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**SECOND LAW BASED OPTIMISATION OF CROSSFLOW PLATE-FIN HEAT EXCHANGER DESIGN USING GENETIC ALGORITHM**

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## ABSTRACT

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well as discrete variables in the presence of given constraints. The optimisation program aims at minimising the number of entropy generation units for a specified heat duty under given space restrictions. The results have also been obtained and validated through graphical contours of the objective function in the feasible design space. The effect of variation of heat exchanger dimensions on the optimum solution has also been presented.

**Key words:** crossflow heat exchanger, entropy generation number, genetic algorithm, optimisation, plate-fin.

## Nomenclature

$A, A_{HT}$  = heat transfer area,  $m^2$

$A_{ff}$  = free flow area,  $m^2$

$C$  = heat capacity rate ( $mC_p$ ),  $W K^{-1}$

$C_p$  = specific heat of fluid,  $W kg^{-1} K^{-1}$

$Cr = C_{min}/C_{max}$

$D_h$  = hydraulic diameter,  $m$

$f$  = Fanning friction factor

$f(X)$  = objective function

$f_{max}$  = fitness parameter

$g(X)$  = constraint

$G$  = mass flux velocity ( $m/A_{ff}$ ),  $kg m^{-2} s^{-1}$

$h$  = heat transfer coefficient,  $W m^{-2} K^{-1}$

$H$  = height of the fin,  $m$

$j$  = Colburn factor

$l$  = lance length of the fin,  $m$

$L$  = heat exchanger length,  $m$

$m$  = mass flow rate of fluid,  $\text{kg s}^{-1}$   
 $n$  = fin frequency, fins per meter  
 $N$  = number of years  
 $N_a, N_b$  = number of fin-layers for fluid a and b  
 $N_s$  = number of entropy generation units, dimensionless  
 $\text{NTU}$  = number of transfer units, dimensionless  
 $P$  = pressure,  $\text{N m}^{-2}$   
 $p_c$  = crossover probability  
 $p_m$  = mutation probability  
 $\text{Pr}$  = Prandtl number  
 $\Delta P$  = pressure drop,  $\text{N m}^{-2}$   
 $Q$  = rate of heat transfer,  $\text{W}$   
 $R$  = specific gas constant,  $\text{J kg}^{-1} \text{K}^{-1}$   
 $R1$  = penalty parameters  
 $\text{Re}$  = Reynolds number  
 $s$  = fin spacing ( $1/n-t$ ),  $\text{m}$   
 $\dot{S}$  = rate of entropy generation,  $\text{W K}^{-1}$

$\Delta S$  = entropy difference,  $\text{W/kg-K}$   
 $\text{St}$  = Stanton number [ $h/(GC_p)$ ]  
 $t$  = fin thickness,  $\text{m}$   
 $T$  = Temperature,  $\text{K}$   
 $U$  = overall heat transfer coefficient,  $\text{W m}^{-2} \text{K}^{-1}$   
 $x_i$  = variable  
 $X = (x_1, x_2, \dots, x_k)$

***Greek symbols***

$\varepsilon$  = effectiveness  
 $\rho$  = density,  $\text{kg m}^{-3}$   
 $\mu$  = viscosity,  $\text{N m}^{-2} \text{s}^{-1}$   
 $\phi(.)$  = penalty function

***Subscripts***

$a, b$  = fluid a and b  
 $i$  = variable number  
 $1$  = inlet  
 $\text{max}$  = maximum  
 $\text{min}$  = minimum  
 $2$  = exit

**1. Introduction**

Compact heat exchangers are characterised by a large heat transfer surface area per unit volume. This leads to reduced space, weight, support structure, and footprint; reduced energy requirement and cost; and improved process design compared to conventional heat exchangers. Amongst different varieties of compact heat exchangers crossflow plate-fin heat exchangers are widely used in aerospace, automobile, cryogenic and chemical process plants for their low weight and volume, high efficiency and ability to handle many streams. However, the superior thermal performance of compact heat exchangers is in general associated with a high pressure-drop and its related aspects. Therefore, it often becomes necessary to find a trade-off between the increased rate of heat exchange and the power consumption due to higher pressure-drop within the constraints of specified performance requirements with available resources. Also, analysis based on second law of thermodynamics is applied for this purpose and can best deal with this situation.

Second-law based optimisation by entropy generation minimisation (EGM) is the method of thermodynamic optimisation of real systems that owe thermodynamic imperfection to the irreversibilities due to heat transfer, fluid flow and mass transfer. The thermodynamic irreversibility or number of entropy generation units ( $N_s$ ) indicates the amount of lost useful power, which is not available due to system irreversibilities. In a heat exchanger, irreversibilities are generated due to finite temperature difference heat transfer in the fluid streams and the pressure drops along them. Optimising heat exchanger or any other system on this basis means minimising the amount of lost or unavailable power by accounting for the finite size constraints of actual devices and finite time constraints of actual process [1, 2]. London [3] has discussed in detail about the entropy generation, irreversibility evaluation and the relationship between irreversibility and the economics by taking an example of a condenser. An operationally

convenient methodology has also been presented by [London and Shah \[4\]](#) for relating economic costs to entropy generation. This methodology allows the designer to determine the trade-offs between the individual irreversibilities due to flow friction, heat transfer, heat leakage and mixing in context to a heat exchanger.

[Bejan \[5\]](#) presented the design of a gas-to-gas counterflow heat exchanger for minimum irreversibility and the design of a regenerative heat exchanger for minimum heat transfer area with fixed irreversibility. [Seculic and Herman \[6\]](#) have presented the optimisation of a compact crossflow heat exchanger for the minimum enthalpy exchange irreversibility (EEI) using numerical method. Instead of optimising single component, global performance of the installation was used by [Vargas et al. \[7\]](#) for optimisation of total component volume and wall material volume by taking an example of crossflow heat exchanger used in environmental control system of an aircraft. [Vargas and Bejan \[8\]](#) again used the concept of optimising global performance by selecting finned and /or smooth parallel plate type crossflow heat exchanger of the environmental control system of an aircraft.

Different search techniques can be good alternatives for optimisation problems containing discrete or discrete-continuous variables. However, the conventional techniques become very cumbersome and laborious when the extremum is sought for a multivariable problem having a number of constraints [\[9\]](#). There are a few classical techniques of optimisation, which can handle a combination of continuous and discrete variables, the solution procedure becomes rather complex [\[10\]](#). A compact plate-fin type crossflow heat exchanger possesses a large number of design variables. The performance parameters of the heat exchanger bear complex functional relationships with these variables. Further, some of these variables are often discrete in nature. These render the optimisation of such equipment a rather difficult task. In

recent times, some non-traditional probabilistic search algorithms, namely genetic algorithm (GA) and simulated annealing (SA) are being applied to the optimisation of various engineering systems in general and to thermo-processes and fluid applications in particular. These techniques can overcome the above-mentioned difficulties to a large extent. Genetic algorithm mimics the principle of natural genetics and natural selection to constitute search and optimisation procedures. Genetic Algorithm (GA) based on evolutionary global search technique is particularly suitable for such problems [11]. Genetic algorithm has been applied successfully for the optimum design of different thermal systems and components namely convectively cooled electronic components [12] and cooling channels [13], fin profiles [14], finned surface and finned annular ducts [15], compact high performance coolers [11], shell and tube heat exchangers [10] and compact plate-fin heat exchangers [16]. Further, optimisation of crossflow plate-fin heat exchangers have been done by minimising total annual cost [17] and total thermoeconomic cost [18] of the exchanger.

In this work a GA based optimisation technique for crossflow plate-fin heat exchangers has been developed, which minimises the total number of entropy generation units [1, 2] for a specified heat duty under given space restrictions. The solution has been obtained in terms of optimising the heat exchanger dimensions as well as fin specifications. The optimum result compares well with that obtained by the graphical technique. The selection of optimum GA parameters for the present problem has also been done to achieve the faster and better result.

## 2. Outline of the Scheme of Optimisation

Genetic algorithm is an evolutionary search procedure based on the principles of genetics and natural selection. An elaborate description of this technique is available in a number of references [19-22].

The genetic search is started with an initial set of *population*. The members of population can conveniently be represented by a binary coding consisting of 0's and 1's. The value of objective function for a particular member decides its merit (competitiveness) in comparison with its counterparts. In GA language this is termed as *fitness function*. After creating an initial population, a simple GA works with three operators: *reproduction*, *crossover* and *mutation*. Reproduction, which constitutes a selection procedure whereby individual strings are selected for mating based on their fitness values relative to the fitness of the other members. Individuals with higher fitness values have a higher probability of being selected for mating and for subsequent genetic production of offsprings. This operator, which weakly mimics the Darwinian principal of survival of the fittest, is an artificial version of natural selection, where the selection is done stochastically.

After reproduction, the crossover operator alters the composition of the offspring by exchanging part of strings from the parents and hence creates new strings. Crossover is also achieved stochastically using a suitable crossover probability. The need for mutation is to create point in the vicinity of the current point, thereby achieving a local search around the current solution, which sometimes is not possible by reproduction and crossover. Mutation increases the variability of the population. For a GA using binary alphabet to represent a chromosome, mutation provides variation to the population by changing a bit of the string from 0 to 1 or vice versa with a small mutation probability.

GA does not guarantee convergence to global optimum solution and so require suitable stopping criteria. The GA can be terminated when there is no improvement in the objective function (fitness) for a defined number of consecutive generations within a prescribed tolerance range, or when it covers a pre-specified maximum number of generations.

In the simplest form GA can be formulated as unconstrained maximisation [22]. For the present problem GA has been used for constrained minimisation. If there are number of constraint conditions and the objective function needs to be minimised the problem can be stated as follows:

$$\text{Minimise } f(\mathbf{X}), \quad \mathbf{X}=[x_1, \dots, x_k] \quad (1)$$

Where constraints are given by

$$g_j(\mathbf{X}) \leq 0, \quad j=1, \dots, m \quad (2)$$

and

$$x_{i, \min} \leq x_i \leq x_{i, \max}, \quad i=1, \dots, k. \quad (3)$$

For implementation in GA, the first step is to convert the constrained optimisation problem into an unconstrained one by adding a penalty function term.

$$\text{Minimise } f(\mathbf{X}) + \sum_{j=1}^m \Phi(g_j(\mathbf{X})), \quad (4)$$

subject to

$$x_{i, \min} \leq x_i \leq x_{i, \max}, \quad i=1, \dots, k. \quad (5)$$

Where  $\Phi$  is a penalty function defined as,

$$\Phi(g(\mathbf{X})) = R1 \cdot \langle g(\mathbf{X}) \rangle^2. \quad (6)$$

Here R1 is the penalty parameter having an arbitrary large value.



The second step is to convert the minimisation problem to a maximisation one. This is done redefining the objective function such that the optimum point remains unchanged. The conversion used in the present work is as follows

$$\text{Maximise } F(\mathbf{X}), \quad (7)$$

where,

$$F(\mathbf{X}) = 1 / \{ f(\mathbf{X}) + \sum_{j=1}^m \Phi(g_j(\mathbf{X})) \}. \quad (8)$$

More details regarding the scheme and the algorithm are given by [Mishra et al. \[17\]](#).

### 3. Thermodynamic Optimisation

[Figure 1](#) depicts a schematic view of a crossflow plate-fin heat exchanger with offset-strip fins. Following assumptions are made for the analysis.

1. The heat exchanger is operating under steady state condition.
2. Offset-strip fins of the same specifications are used for both the fluids.
3. Both the fluids are assumed to be ideal gases.
4. Heat transfer coefficients and the area distribution are assumed to be uniform and constant.
5. Physical property variation of the fluids with temperature is neglected.
6. Number of fin layers for fluid b is assumed to be one more than that of fluid a ( $N_b=N_a+1$ )

Rate of entropy generation for the two fluid streams is

$$\dot{S} = m_a(\Delta S_a) + m_b(\Delta S_b) \quad (9)$$

Following the methodology of [Bejan \[5\]](#),  $\dot{S}$  can be expressed in terms of temperature and pressure.

$$\dot{S} = m_a \left[ Cp_a \ln \frac{T_{a,2}}{T_{a,1}} - R_a \ln \frac{P_{a,2}}{P_{a,1}} \right] + m_b \left[ Cp_b \ln \frac{T_{b,2}}{T_{b,1}} - R_b \ln \frac{P_{b,2}}{P_{b,1}} \right] \quad (10)$$

$$\text{Now } \varepsilon = \frac{C_a(T_{a,1} - T_{a,2})}{C_{\min}(T_{a,1} - T_{b,1})} = \frac{C_b(T_{b,2} - T_{b,1})}{C_{\min}(T_{a,1} - T_{b,1})} \quad (11)$$

$$\text{So, } T_{a,2} = T_{a,1} - \varepsilon \frac{C_{\min}}{C_a}(T_{a,1} - T_{b,1}) \quad (12)$$

$$T_{b,2} = T_{b,1} + \varepsilon \frac{C_{\min}}{C_b}(T_{a,1} - T_{b,1}) \quad (13)$$

$$\text{and, } P_{a,2} = P_{a,1} - (P_{a,1} - P_{a,2}) = P_{a,1} - \Delta P_a \quad (14)$$

$$P_{b,2} = P_{b,1} - (P_{b,1} - P_{b,2}) = P_{b,1} - \Delta P_b \quad (15)$$

Finally number of entropy generation units ( $N_s = \frac{\dot{S}}{C_{\max}}$ ) is defined as follows.

$$N_s = \frac{C_a}{C_{\max}} \left[ \ln \left\{ 1 - \varepsilon \frac{C_{\min}}{C_a} \left( 1 - \frac{T_{b,1}}{T_{a,1}} \right) \right\} - \frac{R_a}{Cp_a} \ln \left\{ 1 - \frac{\Delta P_a}{P_{a,1}} \right\} \right] \\ + \frac{C_b}{C_{\max}} \left[ \ln \left\{ 1 + \varepsilon \frac{C_{\min}}{C_b} \left( \frac{T_{a,1}}{T_{b,1}} - 1 \right) \right\} - \frac{R_b}{Cp_b} \ln \left\{ 1 - \frac{\Delta P_b}{P_{b,1}} \right\} \right] \quad (16)$$

For crossflow heat exchanger with both fluids unmixed, effectiveness [23] is given by

$$\varepsilon = 1 - \exp \left[ \left( \frac{1}{Cr} \right) NTU^{0.22} \left\{ \exp[-Cr \cdot NTU^{0.78}] - 1 \right\} \right], \quad (17)$$

where,  $Cr = C_{\min} / C_{\max}$ ,

$$\text{and } \frac{1}{NTU} = \frac{C_{\min}}{UA} = C_{\min} \left[ \frac{1}{(hA)_a} + \frac{1}{(hA)_b} \right] \quad (18)$$

Introducing the expressions for heat transfer coefficients,

$$\frac{1}{NTU} = C_{\min} \left[ \frac{1}{j_a Cp_a Pr_a^{-2/3} m_a} \frac{A_{ff a}}{A_a} + \frac{1}{j_b Cp_b Pr_b^{-2/3} m_b} \frac{A_{ff b}}{A_b} \right]. \quad (19)$$

For the geometrical details shown in [Figure 1](#), one may get the free flow areas as

$$A_{ff_a} = (H_a - t_a)(1 - n_a t_a) L_b N_a, \quad (20)$$

$$A_{ff_b} = (H_b - t_b)(1 - n_b t_b) L_a N_b. \quad (21)$$

Similarly heat transfer areas for the two sides can be obtained as given below.

$$A_a = L_a L_b N_a [1 + 2 n_a (H_a - t_a)] \quad (22)$$

$$A_b = L_a L_b N_b [1 + 2 n_b (H_b - t_b)] \quad (23)$$

$$\text{Total heat transfer area, } A_{HT} = A_a + A_b = L_a L_b [N_a \{1 + 2 n_a (H_a - t_a)\} + N_b \{1 + 2 n_b (H_b - t_b)\}] \quad (24)$$

Rate of heat transfer may be calculated as follows

$$Q = \varepsilon C_{\min} (T_{a,1} - T_{b,1}) \quad (25)$$

Also, frictional pressure drop [\[24\]](#) for the two fluid streams can be calculated readily as

$$\Delta P_a = \frac{4 f_a L_a G_a^2}{2 \rho_a D_{h,a}} = \frac{2 f_a m_a^2}{\rho_a} \frac{L_a}{D_{h,a} L_b^2 N_a^2 (H_a - t_a)^2 (1 - n_a t_a)^2}, \quad (26)$$

$$\Delta P_b = \frac{4 f_b L_b G_b^2}{2 \rho_b D_{h,b}} = \frac{2 f_b m_b^2}{\rho_b} \frac{L_b}{D_{h,b} L_a^2 N_b^2 (H_b - t_b)^2 (1 - n_b t_b)^2}. \quad (27)$$

*j* and *f* factors may be evaluated from available correlations [\[25\]](#).

For laminar flow ( $Re \leq 1500$ )

$$j = 0.53(Re)^{-0.5} (l/D_h)^{-0.15} \{s/(H-t)\}^{-0.14} \quad (28)$$

$$f = 8.12(Re)^{-0.74} (l/D_h)^{-0.41} \{s/(H-t)\}^{-0.02} \quad (29)$$

For turbulent flow ( $Re > 1500$ )

$$j = 0.21(Re)^{-0.4} (l/D_h)^{-0.24} (t/D_h)^{0.02} \quad (30)$$

$$f = 1.12(Re)^{-0.36} (l/D_h)^{-0.65} (t/D_h)^{0.17} \quad (31)$$

Where,

$$Re = \frac{GD_h}{\mu} = \frac{m D_h}{Aff \mu}. \quad (32)$$

For the given fin geometry the hydraulic diameter  $D_h$  is given by,

$$D_h = \frac{2(s-t)(H-t)}{\{s+(H-t)\} + \frac{(H-t)t}{l}}, \quad (33)$$

where

$$s = (1/n - t). \quad (34)$$

Now the statement of optimisation problem in terms of the variables defined above is as follows.

$$\text{Minimise } f(X) = N_s, \quad (35)$$

subjected to the constraints:

$$\left. \begin{aligned} g_1(X) &\Rightarrow 0.1 \leq L_a \leq 1; \\ g_2(X) &\Rightarrow 0.1 \leq L_b \leq 1; \\ g_3(X) &\Rightarrow 0.002 \leq H \leq 0.01; \\ g_4(X) &\Rightarrow 100 \leq n \leq 1000; \\ g_5(X) &\Rightarrow 0.0001 \leq t \leq 0.0002; \\ g_6(X) &\Rightarrow 0.001 \leq l \leq 0.010; \\ g_7(X) &\Rightarrow 1 \leq N_a \leq 10; \\ g_8(X) &\Rightarrow \xi(X) - Q = 0. \end{aligned} \right\} \quad (36)$$

It may be noted that  $g_8(x)$  is the equality constraint obtained from modifying the Eq. (25), where  $\xi(X)$  represents the left hand side of the equation, and  $Q$  is the heat duty requirement of the exchanger mentioned as 160 kW in the present example .

Different operating variables selected for the present example are from a sizing problem [26] modified to a constrained minimization problem.

A gas-to-air crossflow plate-fin heat exchanger having minimum heat duty 160 kW needs to be designed and optimized for minimum entropy generation. The gas and air have inlet temperatures as 240 °C and 4 °C respectively, and flow rates as 0.8962 and 0.8296 kg/s respectively. The fin surfaces on both sides of exchanger are assumed to be plate-fins, having same specifications. Both the fluids are assumed to be air behaving as ideal gas. Maximum dimension of the exchanger is limited to 1 m x 1 m, and maximum number of fin-layer for gas side is to be 10. The range of fin parameters (fin height, fin frequency, fin thickness and lance length) are also defined and shown in eq. (36). Thus the objective is to find out the heat exchanger dimensions ( $L_a$  and  $L_b$ ), number of fin layers ( $N_a$  and or  $N_b$ ) and other fin parameters ( $H$ ,  $n$ ,  $t$  and  $l$ ) giving the required heat duty for minimum entropy generation.

The basic parameters and the property values considered for the two fluids are shown in Table 1.

#### **4. Results**

Though the designer has some independence in selecting the GA parameters, it has been observed that selection of proper GA parameters renders a quick convergence of the algorithm and the proper GA parameters are problem specific [13, 27]. Therefore initially an exercise has been made following the methodology of Wolfersdorf et al. [13] to select the optimum GA

parameters for the present problem. Figure 2 (a) to (d) shows the variation of maximum fitness function  $f_{\max}$ , number of entropy generation units  $N_s$ , and effectiveness  $\epsilon$ , with the population size, crossover and mutation probabilities and penalty parameter, respectively. Taking minimum entropy generation units,  $N_s$  as the selection criteria following parametric values are selected for GA, population size 40, crossover probability 0.4, mutation probability 0.01, and penalty parameter  $R1=500$ . Though it differs slightly for maximum  $f_{\max}$  or for maximum  $\epsilon$ .

The optimum solution using the above values of selected parameters is given in the Table 2.

The variation of heat duty generated in the solution space and the total entropy generation units,  $N_s$  with heat exchanger dimensions  $L_a$  and  $L_b$  keeping other parameters fixed at their optimum values, are given in Figure 3. For the specified heat duty i.e. 160 kW, corresponding  $N_s$  (a little higher than 0.063) is given by the curve AB (top right hand corner of Figure 3). This clearly agrees with the solution ( $L_a$ ,  $L_b$  and  $N_s$ ) obtained by GA in Table 2.

Next, an effort has been made to determine the optimum design due to imposition of an additional constraint along with those specified earlier. In practice a heat exchanger is to be designed for a given length restriction or total number of finned layers. Accordingly,  $L_a$ ,  $L_b$  or  $N_a$  is to be taken constant individually in the exercise for optimisation. Results of such exercises are shown in Figures 4 (a), (b) and (c) respectively. For example, keeping  $L_a$  fixed at different values, optimum solution for  $N_s$  and corresponding pressure drops have been calculated and depicted in Figure 4 (a). Thus these figures do not simply show the variation of  $N_s$  with  $L_a$ ,  $L_b$  and  $N_a$  respectively, they actually show how the optimum result is changing by enforcing an additional constraint. Again, the minimum value of  $N_s$  is coinciding with the similar parametric values of  $L_a$ ,  $L_b$  and  $N_a$  obtained at the overall optimum solution given in Table 2. These figures

also give additional information regarding the variation of pressure drop and hence the power requirement for both the fluids with the introduction an additional constraint. From all these figures it is obvious that introduction of additional constraint increases the irreversibility. It is also interesting to note that the optimum design is highly sensitive to some of these geometric parameters and a small deviation from the optimum value may give a large degradation in performance.

## **5. Conclusion**

A model for optimisation of crossflow plate-fin heat exchanger having large number of design variables of both discrete and continuous type has been developed using genetic algorithm. The case of multilayer plate-fin heat exchanger has been solved for minimum entropy generation units. The study shows the application and importance of design approach based on second law of thermodynamics and also the suitability of genetic algorithm for optimisation of such complex problems. The effect of some selected design variables on the optimum result, i.e. on irreversibilities associated and the pressure drops on the two sides, is anticipated. The result shows the effect of an additional constraint on the optimum solution and the corresponding power requirement in terms of pressure drops. The results can well be used for designers to start with or to have an initial guess.

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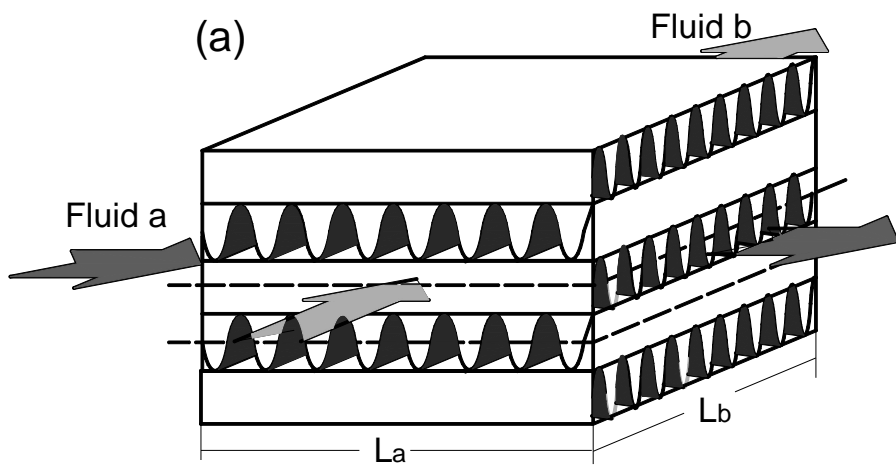
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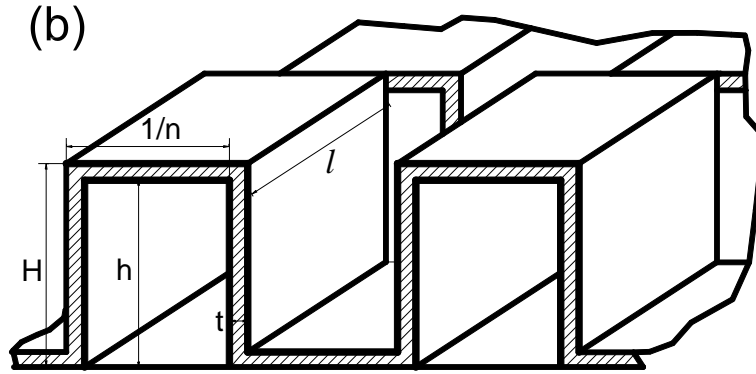
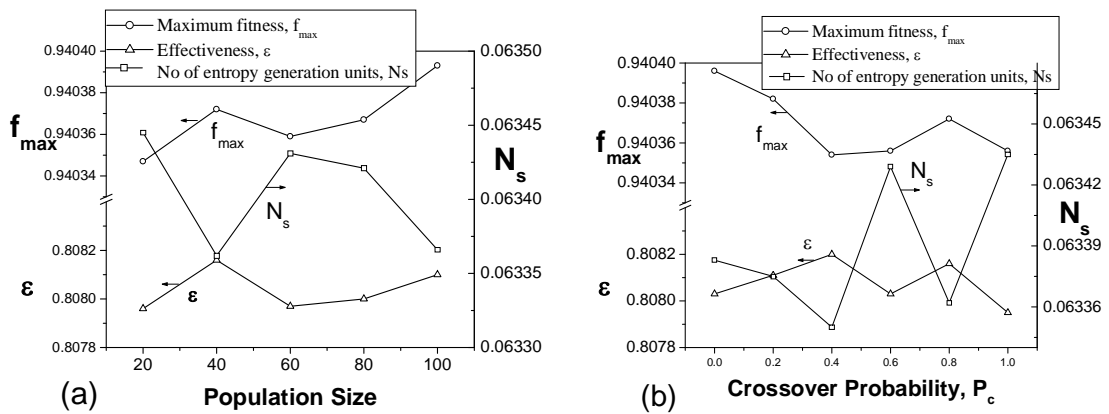


FIGURE 1. (a) Schematic representation of crossflow plate-fin heat exchanger, and (b) detailed view of offset-strip fin.



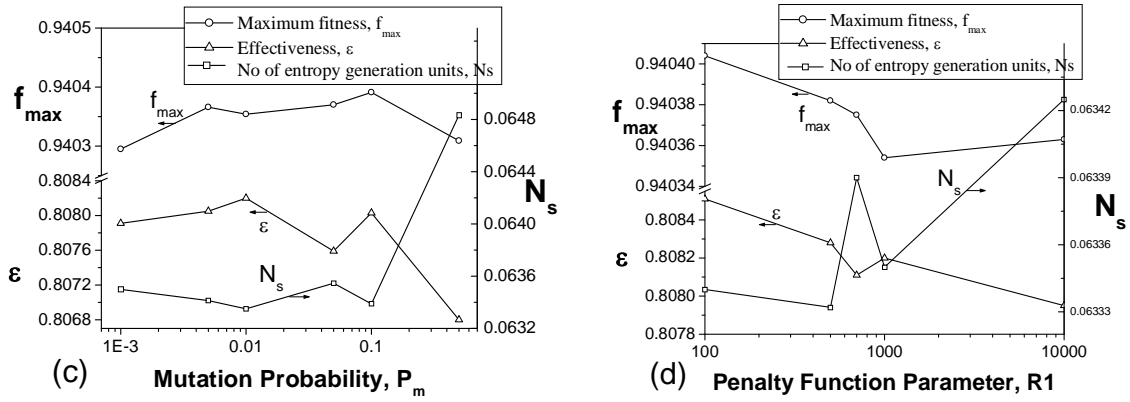


FIGURE 2. Effect of different GA parameters, (a) population (b) crossover probability (c) mutation probability, and (d) penalty parameters on maximum fitness and total annual cost.

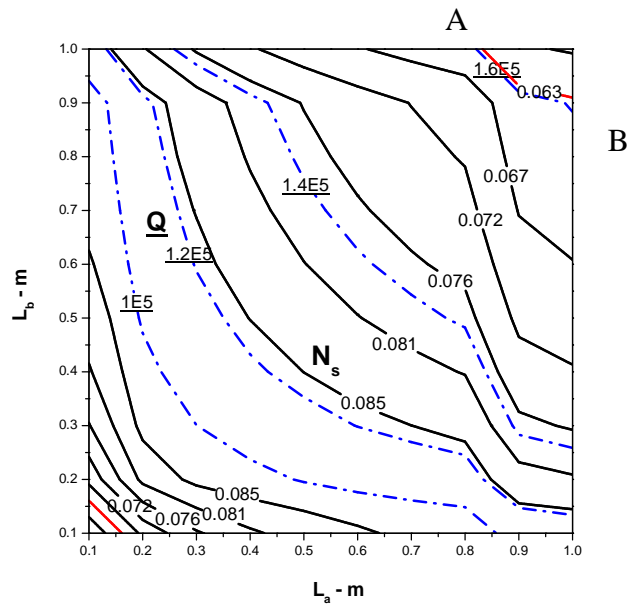
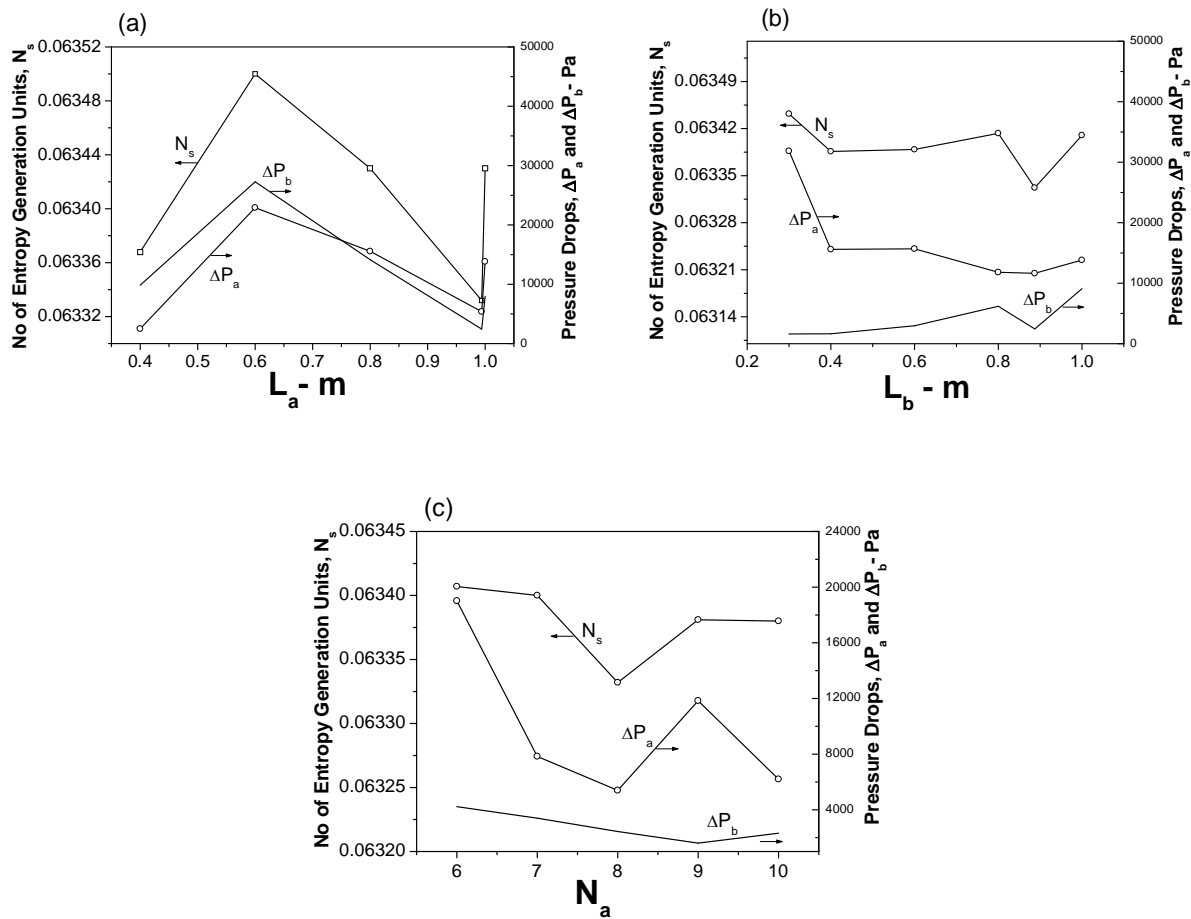


FIGURE 3. Contours for number of entropy generation units  $N_s$ , and heat duty  $Q$ , in the design space.



**FIGURE 4.** Effect of variation of (a)  $L_a$ , (b)  $L_b$  and (c)  $N_a$  on number of entropy generation units-  $N_s$  and pressure drops on the two sides.

### Figure captions

**Figure 1** (a) Schematic representation of crossflow plate-fin heat exchanger, and (b) detailed view of offset-strip fin.

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### **Table captions**

Table 1. Different operating parameters selected for the present example.

Table 2. The optimum solution using the selected values of parameters from Figure 2.



Table 1. Different operating parameters selected for the present example.

Parameters	Fluid a	Fluid b
Mass flow rate, $m$ ( $\text{kg s}^{-1}$ )	0.8962	0.8296
Inlet temperature, $T_1$ (K)	513	277
Inlet pressure, $P_1$ (Pa)	$10^5$	$10^5$
Specific heat, $C_p$ ( $\text{J kg}^{-1} \text{K}^{-1}$ )	1017.7	1011.8
Density, $\rho$ ( $\text{kg m}^{-3}$ )	0.8196	0.9385
Dynamic viscosity, $\mu$ ( $\text{N s m}^{-2}$ )	241.0	218.2
Prandtl number, $Pr$	0.6878	0.6954
Heat duty of the exchanger, $Q$ (kW)	160	

**Table 2.** The optimum solution using the selected values of parameters from Figure 2.

<b>L<sub>a</sub>, m</b>	<b>L<sub>b</sub>, m</b>	<b>H, mm</b>	<b>n, fins/m</b>	<b>t, mm</b>	<b>l, mm</b>	<b>N<sub>a</sub></b>	<b>N<sub>s</sub></b>	<b>Q, kW</b>
0.994	0.887	9.53	534.9	0.146	6.3	8	0.063332	159.99