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THERMO ECONOMICAL OPTIMIZATION OF PLATE TYPE OF HEAT EXCHANGERS FOR WASTE HEAT RECOVERY

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Abstract: A thermo economic optimization analysis is presented yielding simple algebraic formula for estimating the optimum area of heat exchangers of plate type (both fluids unmixed) which are applied in industrial applications. An economic analysis method is used in the present study, together with the thermal analyses of plate heat exchangers, for thermo economic optimization. The validity of the optimization formulations was checked.
Keywords: Thermo economics, plate heat exchanger, unmixed type, optimization.

ATIK ISI GERİDÖNÜŞÜMÜ İÇİN PLAKALI TİP ISI DEĞİŞTİRİCİLERİNİN TERMO EKONOMİK OPTİMİZASYONU

Özet: Sanayide kullanılan çapraz akımlı, karışımsız tip plakalı ısı değiştiricilerinin optimum alan hesabı için bir termo ekonomik optimizasyon basit matematiksel denklem ile ifade edilmiştir. Bu çalışmada, plakalı ısı eşanjörlerinin hem ısıl ve hem de ekonomik yönden optimizasyonu için yeni bir ekonomik analiz ile ısıl analiz yöntemleri bir arada kullanılmıştır. Optimizasyon denklemlerinin geçerliliği kanıtlanmıştır.
Anahtar Kelimeler: Termo ekonomi, plakalı ısı eşanjörü, karışımsız tip akım, optimizasyon.

NOMENCLATURE

a	Constant depending on values of fixed operating parameters as defined in Eqn. (17),
A	Area of heat exchanger, [m^2]
A_{crit}	Critical area of heat exchanger, [m^2]
b	Constant depending on values of fixed operating parameters as defined in Eqn. (18),
c	Constant depending on values of fixed operating parameters as defined in Eqn. (20),
C	Ratio of minimum to maximum heat capacity rates of two streams in heat exchanger,
C_A	Area dependent first cost of the heat exchanger, [\$/m^2]
C_E	Cost of energy saved, [\$/W.hr]
C_{max}	Higher heat capacity rate in heat exchanger, [= mC_{max}], [W/K]
C_{min}	Lower heat capacity rate in heat exchanger, [= mC_{min}], [W/K]
C_p	Specific heat of circulating fluid having minimum heat capacity rate, [J/(kg.K)]
d	Market discount rate in fraction,
H	Annual time of operation, [h/yr]
i	Energy price escalation rate in fraction,
m	Mass rate of flow of circulating fluid having minimum heat capacity rate, [kg/s]
M_s	Ratio of annual maintenance and operation cost into first original cost,
N	Technical life of heat exchanger, [yr]
N_p	Payback time, [yr]
NTU	Number of transfer units,
P_1	Ratio of the life cycle energy cost savings to the first year energy cost savings, [yr]
P_2	Ratio of the life cycle expenditures incurred because of the additional capital investment to the initial investment,
R_v	Ratio of resale value into the first original cost,
S	Net overall life cycle savings of the heat exchanger, [\$]
U	Overall heat transfer coefficient, [W/(m^2.K)]
Δ	Constant depending on values of fixed operating parameters as defined in Eqn. (19),
ΔT	Maximum temperature difference between hot and cold fluid inlet in plate type heat exchanger, [^{o}K]
ε_{crit}	Critical effectiveness of plate type heat exchanger
ε	Effectiveness of plate type heat exchanger.

INTRODUCTION

Economics of heat exchanger operation is vitally significant. So, optimum operating temperatures for plate type heat exchanger of both fluids unmixed type as shown in Fig. 1 is extremely important in order to have maximum overall life cycle earnings for these heat exchangers. The optimum values of heat transfer area must be calculated at which minimum cost and so maximum savings occur for plate type heat exchangers that are widely applied for industrial applications for that reason. There exist many parameters for optimizing such heat exchangers in a thermo economical manner. Fixing and, so eliminating all of these thermal and economical parameters depending on the certainty of operating characteristics of applications and the most efficient operating condition of the heat exchangers can determine the optimum heat transfer area for plate heat exchangers. It is known that the effectiveness of the heat exchanger is directly related to its size which affects with its initial cost. A thermo economic feasibility study is necessary before installing the heat exchanging systems. The basic topic of the present work depends upon this idea. A new thermo economic optimization technique is realized and presented for this purpose. Original formulae are developed for calculating the optimum heat transfer area at which the maximum total life cycle savings occur. A thorough search of the current literature showed that there was no previous study on optimizing the life cycle savings of a plate type heat exchanger in detail. A well known and practical method, P_1-P_2 method, which is offered by Duffie and Beckman (1980), is used for optimizing the size and operating conditions of heat exchanger, and original interesting results are presented. Variable parameters used in formulating the optimization problem are listed as technical life of the heat exchanger, first cost of the heat exchanger per unit heat transfer area, annual interest rate, present net price of energy, annual energy price escalation rate, annual average operating time, ratio of minimum heat capacity rate into maximum heat capacity rate, design values of the difference of maximum and minimum temperatures of hot and cold fluids for heat exchanger, overall heat transfer coefficient of the heat exchanger, resale value and the ratio of annual maintenance and operation cost to the first original cost. Optimum heat transfer area of a plate type of heat exchanger and optimum value of net life cycle savings can be calculated easily in a few minutes with the help of practical formulae. A thorough search of the present literature showed that there were several studies about the heat exchangers (Vojtech et. al. 2011, Chung et al. 2002, Grazzini and Rinaldi 2001, Cornelissen and Hirs 1999, Georgiadis 1998, Şahin 1997, Edwards and Matavosian 1982). On the other hand, Aktürk F. at all experimentally investigated the thermal and hydrodynamic performance analyses of sa elected gasketed-plate heat exchanger with different number of plates are performed experimentally for a wide range of Reynolds numbers. Can A. and Kantürer T theoretically and practically investigated about increasing the capacity of autoclave exchanger. All of these studies are not directly related to the present work. Original formulae are developed and presented finally.

MATHEMATICAL FORMULATION

The net savings of a heat exchanger as shown in Fig. 1 can be calculated by using the cost data (Burmeister 1998) as:

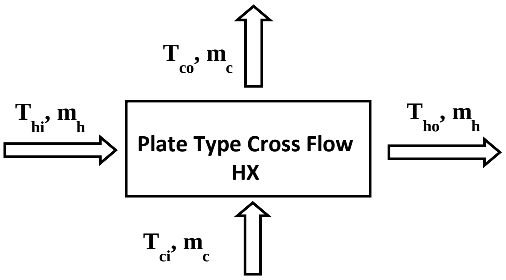

Fig. 1. Schematic figure of a typical plate type heat exchanger.

$$S = P_1 \cdot C_E \cdot \varepsilon \cdot m \cdot C_P \cdot H \cdot \Delta T - P_2 \cdot C_A \cdot A \qquad (1)$$

The effectiveness of plate heat exchanger of both fluids unmixed type as a function of number of transfer units and the ratio of heat capacity rates can be correlated by:

$$\begin{aligned}\varepsilon &= 0.1835 + 0.4067 \cdot NTU - 0.0443 \cdot C \\ &- 0.0529 \cdot NTU^2 - 0.1114 \cdot C^2\end{aligned} \qquad (2)$$

Where both of the number of transfer units of a plate type exchanger and ratio of heat capacity rates must be greater than zero in this equation, and they can be calculated by the followings.

$$NTU = \frac{U \cdot A_{HX}}{m \cdot C_P} \quad \text{and} \quad C = \frac{C_{\min}}{C_{\max}} \qquad (3)$$

The total net life cycle savings of the plate type of heat exchanger as presented in Eqn. (1) can be derived with respect to area of itself and setting it into zero to get optimum value of heat transfer area as in the form of the following function.

$$\frac{\partial S}{\partial A} = P_1 \cdot C_E \cdot m \cdot C_P \cdot H \cdot \Delta T \cdot \frac{\partial \varepsilon}{\partial A} - P_2 \cdot C_A = 0 \qquad (4)$$

Where the derivative of the effectiveness with respect to area can be evaluated from Eqn. (2) as:

$$\frac{\partial \varepsilon}{\partial A} = \frac{0.4067 \cdot U}{m \cdot C_P} - \frac{0.1058 \cdot U^2 \cdot A}{m^2 \cdot C_P{}^2} \qquad (5)$$

And Eqn. (4) can now be rewritten as follows.

$$\frac{\partial S}{\partial A} = P_1 \cdot C_E \cdot H \cdot \Delta T \cdot \left[0.4067 \cdot U - \frac{0.1058 \cdot U^2 \cdot A_{opt}}{m \cdot C_P}\right] - P_2 \cdot C_A = 0 \qquad (6)$$

Then as in follow is written:

$$A_{opt} = \frac{3.844 \cdot m \cdot C_P}{U} - \frac{9.452 \cdot P_2 \cdot C_A \cdot m \cdot C_P}{U^2 \cdot P_1 \cdot C_E \cdot \Delta T \cdot H} \qquad (7)$$

The second derivative of the net total life cycle savings function is always negative which indicates a local maximum certainly.

$$\frac{\partial^2 S}{\partial A^2} = \frac{-0.1058 \cdot P_1 \cdot C_E \cdot \Delta T \cdot H \cdot U^2}{m \cdot C_P} \langle 0 \qquad (8)$$

The economic parameters P_1 and P_2 are defined as in following equations (Duffie and Beckman 1980). **If i is equal to d**:

$$P_1 = \frac{N}{1+i} \qquad (9)$$

And for i $\neq$ d:

$$P_1 = \frac{1}{(d-i)} \cdot \left\{ 1 - \left[\frac{1+i}{1+d} \right]^N \right\} \tag{10}$$

and

$$P_2 = 1 + P_1 \cdot M_S - R_V \cdot (1+d)^{-N} \tag{11}$$

For waste heat recovery application purpose the payback period can be estimated by setting the net savings into zero for optimal area as:

$$S = 0 \Rightarrow P_1 = \frac{P_2 \cdot C_A \cdot A_{opt}}{C_E \cdot \varepsilon \cdot m \cdot C_P \cdot \Delta T \cdot H} \tag{12}$$

for i = d case:

$$N_P = \frac{P_2 \cdot C_A \cdot A_{opt} \cdot (1+i)}{C_E \cdot \varepsilon \cdot m \cdot C_P \cdot \Delta T \cdot H} \tag{13}$$

For i # $\neq$ d case the value of payback is calculated via using the following equality.

$$N_P = \frac{\ln\left[\dfrac{P_2 \cdot C_A \cdot A_{opt} \cdot (d-i)}{C_E \cdot \varepsilon \cdot m \cdot C_P \cdot \Delta T \cdot H} \right]}{\ln\left[\dfrac{1+i}{1+d} \right]} \tag{14}$$

The critical value of effectiveness of the plate type heat exchanger can be determined by setting S function into zero as in the following equality:

$$\varepsilon_{crit} = \frac{P_2 \cdot C_A \cdot A_{crit}}{P_1 \cdot C_E \cdot m \cdot C_P \cdot \Delta T \cdot H} \tag{15}$$

The critical value of the heat transfer area can be found by using Eqns. (2) and (15) together as in Eqn. (16).

$$A_{crit} = \frac{-b + \sqrt{\Delta}}{2 \cdot a} \tag{16}$$

where as:

$$a = \frac{0.0529 \cdot U^2}{m^2 \cdot C_P{}^2} \tag{17}$$

$$b = \frac{P_2 \cdot C_A}{P_1 \cdot C_E \cdot m \cdot C_P \cdot \Delta T \cdot H} - \frac{0.4067 \cdot U}{m \cdot C_P} \tag{18}$$

$$\Delta = b^2 - 4 \cdot a \cdot c \tag{19}$$

and

$$c = 0.1114 \cdot C^2 + 0.0443 \cdot C - 0.1835 \tag{20}$$

RESULTS AND DISCUSSION

For a typical single fluid heat exchanger problem (Stoecker, 1989), it is given that i = 0, d = 0.08, U = 23 W/(m^2.K), m = 7.5 kg/sec, C_P = 1050 J/kg K, H = 8760 hr/yr, N = 5 yr, ΔT = 155 ^oK, C_A = 90 \$/m^2, C_E = $5.4.10^{-6}$ \$/(Whr), M_S = 0, R_V = 0, and C = 1. The values of total net life cycle savings for this specific example are depicted in Fig. 2. There exist specific local maximum saving points in this figure. The optimum heat transfer area of plate type heat exchanger is calculated as 884.36 m^2 (optimum NTU is 2.582) and for single fluid heat exchanger the optimum value of heat exchanger area was determined as in between 686 and 726 m^2 for this sample problem by using Fibonacci Search Method. The main reason of this difference is as a result of different effectiveness-NTU relationship variation characteristics for single fluid and plate type heat exchangers.

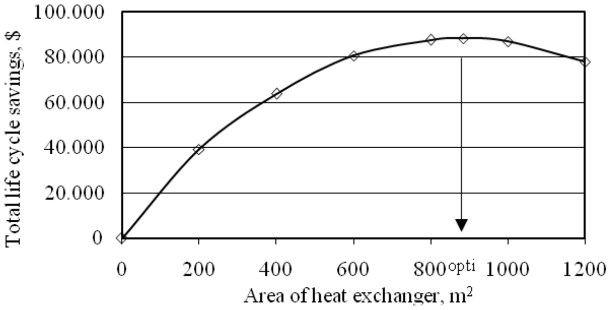

Fig. 2. Total life cycle savings versus heat transfer area for plate type heat exchanger.

The values of net life cycle savings with corresponding optimum heat transfer area, number of transfer units, and effectiveness values are presented in Table 1 and Fig. 3.

Table 1. Thermo economic Performance Values for Plate Type Heat Exchanger.

A (m^2)	200	400	600	800	884	1000	1200
NTU	0.584	1.168	1.752	2.337	2.583	2.921	3.505
ε	0.247	0.431	0.581	0.689	0.725	0.764	0.803
S (\$)	39252	63901	80670	87703	88455	87087	78127

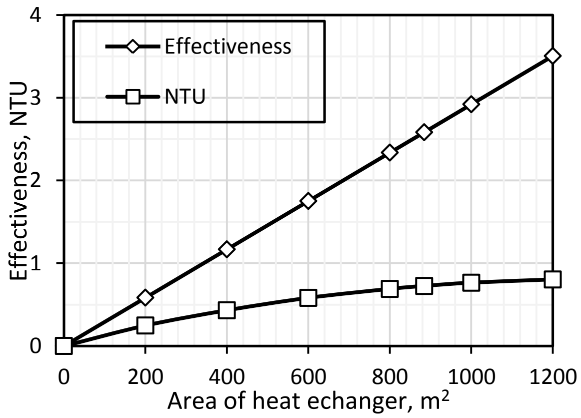

Fig. 3. NTU and □ versus heat transfer area for plate type heat exchanger.

Payback period is calculated as 2.14 years by means of Eqn. (14) for this sample problem. On the other hand, critical area of heat transfer is calculated as 2655.44 m^2 (critical NTU is 7.752) by using Eqn. (16) through Eqn. (20). There exist so many design tools such as computer codes for designing the heat exchangers. These codes are helpful for speedy estimation of heat exchangers and they are used in industry. These computer codes select the standardized heat exchanger due to pre designed operating temperature levels by using thermo-hydraulic data without considering economics.

A typical application for a plate type heat exchanger industrial from Barış Lt. Co. It is given that i=d=0.10, U=39.50 W/m^2K, m=3.92 kg/sec (waste exhaust gas), Cp=1126 J/kgK, H=4800 hr/yr, N = 10 years, □T=128 K, C_A=160 $\$/m^2$, C_E=5.21x10^{-5} \$/(Whr), M_s= 0.125, R_v=0.0625 and C=0.313.

Barış Lt. Co. used 249.50m^2 of heat exchanger area for this application. The area is compared with present optimization equation (7) and the optimum area is calculated for this sample application as 398.438 m^2. They saved 786787.45 \$ for 249.50 m^2 of heat exchanger area whereas if optimum area were used this saving would be 907470.81 \$ with a 1.29 year payback time for the life cycle of 10 years. This indicated us that the thermo-economic optimization is vitally significant for plate type heat exchanger applications.

CONCLUSION

It can be deduced that there exists always a local maximum heat transfer area value in plate type heat exchanger applications as presented in two samples of heat exchanger optimization problems. Excessive heat transfer area will not be cost effective beyond the optimum values in spite of a greater heat transfer recovery potential. The number of transfer units and the ratio of heat capacity rates of heat exchanging fluids have dominant effect on the amount of total life cycle earnings of a plate type heat exchanger. It is clear that there exist good thermal and economic performances all together at the optimum point for plate type heat exchanger. The plate type of heat exchangers must be designed close to this optimum point. The present formulae may seem to be helpful for cross flow heat exchanger designers and manufacturers when using for waste heat recovery.

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