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ORIGINAL ARTICLE

CROSS FLOW PLATE FIN HEAT EXCHANGER ENTROPY GENERATION MINIMIZATION USING PARTICLE SWARM OPTIMIZATION ALGORITHM

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ABSTRACT: In the present paper parallel particle swarm optimization algorithm is used for optimization of a cross flow plate fin heat exchanger with consideration of entropy generation. A total number of seven geometrical design parameters, namely, hot fluid length, cold fluid length, length of the fin, height of the fin, frequency of the fin, thickness of the fin, and number of layers of hot side, are considered as the optimization variables. The accuracy of the algorithm is verified via an applied example. After the verification the computer code is implemented for minimization of entropy generation in the plate fin heat exchanger.

KEYWORDS: Optimization, Plate Fin Heat Exchanger, Entropy Generation, Efficiency.

INTRODUCTION

An efficient heat transfer between process fluids is an important step in most of thermo-fluid processes such as industrial power generation plants, chemical, petrochemical, and petroleum industries. In the cases which both of the process fluids are gas, cross flow plate fin heat exchangers, which are a typical compact heat exchanger, are widely used because of their low weight and volume and high efficiency (Shah and Sekulic, 2003). The traditional time-consuming scheme of design for heat exchangers involves selection of a large number geometrical parameters and survey that those satisfy a given objective (rate of heat transfer, pressure drop, heat transfer area, etc.). This cumbersome approach does not guarantee an optimum solution. There are some earlier works in which the traditional methods have been used to optimize plate fin heat exchangers (Reneaume *et al.*, 2000; Muralikrishna and Shenoy, 2000). As a newer and more effective tool, genetic algorithm, which is stochastic global search algorithms, has been widely employed in optimization of plate fin heat exchangers. Some works include structure sizes optimization (Xie *et al.*, 2008), second low based optimization (Mishra *et al.*, 2009), Thermal-economic optimization (Sanaye and Hajabdollahi, 2010), cast minimization (Ahmadi *et al.*, 2011), and stacking pattern optimization (Ghosh *et al.*, 2011) of plate fin heat exchangers using genetic algorithm. Recently, Particle Swarm Optimization (PSO), which is a population-based search algorithm (Engelbrecht, 2007), has been used to optimize heat exchangers problems such as optimization of heat exchangers networks (Silva *et al.*, 2008; Silva *et al.*, 2009) and optimization of shell and tube heat exchangers (Patel and Rao, 2010). Also there are very few recently published studies on optimization of plate fin heat exchangers. Rao and Patel (Rao and Patel, 2010) employed the particle swarm optimization algorithm for thermodynamic optimization of a cross flow plate fin heat exchanger. Yousefi *et al.*, (2012) explored application of a Genetic Algorithm hybrid with Particle Swarm Optimization for design optimization of a plate fin heat exchanger. In PSO any individuals, which are named particles, move towards optimal solution. The present paper, aims to employ the PSO algorithm for entropy generation minimization in cross flow plate fin heat exchangers.

OPTIMIZATION METHODOLOGIES

2.1. Particle swarm optimization algorithm

The particle swarm optimization algorithm is a population-based search algorithm based on the simulation of the social behavior of birds within a flock. In PSO each particle is an individual in the search space. These particles are distributed through the search space. In an *n*-dimensional space the position and velocity of each particle is shown by $X_i = (x_{i1}, x_{i2}, \ldots, x_{in})$ and $V_i = (v_{i1}, v_{i2}, \ldots, v_{in})$, respectively. The position and velocity of each particle within the search is calculated based on the information obtained by the particle and the neighborhood particles. Therefore the variation in position of each particle depends on own information and those of others. The fitness function, *f(t)*, measures

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how close the corresponding solution is to the optimum or quantifies the solution of a particle. Updating velocity of particle *i* is according to the following equation:

$$V_{ij}(\tau+1)=WV_{ij}(\tau)+c_1r_1(\tau)\big(pbest_{ij}-x_{ij}(\tau)\big) \\ -c_2r_2(\tau)\big(gbest_j-x_{ij}(\tau)\big),\ j=1,2,\ldots,n \qquad (1)$$

Where x_{ij} is the position of particle *i* in dimension *j*, the coefficient r_1 and r_2 are random numbers between 0 and 1, c_1 and c_2 are weighting number between 0 and 2, and *W* is an arbitrary weighting coefficient in the rage of 0 to 1. The personal best position, *pbest*, associated with particle *i* is the best position the particle has visited since the first time step. Also the global best position, *gbest*, is best position visited in the search space since the first time step. After updating the velocity of particles, their new positions are obtained according to the following:

$$X_{ij}(\tau+1)=x_{ij}(\tau)+V_{ij}(\tau+1),\ j=1,2,\ldots,n \qquad (2)$$

When a desirable value for *gbest* is obtained or maximum iteration is done, the search process is finished. The flowchart of PSO for an optimization problem is shown in Figure 1.

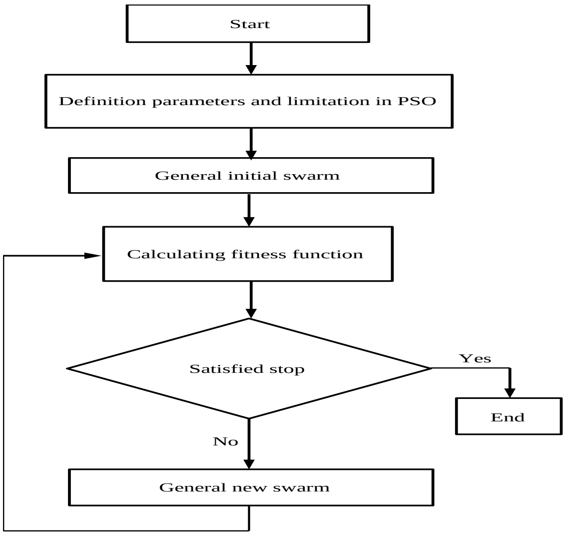

Figure 1: Flowchart of particle swarm optimization algorithm

THERMAL MODELING AND ENTROPY GENERATION IN PLATE FIN HEAT EXCHANGERS

The cross flow plate fin heat exchanger with rectangular offset strip fins considered in the present study is shown in Figure 2. The variation of thermophysical properties of fluids with temperature is neglected and the fluids are considered as ideal gas. It is assumed that the heat exchanger works under steady-state conditions and the thermal resistance of the walls and influence of fouling is neglected. Also the heat transfer coefficient and the area of distribution are uniform and constant. Moreover to avoid heat waste to the ambient, number of fin layers of the cold fluid is considered one more than that of the hot fluid.

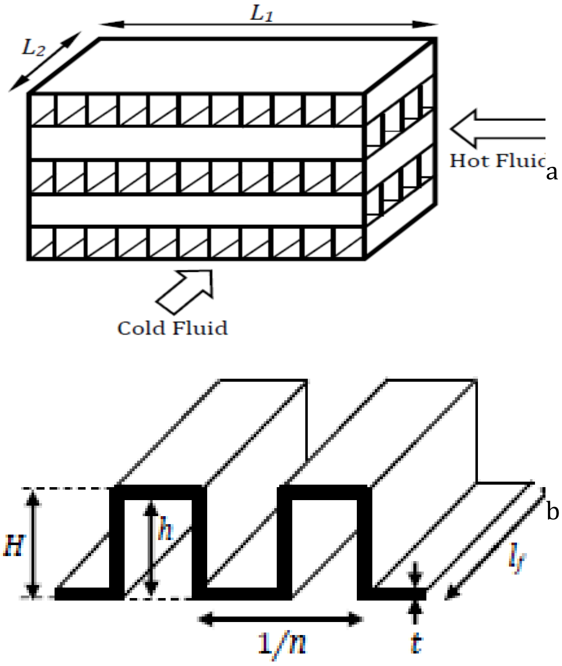

Figure 2: Diagram of typical cross flow plate fin heat exchanger (a) and offset strip fin (b)

The entropy generation is caused by finite temperature difference and fluid friction which can be written as follows:

$$\begin{aligned}\dot{S}&=\dot{S}_{\Delta T}+\dot{S}_{\Delta P}\\ &=\int_i^o\left(\frac{\dot{m}c_p\,dT}{T}\right)_{1,2}+\left(\dot{m}\frac{\Delta P}{\rho}\frac{\ln\frac{T_o}{T_i}}{T_o-T_i}\right)_{1,2}\\ &=\left(\dot{m}c_p\right)_1\ln\frac{T_{o,1}}{T_{i,1}}+\left(\dot{m}c_p\right)_2\ln\frac{T_{o,2}}{T_{i,2}}+\\ &\dot{m}_1\frac{\Delta P_1}{\rho_1}\frac{\ln\frac{T_{o,1}}{T_{i,1}}}{T_{o,1}-T_{i,1}}+\dot{m}_2\frac{\Delta P_2}{\rho_2}\frac{\ln\frac{T_{o,2}}{T_{i,2}}}{T_{o,2}-T_{i,2}}\end{aligned} \qquad (2)$$

The outlet temperature of the fluid is obtained by definition of heat exchanger effectiveness as following:

$$\varepsilon = \frac{C_1\left(T_{i,1}-T_{o,1}\right)}{C_{min}\left(T_{i,1}-T_{i,2}\right)} = \frac{C_2\left(T_{o,2}-T_{i,2}\right)}{C_{min}\left(T_{i,1}-T_{i,2}\right)} \tag{3}$$

$$T_{o,1} = T_{i,1} - \varepsilon\frac{C_{min}}{C_1}\left(T_{i,1}-T_{i,2}\right) \tag{4}$$

$$T_{o,2} = T_{i,2} + \varepsilon\frac{C_{min}}{C_2}\left(T_{i,1}-T_{i,2}\right) \tag{5}$$

Where, $C_1 = \left(\dot{m}c_p\right)_1$, $C_2 = \left(\dot{m}c_p\right)_2$, and $C_{min} = min\left\{\left(\dot{m}c_p\right)_1, \left(\dot{m}c_p\right)_2\right\}$. Number of entropy generation unites is defined as follows:

$$N_S = \frac{\dot{S}}{C_{max}} \tag{6}$$

Substituting Eqs. 2, 4, and 5 in Eq. 8 yields:

$$\begin{aligned} N_S = {} & \frac{C_1}{C_{max}}\left\{\ln\left(T_{i,1}-\varepsilon\frac{C_{min}}{C_1}\left(T_{i,1}-T_{i,2}\right)\right)-\ln T_{i,1}\right\}\times \\ & \left[1+\frac{\Delta P_1}{c_{p1}\,\rho_1}\frac{1}{T_{i,1}-\varepsilon\frac{C_{min}}{C_1}\left(T_{i,1}-T_{i,2}\right)-T_{i,1}}\right]+ \\ & \frac{C_2}{C_{max}}\left\{\ln\left(T_{i,2}+\varepsilon\frac{C_{min}}{C_2}\left(T_{i,1}-T_{i,2}\right)\right)-\ln T_{i,2}\right\}\times \\ & \left[1+\frac{\Delta P_2}{c_{p2}\,\rho_2}\frac{1}{T_{i,2}+\varepsilon\frac{C_{min}}{C_2}\left(T_{i,1}-T_{i,2}\right)-T_{i,2}}\right] \end{aligned} \tag{7}$$

For cross flow plate fin heat exchangers the effectiveness is (Shah and Sekulic, 2003):

$$\varepsilon = 1 - exp\left[\frac{1}{C^*}NTU^{0.22}\left\{exp\left(-C^*NTU^{0.78}\right)-1\right\}\right] \tag{8}$$

Where,

$$C^* = \frac{C_{min}}{C_{max}} \tag{9}$$

and *NTU* is number of transfer unit and is given by:

$$\frac{1}{NTU} = \frac{C_{min}}{AU} = C_{min}\left(\frac{1}{(hA)_1}+\frac{1}{(hA)_2}\right) \tag{10}$$

In the above equation the heat transfer coefficient, *h*, is calculated from:

$$h = jc_pG\,Pr^{-\frac{2}{3}} \tag{11}$$

Where, the Colburn coefficient, *j*, is calculated by the correlation reported by Manglik and Bergles, (1995):

$$\begin{aligned} & j = 0.6522(Re)^{-0.5403}(\alpha)^{-0.1541}(\delta)^{0.1499}(\gamma)^{-0.0678}\times \\ & \left[1+5.3\times10^{-5}(Re)^{1.34}(\alpha)^{0.504}(\delta)^{0.456}(\gamma)^{-1.055}\right]^{0.1} \end{aligned} \tag{12}$$

The following parameters are used in the above equation:

$$\alpha = \frac{s}{H-t},\ \delta = \frac{t}{l},\ \gamma = \frac{t}{s},\ s = \frac{1}{n}-t \tag{13}$$

Also the mass flux, *G*, used in Eq. 11 is:

$$G = \frac{m}{A_{ff}} \tag{14}$$

Where, A_{ff} is free flow cross sectional area and is calculated from:

$$A_{ff,1} = \left(H_1-t_1\right)\left(1-n_1t_1\right)L_2N_1 \tag{15}$$

$$A_{ff,2} = \left(H_2-t_2\right)\left(1-n_2t_2\right)L_1N_2 \tag{16}$$

The heat transfer area for both sides of the heat exchanger is as following:

$$A_1 = L_1L_2N_1\left[1+\left\{2n_1\left(H_1-t_1\right)\right\}\right] \tag{17}$$

$$A_2 = L_1L_2N_2\left[1+\left\{2n_2\left(H_2-t_2\right)\right\}\right] \tag{18}$$

Also the Reynolds number, *Re*, is defined by:

$$Re = \frac{GD_h}{\mu} = \frac{mD_h}{A_{ff}\,\mu} \tag{19}$$

Where, D_h is hydraulic diameter and is defined as:

$$D_h = \frac{4s(H-t)l}{2\left(sl+\left(H-t\right)l+t(H-t)\right)+ts} \tag{20}$$

The total rate of heat transfer through the heat exchanger is obtained from:

$$Q = \varepsilon C_{min}(T_{i,1}-T_{i,2}) \tag{21}$$

The frictional pressure drop for both sides of the heat exchanger is given by:

$$\Delta P_1 = \frac{2f_1L_1G_1^2}{\rho_1D_{h,1}} \tag{22}$$

$$\Delta P_2 = \frac{2f_2L_2G_2^2}{\rho_2D_{h,2}} \tag{23}$$

Where fanning factor, *f*, for offset strip fin is obtained using the correlation proposed by Manglik and Bergles, (1995):

$$\begin{aligned} & f=9.6243(Re)- \\ & 0.7422(\alpha)^{-0.1856}(\delta)^{0.3053}(\gamma)^{-0.2659}\times \\ & \left[1+7.669\times10^{-8}(Re)^{4.429}(\alpha)^{0.920}(\delta)^{3.767}(\gamma)^{0.236}\right]^{0.1} \end{aligned} \qquad (24)$$

RESULTS AND DISCUSSION

In the present study, number of entropy generation unites, N_S, which is introduced in Eq. 7 is considered as the fitness function. The problem considered in the present paper is gas to air single pass cross flow plate fin heat exchanger which is needed to be designed for the minimum entropy generation. The operating conditions for the problem are listed in Table 1.

Table 1: Operating conditions for the present study

Parameters	Hot side	Cold side
Mass flow rate, m (kg/s)	1.66	2
Inlet temperature, (°C)	900	200
Specific heat, c_p (J/kg K)	1122	1073
Density, ρ (kg/m³)	0.6296	0.9638
Dynamic viscosity, μ (Ns/m²)	401×10^{-7}	336×10^{-7}
Prandtl number, *Pr*	0.731	0.694

In the present paper, seven parameters namely, hot flow length (L_1), cold flow length (L_2), number of fin layer of hot side (N_1), frequency of the fin (*n*), thickness of the fin (*t*), height (*H*) and length (l_f) of the fin are considered as optimization variables (dimensions of the search space). As mentioned before, number of fin layers of the cold fluid is considered one more than that of the hot fluid. It should be noted that all variables with the exception of number of fin layer of hot side are continuous. Ranges of the variables are listed in Table 2. The Additional constraint is required heat transfer.

Table 2: Range of design parameters

Parameters	Min	Max
Hot fluid length, L_1 (m)	0.1	1
Cold fluid length, L_2 (m)	0.1	1
Length of the fin, L_f (mm)	1	10
Height of the fin, *H* (mm)	2	10
Frequency of the fin, *n* (m⁻¹)	100	1000
Thickness of the fin, *t*	0.1	0.2
Number of layers of hot side, N_1	1	200

To verify the developed code, one application examples is considered which was solved previously in literature. This applied example is a gas to air cross flow plat fin heat exchangers with heat duty of 1069.8 KW which is designed for minimum heat transfer area. The obtained results for this example compared with the results of (Yousefi *et al.*, 2012) in Tables 3. It is observed from the table that the obtained results are acceptable.

Table 3: Comparison between the results of present study and those of Yousefi *et al.*, (2012) for minimum heat transfer area

Parameters	Yousefi *et al.*, (2012)	Present study
Hot fluid length, L_1 (m)	0.21	0.22
Cold fluid length, L_2 (m)	0.23	0.23
Length of the fin, L_f (mm)	2.1	2.5
Height of the fin, *H* (mm)	5.9	6.3
Frequency of the fin, *n* (m⁻¹)	1000	921
Thickness of the fin, *t*	0.1	0.1
Number of layers of hot side, N_1	91	83
Heat transfer area (m²)	112.69	109.7

After the applied example, the problem of minimization of entropy generation in the plate fin heat exchanger is studied. Figure 3 depicts the iteration process of particle swarm optimization algorithm. A remarkable reduction in fitness function is observed in the beginning of the iteration process. Moreover after the iteration 18 the change of fitness function become small.

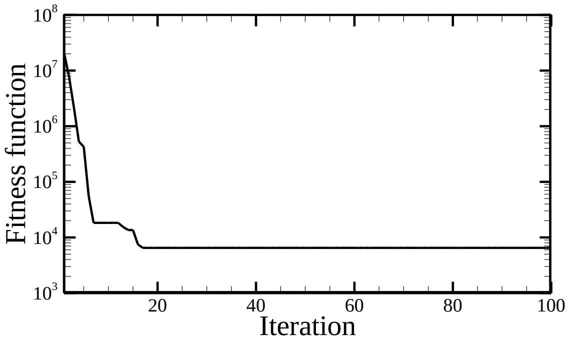

Figure 3: Variation of fitness function with iterations via the particle swarm optimization

The results obtained via the particle swarm optimization are shown in Table 4. It is observed that the length of hot and cold streams approach to their maximum values. A moderate value for fin frequency is obtained while the number of layers of hot side is relatively small.

Table 4: Geometrical parameters obtained by the particle swarm optimization algorithm

Parameters	Min
Hot fluid length, L_1 (m)	0.996
Cold fluid length, L_2 (m)	0.997
Length of the fin, L_f (mm)	4.841
Height of the fin, *H* (mm)	6.593
Frequency of the fin, *n* (m⁻¹)	448.035
Thickness of the fin, *t*	0.123
Number of layers of hot side, N_1	23
Heat Transfer rate (W)	1278419
Number of entropy generation unite	6335.648

CONCLUSION

In the present study a computer program for particle swarm optimization is developed. After verification, the developed code is used for minimization of entropy generation in a cross for plate fin heat exchanger. The obtained results indicate that the proposed version of PSO can converge to the optimum results with an acceptable accuracy. Also the obtained result show that for minimization of the entropy generation the length of the heat exchanger for both hot and cold streams should be near the maximum possible value while the number of layers of heat exchanger has a small value. Also geometrical parameters of the fin are near the moderate possible values.

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