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Design Optimisation of Counter Flow Plate-Fin Heat Exchanger for Minimum Total Annual Cost Using Graphical Technique

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ABSTRACT

This paper presents the minimisation of total annual cost of plate-fin heat exchanger, which comes under the family of compact heat exchanger. Optimisation of objective function is carried out with the application of graphical technique. The minimum value of total annual cost obtained for the feasible design region leads to the optimum solution of the problem. The design variables considered in this paper is heat exchanger lengths as well number of fin layer of heat exchanger.

Keywords: Graphical technique, heat duty, optimisation, Plate-fin heat exchanger, , pressure drop, total annual cost.

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1. INTRODUCTION

Optimization of heat exchangers owing to their vital role in various industries has attracted lots of interests all over the world. Several researches have been performed in this area for various types of heat exchangers, considering different objective functions and design parameters. Since PFHEs design deals with several parameters and nonlinear equations, optimization of these systems faces with rather high complexity. Pressure drops and heat transfer rates are interdependent quantities and both of them essentially influence the capital and operating costs of any heat exchanging system.

2. EARLY WORK OF DESIGN OPTIMIZATION FOR MINIMUM TOTAL ANNUAL COST OF PLATE FIN HEAT EXCHANGER

The objective function minimizes the total annual cost (TAC), which includes the utility consumption cost, the capital investment for the heat exchangers and pumping devices, and the pumping power cost. Non-linear fixed-charge cost models are used for heat exchanger and pump investment costs. Reneaume et al. (2000) proposed a tool for the optimal design of compact heat exchangers. Mathematical programming techniques are integrated in the COLETH program. The resulting program allows optimization of the fins (height, thickness etc.), the core (width) and the distributors (widths). Three test examples are presented. The program abilities are illustrated a 10% capital cost reduction is achieved. Soylemez (2000) perform practical P_1-P_2 method for economical optimisation of parallel and counter flow heat exchanger sizing for heat recovery. Result reveals that the effectiveness of the heat exchanger increases as the area of the exchanger increases. Wang and Sunden (2003) carried out optimal design of plate heat exchanger which consists of two categories the design with fixed allowable pressure drops and the complete optimal design without pressure drop specifications. Four examples are used to demonstrate the proposed method. Crane (2004) shows in his study that a net power output of 1 kW can be achieved for a modestly sized heat exchanger core such that the net power density based on heat exchanger volume is approx.45 kW/m^3 for thermoelectric waste heat recovery. Gebreslassie et al (2010) present thermoeconomic optimization based on the structural method. Objective function is minimum annual total cost. Sanaye and Hajabdollahi (2010) present thermal modelling and optimal design of compact heat exchangers. ε-NTU method was applied to case study and estimate the heat exchanger pressure drop and effectiveness. Fast and elitist non-dominated sorting genetic-algorithm (NSGA-II) was applied to obtain the maximum effectiveness and the minimum total annual cost (sum of investment and operation costs) as two objective functions. Teke et al. (2010) use a non dimensional number E which is based on technical and economical parameter. Net gain in term of cost is calculated for parallel, counter and cross flow taking effectiveness as one criteria, for determining the best type of heat exchangers for heat recovery. Ahmadi et al.(2011) designed PFHE using multi-objective genetic algorithm optimization technique. Number of entropy generation units and total annual cost were considered as the two objective functions. The results depict that higher exergy efficiency leads to have efficient heat exchangers in thermodynamic as well as thermo-economic points of view.

The results of exergy destruction showed that by decrease in exergy destruction, the total annual cost increases, respectively. Also, a set of Pareto optimal front points was presented. Najafi et al. (2011) presented a multi objective genetic algorithm optimization of plate and fin heat exchanger (PFHE) taking both side fluid as air. Two different objective functions including the total rate of heat transfer and the total annual cost of the system are simultaneously optimized to achieve a set of optimal solutions. Several geometric variables including the total length of the hot and cold side of the heat exchanger, fin height, fin frequency, lance length of the fin, fin thickness and the number of fin layers are considered as optimization parameters. A sensitivity analysis is carried out in order to investigate the effect of some geometric variables on each objective function Ghazi et al (2012) used genetic algorithm for thermo-economic optimization to obtain the optimum values of design parameters for a dual pressure Heat Recovery Steam Generator with deaerator evaporator applied in combined cycle power plants. They found that at higher inlet gas enthalpy, the required heat transfer surface area (and its corresponding capital cost) increases Rao and Patel (2013) compared total annual cost optimisation of heat exchanger with GA multi-objective optimization and modified teaching-learning-based optimization algorithm. Plate-fin heat exchanger and shell and tube heat exchanger are considered for the ptimization. Application example of Sanaye, S, and Hajabdollahi, H. (2010) was considered for plate fin heat exchanger optimisation.

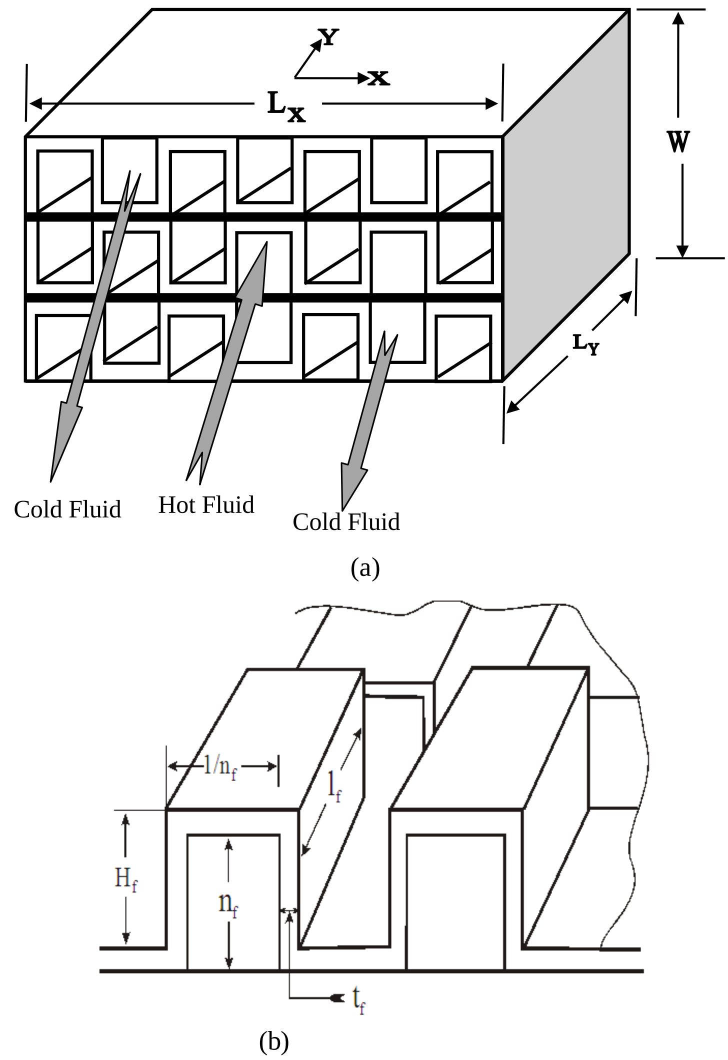

Fig.1: Simplified diagram of (a) Counter flow plate-fin heat exchanger, (b) offset strip fin

3. MATHEMATICAL MODELLING

For counter flow plate fin heat exchanger following assumptions are made for analysis and thermodynamic optimization:

1. The Steady state condition established in heat exchanging process.
2. Mass flow rate of fluids are constant.
3. The heat energy lost due to radiation is ignored.
4. Offset strip fins with same specifications are used for two fluid flow side of the exchanger.
5. Same specifications of offset strip fins are used in hot as well as cold side fluid.
6. Behaviour of two fluids during process is as ideal fluid.
7. The variation of property of two fluids along the temperature is ignored.
8. When the design consists of more than two layers of finned passages, number of fin layer for cold fluid b is assumed to be one more than that of hot fluid 'a' ($N_{L,b} = N_{L,a} + 1$).

Analytical Expression

$$A_{ff_a} = (H_{f,a} - t_{f,a})(1 - n_{f,a}t_{f,a})L_x N_{L,a} \quad (1)$$

$$A_{ff_b} = (H_{f,b} - t_{f,b})(1 - n_{f,b}t_{f,b})L_x N_{L,b} \quad (2)$$

$$A_a = L_x L_y N_{L,a}[1 + 2n_{f,a}(H_{f,a} - t_{f,a})] \quad (3)$$

$$A_b = L_x L_y N_{L,b}[1 + 2n_{f,b}(H_{f,b} - t_{f,b})] \quad (4)$$

Then total heat transfer area can be given as

$$A = A_x = A_a + A_b \quad (5)$$

The hydraulic diameter for given fin geometry can be calculated as follow

$$D_h = \frac{2(s_f - t_f)(H_f - t_f)}{\{s_f + (H_f - t_f)\} + \frac{(H_f - t_f)t_f}{l_f}} \quad (6)$$

Where

$$s_f = \left(1/n_f - t_f\right) \quad (7)$$

$$h_f = H_f - t_f \quad (8)$$

Mass flux velocity G

$$G = \frac{m}{A_{ff}} \quad (9)$$

$$Re = \frac{GD_h}{\mu} \quad (10)$$

Heat transfer coefficient computed in term of Colburn j factor by

$$j = \frac{h}{GC_P}(P_r)^{2/3} \quad (11)$$

Assuming critical Reynolds number to be 1500, the characteristics 'j' and 'f' of the offset-strip fins are given as follows (Joshi and Webb [1987]
for laminar flow (Re ≤ 1500)

$$j = 0.53(Re)^{-0.5}\left(l_f/D_h\right)^{-0.15}\{s_f / H_f - t_f\}^{-0.14} \quad (12)$$

$$f = 8.12(Re)^{-0.74}\left(l_f/D_h\right)^{-0.41}\{s_f / H_f - t_f\}^{-0.02} \quad (13)$$

For turbulent flow (Re ≥ 1500)

$$j = 0.21(Re)^{-0.4}\left(l_f/D_h\right)^{-0.24}\{t_f / D_h\}^{0.02} \quad (14)$$

$$f = 1.12(Re)^{-0.36}\left(l_f/D_h\right)^{-0.65}\{t_f / D_h\}^{0.17} \quad (15)$$

$$h = 0.21(m)^{0.6}(D_h)^{-0.18}(h_f)^{-0.6}(1 - n_f t_f)^{-0.6}(L_X)^{-0.6}(N_L)^{-0.6}(\mu)^{0.4}(l_f)^{-0.24}(t_f)^{0.02}C_P(P_r)^{-2/3} \quad (16)$$

$$\frac{1}{UA} = \frac{1}{(hA)_a} + \frac{1}{(hA)_b} \quad (17)$$

$$\mathrm{LMTD} = \frac{(T_{a\,1}-T_{b\,2})-(T_{a\,2}-T_{b\,1})}{\ln\left\{\frac{(T_{a\,1}-T_{b\,2})}{(T_{a\,2}-T_{b\,1})}\right\}} \tag{18}$$

$$Q = UA\,LMTD \tag{19}$$

$$Q = 7.65451 * (10)^5 (L_X)^{0.4} L_y \left[\left(N_{L,a}\right)^{-0.4} + 1.136638\left(N_{L,a}+1\right)^{-0.4}\right]^{-1} \tag{20}$$

$$\Delta P_a = \frac{4 f_a L_X G_a^2}{2\rho_a D_{h\,a}} \tag{21}$$

$$\Delta P_b = \frac{4 f_b L_X G_b^2}{2\rho_b D_{h\,b}} \tag{22}$$

$$\Delta P_a = 132986.44 L_y (L_X)^{-1.64} \left(N_{L,a}\right)^{-1.64} \tag{23}$$

$$\Delta P_b = 91185.01 L_y (L_X)^{-1.64} \left(N_{L,a}+1\right)^{-1.64} \tag{24}$$

Design for Minimum Total Annual Cost

The method of defining the total annual cost may vary depending upon the application. However, it should comprise of the initial cost of the equipments namely the heat exchanger and the prime movers for the fluid streams and the running cost of these equipments. Cost of both the heat exchanger and the prime movers will have a fixed and a variable component as $Z=kA+k_o$ (Zubair et. al., 1987). The variable component (kA) for the heat exchanger may be assumed to depend upon the total heat transfer area as the type of heat transfer surface has been specified. In case of prime movers initial as well as variable component of it depends upon the power consumed by them. Such basis for cost estimation has also been taken by Muralikrishna and Shenoy (2000).

Total annual cost TAC = Initial cost of (heat exchanger core + pump a + pump b) + operating cost of (pump a + pump b) (25)

Thus total annual cost TAC is given by (Muralikrishna and Shenoy, 2000)

$$\mathrm{TAC} = Af\left\{C_a + C_b (A_x)^{C_x} + C_c + C_f\left(\frac{m_a \Delta P_a}{\rho_a}\right)^r + C_c + \left(\frac{m_b \Delta P_b}{\rho_b}\right)^r\right\} + \frac{C_{pow}\left(\mathrm{time}/\mathrm{year}\right)}{\eta_{pump}}\left[\frac{m_a \Delta P_a}{\rho_a} + \frac{m_b \Delta P_b}{\rho_b}\right] \tag{26}$$

$$\begin{aligned}\mathrm{TAC} = {} & 10948 + 1974.5335 (L_X)^{0.81} \left(L_y\right)^{0.81} \left(2N_{L,a}+1\right)^{0.81} + 4495.282\left(L_y\right)^{0.68} (L_X)^{-1.1152} \left(N_{L,a}\right)^{-1.1152} \\ & + 2856.4792\left(L_y\right)^{0.68} (L_X)^{-1.1152} \left(N_{L,a}+1\right)^{-1.1152} + 66578.437 L_y (L_X)^{-1.64} \left(N_{L,a}\right)^{-1.64} \\ & + 34269.737 L_y (L_X)^{-1.64} \left(N_{L,a}+1\right)^{-1.64}\end{aligned} \tag{27}$$

The objective functions and the range of different design variables with upper and lower bounds are as follows:

$$\text{Minimise } f(x) = \mathrm{TAC} \tag{28}$$

For number of hot layer limits are

$$5 \le N_{L,a} \le 35 \tag{29}$$

For lengths of heat exchanger limits are

$$0.1 \le L_X \le 0.29 \tag{30}$$

$$0.0855 \le L_y \le 0.115 \tag{31}$$

4. CALCULATION PROCEDURE

Total annual cost and heat duty are functions of $N_{L,a}$, L_x and L_y. Using equation (20) and (27) and eliminating L_y, TAC can be made a function of only two parameters $N_{L,a}$ and L_x for a specified heat duty, and corresponding pressure drops (ΔP_a and ΔP_b) can be obtained [equation (23) and (24)]. Further, for a particular value of TAC, different combinations of $N_{L,a}$ and L_x can be assumed. For each combination, pressure drops can be calculated for the two fluids and plotted on the pressure drop diagram (ΔP_a Vs ΔP_b), which leads to equi-TAC curve. The minimum value of TAC obtained for the feasible design region leads to the optimum solution of the problem.

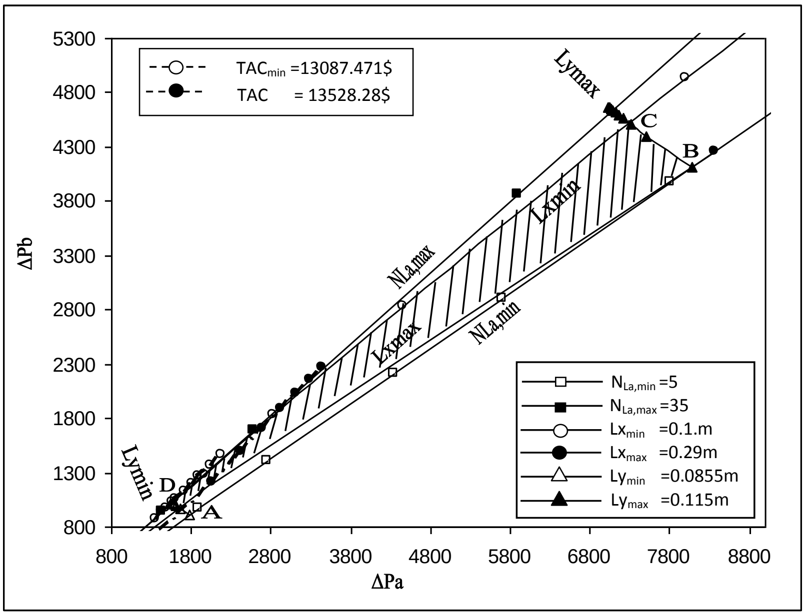

Fig. 2: Feasible design space and optimum solution corresponding to minimum total annual cost (TAC).

Figure 2 shows a feasible design space ABCD satisfying different design constraints and the variation of iso-TAC curve. The optimum solution for minimum TAC is obtained at point D and is given in table-1.

Table 1: Optimum solution for minimum total annual cost (TAC).

$N_{L,a}$	$N_{L,b}$	Lx m	Ly m	W m	ΔP_a Pa	ΔP_b Pa	TAC_{min} $
33	34	0.1015	0.0855	0.7138	1565	1021.77	13087.47

5. EXAMPLE APPLICATION

Numerical example selected for the present problem is as given under.

A counter flow plate-fin heat exchanger has been selected for cooling of hot air. Hot air has mass flow rate of 2070 kg/hr and a temperature drop from 581.26 K to 497.7 K. The cold air has mass flow rate 1962 kg/hr with an inlet temperature of 380.6 K.

The heat exchanger has to be designed based on minimum total annual cost.
The characteristics of the fins for both sides of the fluids are as under –

Fin pitch = 598.4 fins/m
Plate spacing = 10.5 mm
Lance length of fin = 3.175 mm
Fin thickness = 0.152 mm

Following data have been considered for the analysis –

Cp_a	= 1040.97 J/kg K	Cp_b	= 1010.4 J/kg K
C_{max}	= C_a = 598.55 J/K	$C_{min} = C_b$	= 550.66 J/K
m_a	= 0.575 kg/s	m_b	= 0.545 kg/s
$P_{a,in}$	= 160.5 kPa	$P_{b,in}$	= 160.5 kPa
Pr_a	= 0.676	Pr_b	= 0.682
R_a	= 287 J/kg K	R_b	= 287 J/kg K
ρ_a	= 0.6545 kg/m^3	ρ_b	= 0.8286 kg/m^3
υ_a	= 43.11 x 10^{-6} m^2/s	υ_b	= 29.28 x 10^{-6} m^2/s
μ_a	= 28.13 x 10^{-6} N-s/m^2	μ_b	= 24.24 x 10^{-6} N-s/m^2

$T_{b,out}$ = 471.4 K

LMTD = 113.46°C
Af = 0.332
C_a = 30000
C_b = 750
C_c = 2000
C_f = 5
C_{pow} = 0.00005 $/kW-Hr
C_x = 0.81
N = 10 year
r = 0.68
operational Time/Year = 8000 hrs.
η_{pump} = 0.7

6. NOMENCLATURE

A	heat transfer area, m^2
Af	annualisation factor for capital cost, $
A_{ff}	free flow area, m^2
A_x	total heat transfer area, m^2
C	heat capacity rate (mC_p), J/K
C_a, C_b	cost coefficients, eq. (26)
C_c, C_f	cost coefficients, eq. (26)
C_p	specific heat of fluid, J/kg.K
C_{pow}	cost of power, $/kW-Hr
C_r	C_{min} / C_{max}
C_x	cost coefficient (exponent), eq. (26)
D_h	hydraulic diameter, m
f	fanning friction factor
G	mass flux velocity (= m/A_{ff}), Kg/m^2s)
H_f	plate spacing of fin, m
h	convective heat transfer coefficient, (W/m^2K)
h_f	fin height (= H_f - t_f), m
j	Colburn factor [= St $(Pr)^{2/3}$]
k	thermal conductivity, (W/mK)
L_x	heat exchanger length in X direction, m
L_y	heat exchanger length in Y direction, m
l_f	lance length of fin, m
LMTD	logarithmic mean temperature difference, °C
m	mass flow rate of fluid, λg/s
N	number of years of operation
N_L	number of fin layers
n_f	fin frequency, fins per meter
P	pressure, N/m^2
ΔP	Pressure drop, N/m^2
Pr	Prandtl number
Q	rate of heat transfer, W
R	specific gas constant, J/kg. K
r	cost coefficient (exponent), eq. (4.27)
Re	Reynolds number
St	Stanton number [= h / GCp]
S_f	fin spacing , m
T	temperature, K
To	ambient temperature, K
t_f	fin thickness, m
TAC	total annual cost, $

Time/Year	yearly operational time, hours
U	overall heat transfer coefficient, (W/m^2K)
a	hot fluid
b	cold fluid
max	maximum
min	minimum
1	inlet
2	outlet
W	width of heat exchanger, m
ε	effectiveness of heat exchanger
ρ	density, Kg/m^3
μ	viscosity, $N\text{-}s/m^2$
η_{pump}	efficiency of pump

Subscript:

CONCLUSIONS

The present work establishes the capability of graphical technique to find optimal solution of multi-variable, non linear complex optimisation problem. In this paper optimum result of design variables for minimum total annual cost as objective function for the plate fin heat exchanger are determined. Minimum total annual cost occur at maximum number of fin layers in other words at large size of heat exchanger. Pressure drops and heat transfer rates are interdependent quantities and both of them affect the capital and operating costs of any heat exchanging system. The result of this work exhibit the impact of cost component for heat exchanger selection.

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