JCAMECH

Vol. 50, No. 2, December 2019, pp 246-255
DOI: 10.22059/jcamech.2018.255831.263

Minimization of Entransy Dissipations of a Finned Shell and Tube Heat Exchanger

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ARTICLE INFO

Article history:
Received: 14 April 2018
Accepted: 23 May 2018

Keywords:
Entransy dissipation
Genetic algorithm
Optimization
Shell and tube heat exchanger

ABSTRACT

Improving heat transfer and performance in a radial, finned, shell and tube heat exchanger is studied in this study. According to the second law of thermodynamics, the most irreversibilities of convective heat transfer processes are due to fluid friction and heat transfer via finite temperature difference. Entransy dissipations are due to the irreversibilities of convective heat transfer. Therefore, the number of entransy dissipation is considered as the optimization objective. Thirteen optimization variables are considered, including the number of tubes, tube diameter, tube length, fin height, fin thickness, the number of fins per inch length of tube and baffle spacing ratio. The "Delaware modified" technique is used to determine heat transfer coefficients and the shell-side pressure drop. In this technique, the baffle cut is 20 percent. The results show that using genetic algorithm the optimization can improve the heat transfer by 13percent and performance of heat exchanger increased by 18percent. In order to show the accuracy of the algorithm the results compared to the particle swarm optimization.

1. Introduction

Respective to the increase in energy demand and the intense reduction of fossil fuels such as oil and coal, efficient energy use, leads to saving the resources. Reducing unnecessary energy losses and improving heat exchangers performance is important since heat exchangers are one of the most important devices in thermal systems and are widely used in the chemical industry, oil refineries, power engineering, etc.

Criteria for performance evaluation of the heat exchanger are generally classified into two categories: one based on the first law of thermodynamics and the other, based on the combination of the first and second laws of thermodynamics. In a heat exchanging process through a heat exchanger heat transfer via a limited temperature difference, fluid friction under pressure drop and mixing of two fluids can usually happen. These processes are considered as irreversible thermodynamic processes. Thus, in the last decades, the study on the second category has attracted a lot of attention. Some of them have been argued in the following passage.

Bejen [1] developed the method of entropy generation minimization for the optimal design of heat exchangers. Also, identifying the factors of entropy generation, it attempted to minimize entropy generation. He calculated and evaluated two types of irreversibilities in heat exchangers as follows:

One is the thermal conduction due to the limited temperature difference and the other is the frictional pressure drop due to fluid circulation in the system. Therefore, the overall rate of entropy generation is due to the total entropy generated by the heat conduction and fluid friction. However, among the various principles in thermodynamics, most investigations are still underway on the principle of entropy generation minimization. Accordingly, the method of entropy generation minimization is widely used in the modeling and optimization of thermal systems due to thermodynamic defects in heat transfer, mass transfer and irreversible flow of fluids. This shows some contradictions in the design of heat exchangers [2] because in the design of heat exchangers, efficiency is crucial. Therefore, the method of entropy generation minimization is more useful in the processes of conversion heat to work.

Guo et al. [3] introduced a new concept, comparing the electrical and thermal conductivity and establishing a one-to-one correspondence between thermal and electrical concepts called "entransy" that describes the ability of heat transfer of body. On the basis of entransy, the heat transfer efficiency is defined, and according to the efficiency, the optimal design of the heat

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exchanger is discussed. They found that in the irreversible processes, the entransy is wasted and ability of heat transfer is reduced [4]. It means that, the higher entransy losses or dissipations, implying a higher degree of irreversibility in the heat transfer processes. Therefore, it can be used as a benchmark for performance evaluation of heat exchanger.

Wang et al. [5] extracted the entransy transfer equations, and studied the entransy transmission processes for multimodal viscous fluids that exposed to conduction and convection heat transfer, mass transfer, and chemical reactions.

Chen and Ren [6] considering the ratio of the temperature difference to heat flow, defined a concept called general thermal resistance for convection heat transfer processes and developed a theory of minimizing thermal resistance to optimize the convection heat transfer processes. They also found that the principle of minimizing thermal resistance is equivalent to the principle of minimizing entransy dissipation.

Xia et al. [7] considering the minimum entransy dissipations as the objective function, investigated the optimal parameters for a heat exchanger with two fluids at conditions of constant heat transfer rate.

Guo et al. [8] showed that the overall entransy dissipation rate is minimized when the local entransy loss rate is distributed uniformly throughout the heat exchanger. They also found that the results of the optimization of the principle of uniformity of the field of temperature difference and the principle of uniformity of the distribution of entransy dissipations are compatible with each other, when the heat transfer rate, the heat transfer surface and the heat transfer coefficient between the fluids are known.

Wei et al. [9] presented an optimal design approach for the dry cooling system of a power plant by entransy method. The entransy dissipation equations include that the ambient air heated by heat exchangers and circulating water heated by the exhausted steam of turbine, were derived. Combined with the force balance equation of natural draft dry cooling tower, the effective parameters on performance and annual cost of the cooling system were considered. Finally, the entransy based optimization model was performed to minimization of annual cost of the system.

Xu et al. [10] presented entransy and entransy dissipation theory for simplification of analysis of heat exchanger networks. In their work the entransy balance equation was considered as a constraint to replace the conventional constraints of the heat transfer and energy balance for the heat exchangers, to simplify the calculation of Lagrange multiplier method for the optimization of heat exchanger networks. They showed that, the number of constraints and variables were decreased and the solution method was simplified and also they studied about the calculation time of their method.

Abed et al. [11] presented an optimization method for design of shell and tube heat exchangers. An electromagnetism-like algorithm was used to optimize the capital cost of the heat exchanger and designing a high operating performance for it. The optimization algorithm was proposed consideration of geometric parameters and maximum allowable pressure drop. A mathematical model of the heat exchanger design with using of a computer code was developed for the optimal conditions of the heat exchanger. The results showed that the optimization method lead to a significant decrease in the heat transfer area.

Liu et al. [12] investigated the principles of extremum entropy and entransy generation for the optimization of heat exchanger and concluded that the principle of entropy generation minimization for those heat exchangers that work in the Brayton cycle is better than principles of extremum entransy dissipation, while for principles of extremum entransy dissipations, the results will be better for those heat exchangers that are used for heating and cooling.

Xu et al. [13, 14] extracted the term of entransy dissipation due to thermal conduction and fluid friction in heat exchangers. They found that in performance evaluation and optimal design of heat exchangers, it is essential that the entransy dissipation be dimensionless. Then, the dimensionless method was introduced for the entransy dissipations in the heat exchangers and used to performance evaluation of the heat exchangers. Therefore, changing entransy dissipations in dimensionless form and considering it as the objective function, they changed the geometry of heat exchanger and optimized it.

Venkata et al. [15] optimized the shell and tube heat exchanger using Jaya algorithm. The optimization results showed that using Jaya algorithm has better results compared to PSO and GA methods.

Mirzaei et al. [16] using genetic algorithm presented a multi-objective optimization algorithm. Cost and thermal efficiency were considered as the objective functions in this research. In this study, the shell and tube efficiency increased by 20%. The results showed that in industrial applications where high efficiency is not needed, the constructal heat exchanger is not useful.

Lemos et al. [17] optimized the shell and tube heat exchanger by taking into account the fouling factor. The objective function in this study was to minimize the heat transfer area or the total cost. In this research, an optimal heat exchanger using linear method was designed.

Lei et al. [18] used Numerical simulations to design a shell and tube heat exchanger with louver and conventional Segmental baffles. The comparisons of shell and tube heat exchanger with louver baffles and shell and tube heat exchanger with conventional Segmental baffles showed that, at the similar conditions, the shell and tube heat exchanger with louver baffles requires lower pumping power.

Bichkar et al. [19] Examined the effect of baffle types on the shell and tube heat exchangers. The results showed that helical baffles causes a lower pressure compared to single segmental baffles. It was also observed that the use of helical baffles in comparison with Single segmental baffles and double segmental baffles would increase overall efficiency.

Gu et al. [20] used entransy theory and genetic algorithm to optimize shell and tube heat exchangers. Results showed that increasing the shell-side velocity, decreases the entransy dissipation thermal resistance. In this research, the entransy dissipation thermal resistance of 7% was reduced by optimizing the shell and tube heat exchanger.

In the present study, minimization the entransy dissipations based on thermal conduction and fluid friction in dimensionless form and considering it as an objective function, the optimization of the finned, shell and tube heat exchangers has been performed. With the help of MATLAB software [21], an optimization algorithm has been designed and implemented. Most prior optimizations use the Kern method for calculating the heat transfer coefficient and the shell-side pressure drop, but in this research a Simplified Delaware method used which has better accuracy than the Kern method. The investigation is based on an industrial case. In addition, an extensive analysis of shell and tube heat exchanger using the principles of entransy has been presented. For this purpose, the effect of different parameters such as the inlet temperature, the number of fins per unit length of tube, the number of baffles in the shell and the mass flow rate, on the entransy and etc., have been investigated. Considering that such an extensive research has not been carried out in previous work, this study seems to be necessary.

2. Thermal modeling of finned shell and tube heat exchanger

Regarding the usual assumptions such as the absence of any longitudinal thermal conductivity, and negligible kinetic and potential energy variations, and without heat loss between the heat exchanger and its surroundings, the equilibrium equation for the heat exchanger as shown in Figures 1 and 2, can be calculated as [22]:

$$\dot{Q} = \left(\dot{m}C_p\right)_h \left(T_{h,i} - T_{h,o}\right) = \left(\dot{m}C_p\right)_c \left(T_{c,o} - T_{c,i}\right) \tag{1}$$

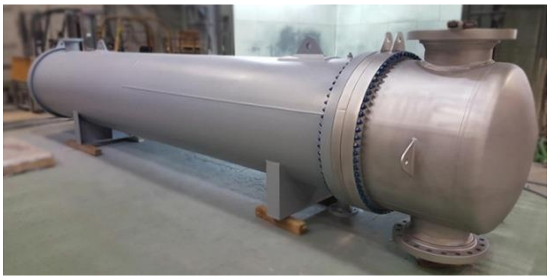

Figure 1. Schematic of shell and tube heat exchanger, type AES [23].

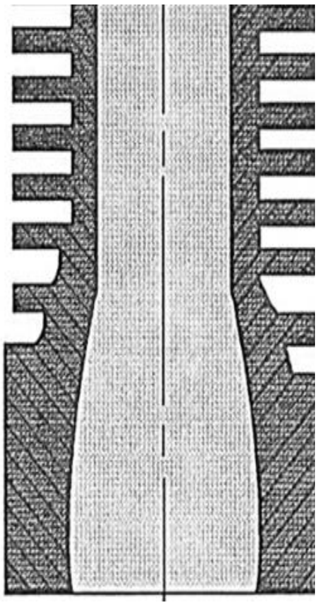

Figure 2. Schematic illustration of a radial low-fin tube [24].

In Eq. (1) $\dot{Q}$ is the actual heat transfer rate, $\dot{m}$ is the mass flow rate of the fluid, C_p is the specific heat of the fluid at constant pressure and T represents the temperature. The indices of h and c, indicate a hot and cold fluid, respectively, and the indices i and o refer to the input and output of the heat exchanger, respectively. The effectiveness of the heat exchanger is also can be expressed as follows [25]:

$$\varepsilon = \frac{2}{\left(1+c^*\right) + \left(1+c^{*2}\right)^{0.5} \coth\left(\frac{NTU}{2}\left(1+c^{*2}\right)^{0.5}\right)} \tag{2}$$

$$NTU = \frac{U_0 A_{tot}}{C_{\min}} \tag{3}$$

$$C^* = \frac{C_{\min}}{C_{\max}} = \frac{\min\left(\left(\dot{m}C_p\right)_s, \left(\dot{m}C_p\right)_t\right)}{\max\left(\left(\dot{m}C_p\right)_s, \left(\dot{m}C_p\right)_t\right)} \tag{4}$$

Here $c^*$is the thermal capacity rate andNTU is the number of thermal units. $A_{tot}$is the total external surface area of a finned tube heat exchanger and$U_0$is the overall heat transfer coefficient [24].

$$A_{tot} = A_{fins} + A_{prime} \tag{5}$$

$$A_{tot} = \left[2n_f \pi\left(r_{2c}^2 - r_1^2\right) + 2r_1\pi\left(1 - n_f \tau\right)\right] L.N_t \tag{6}$$

$$U_0 = \left[\frac{A_{tot}}{A_i h_i} + \frac{R_{Di} A_{tot}}{A_i} + \frac{A_{tot} \ln\frac{d_o}{d_i}}{2\pi K_{tube} L} + \frac{1}{h_o \eta_w} \frac{R_{Do}}{\eta_w}\right] \tag{7}$$

Here L, N_t, d_i, d_o, R_{Di}, R_{Do}, K_w, r_{2c}, r_1 and n_f are tube length, number of tubes, inside and outsidediameters of the tube, tube and shell side fouling resistances, thermal conductivity of tube wall, corrected fin radius, external radius of root tube and number of fins per unit length of tube, respectively. Also, A_i denotes the internal surface area of the tube and can be written as follow [24, 26]:

$$A_i = \pi d_i L N_t \tag{8}$$

$$\eta_w = \left(\frac{A_{prime}}{A_{tot}}\right) + \eta_f \left(\frac{A_{fins}}{A_{tot}}\right) \tag{9}$$

$$\eta_f = \frac{\tanh\left(m\varphi\right)}{\left(m\varphi\right)} \tag{10}$$

$$\varphi = \left(r_{2c} - r_1\right)\left[1 + 0.35\ln\left(\frac{r_{2c}}{r_1}\right)\right] \tag{11}$$

$$m = \left(\frac{2h_o}{K\tau}\right)^{0.5} \tag{12}$$

$$r_{2c} = r_2 + \frac{\tau}{2} \tag{13}$$

Where, η_w denotes the weighted efficiency of the finned surface and η_f is the fin efficiency. φ is a parameter in the equation for efficiency of annular fins, τ is the fin thickness and k is the thermal conductivity.

The tube-side heat-transfer coefficient, hi, is computed using the following equation [27-29]:

$$
\begin{aligned}
h_i &= \left(\frac{k_t}{d_i}\right)0.116\left(\mathrm{Re}_t^{2/3}-125\right)\mathrm{Pr}_t^{1/3}\left(1+\frac{d_i}{L}\right)^{2/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} && \text{for } \mathrm{Re}_t > 10^4 \\
h_i &= \left(\frac{k_t}{d_i}\right)0.027\,\mathrm{Re}_t^{0.8}\,\mathrm{Pr}_t^{0.4}\left(\frac{\mu}{\mu_w}\right)^{0.14} && \text{for } 2100 < \mathrm{Re}_t < 10^4 \\
h_i &= \left(\frac{k_t}{d_i}\right)1.86\left(\frac{\mathrm{Re}_t\,\mathrm{Pr}_t\,d_i}{L}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} && \text{for } \mathrm{Re}_t < 2100
\end{aligned} \tag{14}
$$

The above equation is acceptable for $\left(\frac{\mathrm{Re}_t\,\mathrm{Pr}_t\,d_i}{L}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} > 2$. For $\left(\frac{\mathrm{Re}_t\,\mathrm{Pr}_t\,d_i}{L}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} < 2$ the tube-side heat-transfer coefficient calculated as follow [29]:

$$h_i = 3.66\frac{k_t}{d_i} \tag{15}$$

Here, k_t, Pr_t, μ and μ_w are the tube inside fluid thermal conduction coefficient, the tube side Prandtl number, the viscosity and fluid viscosity evaluated at average temperature of tube wall. Also, the tube side Reynolds number represented as follow [24]:

$$\mathrm{Re}_t = \frac{4\dot{m}_t\left(\frac{n_p}{N_t}\right)}{\pi d_i \mu} \tag{16}$$

Where $\dot{m}_t$ and n_p are the mass flow rate and the number of tube passes, respectively.

The pressure drop due to fluid friction in the tubes is given by following equations [24, 27, 30]:

$$\Delta p_f = f\,\frac{L.n_p}{2000 d_i}\,\frac{G_t^2}{s.\left(\frac{\mu}{\mu_w}\right)^{0.14}} \tag{17}$$

$$G_t = \frac{4\dot{m}_t\left(\frac{n_p}{N_t}\right)}{\pi d_i^2} \tag{18}$$

$$
\begin{aligned}
f &= \frac{64}{\mathrm{Re}_t} && \mathrm{Re}_t < 2300 \\
f &= 0.4137\,\mathrm{Re}_t^{-0.2585} && \mathrm{Re}_t > 2300
\end{aligned} \tag{19}
$$

The pressure drop due to tube entrance and exit effects is given by following equation [24, 27, 30]:

$$\Delta p_r = 5\times10^{-4}\frac{\alpha_r.G_t^2}{s} \tag{20}$$

α_r is the number of velocity heads allocated for the tube-side minor pressure losses, which can be inferred from Table 1. [24].

Table 1. Number of velocity heads allocated for minor losses on tube side.

Flow regime	Regular tubes	U-tubes
Turbulent	$2n_p - 1.5$	$1.6n_p - 1.5$
Laminar, Re ≥ 500	$3.25n_p - 1.5$	$2.38n_p - 1.5$

The pressure drop in the nozzle is given by following equation [24, 27, 30]:

$$G_n = \frac{4\dot{m}_t}{\pi d_n^2} \tag{21}$$

$$\mathrm{Re}_n = \frac{4\dot{m}_t}{\pi d_n \mu_t} \tag{22}$$

$$
\begin{aligned}
\Delta p_n &= 7.5\times10^{-4}\frac{N_s.G_n^2}{s} && \text{for turbulent flow} \\
\Delta p_n &= 1.5\times10^{-3}\frac{N_s.G_n^2}{s} && \text{for laminar flow}
\end{aligned} \tag{23}
$$

Where G_n the mass flux in the nozzle, and N_s is the number of shells connected in series. The inner diameter of the shell is also calculated from Eq. (24)[31]:

$$D_s = 0.637 p_T\sqrt{\pi.N_t.\frac{CL}{CTP}} \tag{24}$$

Where p_T, is the tube pitch and CL is tube layout constant, which equals 1 for 45° and 90° and 0.87 for 30° and 60°. CTP is also proposed 0.9, based on the choice of shell and fixed tube plate, for two pipe passes. The sindex also points to the shell.
The shell-side heat-transfer coefficient, h_o is computed using the Simplified Delaware method [24, 27]:

$$h_o = j_H\left(\frac{K}{d_e}\right)pr^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{25}$$

$$j_H = 0.5\left(1+\frac{B}{D_s}\right)\left(0.08\,\mathrm{Re}_s^{0.6821}+0.7\,\mathrm{Re}_s^{0.1772}\right) \tag{26}$$

$$
\begin{aligned}
d_e &= \frac{4p_T^2-\pi d_r^{*2}}{\pi d_r^*} && \text{for a square pitch} \\
d_e &= \frac{4\times0.86p_T^2-\pi d_r^{*2}}{\pi d_r^*} && \text{for a triangular pitch}
\end{aligned} \tag{27}
$$

$$d_r^* = \left[d_r^2 + 4n_f \times b\times\tau\left(d_r+b\right)\right] \tag{28}$$

Where j_H , d_e ,d_r^*,d_r , n_f , b and τ are the modified Colburn factor for the shell-side heat transfer and the equivalent diameter, the effective root tube diameter, the external diameter of the tube root, the number of fins per unit length of tube, the fin height, the fin thickness, respectively.

In the Simplified Delaware method the shell-side pressure drop is computed using the following equation [27]:

$$\Delta p_s = \Delta p_{f,s} + \Delta p_n \tag{29}$$

$$\Delta p_{f,s} = \frac{f \times G_s^2 \times D_s \times (n_b + 1)}{2000 \times d_e \times s \times \left(\frac{\mu}{\mu_w}\right)^{0.14}} \tag{30}$$

Where G_s is the shell side mass velocity and can be obtained as follow [24, 27]:

$$G_s = \frac{\dot{m}_s}{A_s} \tag{31}$$

$$A_s = \frac{D_s CB}{p_T} \tag{32}$$

Here A_s denotes flow area across the tube bundle, C clearance between tubes in the bundle, n_b number of baffles, s specific gravity and p_T tube pitch (replaced by $\frac{p_T}{\sqrt{2}}$ for 45^o tube layout).

The shell-side friction factor is given in following equation [24]:

$$f = 144\left[f_1 - 1.25\left(1 - \frac{B}{D_s}\right)(f_1 - f_2)\right] \tag{33}$$

If $0.2032 \le D_s \le 0.59$ and $\mathrm{Re}_t > 1000$:

$$\begin{aligned} f_1 &= \left(0.0076 + 0.000166\frac{D_s}{0.0254}\right)\mathrm{Re}^{-0.125} \\ f_2 &= \left(0.0016 + 5.8\times10^{-5}\frac{D_s}{0.0254}\right)\mathrm{Re}^{-0.157} \end{aligned} \tag{34}$$

If $0.2032 \le D_s \le 0.59$ and $\mathrm{Re}_t < 1000$:

$$\begin{aligned} f_1 &= \exp\left[0.092(\ln \mathrm{Re})^2 - 1.48\ln \mathrm{Re} - 0.000526\left(\frac{D_s}{0.0254}\right)^2 + 0.0478\frac{D_s}{0.0254} - 0.338\right] \\ f_2 &= \exp\left[0.123(\ln \mathrm{Re})^2 - 1.78\ln \mathrm{Re} - 0.00132\left(\frac{D_s}{0.0254}\right)^2 + 0.0678\frac{D_s}{0.0254} - 1.34\right] \end{aligned} \tag{35}$$

Note that for $D_s > 0.59$ we take $D_s = 0.59$ based on the relationship above. The pressure drop due to the shell-side nozzles can be estimated in the same manner as for the tube-side nozzles.

3. Single-objective optimization using Genetic Algorithm

Entransy is a physical quantity to describe the ability of heat transfer. In fact, considering the reversible heat process for a body with a temperature *T* and a constant volume heat capacity (cv), its physical meaning can be understood. For a reversible process, the temperature difference between the body and the thermal source is extremely small and the heat added to the body is extremely small, too. Giving a very small amount of heat to any source, the temperature of this heat source increases very small. The temperature represents a thermal potential because at different temperatures, the availability of the heat is changed. When a very small amount of heat is added to the body, like the evacuation of electrical energy in a capacitor, an increase in the energy of the thermal energy potential resulting from the thermal load and the potential of the partial temperature is seen. If absolute zero is considered as the potential of zero temperature, then the potential of thermal energy for an object at temperature T or entransy of the object can be calculated from Eq. (36):

$$E_h = \int_0^T Q_{vh} dT = \frac{1}{2} m c_v T^2 \tag{36}$$

Thermal energy is stored in the heat transfer process, while entransy is wasted by irreversible heat transfer processes [32]. In heat transfer processes, the higher degree of reversibility, make a lower entransy dissipations or losses. Therefore, it is important to minimize the entransy dissipations in the heat exchangers in order to achieve its optimal performance. In heat exchangers, heat transfer via the limited temperature difference and fluid friction are two main irreversible processes that lead to entransy dissipations. Accordingly, at first, the entransy dissipations associated with these irreversible factors are calculated and then the optimal design of the heat exchanger is performed by minimizing the number of entransy dissipations.

According to the entransy definition, the entransy dissipations due to the thermal conduction in the heat exchanger are expressed by Eq. (37)[33].

$$G_{\Delta T} = -\int_i^o \left(\dot{m} C_p T dT\right)_{h,c} = \frac{1}{2}\left(\dot{m}C_p\right)_h \left(T_{h,i}^2 - T_{h,o}^2\right) + \frac{1}{2}\left(\dot{m}C_p\right)_c \left(T_{c,i}^2 - T_{c,o}^2\right) \tag{37}$$

Number of entransy dissipations caused by thermal conduction can be defined as the ratio of the actual entransy dissipations to the maximum entransy dissipations in the heat exchanger. Therefore, it can be obtained by dividing the Eq. (22) to $Q\left(T_{h,i} - T_{c,i}\right)$ [33].

$$g_{\Delta T} = \frac{G_{\Delta T}}{Q\left(T_{h,i} - T_{c,i}\right)} \tag{38}$$

Entransy dissipations due to fluid friction, for an incompressible fluid, can be calculated from Eq. (24)[10, 13].

$$G_{\Delta p} = -\int_i^o \frac{\dot{m}T}{\rho} dp = \left(\frac{\dot{m}\Delta p}{\rho} \frac{T_o - T_i}{\ln\left(\frac{T_o}{T_i}\right)} \right)_{h,c} = \frac{\dot{m}_t \Delta p_t}{\rho_t} \frac{T_{h,o} - T_{h,i}}{\ln\left(\frac{T_{h,o}}{T_{h,i}}\right)} + \frac{\dot{m}_s \Delta p_s}{\rho_s} \frac{T_{c,o} - T_{c,i}}{\ln\left(\frac{T_{c,o}}{T_{c,i}}\right)} \quad (39)$$

The number of entransy dissipations due to fluid friction is also given by Eq. (25).

$$g_{\Delta p} = \frac{G_{\Delta p}}{Q\left(T_{h,i} - T_{c,i}\right)} \quad (40)$$

Therefore, the entransy dissipations caused by thermal conduction and fluid friction are converted to the entransy dissipations number, by dimensionless method. The total number of entransy dissipations is derived from sum of numbers of entransy dissipations due to thermal conduction and fluid friction, which is defined in the form of Eq. (26).

$$g^* = g_{\Delta T} + g_{\Delta p} \quad (41)$$

Now g^* is considered as the objective function for optimal design. The given data for the design of the heat exchanger at constant heat rate is given in Table 2.

Table 2. Data for the heat exchanger.

Thermophysical and process data	Tube side (water)	Shell side (nitrogen)
Specific gravity (-)	0.99	0.97
Specific heat (J/kg K)	4190	1040
Dynamic viscosity (Pa s)	0.0007128	0.000021
Thermal conductivity (W/m K)	0.62	0.03
Prandtl number (-)	4.82	0.72
Fouling factor (m^2W/K)	0.00035	0.0004

The tube arrangement (30°, 45°, 90°), the number of fins per unit length and the diameter (among the 23 available in the TEMA standard listed in Table 4) are three discrete design variables [24]. The bounds for the decision variables involved in optimization of the objective functions (minimum entrancy dissipation) are listed in Table 3.

Table 3. Bounds for design parameters.

Variable	Lower value (or values considered)	Upper value
Tube arrangement	(30°, 45°, 90°)	-
Tube pitch rate	1.25	2
Tube length (m)	3	8
Tube number	100	700
Baffle spacing ratio	0.2	0.4
Fin height (m)	0.00127	0.003175
Fin thickness (m)	0.000254	0.000304
Number of fins per inch length of tube	(16, 19, 26)	-
Inlet temperature of tube (°C)	15	35
Inlet temperature of shell (°C)	100	135
Mass flow rate on the tube side	8	20
Mass flow rate on the shell side	2	9

Table 4. Inner and outer diameters and external diameter of root tube (di, do, dr) in inches for 23 standard tubes (for Radial Low-Fin Tubing (Type S/T True fin):19 fins per tube inch).

	d_i (in)	d_o (in)	d_r (in)
1	0.291	1/2	0.375
2	0.384	5/8	0.5
3	0.459	3/4	0.625
4	0.584	7/8	0.75
5	0.709	1	0.875
6	0.277	1/2	0.375
7	0.356	5/8	0.5
8	0.495	3/4	0.625
9	0.560	7/8	0.750
10	0.685	1	0.875
11	0.259	1/2	0.375
12	0.527	3/4	0.625
13	0.634	7/8	0.75
14	0.759	1	0.875
15	0.657	1	0.875
16	0.370	5/8	0.5
17	0.509	3/4	0.625
18	0.620	7/8	0.75
19	0.745	1	0.875
20	0.402	5/8	0.5
21	0.481	3/4	0.625
22	0.606	7/8	0.75
23	0.731	1	0.875

Due to the non-convexity of the problem, intelligent algorithms that based on population are used. One of these methods is the direct search algorithm. Although this method does not require information regarding to the gradient of the objective function, but it is strongly dependent on the selection of the initial population [34]. Therefore, for solving the problem the genetic algorithm method is used in the MATLAB software [27]. The genetic algorithm starts the search from a set of points. It also provides a high level of capability by simulating nature-compatible in an evolutionary process [34]. More importantly, the genetic algorithm has a great ability to achieve optimal points [35, 36]. For the reasons given, a genetic algorithm [37] has been used to search for optimal solution for heat exchangers. The primary population that satisfies the constraints is generated randomly. In the method of genetic algorithm, according to the objective function, each potential solution is evaluated quantitatively. So from a random initial population in the range of design variables, the algorithm creates a sequence of new generations and repeats until the stop criterion is met. In this process, integrating two parents from the current generation, by the action of crossover, or changing a chromosome by the action of mutations, new children are produced. The new generation is formed by some parents and the children on the basis of fitness function. Also, the population size is kept constant eliminating children who have less value on the fitness function. Those chromosomes that have a higher value on the fitness function have a greater chance of survival. Finally, this guarantee of convergence to the best person, after a certain number of generations (repetitions), indicates optimal solution for the problem [38]. The size of the primary population and the maximum number of generations are 40 and 1000, respectively.

4. Results and discussion

Fig. 3 shows the variations in the best values for entransy dissipations versus the number of generations towards the optimum value (minimum value). The value of objective function was calculated at each iteration and the minimum value at each iteration is selected. As evident in this figure, initially the number of entransy dissipations caused by thermal conduction and fluid friction decreases and be constant after several times repetition. Considering the unchanged value of the objective function in the next iteration, the optimization computation was terminated. It can also be found from this figure that the genetic algorithm has a very high application in finding an optimal overall solution.

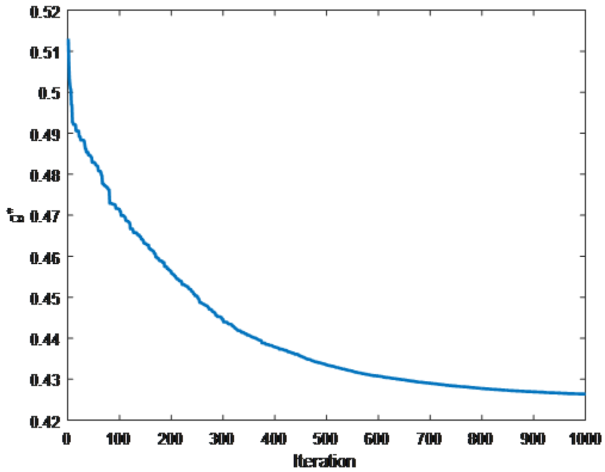

Figure 3. The variations of number of entransy dissipations versus the number of generations.

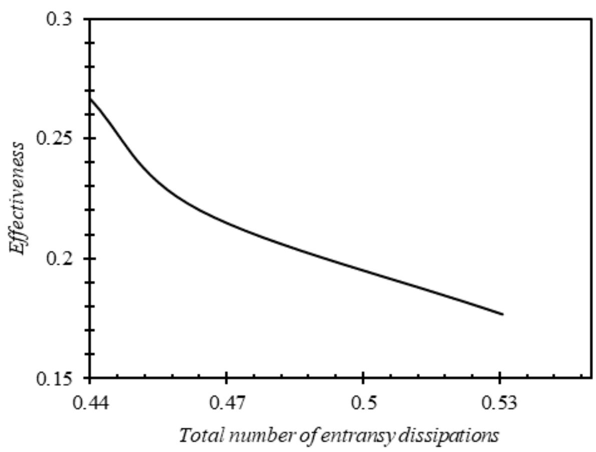

Figure 4. Variation of effectiveness based on the total number of entransy dissipations.

In order to further demonstrate the advantages of the single-objective optimization design, the comparison between a randomly generated initial design and the optimal one is shown in Table 5. From this table, it is clear that the exchanger effectiveness increases from 0.18 to 0.35, and the heat transfer increases by 13%. The results of the genetic algorithm (GA) is compared with particle swarm optimization (PSO). As it is seen the results obtained from GA are in good agreement with the PSO algorithm. With regard to Table 5, it can be concluded that the number of entransy dissipations due to fluid friction is about five order lower than the entransy dissipations due to thermal conduction. In fact, often, the irreversibility due to fluid friction is far less than the irreversibility due to heat conduction for liquids [39].

Table 5. Comparison between primary and optimal design.

Specifications	Primary	Optimum (GA)	Optimum (PSO)
L	2.35	4.5	4.6
n_t	100	536	700
B	0.24	0.34	0.396
d_0	0.019	0.0254	0.0254
n_f	16	19	26
NTU	0.2	0.46	0.44
Q	737	838.39	842.4
ε	0.18	0.35	0.34
$g_{\Delta T}$	0.44	0.4251	0.429
$g_{\Delta p}$	0.09	0.001	0.001
g^*	0.53	0.4261	0.43

Figs. 4 shows, the changes in the effectiveness of the heat exchanger in terms of the total number of entransy dissipations under constant heat transfer rate conditions .As can be seen from these figures, increasing the total number of entransy dissipations, the effectiveness of the heat exchanger decreases. Thus, through the process of optimization, selecting the decision variables in order to minimizing the total number of entransy dissipations, the performance of the heat exchanger is improved considerably. Also, the number of transfer units increases 2.3 times larger than initial state. As the number of transfer units increases, the performance of the heat exchanger improves with increase of the heat transfer surface.

Fig. 5 shows the fluid mass flow rate variations of the tube side on the total number of entransy dissipations in terms of the constant heat transfer rate. Due to the fact that the fluid mass flow rate of shell side has a direct relation to the entransy dissipations, decreasing this parameter the number of entransy dissipations decreases.

Fig. 6 shows the variation in the number of baffles in the shell on the total entransy dissipations number in the conditions of the constant heat transfer rate. Physically, the pressure drop increases as the number of baffle increases. Due to the fact that the pressure drop has a direct relation to the entransy dissipations, decreasing this parameter the number of entransy dissipations decreases.

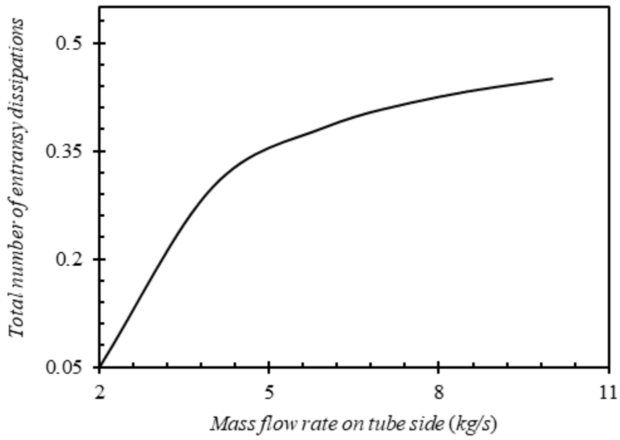

Figure 5. Variations of total entransy dissipation number with fluid mass flow rate

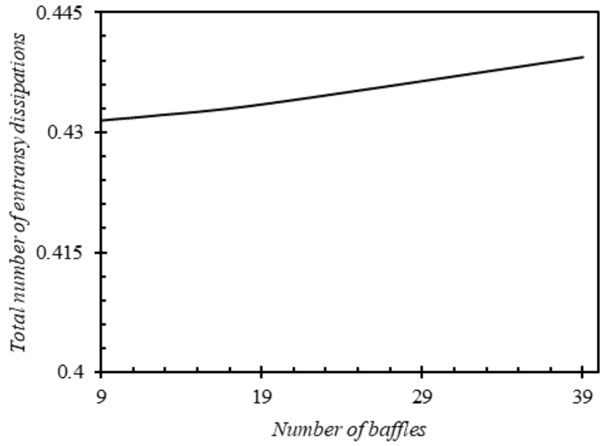

Figure 6. Variations of total entransy dissipation number with the number of baffles.

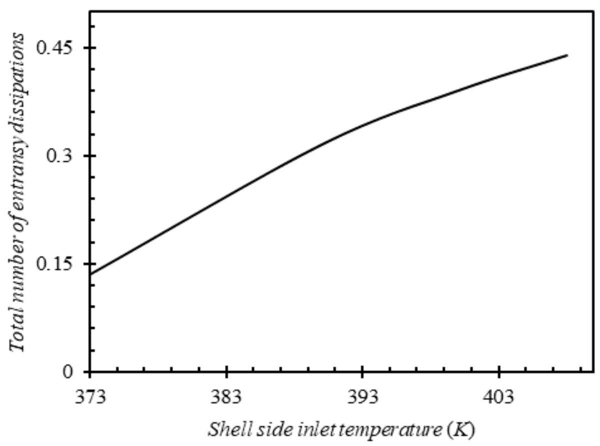

Figure 7. Variations of total entransy dissipation number with the shell side inlet temperature.

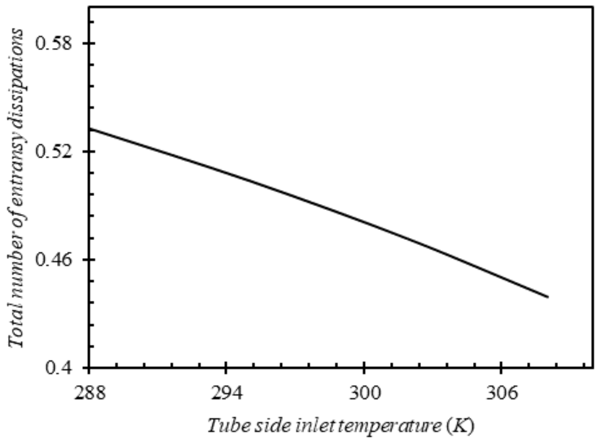

Figure 8. Variations of total entransy dissipation number with the tube side inlet temperature.

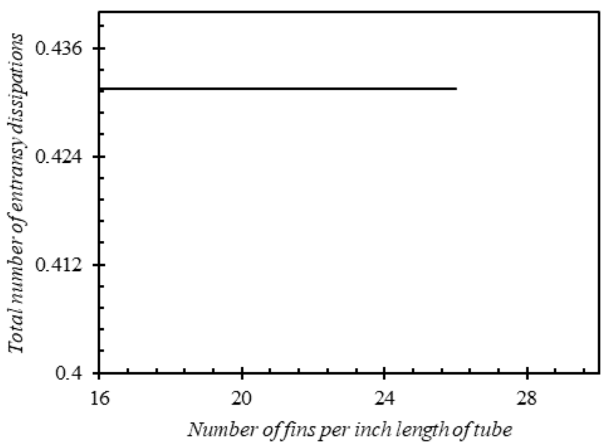

Figure 9. Variations of total entransy dissipation number with the number of fins per inch length of tube.

Fig. 7 shows the variation in the shell side inlet temperature on the total entransy dissipations number in the conditions of the constant heat transfer rate. Due to the fact that entransy dissipations due to thermal conduction is the main term of total entransy dissipations number and entransy dissipations due to thermal conduction has a direct relation to the shell side inlet temperature, decreasing this parameter the number of entransy dissipations decreases.

Fig. 8 shows the variation in the tube side inlet temperature on the total entransy dissipations number in the conditions of the constant heat transfer rate. Due to the fact that entransy dissipations due to thermal conduction is the main term of total entransy dissipations number and entransy dissipations due to thermal conduction has a relation to the negative tube side inlet temperature, increasing this parameter the number of entransy dissipations decreases.

Fig. 9 shows the number of fins per inch length of tube on the total number of entransy dissipations in terms of the constant heat transfer rate. Due to the fact that number of fins per inch length of tube has no relation to the inlet and outlet temperature, pressure drop and mass flow rate, this parameter has no relation to the entransy dissipations, so changing it, the number of entransy dissipations does not change.

5. Conclusion

In this study, the performance of finned shell and tube heat exchangers were investigated. According to the theory of entransy dissipations, the irreversibilities of heat transfer and fluid friction, which are considered as two important factors in decreasing the performance of the heat exchanger, have been introduced by a non-dimensional parameter that called the number of entransy dissipation. Therefore, the present research was based on the theory of entransy dissipations.

A single-objective optimization approach has been developed in which the total entransy dissipation has been considered as an objective function. The main results of this study can be summarized as below:

- When the heat transfer rate is constant, optimal single-objective design can significantly improve the performance of the heat exchanger.
- The minimum of total number of entransy dissipations as an objective function is almost equivalent to minimizing the entransy dissipations due to thermal conduction, whereas the impact of number of entransy dissipations caused by fluid friction can be almost ignored.
- total number of entransy dissipations will increase under the effect of increasing shell side inlet temperature. Whereas total number of entransy dissipations will decrease under the effect of increasing tube side inlet temperature.
- total number of entransy dissipations will increase under the effect of increasing mass flow.
- total number of entransy dissipations will increase under the effect of increasing the number of baffles whereas changing the number of fins per inch length of tube has no effect on the total number of entransy dissipations.

6. References

[1] A. Bejan, 1982, *Entropy generation through heat and fluid flow*, Wiley,

[2] J. Hesselgreaves, Rationalisation of second law analysis of heat exchangers, *International Journal of Heat and Mass Transfer*, Vol. 43, No. 22, pp. 4189-4204, 2000.

[3] Z.-Y. Guo, H.-Y. Zhu, X.-G. Liang, Entransy—a physical quantity describing heat transfer ability, *International Journal of Heat and Mass Transfer*, Vol. 50, No. 13-14, pp. 2545-2556, 2007.

[4] G.-Z. Han, Z.-Y. Guo, Physical mechanism of heat conduction ability dissipation and its analytical expression, in *Proceeding of*, 98-102.

[5] S. Wang, Q. Chen, B. Zhang, An equation of entransy transfer and its application, *Chinese science bulletin*, Vol. 54, No. 19, pp. 3572, 2009.

[6] Q. Chen, J. Ren, Generalized thermal resistance for convective heat transfer and its relation to entransy dissipation, *Chinese science bulletin*, Vol. 53, No. 23, pp. 3753-3761, 2008.

[7] S. Xia, L. Chen, F. Sun, Optimization for entransy dissipation minimization in heat exchanger, *Chinese science bulletin*, Vol. 54, No. 19, pp. 3587, 2009.

[8] J. Guo, M. Xu, L. Cheng, Principle of equipartition of entransy dissipation for heat exchanger design, *Science China Technological Sciences*, Vol. 53, No. 5, pp. 1309-1314, 2010.

[9] H. Wei, X. Du, L. Yang, Y. Yang, Entransy dissipation based optimization of a large-scale dry cooling system, *Applied Thermal Engineering*, Vol. 125, pp. 254-265, 2017.

[10] Y.-C. Xu, Q. Chen, Z.-Y. Guo, Optimization of heat exchanger networks based on Lagrange multiplier method with the entransy balance equation as constraint, *International Journal of Heat and Mass Transfer*, Vol. 95, pp. 109-115, 2016.

[11] A. M. Abed, I. A. Abed, H. S. Majdi, A. N. Al-Shamani, K. Sopian, A new optimization approach for shell and tube heat exchangers by using electromagnetism-like algorithm (EM), *Heat and Mass Transfer*, Vol. 52, No. 12, pp. 2621-2634, 2016.

[12] X. Liu, J. Meng, Z. Guo, Entropy generation extremum and entransy dissipation extremum for heat exchanger optimization, *Chinese science bulletin*, Vol. 54, No. 6, pp. 943-947, 2009.

[13] M. Xu, J. Guo, L. Cheng, Application of entransy dissipation theory in heat convection, *Frontiers of Energy and Power Engineering in China*, Vol. 3, No. 4, pp. 402, 2009.

[14] L. Chen, Progress in entransy theory and its applications, *Chinese science bulletin*, Vol. 57, No. 34, pp. 4404-4426, 2012.

[15] R. V. Rao, A. Saroj, Economic optimization of shell-and-tube heat exchanger using Jaya algorithm with maintenance consideration, *Applied Thermal Engineering*, Vol. 116, pp. 473-487, 2017.

[16] M. Mirzaei, H. Hajabdollahi, H. Fadakar, Multi-objective optimization of shell-and-tube heat exchanger by constructal theory, *Applied Thermal Engineering*, Vol. 125, pp. 9-19, 2017/10/01/, 2017.

[17] J. C. Lemos, A. L. Costa, M. J. Bagajewicz, Linear method for the design of shell and tube heat exchangers including fouling modeling, *Applied Thermal Engineering*, Vol. 125, pp. 1345-1353, 2017.

[18] Y. Lei, Y. Li, S. Jing, C. Song, Y. Lyu, F. Wang, Design and performance analysis of the novel shell-and-tube heat exchangers with louver baffles, *Applied Thermal Engineering*, Vol. 125, pp. 870-879, 2017.

[19] P. Bichkar, O. Dandgaval, P. Dalvi, R. Godase, T. Dey, Study of Shell and Tube Heat Exchanger with the Effect of Types of Baffles, *Procedia Manufacturing*, Vol. 20, pp. 195-200, 2018.

[20] X. Gu, M. Wang, Y. Liu, S. Wang, Multi-parameter optimization of shell-and-tube heat exchanger with helical baffles based on entransy theory, *Applied Thermal Engineering*, Vol. 130, pp. 804-813, 2018.

[21] W. Gander, *Starting and Using Matlab*, in: *Learning MATLAB*, Eds., pp. 1-9: Springer, 2015.

[22] R. K. Shah, D. P. Sekulic, 2003, *Fundamentals of heat exchanger design*, John Wiley & Sons,

[23] H. Sadeghzadeh, M. Aliehyaei, M. A. Rosen, Optimization of a Finned Shell and Tube Heat Exchanger Using a Multi-Objective Optimization Genetic Algorithm, *Sustainability,* Vol. 7, No. 9, pp. 11679-11695, 2015.

[24] R. W. Serth, T. Lestina, 2014, *Process heat transfer: Principles, applications and rules of thumb*, Academic Press,

[25] S. Kakac, H. Liu, A. Pramuanjaroenkij, 2012, *Heat exchangers: selection, rating, and thermal design*, CRC press,

[26] F. McQuiston, D. Tree, Optimum space envelopes of the finned tube heat transfer surface, *ASHRAE Trans,* Vol. 78, No. 2, pp. 144-152, 1972.

[27] D. Q. Kern, 1950, *Process heat transfer*, Tata McGraw-Hill Education,

[28] H. Hausen, Darstellung des Warmeuberganges in Rohren durch verallgemeinerte Potenzbeziehungen, *Z. VDI Beih. Verfahrenstech,* Vol. 4, pp. 91-98, 1943.

[29] E. N. Sieder, G. E. Tate, Heat transfer and pressure drop of liquids in tubes, *Industrial & Engineering Chemistry,* Vol. 28, No. 12, pp. 1429-1435, 1936.

[30] J. Henry, Headers, nozzles, and turnarounds, *Heat Exchanger Design Handbook,* Vol. 2, 1982.

[31] J. Taborek, 1991, *Industrial heat exchanger design practice*, Wiley, New York,

[32] Z. Guo, X. Cheng, Z. Xia, Least dissipation principle of heat transport potential capacity and its application in heat conduction optimization, *Chinese science bulletin,* Vol. 48, No. 4, pp. 406-410, 2003.

[33] J. Guo, L. Cheng, M. Xu, Entransy dissipation number and its application to heat exchanger performance evaluation, *Chinese science bulletin,* Vol. 54, No. 15, pp. 2708-2713, 2009.

[34] Y.-H. Oh, T.-K. Chung, M.-K. Kim, H.-K. Jung, Optimal design of electric machine using genetic algorithms coupled with direct method, *IEEE Transactions on Magnetics,* Vol. 35, No. 3, pp. 1742-1745, 1999.

[35] S. Sanaye, M. Chahartaghi, Thermal—economic modelling and optimization of gas engine-driven heat pump systems, *Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy*, Vol. 224, No. 4, pp. 463-477, 2010.

[36] A. Fanni, M. Marchesi, A. Serri, M. Usai, A greedy genetic algorithm for continuous variables electromagnetic optimization problems, *IEEE Transactions on Magnetics,* Vol. 33, No. 2, pp. 1900-1903, 1997.

[37] C. R. Houck, J. Joines, M. G. Kay, A genetic algorithm for function optimization: a Matlab implementation, *Ncsu-ie tr,* Vol. 95, No. 09, pp. 1-10, 1995.

[38] G. Cammarata, A. Fichera, D. Guglielmino, Optimization of a liquefaction plant using genetic algorithms, *Applied energy,* Vol. 68, No. 1, pp. 19-29, 2001.

[39] J. Guo, L. Cheng, M. Xu, Multi-objective optimization of heat exchanger design by entropy generation minimization, *Journal of Heat Transfer,* Vol. 132, No. 8, pp. 081801, 2010.