Minimizing the Entropic Potential Losses Number in a Gasket-Plate Heat Exchanger

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ABSTRACT: *The entropic potential losses number shows the reduction of energy quality as a function of the entropy produced in a process. In this research, using the Harris Hawks Optimization (HHO) method, the minimization of the entropic potential losses in a gasket-plate heat exchanger has been studied. Considering six design variables: number of plates, cold inlet fluid temperature, mass flow rate of hot fluid, hot inlet fluid temperature, plate pitch, and plate width. The results showed that minimizing the entropic potential losses number increases the heat transfer rate by 103% and the efficiency of the heat exchanger by 27%. Also, the effect of some geometric and process parameters on the entropic potential loss number has also been investigated in this research. It was observed that by increasing the mass flow rate of the hot fluid in the gasket-plate heat exchanger, the entropic potential losses number will increase. It was also seen that the entropic potential loss number will also increase with the increase of hot outlet fluid temperature. It was found that with a decrease of 20 degrees Kelvin in the outlet temperature of the hot fluid, the entropic potential losses number decreased by about 0.06.*

KEYWORDS: *Gasket-plate heat exchanger; Entropic potential losses number; Energy quality; Optimization; Harris Hawks.*

INTRODUCTION

Considering the limitation of energy resources, the increase in demand for energy consumption, and the presence of significant losses in thermal systems, it is necessary to provide a solution to reduce energy loss from these systems. There are two important irreversibility factors in heat exchangers; one is heating conduction due to limited temperature difference and the other is fluid friction. The number of entropic potential losses is caused by the irreversibility of the heat transfer process [1]. *Guo et al.* [2] compared electrical conductivity and thermal conductivity in their research and proposed the concept of entransy to describe the heat transfer ability of an object. Heat transfer efficiency was defined based on the concept of entransy and was introduced as a criterion for optimizing heat exchangers. The results showed that the heat transfer ability decreases in irreversible processes. In other words, entransy is wasted in these processes [3]. Therefore, with the increase of irreversibility in the heat transfer process, the entransy dissipations will also increase. Bejan [4,5] defined and minimized the total rate of entropy generation as the sum of the entropy produced due to the limited temperature difference and frictional

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1021-9986/2023/3/1006-1016 11/$/6.01

pressure drop of the fluid. Minimizing the produced entropy is used in many thermodynamic analyses and heat exchanger optimization. Since in the analysis of heat exchangers, efficiency is a more important parameter than heat transfer, in some cases contradictions are observed. The results of this research showed that the principle of minimizing entropy generation is more useful in optimizing heat exchangers whose purpose is to convert heat into work [6].

Chen et al. [7] minimized entropy generation and entransy dissipations separately and minimized the generation entropy and entropy losses separately for a four-square cavity. They found that minimizing the entropy generation causes the maximum amount of heat to be converted into work in the heat exchangers, while minimizing the entransy dissipations maximizes the efficiency. *Ghodoossi et al.* [8] using the method of minimizing entropy generation analyzed the effect of the complex levels of tree networks on the heat conduction paths and concluded that if the complex levels of heat generation increase, basically the performance of the heat flow does not improve. *Escher et al.* [9] showed in their research that the performance coefficient of parallel channel networks is more than 5 times that of tree networks with constant mass flow rate. This is while almost 4 times more heat is taken from tree networks in the constant pressure gradient. Therefore, the efficiency of the heat exchanger does not always increase by reducing the entropy generation. In their research, *Liu et al.* [10] minimized entropy generation and entransy dissipations. From the comparison of the results, they found that the optimization of heat exchangers in heat transfer processes such as the Brayton cycle, whose main goal is to convert heat into work, by minimizing the entropy generation, has better results.

Shah and *Skiepko* [11] studied the relationship between entropy generation and efficiency for 18 types of heat exchangers. The results showed that in some cases minimizing the entropy generation, not only the efficiency of the heat exchanger did not increase but also decreased. Therefore, they found that minimizing the entropy generation does not necessarily optimize the heat exchanger. *Cao et al.* [12] numerically simulated the effect of the spiral baffle angle of heat exchangers on the resistance to fluid flow and heat transfer. The experimental results showed that the performance of the heat exchanger increases with the increase of the spiral angle of the baffle. *Jamil et al.* [13] presented a numerical model In their research and analyzed gasket-plate heat exchangers with a thermal-hydraulic view. They showed that the flow rate and Chevron angle are very influential on the pressure drop and heat transfer. Also, they found that they made the fooling resistance very accurate in the design of the heat exchanger.

Jutapatet et al. [14] investigated the effect of surface roughness on the condensation of R-134A in gasket-plate heat exchangers. The results showed that the heat transfer coefficient of the rough surface is 31 to 41% higher than that of the smooth surface. While the frictional pressure gradient of the rough surface is about 14 to 29% higher than that of the smooth surface. Finally, they introduced the rough surface as a suitable alternative to smooth surfaces. *Nahes et al*. [15] presented an optimal design method for gasket-plate heat exchangers called Set Trimming for the first time. The innovation of the method introduced in this research was the reduction of the search space, which guaranteed convergence and did not depend on good initial guesses. In their research, they showed that the set-trimming method is faster than other methods in designing a plate heat exchanger. *Soman et al.* [16] using Solidworks software studied the effect of hot and cold fluid flow on heat transfer parameters. In this research, water was chosen as the working fluid, and firstly, the effect of hot fluid flow rate on Nusselt number was studied. Next, the effect of Reynolds number of hot fluid on Nusselt number of cold fluids was investigated. The results showed that increasing the speed of the hot fluid flow increases the Nusselt number. It was also observed that increasing the Reynolds number of hot fluid increases the Nusselt number of cold fluids.

Kumar et al. [17] studied the effect of geometrical parameters on the performance of Chevron plate heat exchangers. In this research, the effect of the Chevron angle on the friction factor, pressure drop and efficiency of the heat exchanger was investigated experimentally. The results showed that a higher Chevron angle leads to a uniform distribution of the fluid flow in the channel and an increase in the efficiency of the heat exchanger. *Zhong et al.* [18] experimentally and numerically investigated the hydraulic performance of plate heat exchangers at low Reynolds numbers. The results showed that the pressure drop of the studied plate heat exchanger increases with the increase of Reynolds number and temperature, while the friction factor of the plate heat exchanger decreases with the increase of Reynolds number, but is not affected by temperature.

Guo et al. [19] modeled the plate heat exchanger based on sensitivity analysis. Compared to other models, the model presented in this research is modeled in less time and also has the ability to track the temperature of the outlet cooling water with high accuracy in different operating conditions of the heat exchanger. The presented model can be used to save and optimize the energy of the circulating cooling water systems.

The highest rate of heat transfer and the lowest value of losses are the main goals of heat transfer processes. Therefore, these goals could be achieved by minimizing the number of entropic potential losses. In irreversible heat transfer processes, entropy is generated. In these processes, the higher the reversibility, the lower the entropy generation. Therefore, it is important to minimize the number of entropic potential losses in energy systems in order to achieve its optimal performance. A review of the previous researches shows that the number of entropic potential losses in the gasket-plate heat exchangers has not been investigated so far. In this study, the number of entropic potential losses was chosen as the objective function of optimization in order to achieve the highest heat transfer rate and the lowest production entropy value. The effect of geometric and process parameters on the objective function is also studied in this research.

Thermodynamic modeling

In this research, a gasket- plate heat exchanger with opposite flow direction and Chevron plates is investigated; a schematic of its plates is shown in Fig. 1.

The research assumptions are:

- The condition is steady state.
- The overall heat transfer coefficient is constant throughout the length of the heat exchanger.
- The flow of hot and cold fluids is uniform in all directions.
- Heat loss is negligible.
- The temperature of hot and cold fluids is uniform.
- The velocity in the cross section of the inlet and outlet flow is uniform.
- The fooling resistance is assumed to be constant.

The efficiency of a counter flow heat exchanger is expressed as follows [20]:

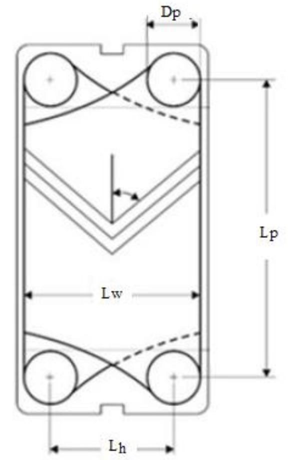

Fig. 1: Schematic of gasket- plate heat exchanger.

$$\varepsilon = \frac{1 - e^{-NTU(1-C^*)}}{1 - C^* e^{-NTU(1-C^*)}} \tag{1}$$

In equation (1), the number of transfer units and the

$$NTU_{max} = \frac{UA_{tot}}{C_{min}} \tag{2}$$

$$C^* = \frac{C_{min}}{C_{max}} \tag{3}$$

The overall heat transfer coefficient in the heat exchanger is [21]:

$$U = R_{f,h} + R_{f,c} + \left[\frac{1}{h_h} + \frac{1}{h_c}\right] + \left(\frac{t}{k}\right)_w \tag{4}$$

Where w, t and k refer to wall thickness and wall conductivity coefficient, respectively. The Nusselt number for both hot and cold fluids is obtained from the following equation [20]:

$$Nu = (Re)^{0.663} Pr^{1/3} \tag{5}$$

The heat transfer coefficient for both sides is obtained using the definition of Nusselt number as follows [22]:

$$h = \frac{Nu \times k}{de} \tag{6}$$

In the above equation, k is the heat transfer coefficient of the fluid conduction and de is the diameter of the Chevron plate, which obtained from the following relation [21]:

$$d_e = 2 \times b \tag{7}$$

$$b = p - t \tag{8}$$

Where p and b are the pitch of the plate and average

Table 1: Characteristics of hot and cold fluids.

Process information	Hot fluid (water)	cold fluid (water)
fluid density(kg/m^3)	985	995
specific heat$(j/kg.K)$	4183	4178
fluid viscosity$(pa.s)$	0.000509	0.000776
fluid thermal $(W/m.K)$ conductivity	0.645	0.617
Prandtl number	3.31	5.19
Fouling Factor$(m^2.W/K)$	0.00005	0

Table 2: Range of optimization variables.

variable	Lower limit	Upper limit
n_t	100	500
$L_w(mm)$	0.5	1
p(mm)	0.001	0.004
$\dot{m}_h(kg/s)$	100	150
$T_{c,i}(K)$	275	285
$T_{h,0}(K)$	300	310

distance, respectively. The Reynolds number of the cold and hot sides is obtained from the following equation [21]:

$$Re = \frac{G_{ch} \times d_e}{\mu} \tag{9}$$

G_{ch} is the mass flux that passes through each channel and is defined by the following equation [21]:

$$G_{ch} = \frac{\dot{m}_{ch}}{A_{ch}} \tag{10}$$

$$A_{ch} = b \times L_w \tag{11}$$

$$L_w = L_h + d_p \tag{12}$$

L_h and d_p are the horizontal distance taken from the center of the port and port diameter respectively. $\dot{m}_{ch}$ is the mass flow rate that passes through each channel and is calculated from the following equation [21]:

$$\dot{m}_{ch} = \frac{\dot{m}}{n_{cp}} \tag{13}$$

$$n_{cp} = \frac{n_t - 1}{2 \times n_p} \tag{14}$$

n_{cp} is the number of channels in each pass. n_p and $n_t$are passes number and plates number, respectively.

Length intensive plate is obtained from the following equation [21]:

$$L_c = n_t \times p \tag{15}$$

The total pressure drop in hot and cold fluids is obtained from the sum of the frictional pressure drop in the channel and the pressure drop in the port [21]:

$$\Delta p_{tot} = \Delta p_c + \Delta p_p \tag{16}$$

The frictional pressure drop in the channel is calculated from the following equation [21]:

$$\Delta p_c = \frac{4fL_v n_p G_{ch}^2}{2\rho d_e} \tag{17}$$

The friction factor for both hot and cold fluids is [21]:

$$f = \frac{1.441}{Re^{0.206}} \tag{18}$$

The pressure drop in the port expressed as [21]:

$$\Delta p_p = \frac{1.4 n_p G_p^2}{2\rho} \tag{19}$$

$$L_v = L_p + d_p \tag{20}$$

$$L_p = \frac{A_{1p}}{L_w} \tag{21}$$

$$A_{1p} = \frac{A_1}{\emptyset} \tag{22}$$

$$G_p = \frac{4\dot{m}}{\pi d_p^2} \tag{23}$$

d_p is port diameter and A_1, $\emptyset$, G_p and $L_v$plate heat transfer surface area, enlargement coefficient, port mass flux and chevron plate height, respectively.

Entropy generation due to fluid friction in the gasket-plate heat exchanger is expressed as the following equation [1].

$$\dot{S}_{gT} = \left(\frac{\dot{m}\Delta P}{\rho}\right)_h \frac{\ln(T_{h,o}/T_{h,i})}{T_{h,o} - T_{h,i}} + \left(\frac{\dot{m}\Delta P}{\rho}\right)_c \frac{\ln(T_{c,o}/T_{c,i})}{T_{c,o} - T_{c,i}} \tag{24}$$

The entropy generation due to the limited temperature difference in the gasket-plate heat exchanger is [1]:

$$\dot{S}_{gP} = \dot{m}_h C_{p,h} \ln(T_{h,o}/T_{h,i}) + \dot{m}_c C_{p,c} \ln(T_{c,o}/T_{c,i}) \tag{25}$$

Total entropy generation is the sum of entropy generation caused by limited temperature difference and fluid friction:

$$\dot{S}_g = \dot{S}_{gT} + \dot{S}_{gP} \tag{26}$$

The entropic potential losses number is defined as equation (27) [1].

$$N_{EPL} = \frac{\dot{S}_g T_\infty}{q} \tag{27}$$

Harris Hawks optimization method [23]

The Harris Hawks optimization method is a population-based optimization method and consists of three steps. The first step is called exploration. Candidate solutions in this method are Harris Hawks. The best candidate solution in each step is considered as the near-optimal answer. Harris Hawks randomly settle in some places to search for the target. In the Harris Hawks optimization method, prey is identified based on two techniques. A: hawks are sitting and waiting according to the position of other hawks and rabbits. B: hawks are randomly sitting on tall trees and waiting. The second step is the transition from exploration to exploitation. A prey's energy is significantly reduced during escape. To model this fact, the energy of a prey is modeled as follows:

$$E = 2E_0\left(1 - \frac{t}{T}\right) \tag{28}$$

In the above equation,E, T and E_0 represent the escape energy of the prey, the maximum number of repetitions and the initial energy of the prey, respectively. In Harris Hawks optimization algorithm, in any iteration, E_0 changes randomly in the interval (-1, 1).As the value of $E_0$is less than zero, the physical concept is that the rabbit is physically more tired, and as its value is greater than zero, it means that the physical strength of the rabbit has increased. The amount of dynamic escape energy decreases during iterations. In this algorithm, if the escape energy is less than one, during the exploitation steps, the hawks use the neighborhood of the solutions to find the location of the rabbit. If the escape energy is not less than one, the exploration step is repeated.

The third step in Harris Hawks optimization method is exploitation. At this step, Harris Hawks make a surprise attack on the intended prey that was discovered in the previous step. Since the prey tries to escape from dangerous situations, hawks use different techniques for chasing. According to the behavior of the prey in pursuit and its skill in escaping, the attack finally ends by catching the prey that is surprised in a very short time.

Since the optimization method of Harris Hawks is a new and capable algorithm in optimization and compared to many older methods, it achieves the optimal value with higher accuracy in fewer iterations, so in this research this method has been used to optimize the objective function.

In this study, the objective function is the entropic potential losses number. The characteristics of hot and cold fluids and the range of design variables are shown in tables (1) and (2), respectively.

$$\text{Objective function} = N_{EPL} \tag{29}$$

RESULTS AND DISCUSION

In order to validate and confirm the simulated code for the gasket-plate heat exchanger, the obtained thermal and hydraulic parameters have been compared with the results of two references in Table 3 with the same input data. The results of the present research and reference (20) were extracted assuming that the fluid properties are constant. The results of reference (24) were extracted assuming that the fluid properties are a function of temperature. As can be observed from the table, the results of the present study compared to the results of both reference has a good agreement.

Fig. 2 using HHO and GA methods shows the changes of the best values of the entropic potential losses number versus the number of iterations. The value of the objective function is calculated in any iteration and its lowest value is selected. As can be seen, the value of entropic potential losses number, which is caused by frictional losses and limited temperature difference, decreases in any iteration until it remains constant after several iterations. Since the value of the objective function does not change in the next iterations, the calculation of the optimal objective function is finished. From the comparison of HHO and GA optimization methods, it can be seen that HHO method has a much higher convergence speed and its response has converged in 100 repetitions, while the response of the GA method has converged after about 700 repetitions. Therefore, HHO method is more efficient in optimizing the present work.

Table 3: Comparison of the results of this research with the results of two other references.

Parameter	present study	Ref [20]	Ref [24]
$\Delta P_{t,h}(kPa)$	281.06	286.214	298.810
$\Delta P_{t,c}(kPa)$	301.45	305.784	325.890
$h_h(W/m^2K)$	32630	32844	29186
$h_c(W/m^2K)$	27655	27811	24361
f_h	0.2035	0.2040	0.2131
f_c	0.2214	0.2210	0.2359

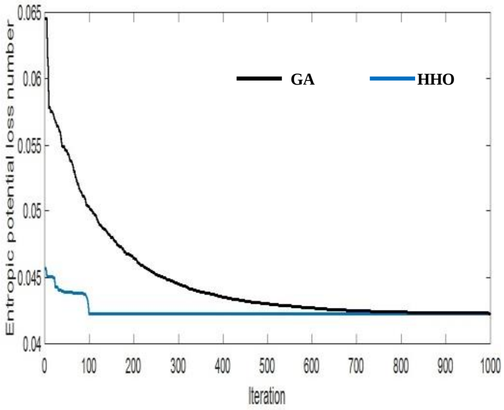

Fig. 2: Entropic potential losses number versus the number of iterations.

The optimization results are compared with the initial results of the gasket-plate heat exchanger in Table 4. It can be seen that the minimization of the entropic potential losses number with the Harris Hawks optimization method has increased the heat transfer rate and the efficiency of the heat exchanger by 103% and 27%, respectively.

Fig.3. shows the effect of increasing the plate number of gasket-plate heat exchanger on the entropic potential losses number. As can be seen, increasing the number of plates reduces the entropic potential losses number. In fact, increasing the number of plates decreases the mass flow rate in each channel and also decreases the friction factor. Therefore, the fluid pressure drop decreases and the entropic potential losses number also decreases. Fig.4. shows the effect of increasing the heat transfer area on the entropic potential losses number. If the increase in the heat transfer area is due to the increase in the number of plates and other geometrical parameters are assumed to be constant, it can be seen that with the increase in the heat transfer area, the entropic potential losses number decreases. Because in this case, the fluid pressure drops decreases and therefore the entropy generation and the entropic potential losses number also decreases.

The effect of increasing the plate pitch on the number of entropic potential losses is shown in Fig. 5. It can be seen that with the increase of plate pitch, the entropy potential losses number decreases with a large slope at first, and then the increase of the plate pitch has no noticeable effect. Increasing the plate pitch up to 2 mm will cause a significant decrease in the fluid pressure drop, and its further increase will reduce the pressure drop with a smaller slope. Since the pressure drop has an inverse relationship with the plate pitch and a direct relationship with the entropic potential losses number, so the plate pitch has an inverse relationship with the entropic potential losses number.

The relationship between the mass flow rate of hot fluid and the entropic potential losses number is shown in Fig. 6. The diagram shows that with the increase in the mass flow rate of the hot fluid, the entropic potential losses number increases. As it is clear from the Equations (24) and (25), the entropic potential losses number has a direct relationship with the fluid flow rate. Also, with the increase in the mass flow rate of the hot fluid, the pressure drop of the hot fluid increases and therefore the entropy generation due to the friction of the fluid also increases. Since we know that with the increase in the mass flow rate of hot fluid, the entropy generation increases, an increase in the mass flow rate of hot fluid will lead to an increase in the entropic potential losses number. Increasing the mass flow rate of the hot fluid, the Reynolds number increases, and considering that the Reynolds number has a direct relationship with the Nusselt number, therefore, increasing the entropic potential losses number increases the Nusselt number. The relationship between the hot fluid Nusselt number and the entropic potential losses number when the mass flow rate of the hot fluid is increased is shown in Fig. 7.

Table 4: Comparison of the initial and optimal results.

variable	Initial value	Optimal value
n_t	105	500
$L_w(mm)$	0.63	1
$p(mm)$	0.0036	0.004
$\dot{m}_h(kg/s)$	140	150
$T_{h,i}(K)$	318	300
$T_{c,i}(K)$	295	285
N_{EPL}	0.0741	0.042
ε	0.55	0.82
$q(kW)$	11712.4	23843

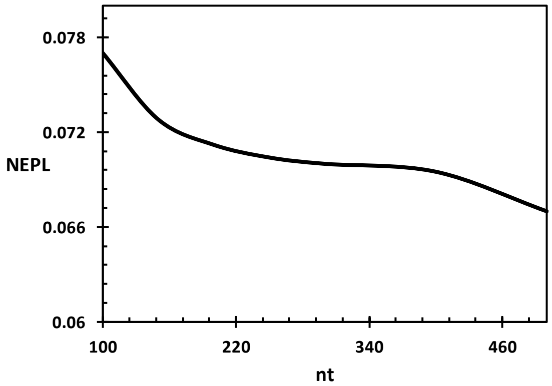

Fig. 3: The effect of increasing the plate number on the entropic potential losses number.

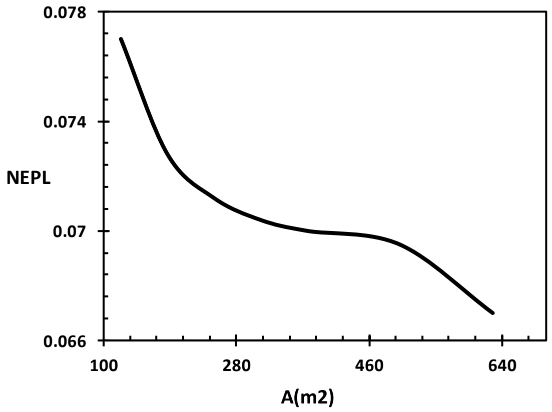

Fig. 4: The effect of increasing the heat transfer area on the entropic potential losses number.

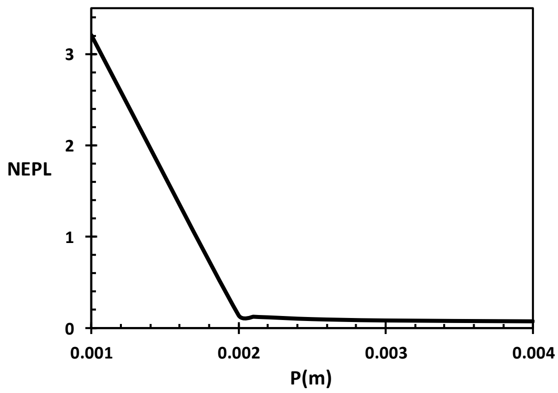

Fig. 5: The effect of increasing the plate pitch on the entropic potential losses number.

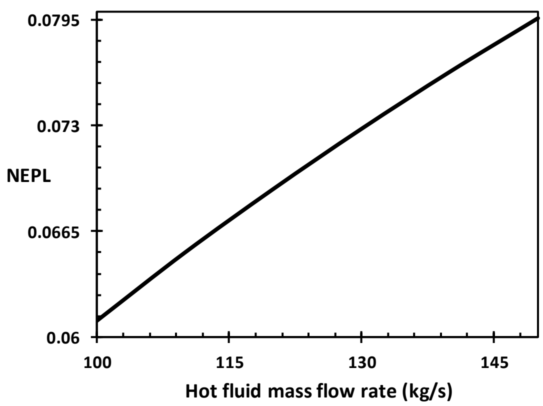

Fig. 6: The effect of the mass flow rate of hot fluid on the entropic potential losses number.

The effect of increasing the width of the plates on the entropic potential losses number is shown in Fig. 8. It can be seen that increasing the width of the plates decreases the entropic potential losses number. Increasing the width of the plates, the area of each channel increases, and the mass flux in each channel also increases. So, the Reynolds number and the fluid pressure drop decrease and it leads to decrease in the entropic potential losses

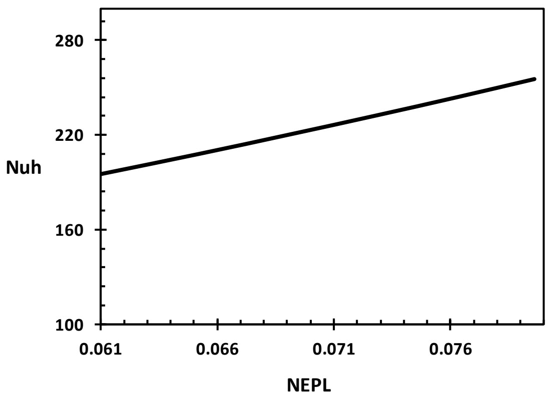

Fig. 7: The effect of the entropic potential losses number on the Nusselt number of hot fluid.

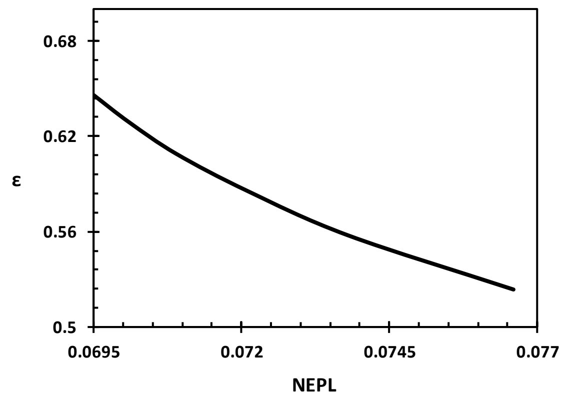

Fig. 9: The effect of the entropic potential losses number on the efficiency of the gasket- plate heat exchanger.

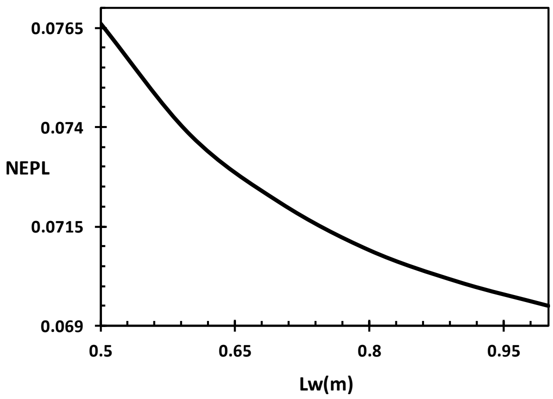

Fig. 8: The effect of the plate width on the entropic potential losses number.

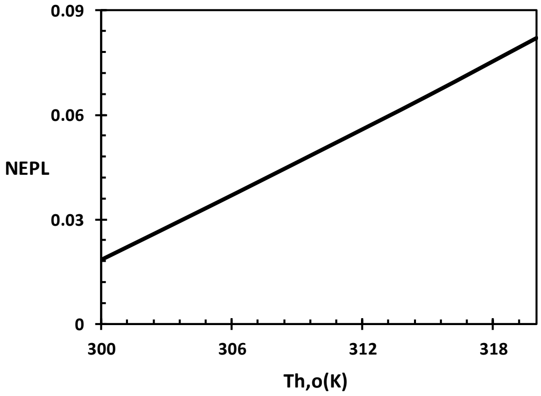

Fig. 10: The effect of the hot fluid outlet temperature on the entropic potential losses number.

number. Fig. 9 shows that the reduction in the width of the plates and the increase in the entropic potential losses number caused by it, reduces the efficiency of the gasket-plate heat exchanger.

As the temperature of the hot fluid outlet increases, the entropic potential losses number increases. This result can be seen from Fig. 10. From the Equation (24), it can be concluded that the entropy generation due to the limited temperature difference has a direct relationship with the outlet temperature of the hot fluid. Therefore, increasing the hot fluid outlet temperature will lead to an increase in the entropic potential losses number. In other words, the results show that with an increase of 20 degrees Kelvin in the hot fluid outlet temperature, the entropic potential losses number has increased about 0.06.

The effect of the height of the heat exchanger plates on the entropic potential losses number is investigated in Fig.11.The results show that increasing the height of the heat exchanger plates causes a slight increase in the entropic potential losses number. Because of increasing the height of the plates slightly increases the pressure drop, and since the entropy produced due to fluid friction has a direct relationship with the fluid pressure drop, the entropic potential losses number also increases slightly.

Fig.12 shows the effect of the enlargement coefficient on the entropic potential losses number. It can be observed that increasing the enlargement coefficient has a negligible effect on the entropic potential losses number. In other words, increasing the enlargement coefficient causes an insignificant increase in the fluid pressure drop, which causes an insignificant increase in the entropic potential losses number. So increasing the enlargement coefficient can be neglected due to its very low effect.

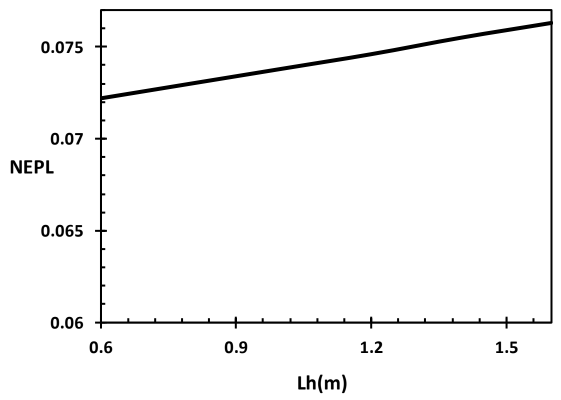

Fig. 11: The effect of the height of the heat exchanger plates on the entropic potential losses number.

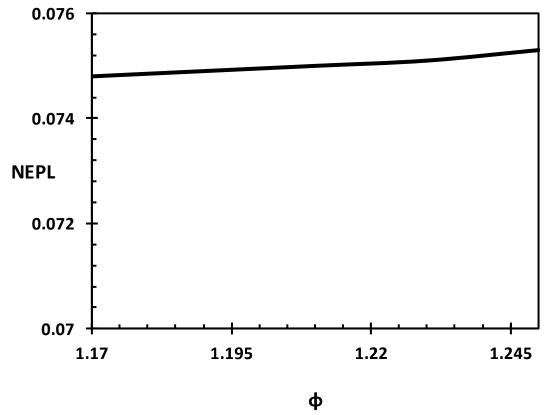

Fig. 12: The effect of enlargement coefficient on the entropic potential losses number.

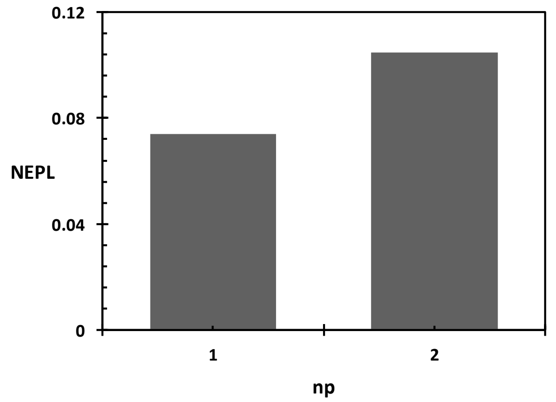

Fig. 13: The effect of the pass numberon the entropic potential losses number.

Fig.13. depicts the relationship between the entropic potential losses number and the number of passes in the gasket-plate heat exchanger. It is clear that for the number of passes equal to 1, the entropic potential losses number is much lower than when the number of passes be equal to 2. In fact, increasing the number of passes, the pressure drop in the heat exchanger will increase significantly. Therefore, the entropy produced due to fluid friction will increase. As a result, the total entropy produced and also the number of entropic potential losses will increase.

CONCLUSIONS

In this research, the entropic potential losses number in gasket-plate heat exchangers was minimized as an objective function using Harris Hawks optimization method. Considering six design variables: number of plates, width of plates, mass flow rate of hot fluid, inlet temperature of cold fluid, outlet temperature of hot fluid and plate pitch. The results showed that minimizing the entropic potential losses number increases the efficiency of the heat exchanger by 27%. Also, it is found that minimizing the entropic potential losses number will lead to a significant increase in the heat transfer rate. The effect of the geometrical and process parameters of the gasket-plate heat exchanger on the entropic potential losses number was also investigated in this study. The relationship between the Nusselt number and the entropic potential losses number was also studied and it was observed that increasing the mass flow rate and the entropic potential losses number, the Nusselt number also increases. In operational conditions and for gasket-plate heat exchangers that are working and it is not possible to change the geometrical parameters, reducing the mass flow rate and the cold fluid inlet temperature are suggested as two solutions to reduce the entropic potential losses number.

Nomenclature

A	Overall heat transfer area, m^2
A_{ch}	Channel flow area, m^2
A_1	Heat transfer area of a plate, m^2
b	Average distance, m
C_p	Specific heat capacity, J/kg.K
C^*	Heat capacity ratio(−)
d_e	Equivalent diameter, m
d_P	Port diameter, m
f	Friction coefficient(−)
G_{ch}	Mass flux in each channel, $kg/m^2.s$
C_P	Mass flux in the ports, $kg/m^2.s$

h	Convection heat transfer coefficient, W/m^2.K
K	Conductive heat transfer coefficient, W/m.K
L_v	Chevron plate height, m
L_h	Horizontal distance in openings, m
L_P	Heat exchanger plate height, m
L_w	Plate width, m
L_c	Length intensive of plates or length of cut, m
$\dot{m}$	Mass flow rate, kg/s
$\dot{m}_{ch}$	Mass flow rate in each channel, kg/s
Nu	Nusselt number (-)
P	Plates pitch, m
Pr	Prandtl number (−)
q	Heat transfer rate, w
Re	Reynolds number(−)
R_f	Fouling resistance, m^2kw^{-1}
T	Temperature, K
t	Plate thickness, m
U	Overall heat transfer coefficient, W/m^2.K

Greek abbreviation

ΔP	Pressure drops, kPa
μ	Viscosity, Pa.s
ρ	Density, kg/m^3
ε	Effectiveness(−)
∅	Enlargement coefficient (−)
β	Chevron angle of plates, degree

Subscrtipts

h	Hot
c	Cold
ch	Channel
i	Input
o	Output

Received : Nove. 3, 2022 ; Accepted : Feb. 20, 2023

REFERENCES

[1] Chahartaghi M., Alavi S.E., Sarreshtehdari A., Investigation of Energy Consumption Reduction in Multistage Compression Process and Its Solutions, **50**: 219-227 (2019).

[2] Guo Z.Y., Cheng X.G., Xia Z.Z., Least Dissipation in Principle of Heat Transport Potential Capacity and Its Application in Heat Conduction Optimization, *Chinese Science Bulletin*, **48**: 406-410 (2003).

[3] Han G.Z., Guo Z.Y., Physical Mechanism of Heat Conduction Ability Dissipation and Its Analytical Expression, *Proceeding of the CSEE*, **27**: 98-102 (2007), [in Chinese].

[4] Bejan A., "*Entropy Generation Through Heat and Fluid Flow*", Wiley, New York,(1982).

[5] Bejan A., "Entropy Generation Minimization". CRC Press, Florida, (1996).

[6] Hesselgreaves J.E. Rationalisation of Second Law Analysis of Heat Exchangers, *International Journal of Heat and Mass Transfer*, **43**: 4159-4204 (2000).

[7] Chen Q., Wang M., Pan N., Guo Z.Y., Optimization Principles for Convectiveheat Transfer, *Energy*, **34**: 1199-1206 (2009).

[8] Ghodoossi L., Entropy Generation Rate in Uniform Heat Generating Area Cooled by Conducting Paths: Criterion for Rating the Performance of Constructal Designs, *Energ. Convers Manage*, **45**: 2951–2969 (2004).

[9] Escher W., Michel B., Poulikakos D., Efficiency of Optimized Bifurcating Tree-Like and Parallel Microchannel Networks in the Cooling of Electronics, *Int. J. Heat. Mass. Transfer*., **52**: 1421–1430 (2009).

[10] Liu X.B., Meng J.A., Guo Z.Y., "Entropy Generation Extremum and Entransy Dissipation Extremum for Heat Exchanger Optimization, *Chinese Science Bulletin*, **54**: 943-947 (2009).

[11] Shah R.K., Skiepko T., Entropy Generation Extrema and their Relationship with Heat Exchanger Effectiveness-Number of Transfer Unit Behavior for Complex Flow Arrangements, *J. Heat Transfer*, **126**: 994-1002 (2004).

[12] Cao X., Du W.J., Ji S., Cheng L., Influence of Overlap Size on Shell Side Performance of Heat Exchangers with Helical Baffles, *Proc. CSEE*, **32**: 78-84 (2012).

[13] Jamil M.A, Zia U.D., Talha S.G., Haseeb Y., Syed M.Z., Thermal-Hydraulic Characteristics of Gasketed Plate Heat Exchangers as a Preheater for Thermal Desalination Systems, *Energy Conversion and Management*, **205**: 112425 (2020).

[14] Jutapat S., Omid M., Ahmet S.D., Somchai W., Effect of Surface Roughness on the Condensation of R-134a in Vertical Chevron Gasketed Plate Heat Exchangers, *Experimental Thermal and Fluid Science*, **91**: 54-63 (2018).

[15] Nahes A.L.M., Martins N.R., Bagajewicz M.J., Bagajewicz M.J., Costa A.L.H., Computational Study of the Use of Set Trimming for the Globally Optimal Design of Gasketed-Plate Heat Exchangers, *Ind. Eng. Chem. Res.*, **60(4)**: 1746–1755 (2021).

[16] Soman P., Radhakrishnan T., Computational Fluid Dynamics Modelling and Analysis of Heat Transfer in Multichannel Dimple Plate Heat Exchanger, *Iranian Journal of Chemistry and Chemical Engineering (IJCCE)*, **41(6)**: 2071-2086 *(*2021).

[17] Kumar B., Soni A., Singh S.N., Effect of Geometrical Parameters on the Performance of Chevron Type Plate Heat Exchanger, *Experimental Thermal and Fluid Science*, 126-133 (2018).

[18] Zhong Y., Deng K., Zhao S., Hu J., Zhong Y., Li Q., Wu Z., Lu Z., Wen Q., Experimental and Numerical Study on Hydraulic Performance of Chevron Brazed Plate Heat Exchanger at Low Reynolds Number, *Processes*, **8**: 1076 (2020)

[19]Guo Y., Wang F., Jia M., Zhang S., Modeling of Plate Heat Exchanger Based on Sensitivity Analysis and Model Updating, *Chemical Engineering Research and Design*, **138**:418-432 (2018).

[20] Mehergan M., Alavi S.E., Thermal and Economic Optimization of an Intercooler of Three-Stage Compressor, *Energy Equipment and Systems*, **9(3)**: 261-278 (2021).

[21] Chahartaghi M., Alavi S.E., Minimization of Entransy Dissipations of a Finned Shell and Tube Heat Exchanger, *Journal of Computational Applied Mechanics*, **50(2)**: 246-255 (2019).

[22] Rodriguez Cabal M.A., Valdes Ortiz M.J., Grisales Noreña L.F., Montoya Giraldo O.D., Ardila Marin J.G., Optimizing the Overall Heat Transfer Coefficient of a Spiral Plate Heat Exchanger Using Gams, *Iranian Journal of Chemistry and Chemical Engineering (IJCCE)*, **41(9)**: 3214-3222 (2022).

[23] Heidari A.A., Mirjalili S., Faris H., Aljarah I., Mafarja M., Chen H., Harris Hawks Optimization: Algorithm and Applications, *Future Generation Computer Systems*, 849-872 (2019).

[24] Hajabdollahi H., Naderi M., Adimi S., A Comparative Study on the Shell and Tube and Gasket-Plate Heat Exchangers: the Economic Viewpoint, *Applied Thermal Engineering*, **92**: 271-282 (2016).