



# Effectiveness improvement and optimization of shell-and-tube heat exchanger with entransy method

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## Abstract

In this study, the optimum design of shell and tube heat exchangers is investigated by applying the entransy dissipation theory and genetic algorithm method. In fact, this article presents solutions to increase performance of heat exchanger with changes in geometry, such as the number of tubes, tube diameter, and baffle geometries. 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 minimization of them has been chosen as objective function, and the corresponding problem optimized under the conditions of the constant heat transfer rate and constant thermal surface. The results show that the single-objective optimization at constant heat transfer rate, can be improve the performance of heat exchanger significantly and in constant heat transfer area increases the effectiveness, but it leads to increase of pump power consumption. It is found that, the fluid friction impact is not fully considered when the working fluid of heat exchanger is liquid in single-objective optimization approach. In order to solving this problem, a multi-objective optimization approach to heat exchanger design is established.

## Nomenclature

### General

$A$	Surface area; (m <sup>2</sup> )
$c_p$	Specific heat at constant pressure; (J/kg. K)
$c_v$	Specific heat at constant volume; (J/kg. K)
$D_s$	Inner diameter of shell; (m)
$d_o$	Outer diameter of tube; (m)
$E_h$	Entransy; (J. K)
$f$	Friction factor; (–)
$G_{\Delta T}$	Entransy dissipation caused by thermal conduction; (J. K)

$g_{\Delta T}$	Number of entransy dissipation caused by thermal conduction; (–)
$G_{\Delta P}$	Entransy dissipation caused by fluid friction; (J. K)
$g_{\Delta P}$	Number of entransy dissipation caused by fluid friction; (–)
$g^*$	Total number of entransy dissipation; (–)
$h_t$	Heat transfer coefficient of tube side; (W/m <sup>2</sup> . K)
$h_s$	Heat transfer coefficient of shell side; (W/m <sup>2</sup> . K)
$K$	Thermal conduction coefficient; (W/m <sup>2</sup> . K)
$L_{hx}$	Heat exchanger length; (m)
$\dot{m}$	Mass flow rate; (kg/s)
$n$	Number of tubes; (–)
$Nu$	Nusselt number; (–)
$N_b$	Number of baffles; (–)
$NTU$	Number of transfer units; (–)
$\Delta P_t$	Pressure drop of tube side; (kPa)
$\Delta P_s$	Pressure drop of shell side; (kPa)
$\dot{Q}$	Heat transfer rate; (W)
$U$	Overall heat transfer coefficient; (W/m <sup>2</sup> . K)
$Pr$	Prandtl number; (–)
$P$	Pressure; (Pa)
$T$	Temperature; (K)
$W$	Pump power consumption; (W)

**Highlights** • The entransy method is used to performance evaluation of shell and tube heat exchanger.

- The number of entransy dissipations caused by thermal conduction and fluid friction were considered as objectives.
- The GA optimization method have been presented to minimization of the total entransy dissipations.
- The optimum performance was happen at minimum thermal entransy dissipation.
- The effectiveness of heat exchanger increases about 42% at optimum condition.

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## Greek symbols

$\eta$	Pump efficiency; (–)
$\varepsilon$	Effectiveness; (–)

$\theta_b$  Central angle of the cutting of the baffles; (rad)

$\mu$  Dynamic viscosity; (kg/m. s)

### Subscripts

b baffle

c (Cold)

h (Hot)

i (Inlet)

o (Outlet)

s Shell side

t Tube side

l (Leakage)

## 1 Introduction

Due to the increase of energy demand and the intense reduction of fossil fuels such as oil and coal, use energy more efficiently, leads to saving the resources. Because 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., reducing unnecessary energy losses and improving the performance of the heat exchanger is important.

The performance evaluation criteria 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 usually heat transfer via a limited temperature difference, fluid friction under pressure drop and mixing of two fluids can be 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. In the following a few of them have been investigated.

Bejen [1] developed the method of minimizing entropy generation for the optimal design of heat exchangers. Also, by 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 minimizing entropy generation. Accordingly, the method of minimizing entropy generation 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 very important. 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, by 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 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 heat transfer, mass transfer, and chemical reactions. They proposed the entransy and entransy transfer equation to study heat transfer optimization in engineering applications. In addition, they showed that in combination with mass, momentum, and energy transfer equations, by solving the entransy transfer equation, the entransy flux and entransy dissipation could be illustrated. They used the entransy transfer equations for convection and conduction heat transfer mechanisms at steady state condition.

Chen and Ren [6], by 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 for minimizing the thermal resistance in order to optimize the convection heat transfer processes. In addition, in order to illustrate the applicability of the generalized thermal resistance a two-dimensional convective heat transfer process with constant wall temperature was considered for analysis and optimization of the convective heat transfer process. They also expressed that the principle of minimizing thermal resistance can be equivalent to the principle of minimizing entransy dissipation.

Xia et al. [7], by 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. The heat transfer between the hot and cold fluids was modeled based on the Newton’s heat transfer law and the heat transfer coefficient was assumed to be constant. In addition, the results for entransy dissipation minimization were also compared with those for entropy generation minimization.

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 including 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.

Yuan and Chen [10] presented an optimization method based on the entransy theory in order to improve the energy efficiency of indirect evaporative cooling systems with coupled heat and mass transfer processes. They studied the irreversible heat and mass transfer processes and presented the entransy dissipation of each irreversible process and the total entransy dissipation in the system. They illustrated the theoretical and mathematical relations for the thermal conductance of each heat exchanger and the mass flow rates of water in each branch to estimation of the optimal system performance. The system was analyzed at conditions of fixed total circulating water flow rate and fixed total thermal conductance.

Xu et al. [11] 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 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. [12] 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 by 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. [13] investigated the principles of extremum entropy and entransy generation for the optimization of heat exchanger and concluded that the principle of minimizing

entropy generation 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.

In order to reach a maximum value of heat transfer, Jia et al. [14] proposed the minimum entransy dissipation as objective function with constraint of power consumption for developing the optimization method. In addition, they performed a numerical analysis for convective heat transfer in a straight circular tube based on optimized governing equations. Furthermore, they showed that the entransy dissipation optimization method could be useful for designing the low resistance and high efficiency heat transfer units of heat exchangers.

Xu et al. [15, 16] 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 evaluation of the performance of the heat exchangers. Therefore, by changing entransy dissipations in dimensionless form and considering it as the objective function, they changed the geometry of heat exchanger and optimize it.

Li et al. [17] used the entransy dissipation minimization method for optimal design of a water to water counter flow heat exchanger. They presented the overall entransy dissipation number as an objective function according to flow friction under finite pressure drops and heat conduction under finite temperature differences. In addition, they expressed analytical relations for the optimal duct aspect ratio and mass velocity of the heat exchanger that were useful for design optimization. Their results showed that in order to decrease of the irreversible dissipations in heat exchangers, the largest heat transfer areas and lowest mass velocities should be chosen if possible.

Qian et al. [18] for design optimization of two phase heat exchangers, presented the applicability of entransy dissipation based thermal resistance. They used the optimization method for maximizing the heat exchanger capacity and minimizing the cost in heating, ventilation, air conditioning and refrigeration applications. In their study, based on a heat exchanger modeling tool, the two phase entransy was presented by optimizing two different types of tube-fin and micro-channel heat exchangers. They illustrated that the entransy dissipation was an effective method to optimize heat exchangers with air flow rate as one design variable but was limited for optimizing heat exchangers with fixed flow rate.

Although different studies have been conducted on the use of entransy method to improve the performance of thermal systems and some types of heat exchangers, but present of a research which performs a comprehensive study in order to modeling and optimization of shell and tube heat exchangers

with consideration of different operating and geometric parameters can be useful in addition to the previous works.

In this research, the performance of shell and tube heat exchangers at constant heat transfer rate and constant heat transfer surface area have been 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 exchangers, have been introduced by a dimensionless parameter that called the number of entransy dissipation.

In addition, by changing the entransy dissipations based on thermal conduction and fluid friction in dimensionless forms and considering the total entransy dissipation as an objective function, the optimization of the shell and tube heat exchangers in both single and multi-objective modes, at conditions of constant heat transfer rate and constant area have been carried out. The optimization has been performed using the Genetic Algorithm method by MATLAB software [19]. Furthermore, different effective parameters such as size and number of tubes, the number of baffles in the shell side, the geometry and positions of baffles, and the mass flow rates have been used for model performing. In addition, effect of main decision variables including the number of tubes, the length of the tubes, the number of baffles, and the central angle of the cutting of the baffles on objective functions have been analyzed and the pump power consumption, the heat exchanger effectiveness, and the entransy dissipations number have been illustrated at different conditions.

A summary of some main contributions of this paper relative to previous studies are:

- Presenting an comprehensive model for analyzing the performance of shell and tube heat exchangers at constant heat transfer rate and constant heat transfer surface area according to the theory of entransy dissipations with consideration of thermal and geometrical operating parameters
- Considering the irreversibilities of heat transfer and fluid friction as two important factors in decreasing the performance of the heat exchangers by introducing a dimensionless parameter that called the number of entransy dissipation
- Presenting the entransy dissipations based on thermal conduction and fluid friction in dimensionless forms and considering the total entransy dissipation as an objective function
- Performing the optimization process with using the Genetic Algorithm method in both single and multi-objective modes, at conditions of constant heat transfer rate and constant surface area
- Estimation of optimum values of decision variables and presenting the appropriate values for system performance parameters

Considering that such an extensive research has not been carried out in the previous works and the effects of different parameters on entransy have not been investigated in this way, this study can be useful.

## 2 Heat calculations of shell and tube heat exchanger

In this section, the basic calculations of heat transfer and pressure drop in shell and tube heat exchanger are discussed. Considering that this type of heat exchanger is one of the most common type of heat exchanger used in thermal cycles, following the optimal design discussion of these type of heat exchangers and the application of the theory of entransy dissipations in them have been presented. In illustrated heat exchanger in Fig. 1, the hot fluid is on the side of the tubes and the cold fluid on the shell side.

### 2.1 Heat transfer calculations

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 [20], the equilibrium equation for the heat exchanger can be calculated as.

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

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 obtained as the ratio of the actual heat transfer rate to the maximum heat transfer rate from Eq. (2) [21].

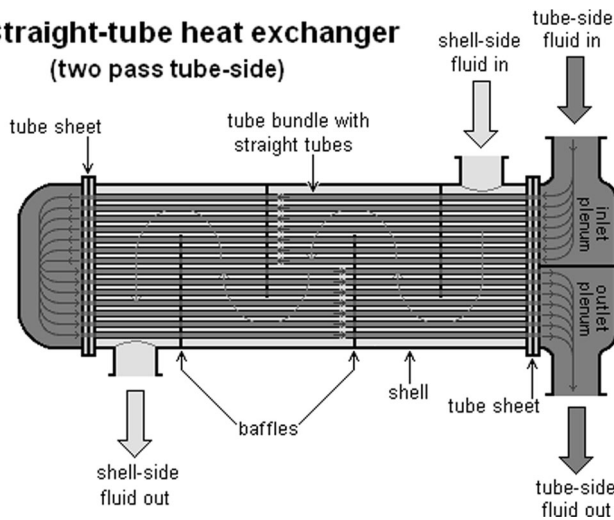
$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} \quad (2)$$

The inner diameter of the shell is also calculated from Eq. (3) [22]:

$$D_s = 0.637 \sqrt{\frac{(P_T^2 \cdot A_o)}{(d_o \cdot L_{hx})}} \sqrt{\frac{CL}{CTP}} \quad (3)$$

In Eq. (3),  $P_T$  is the pipe step,  $CL$  is tube layout constant, and its value for 30 degree layout is 0.87.  $CTP$  is also proposed 0.9, based on the choice of shell and fixed tube plate, for two pipe passes. The  $s$  index also points to the shell.

**Straight-tube heat exchanger**  
(two pass tube-side)



**Fig. 1** Shell and tube heat exchanger with one shell pass and two tube passes [3]

According to the Bell-Delaware method, the convective heat transfer coefficient on the shell side is given by Eq. (4):

$$h_s = h_{id} j_c j_l j_b j_s j_r \tag{4}$$

In Eq. 4,  $j_c$  is the correction factor for baffle cut and spacing between them.  $j_l$  is a correction factor for baffle leakage effects, including the leakage between baffle and tubes, and the leakage between internal surface of the shell and baffle.  $j_b$  is the correction factor for bundle bypassing effects due to the clearance between the exterior tubes in the tube bundle and shell from one side and channel formed in the tube bundle, on the other hand.  $j_s$  is the correction factor for variable baffle spacing at the inlet and outlet of shell. The factor  $j_r$  is the correction factor for shell-side Reynolds number and used when the Reynolds number of the shell side is less than 100. If Reynolds is greater than 100, this factor will be equal to 1 [23].  $h_{id}$  is also the ideal value of the heat transfer coefficient for a completely cross-flow with an ideal tube bundle, and can be calculated from Eq. (5) [21].

$$h_{id} = j_{id} C_{p_s} \left( \frac{m_s}{A_s} \right) \left( \frac{K_s}{C_{p_s} \mu_s} \right)^{2/3} \left( \frac{\mu_s}{\mu_{s,w}} \right)^{0.14} \tag{5}$$

In Eq. 5,  $j_{id}$  is the heat transfer coefficient for an ideal tube set,  $A_s$  is the area of the cross-sectional flow through the shell, in the center of shell,  $m_s$  is the mass flow rate of the shell side,  $K_s$  is the conductivity coefficient on the shell side,  $\mu_s$  is the fluid dynamics viscosity on the shell side, and  $\mu_{s,w}$  is the fluid dynamics viscosity of the shell side at the wall temperature.

The heat transfer coefficient of tube side  $h_t$  can be calculated from the Petukhov- Kirillov relation for Nusselt number, which is given by Eqs. (6) and (7) [21].

$$h_t = Nu_b \frac{K_t}{d_i} \tag{6}$$

$$Nu_b = \frac{(f_t/2) Re_t Pr_t}{1.07 + 12.7(f_t/2)^{0.5} (Pr_t^{2/3} - 1)} \tag{7}$$

$$f_t = (1.58 \ln Re_t - 3.28)^{-2} \tag{8}$$

In Eqs. (6) and (7),  $Nu$ ,  $Re$ , and  $Pr$ , represent the Nusselt, Reynolds, and Prantel numbers, respectively.  $d_i$  represents the internal diameter of the tube and  $f$  represents the friction coefficient. The index  $t$  also points to tube.

According to the heat transfer coefficient on the shell side and the tube, the total heat transfer coefficient  $U_o$  can be calculated from the Eq. (9).

$$U_o = \left( \frac{1}{h_t} \left( \frac{d_o}{d_i} \right) + r_t \left( \frac{d_o}{d_i} \right) + \frac{d_o}{2K_w \ln \left( \frac{d_o}{d_i} \right)} + r_s + \frac{1}{h_s} \right)^{-1} \tag{9}$$

In Eq. (9),  $r_t$  and  $r_s$  indicate the resistance of the fouling on the tube and shell side, respectively.  $K_w$  represents the coefficient of thermal conductivity of the baffles.

**2.2 Pressure drop calculation**

The pressure drop at the tube side is due to the longitudinal pressure drop, the pressure drop in the bends, and the pressure loss in the inlet and outlet areas. With neglecting of the second section, the total pressure drop on the tube side is obtained from Eq. (10) [24].

$$\Delta P_t = \left( 4f_t \frac{L_{hx} N_p}{d_i} + 4N_p \right) \rho_t \frac{V_t^2}{2} \tag{10}$$

In Eq. 10,  $N_p$  is the number of tube passes,  $L_{hx}$  is the heat exchanger’s length,  $f_t$  is the friction coefficient of the tube side, and  $\rho_t$  and  $V_t$  indicate the density and velocity of the fluid in the tube side, respectively.

The pressure drops on the shell side are due to the inlet and outlet areas, and baffle effects. By considering the cross flow this pressure drop can be calculated over the entire length of the shell Eq. (11).

$$\Delta P_c = \Delta P_{b,id} (N_b - 1) R_b R_l \tag{11}$$

$R_b$  and  $R_l$  are the correction factor for bypass flow and the correction factor for baffle leakage effects, respectively [24, 25].

$N_b$  is the number of baffels and  $\Delta P_{b,id}$  is the pressure drop in an equivalent ideal tube bank in one baffle compartment of

central baffle spacing based on the Bell-Delaware theory and can be calculated from Eq. (12) [24, 25].

$$\Delta P_{b,id} = 2f_{id} \frac{\dot{m}_s^2}{\rho_s A_s^2} \left( \frac{\mu_s}{\mu_{s,w}} \right)^{-0.14} N_{r,cc} \quad (12)$$

Where  $N_{r,cc}$  is the number of tube rows crossed during flow through one cross flow in the heat exchanger and can be expressed as:

$$N_{r,cc} = \frac{D_s - 2l_c}{X_l} \quad (13)$$

Where  $X_l$  is the pitch of the tubes parallel with the flow direction, and  $l_c$  is the cutting height of the baffles and are expressed by Eqs. (14) and (15), respectively [23].

$$X_l = \frac{\sqrt{3}}{2P_T} \quad (14)$$

$$l_c = \frac{1}{2} \left( 1 - \cos \frac{\theta_b}{2} \right) D_s \quad (15)$$

Where  $\theta_b$  is the central angle of the cutting of the baffles.

The pressure drop in the window is due to the leakage current but is not affected by the outflow current. The combined pressure drop in the windows is calculated from the Eq. (16).

$$\Delta P_w = N_b \Delta P_{w,id} R_l \quad (16)$$

In Eq. (16),  $\Delta P_{w,id}$  is the pressure drop in an equivalent ideal tube bank in the window section and is obtained from Eq. (17) [25].

$$\begin{aligned} Re_s \geq 100 \quad \Delta P_{w,id} &= \frac{\dot{m}_s^2 (2 + 0.6 N_{r,cw})}{2 \rho_s A_s A_b} \quad Re_s < 100 \quad \Delta P_{w,id} \\ &= \frac{26 \mu_s \dot{m}_s}{\sqrt{\rho_s A_s A_b}} \left( \frac{N_{r,cw}}{P_T d_o} + \frac{L_{b,c}}{D_w^2} \right) + \frac{\dot{m}_s^2}{\rho_s A_s A_b} \end{aligned} \quad (17)$$

In Eq. (17),  $N_{r,cw}$  is the number of tube rows crossed in each baffle window,  $D_w$  is the equivalent diameter of window, and  $A_b$  is the area of the surface flow through the window [25].

The combined pressure drop in the entrance and exit sections is calculated from the Eq. (18).

$$\Delta P_e = 2 \Delta P_{b,id} \frac{N_{r,cc} + N_{r,cw}}{N_{r,cc}} R_b R_s \quad (18)$$

In Eq. (18),  $R_s$  is the correction factor for the entrance and exit sections [24].

So, the overall pressure drop on the shell side for the shell and tube heat exchanger is obtained from Eq. (19).

$$\Delta P_s = \Delta P_c + \Delta P_w + \Delta P_e \quad (19)$$

From Eqs. (10) and (16), the total power of the pumps is expressed as Eq. (20) [26]. In Eq. (20),  $\eta$  is the total efficiency of the pump.

$$W = \frac{1}{\eta} \left( \frac{\dot{m}_t}{\rho_t} \Delta P_t + \frac{\dot{m}_s}{\rho_s} \Delta P_s \right) \quad (20)$$

### 3 The optimum design of shell and tube heat exchanger

#### 3.1 Objective function

Entransy is a physical quantity to describe the ability of heat transfer. In fact, its physical meaning can be understood by considering the reversible heat process for a body with a temperature  $T$  and a constant volume heat capacity ( $c_v$ ). 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. By giving a very small amount of heat to any source, the temperature of these heat sources 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, we see an increase in the energy of the thermal energy potential resulting from the thermal load and the potential of the partial temperature. 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. (21) [3].

$$E_h = \int_0^T Q_{vh} dT = \frac{1}{2} m c_v T^2 \quad (21)$$

Thermal energy is stored in the heat transfer process, while entransy is wasted by irreversible heat transfer processes [27]. 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. (22) [28].

$$G_{\Delta T} = -\int_i^o (\dot{m}C_p T dT)_{h,c} = \frac{1}{2} (\dot{m}C_p)_h (T_{h,i}^2 - T_{h,o}^2) + \frac{1}{2} (\dot{m}C_p)_c (T_{c,i}^2 - T_{c,o}^2) \quad (22)$$

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  $Q(T_{h,i} - T_{c,i})$  [28].

$$g_{\Delta T} = \frac{G_{\Delta T}}{Q(T_{h,i} - T_{c,i})} \quad (23)$$

Entransy dissipations due to fluid friction, for an incompressible fluid, can be calculated from Eq. (24) [15, 16].

$$G_{\Delta P} = -\int_i^o \frac{\dot{m}T}{\rho} dp = \left( \frac{\dot{m} \Delta P}{\rho} \frac{T_o - T_i}{\ln\left(\frac{T_o}{T_i}\right)} \right)_{h,c} = \dot{m}_t \frac{\Delta P_t}{\rho_t} \frac{T_{h,o} - T_{h,i}}{\ln\left(\frac{T_{h,o}}{T_{h,i}}\right)} + \dot{m}_s \frac{\Delta P_s}{\rho_s} \frac{T_{c,o} - T_{c,i}}{\ln\left(\frac{T_{c,o}}{T_{c,i}}\right)} \quad (24)$$

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

$$g_{\Delta P} = \frac{G_{\Delta P}}{Q(T_{h,i} - T_{c,i})} \quad (25)$$

Therefore, the entransy dissipations caused by thermal conduction and fluid friction are converted to the entransy dissipations number, by dimensionless method.

### 3.2 Single-objective optimization

#### 3.2.1 Optimal design of constant heat transfer rate

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).

$$g^* = g_{\Delta T} + g_{\Delta P} \quad (26)$$

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 1. The fluid that used on the shell and tube sides is water.

The decision variables according to the objective function contains the number of tubes, the length of the tubes,

**Table 1** The input data for design of heat exchanger with constant heat transfer rate

Characteristic	Condition of shell side	Condition of tube side
Working fluid	Water	Water
Inlet flow pressure (bar)	50	65
Inlet flow temperature (K)	283.15	368.15
Outlet flow temperature (K)	–	343.15
Mass flow rate (kg/s)	–	50
Flow density (kg/m <sup>3</sup> )	991.15	970
Specific heat at constant pressure (J/kg K)	4174	4200
Kinematic viscosity of flow 10 <sup>-7</sup> . (m <sup>2</sup> /s)	6.96	3.36
Fouling resistance (m <sup>2</sup> K/w)	0.00017	0.000086
Prandtl number	4.5878	2.015

the number of baffles and the central angle of the cutting of the baffles.

The design variables and their range are selected according to the following conditions.

1. A sample of several outer diameter of the tubes  $d_o$  (and step of each diameter are given in Table 2. The total number of heat exchanger tubes are between 50 and 550.
2. The ratio of the distance of the intermediate baffles to inner diameter of the shell is between 0.2 up to 1.
3. The central angle of the baffle's cut is between 1.8546 up to 2.9413 rad.
4. The coolant outlet temperature is between 313.15 up to 343.15 K.

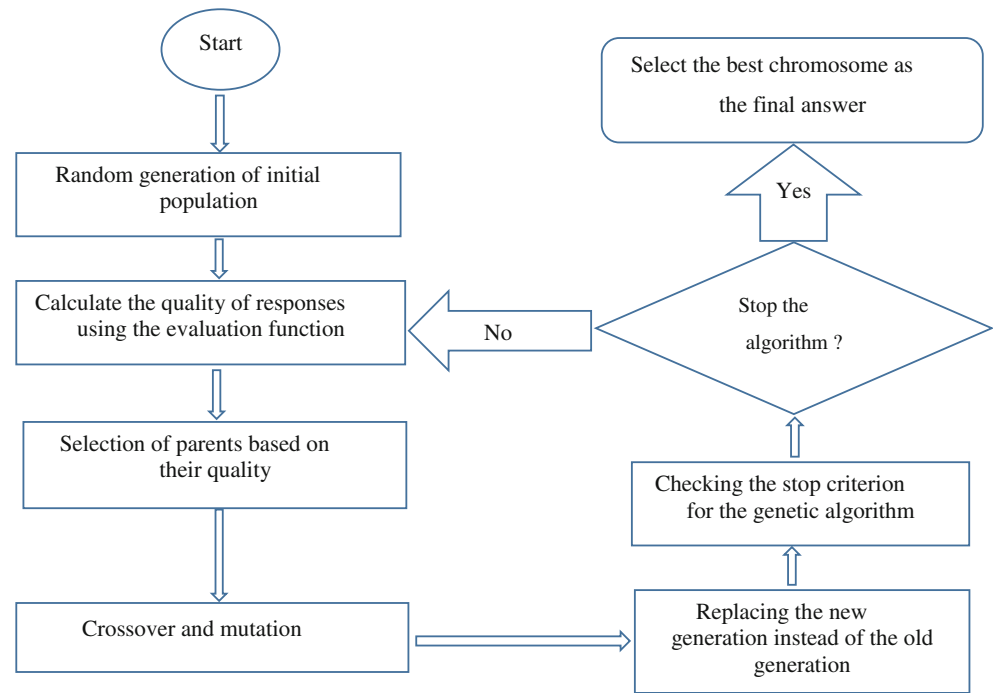
The related constraints for design of the heat exchanger are given as [29]:

1. The tube length ratio to the inner diameter of the shell is between 6 up to 10.
2. The distance between the intermediate baffles should be greater than 50 mm.
3. The tube pressure drop should be less than 50 kPa.
4. The shell pressure drop should be less than 50 kPa.

Due to the non-convexity of the problem, intelligent algorithms that based on population are used. One of these

**Table 2** The external tube diameters and related steps

$d_o$ (mm)	19	20	22	25	30	32	35	38	45
$P_T$ (mm)	25	26	28	32	38	40	44	48	57

**Fig. 2** The flowchart of genetic algorithm

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 [30]. Therefore, for solving the problem the genetic algorithm method is used in the MATLAB software [19]. 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 [30]. More importantly, the genetic algorithm has a great ability to achieve optimal points [31, 32]. For the reasons given, a genetic algorithm [33] 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, by integrating two parents from the current generation, by the action of crossover, or by 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 by eliminating children who are 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

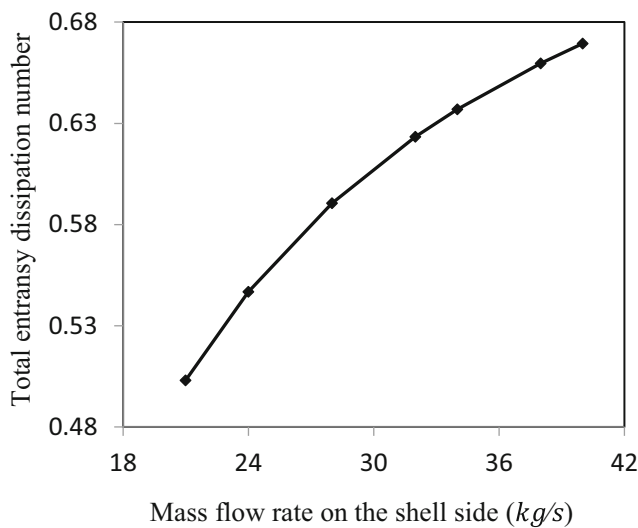
**Table 3** The input data for an optimal exchanger with constant thermal surface

Characteristic	Condition of tube side	Condition of shell side
Working fluid	water	water
Inlet flow pressure (bar)	65	50
Inlet flow temperature (K)	386.15	283.15
Mass flow rate (kg/s)	50	20
flow density (kg/m <sup>3</sup> )	970	991.15
Specific heat capacity at constant pressure (J/kg K)	4200	4174
Kinematic viscosity of flow 10 <sup>-7</sup> . (m <sup>2</sup> /s)	3.36	6.96
Fouling resistance (m <sup>2</sup> K/W)	0.000086	0.00017
Prandtl number	2.015	4.5878

**Table 4** Comparison of the results of validation

Performance parameters	Present study	Results of Guo et al. [36]	Error %
Outlet flow temperature of shell (K)	321.27	321.26	0.003
Heat exchanger effectiveness	0.449	0.448	0.223
Number of transfer units	0.716	0.717	0.140
Heat capacity ratio	0.6559	0.656	0.015
Pump power (W)	1404.4	1403	0.099
The number of entransy dissipations caused by thermal conduction	0.6287	0.63	0.207
The number of entransy dissipations caused by fluid friction	0.0009373	0.000814	13.154





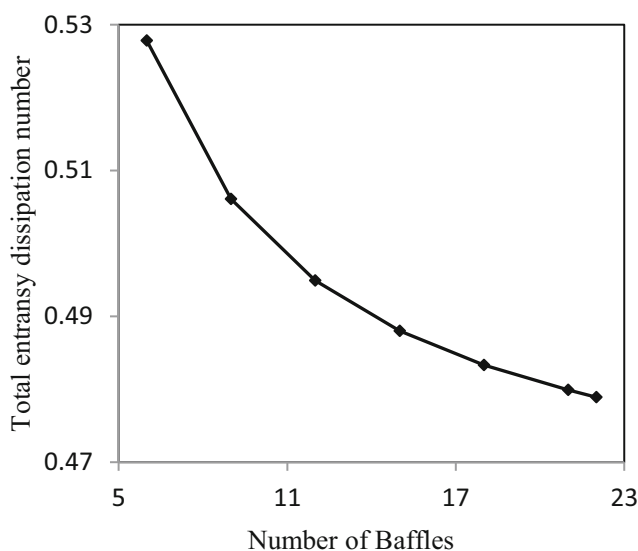
**Fig. 3** Variations of total entransy dissipation number with fluid mass flow rate

problem [34]. The size of the primary population and the maximum number of generations are 40 and 200, respectively. The flowchart of the genetic algorithm is shown in Fig. 2. The condition for stopping the algorithm, which is the same as the change in the objective function in two successive generations, is  $10^{-8}$ .

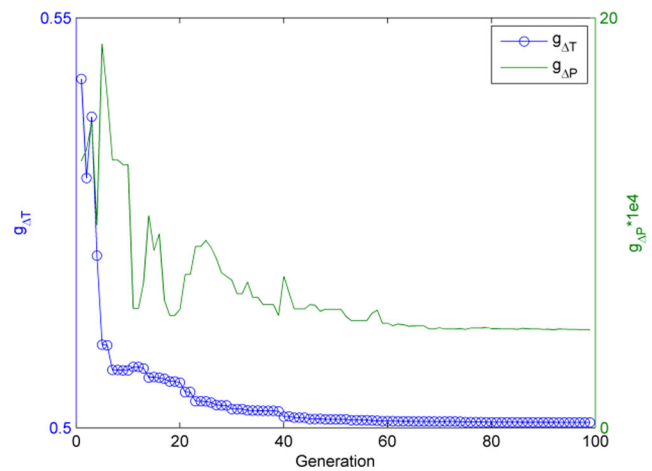
### 3.2.2 Optimal design at constant heat transfer surface area

At this condition, the data for the design of the heat exchanger is given in Table 3.

In this case, the total entransy dissipation numbers are considered as the objective function. Except for the cold flow temperature, the other design parameters and their ranges are the same as in the given table. The heat transfer surface is considered



**Fig. 4** Variations of total entransy dissipation number with the number of baffles



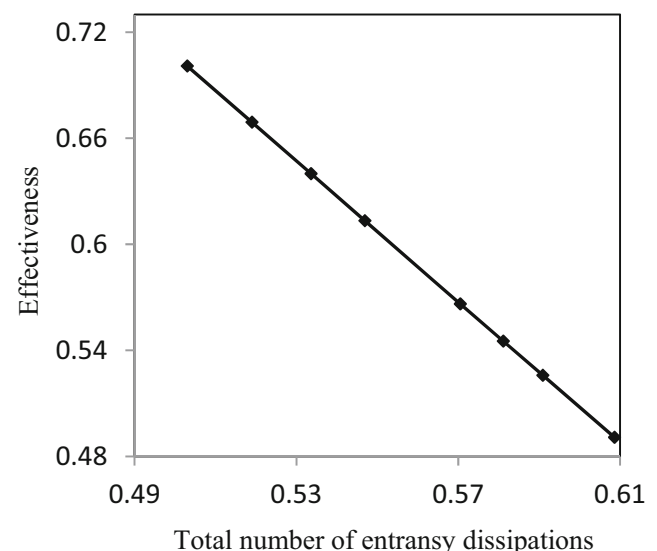
**Fig. 5** The variations of number of entransy dissipations due to the thermal conduction and fluid friction versus the number of generations towards the final optimum value

equal to 70 square meters, and the size of the initial population and the maximum number of generations is 40 and 200, respectively. To solve this optimization problem, the genetic algorithm method is used.

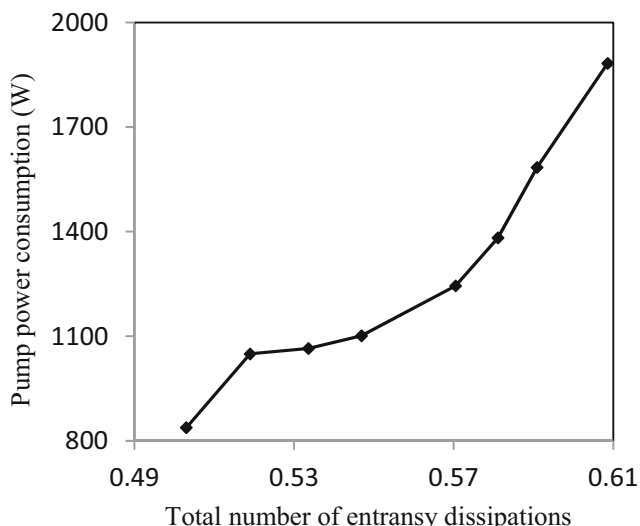
### 3.3 Multi-objective optimization

From the mathematical point of view, multi-objective optimization with respect to equal and unequal constraints, minimizes several objects simultaneously and is shown in Eq. (27).

$$\begin{aligned}
 \min f(x) &= [f_1(x), f_2(x), \dots, f_k(x)] \quad x \in X \quad g_j(x) \\
 &= 0, \quad j = 1, 2, \dots, M \quad h_k(x) \leq 0, \quad k \\
 &= 1, 2, \dots, K
 \end{aligned} \tag{27}$$



**Fig. 6** Variation of effectiveness based on the total number of entransy dissipations



**Fig. 7** Variation of pump power consumption based on the total number of entransy dissipations

$x$  is the independent parameters of the objective function.

$X$  is a parameter space set, if and only if,  $f_i(x) \leq f_i(y)$  for  $i = 1, 2, \dots, k$  and  $f_j(x) < f_j(y)$  for at least one objective function  $j$ ,  $x$  the answer overcomes to all  $y$  's answers. If an answer does not overcome other answers in the possible range, then the answer set is called the optimal solution of the Pareto. The sets of all non-dominant solutions in  $X$  are called optimal Pareto sets ( $P^*$ ). The values of the objective function for the optimal Pareto set are called the Pareto front ( $PF^*$ ) and can be represented as Eq. (28) [35].

$$PF^* := \{f(x) | x \in P^*\} \tag{28}$$

In particular, according to Eq. (26), the entransy dissipations number due to thermal conduction and fluid friction are considered as two separate objective functions, respectively. The input parameters and their ranges are same as that indicated in the optimization of the single-objective with the constant heat transfer surface area. The given data for the heat exchanger is shown in Table 3.

An effective method for solving these problems is a differential algorithm called Pareto optimization [36]. In this method, instead of an optimal response, a set of several optimal answers are obtained for the multi-objective optimization problem, that each of them is equivalent to adopting a special combination of weighted values for different objective

functions. In other words, each pareto answer assigns a particular importance to different functions and provides an optimal response to it. Therefore, the result set of the Pareto method allows the optimal selection to be made according to the importance given to the various objective functions. Therefore, due to the advantages of this method, in this study, the problem of optimization is performed by combination of genetic algorithm with Pareto optimization method which called multi-objective non-dominated Sorting Genetic Algorithm (NSGAI) [35]. This algorithm is one of the most popular and applicable algorithms in the field of multi-objective optimization. In this method, each of the genetic algorithm chromosomes contains a series of proposed values for the variables optimized problem. In each iteration of the genetic algorithm, after calculating the values of the two objective functions for each proposed chromosome, the selection step in the genetic algorithm is performed using the Pareto algorithm and the concept of dominance. According to the Pareto optimization algorithm, in each iteration, in the selection stage of the genetic algorithm, all chromosomes are examined pairwise, and the dominant chromosomes are selected, and they form the first front of the Pareto. Then, the chromosomes related to the first front of the Pareto are discarded from the population and the process is repeated. With the repetition of the process, the second and third and so on, fronts of the Pareto are identified. In other words, the population is partitioned into different fronts. Then, from the resulting fronts, a certain number of chromosomes are selected as parents, so that first the chromosomes of the first front of the Pareto, then the second front, and so on, are selected until the desired number for parental chromosomes are selected. The number of generations of a set is considered 200, which acts as a stopping point to end the repeat process.

## 4 Results and discussion

### 4.1 Validation of results

In order to validate the calculations, for a shell and tube heat exchanger, with one pass of shell and one tube pass in the mode of constant heat transfer rate, the conditions of Table 1 were chosen from the study of Guo et al. [36], and subsequently the results of this study were compared with the results of Ref. [36] and presented in Table 4.

**Table 5** Comparison between primary and optimal design in terms of constant heat transfer rate

Specifications	$n$	$d_o(m)$	$L_{bc}/D_s$	$g^*$	$g_{\Delta P}$	$g_{\Delta T}$	$\epsilon$	W(W)	$C^*$	NTU	$T_{s,o}(K)$	$\theta_b(rad)$
Primary	400	0.019	0.699	0.6086	0.0010068	0.6076	0.4909	1882.7	0.599	0.815	324.8	2.453
Optimum	550	0.02	0.921	0.5030	0.000465	0.5025	0.7009	837.8	0.419	1.479	342.7	2.887

**Table 6** Single-objective optimal design at constant heat transfer surface area

Specifications	$d_o(m)$	$n$	$g^*$	$\varepsilon$	$W(W)$	$NTU$	$\theta_b(rad)$	$L_{bc}/D_s$
Optimized	0.035	50	0.4789	0.7008	820.09	2.767	2.1064	0.2258

As can be seen, the comparison shows that the results are very close to each other. The results of Table 4 are for conditions of the tubes with 0.019 m internal diameter, the number of tubes is 243. In addition, the ratio of the distance between the two intermediate baffles to the inner diameter of the shell is 0.977, and the central angle of baffle's cut is 2.038 rad.

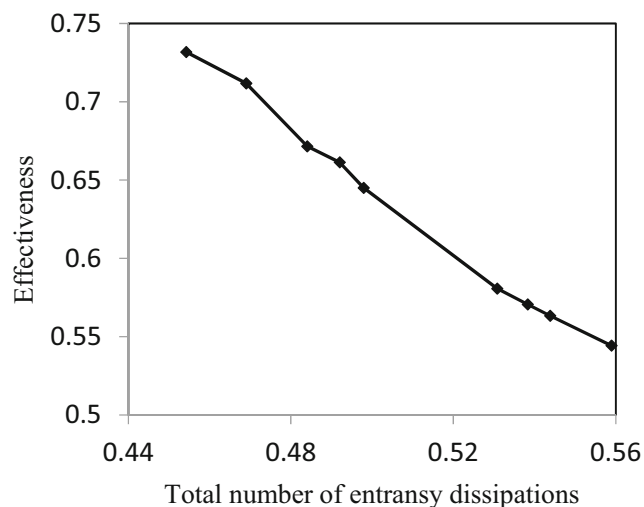
## 4.2 The effect of parameters on entransy dissipations

Figure 3 shows the fluid mass flow rate variations of the shell 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, by decreasing this parameter the number of entransy dissipations decreases.

Figure 4 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 surface. Physically, the heat transfer rate increases as the number of baffle increases. By increasing the heat transfer rate and increasing the effectiveness, the entransy dissipations decrease.

## 4.3 Single-objective optimization results under constant heat transfer rate conditions

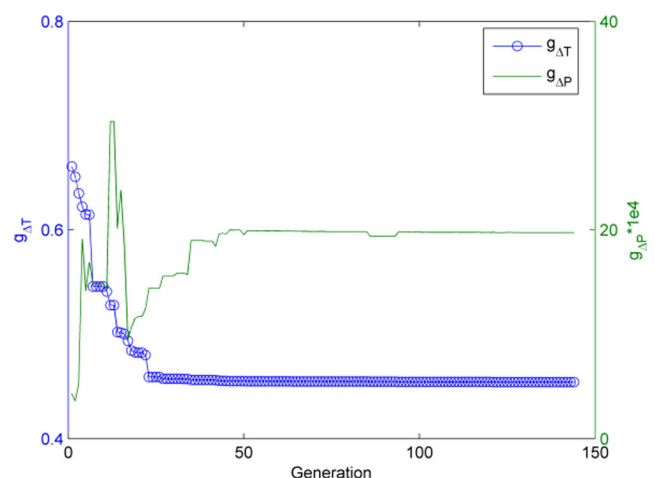
Figure 5 shows the variations in the best values for entransy dissipations versus the number of generations towards the optimum value (minimum value). The value of objective function was calculated at each iteration and the minimum value at

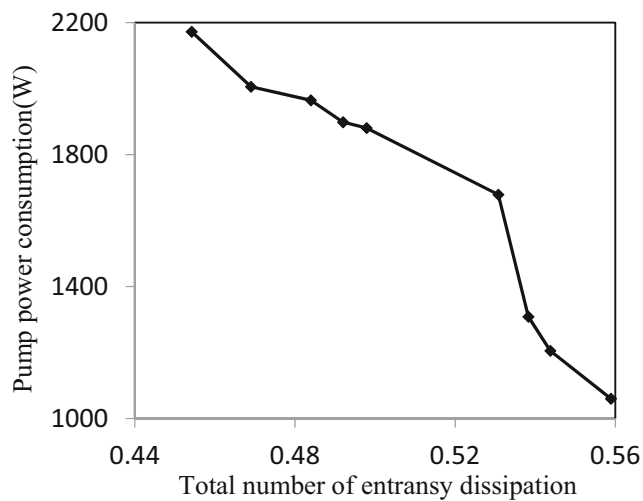
**Fig. 8** Variation of effectiveness based on the total number of entransy dissipations

each iteration is selected. As is evident from this figure, initially the number of entransy dissipations caused by the thermal conduction and fluid friction decreases and be constant after almost sixty times repetition. Considering the unchanged value of the objective function in the next iteration, the optimization computation was terminated. It can also be find from this figure that the genetic algorithm has a very high application in finding an optimal overall solution.

Figures 6 and 7 show, the changes in the effectiveness of the heat exchanger and the power of the pump in terms of the total number of entransy dissipations, respectively. As can be seen from these figures, by increase of the total number of entransy dissipations, the effectiveness of the heat exchanger decreases and the pump power consumption rises sharply.

Thus, through the process of optimization by selecting the decision variables in order of minimizing the total number of entransy dissipations, the performance of the heat exchanger is considerably improved and pump power consumption is reduced. In order to illustrate the advantages of the optimized objective function under the constant heat transfer rate, the comparison between the randomized primary design of the generation and its optimized is presented show in Table 5. In this paper, Table 5 shows that the effectiveness of the heat exchanger is changed from 0.4909 up to 0.7009. That means, the effectiveness of the heat exchanger increases near 42%, whereas, the power consumption of the pump from 1882.7 watts is reduced to 837.8 watts. Therefore, with the reduction of entransy dissipations, the performance of heat exchanger will be improved. Also, the number of transfer units increases 1.8 times larger than initial state. As the number of transfer

**Fig. 9** Variation of the number of entransy dissipations caused by the thermal conduction and fluid friction with the number of generations



**Fig. 10** Variation of pump power consumption with the total number of entransy dissipation

units increases, the performance of the heat exchanger improves with increase of the heat transfer surface. With regard to Table 5, it can be concluded that the number of entransy dissipations due to fluid friction is about three 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 [37].

Therefore, the optimal single-objective design of the heat exchanger for the desired shell and tube heat exchanger is time-consuming. Since the number of entransy dissipations caused by thermal conduction and fluid friction is not co-ordinate, considering these two parameters as objective function may lead to some unintended consequences. This can be clearly seen in the design of the heat exchanger in the state of constant heat transfer surface area.

#### 4.4 Single-objective optimization results at constant heat transfer surface area

The results of single-objective optimization for diameter of 0.035 m can be seen in Table 6.

The relationship between the effectiveness of the heat exchanger and the total number of entransy dissipations is

shown in Fig. 8. From Fig. 8, it can be seen that by reducing the number of entransy dissipations, the effectiveness of the heat exchanger is significantly improved due to the increase in the number of transfer units. Also in Fig. 9, the variations of objective functions which are the number of entransy dissipations caused by the thermal conduction and fluid friction with the number of generations have been shown at optimization procedure.

As shown in Fig. 9, with increasing number of generations, the number of entransy dissipations caused by thermal conduction decreases considerably, while the number of entransy dissipations caused by fluid friction increases, although the order of magnitude of entransy dissipations caused by fluid friction is less than it based on thermal conduction.

Figure 10 shows the relationship between the pump power consumption and the total entransy dissipations. From the Figs. 8 and 10, it can be concluded that by reducing the total number of entransy dissipations, the effectiveness of the heat exchanger is significantly improved, whereas the power consumption of the pump is dramatically increased. Since the heat transfer surface area is constant, in this case, improving the effectiveness of the heat exchanger leads to the higher cost of the pump power consumption. From Figs. 8 and 10 it can be seen that considering 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 because it is much smaller than the number of entransy dissipations caused by thermal conduction. In order to solve this problem, a multi-objective optimal design for shell and tube heat exchangers is presented in the next section.

#### 4.5 Multi-objective optimization results under constant surface heat transfer conditions

Some of the solutions obtained from multi-objective optimization are shown in Table 7. From this table, it can be seen that the higher effectiveness can be obtained at lower pump power consumption. According to Tables 6 and 7, it can be

**Table 7** Pareto front results using multi-objective optimization method at constant heat transfer surface area

$d_o(m)$	$n$	$L_{bc}/D_s$	$\theta_b(rad)$	$g^*$	$g_{\Delta P}$	$g_{\Delta T}$	$\varepsilon$	$W(W)$	$U_o(kW/m^2 \cdot K)$	$Re_t$	$f_t$	$Nu$
0.030	78	0.3787	2.862	0.5	0.0003	0.4997	0.6790	337.74	1.4709	35,360	0.0050	136.1
0.030	78	0.2523	2.934	0.489	0.0003	0.4896	0.6925	459.86	1.5589	35,360	0.0050	136.1
0.030	78	0.2523	2.934	0.488	0.0004	0.4881	0.6947	461.46	1.5738	35,360	0.0050	136.1
0.030	55	0.3005	2.934	0.475	0.0005	0.4753	0.7119	770.084	1.7115	50,148	0.0052	180.9
0.030	55	0.3005	2.934	0.474	0.0006	0.4743	0.7134	773.04	1.7243	50,148	0.0052	180.9
0.030	55	0.3005	2.934	0.474	0.0007	0.4738	0.7141	774.24	1.7305	50,148	0.0052	180.9

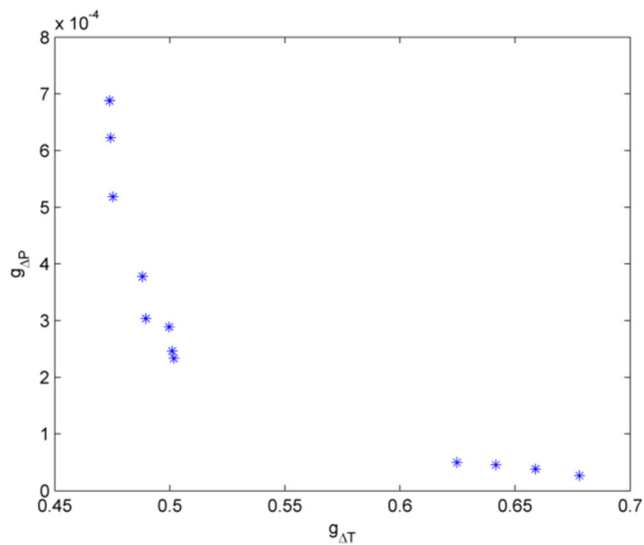


Fig. 11 Pareto front for constant surface heat transfer area

concluded that the optimal solution in Table 7 requires lower pump power consumption with an almost equal effectiveness compared to Table 6. The Pareto front, which includes multi-objective optimization results, has been shown in Fig. 11.

Figure 11 shows the variation in the number of entransy dissipations caused by thermal conductin and fluid friction for different optimal solutions in the pareto optimization set. There are several optimum solution in this method. In addition, the power consumption of the pump and the effectiveness of the heat exchanger for optimal solutions are shown in Fig. 12.

Also, due to the fact that there are a several optimal solutions from multi-objective optimization, for design of the heat exchanger there are more alternatives rather than single-objective solving and each optimum solution can be chosen based on users' needs.

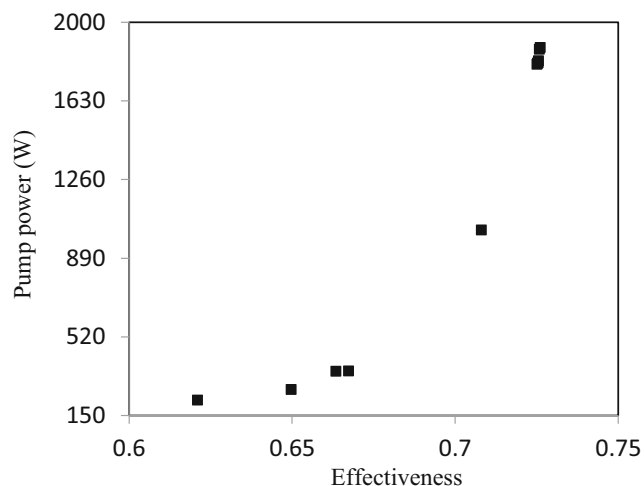


Fig. 12 Pump power consumption in terms of the effectiveness of the heat exchanger at optimum conditions based on Pareto front method

## 5 Conclusion

In this research, the performance of shell and tube heat exchangers at conditions of constant heat transfer rate and constant heat transfer surface area 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.

Two optimization methods have been investigated for optimal design of heat exchangers. In the first method, a single-objective optimization approach has been developed in which the total entransy dissipation has been considered as a 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.
- In the case where the heat transfer surface area is constant, improving the effectiveness of the heat exchanger with the optimization process leads to increase of the cost of pump power consumption. In order to solve this problem, a multi-objective optimization method for heat exchanger was created. In which the number of entransy dissipations caused by thermal conduction and fluid friction were considered as two separate objectives.
- In contrast to the single-objective optimization approach, with the optimal design of the multi-objective for the heat exchangers, it can be achieved approximately the same effectiveness, but with less pump power consumption.
- The minimum of total number of entransy dissipations as an objective function is almost equivalent to minimizing the entransy dissipatins due to thermal conduction, whereas the impact of number of entransy dissipations caused by fluid friction can be almost ignored.
- The multi-objective optimization leads to non-unique optimal solutions, which reflects more flexibility for the optimal design of heat exchangers and allows to select more options.
- In most engineering problems, the targets for optimization are in contrast with each other. As one of the targets improves, the other one goes towards undesirable condition. So, depending on which target is preferable, the optimal answer can be different. Therefore, instead of one answer, there is a set of solutions. This is another reason to demonstrate the more flexibility of multi-objective optimization that used in this study. In addition, the set of answers in the curve is not superior to each other and so-called non-dominant.

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