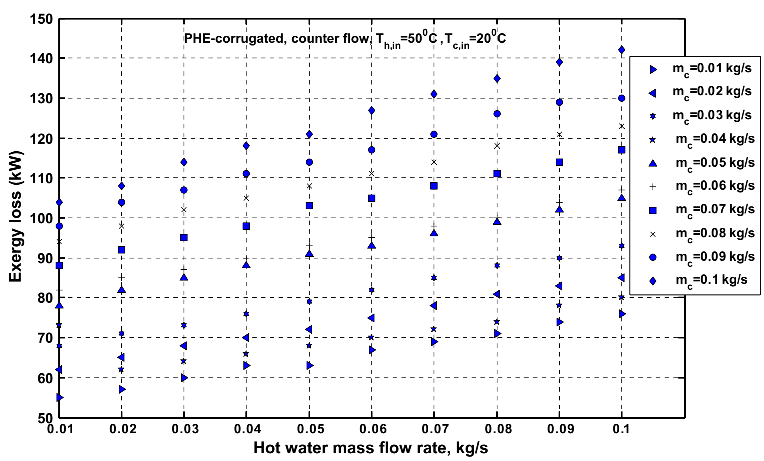


**Fig. 9.** Variations of exergy loss with hot water mass flow rate for different cold water mass flow rate in counter flow heat exchangers.

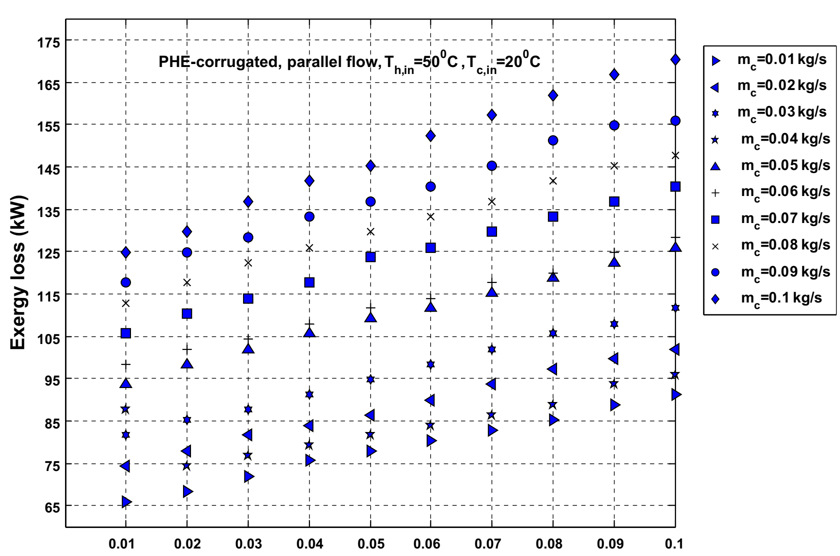


**Fig. 10.** Variations of exergy loss with hot water mass flow rate for different cold water mass flow rate in parallel flow heat exchangers.

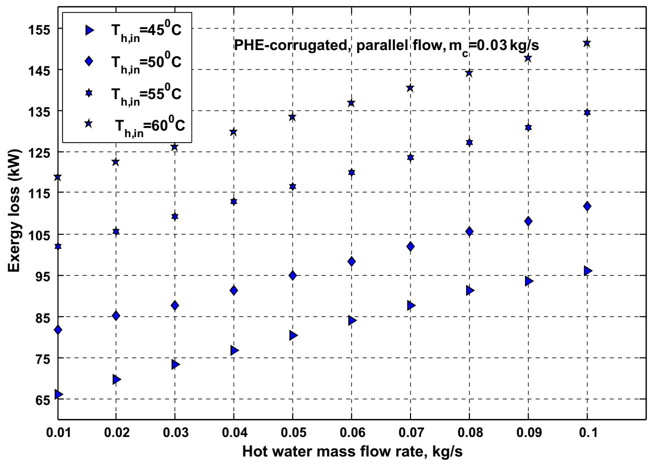


**Fig. 11.** Variations of exergy loss with hot water mass flow rate for different inlet hot water temperatures in counter flow heat exchanger.

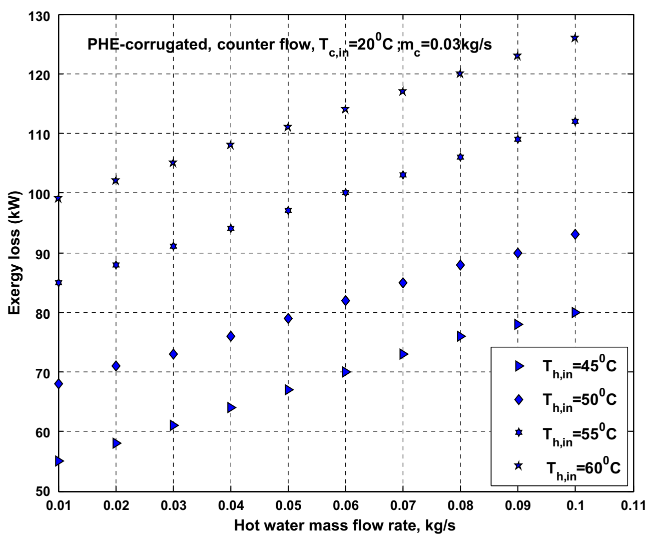


**Fig. 12.** Variations of exergy loss with hot water mass flow rate for different inlet hot water temperatures in parallel flow heat exchanger.

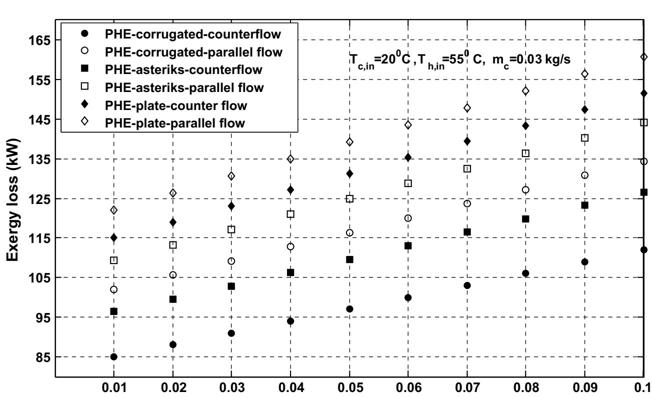


**Fig. 13.** Variations exergy loss with hot water mass flow rate for different PHE.

## 6. Conclusion

In this study, in order to increase heat transfer in plate type heat exchanger by passive method, rectangular fins are located on the plates. In this manner, flow path of the fluid and surface area of the plates is increased.

Nusselt number, the efficiency, pressure drop and exergy loss is discussed comparatively. The conclusions can be drawn from the experimental study of the heat exchanger designed, and show the efficiency of the heat exchanger increases with increasing the fluids' contact surface, pressure drop and mass flow rates due to an enhanced heat transfer to the fluid.

In the experimental results obtained from three different heat exchanger types, it was seen that the heat gained from corrugated type heat exchanger is higher than that of the others. Accordingly, pressure drop increases too. Pressure drop greatly increases the capital costs. Because of this, a thermodynamic optimization should be made between heat transfer and pressure drop. Pressure drop, however, has not an importance for heat exchanger. Because, increasing of the heat exchanger efficiency causes smaller dimensions of the heat exchanger and decreasing of the production costs.

## References

[1] T. Zaleski, K. Klepacke, A plate heat-exchanger-method of calculation, chart and guidelines for selecting plate heat-exchangers configuration, Chem. Eng. Process. 31 (1) (1992) 45–56.
[2] W.M. Rohsenow, J.P. Hartnett, Y.I. Cho, Handbook of Heat Transfer, third ed., McGraw-Hill, New York, 1998.
[3] C.O. Bennet, J.O. Meyers, Momentum Heat and Mass Transfer, third ed., McGraw-Hill, London, 1982.
[4] A.E. Bergles, Techniques to augment heat transfer, in: W.M. Rosenhow, J.P. Hartnett, E.N. Ganic (Eds.), Handbook of Heat Transfer Applications, McGraw-Hill, New York, 1985 (Chapter 3).
[5] A.E. Bergles, The imperative to enhance heat transfer, in: S. Kakac, A.E. Bergles, F. Mayinger, H. Yüncü (Eds.), Heat Transfer Enhancement of Heat Exchanger, Kluwer Academic Publishers, Dordrecht, The Netherlands, 1998.
[6] A. Dewan, P. Mahanta, K.S. Raju, P. Sures Kumar, A review of passive heat transfer augmentation techniques, Proc. Inst. Mech. Eng.: Part A J. Power Energy 218 (2004) 509–527.
[7] H. Baehr, K. Stephan, Warme-und Stoffübertragung, fourth ed., Springer-Verlag, Berlin, 2004.
[8] S. Kakaç, R.K. Shah, W. Aung, Handbook of Single-Phase Convective Heat Transfer, Wiley, New York, 1987.
[9] W.M. Kays, A.L. London, Compact Heat Exchangers, third ed., McGraw-Hill, New York, 1984.
[10] E.M. Sparrow, L.M. Hossfeld, Effect of rounding of protruding edges on heat transfer and pressure drop in a duct, Int. J. Heat Transfer 27 (1984) 1715–1723.
[11] G. Fabbri, Heat transfer optimization in corrugated wall channels, Int. J. Heat Mass Transfer 43 (2000) 4299–4310.
[12] S.G. Kandikar, R.K. Shah, Multi-pass plate heat exchangers-effectiveness-NTU result and guidelines for selecting pass arrangement, J. Heat Transfer 111 (1989) 300–313.
[13] H. Benli, Investigation of the heat transfer of single pass plate type heat exchangers having different surface profiles, M.Sc. Thesis, University of Firat, Elazığ, Turkey 1998 (in Turkish).
[14] A. Durmuş, Heat transfer and exergy loss in a concentric heat exchanger with snail entrance, Int. Commun. Heat Mass Transfer 29 (3) (2002) 303–312.
[15] J.A.W. Gut, J.M. Pinto, Modeling of plate heat exchangers with generalized configurations, Int. J. Heat Mass Transfer 46 (14) (2003) 2571–2585.
[16] J.A.W. Gut, R. Fernandes, J.M. Pinto, C.C. Tadini, Thermal model validation of plate heat exchanger with generalized configuration, Chem. Eng. Sci. 59 (2004) 4591–4600.
[17] İ. Kutbaş, A. Durmuş, H. Eren, E. Turgut, Effect of propeller type swirl generators on the entropy generation and efficiency of heat exchanger, Int. J. Therm. Sci. (in press).
[18] M. Yilmaz, O.N. Sara, S. Karsli, Performance evaluation criteria for heat exchangers based on second law analysis, Exergy Int. J. 4 (2001) 278–294.