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A Point of View on Optimum Design of Shell and Tube Heat Exchangers

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The concept of irreversibility is described based on theory of second law of thermodynamics. Its understanding followed by the entropy generation minimization is challenges for specialists. Nowadays advanced thermodynamics uses the entropy generation rate as a parameter able to quantify the significance of irreversibilities. Heat exchangers play a major role in the safety performance and economy of ships. The aim of this paper is to offer to marine engineers a tool to deal with optimum design of shell and tube heat exchangers by entropy generation minimization method. The modified entropy generation number is considered as the objective function in the minimization process. Thus, marine engineers' goal is to minimize the modified entropy generation number in order to increase effectiveness and to decrease pressure drop.

Keywords: *shell and tube heat exchanger, optimization, minimization, entropy generation.*

1. Introduction

Shell and tube heat exchangers have resistant manufacturing features, show design flexibility and are easy adaptable to operational conditions.

Thus, they are suitable for processing industries, oil refineries, nuclear power plants and other large-scale chemical process. In maritime technologies, they can be also found for cooling oil or hydraulic fluids.

For these reasons, the design of this equipment is an important issue when talking about thermal processes.

However, it is true that this activity should face some challenges coming from the difficulty of heat transfer equations and pressure drops.

This paper deals with the optimization of shell and tube heat exchanger design by the help of entropy generation minimization.

Entropy generation minimization (EGM) is a tool in the hands of specialists aiming modeling and optimization (1).

This analysis starts with the establishment of the entropy generation of a system as a function of physical characteristics as dimensions, materials, constrains.

It is useful to develop a model considering building blocks specific to applied thermodynamics, such as systems, laws, cycles etc, but also basic principles that govern the processes, like heat and mass transfer.

In this way, the proposed model becomes realistic, being considered the inherent irreversibility.

It is found the minimum entropy generation for the model and the entropy generation number expresses the approach of any other design to the limit of realistic ideality.

2. Methods

Any heat transfer calculations starts with the writing of the overall energy balance and the rate equation, resulting the size of heat energy transferred.

Thus, the overall heat transfer coefficient (U_0) depends on the tube side and shell side heat transfer coefficients and fouling resistance (1), so:

$$\frac{1}{U_0} = \frac{1}{h_0} + \frac{\Delta r}{k}\left(\frac{A_0}{A_{lm}}\right) + \frac{1}{h_i}\left(\frac{A_0}{A_i}\right) + R_{f,0} + R_{f,i} \tag{1}$$

In equation (1) h_0 and h_i are heat transfer coefficient for the fluid flowing in the shell and, respectively, through the tubes; "A" is the surface while subscripts "i", "o" and "lm" indicate inside and outside surface areas of a tube and their log mean; for the fouling resistances on one unit area was used "R_f", additionally "o" indicating shell side and "i" tube side.

Usually, tube side heat transfer coefficient is determined by using turbulent flow equations, while shell side heat transfer coefficient assessment is a more laboriously task because of baffles presence.

The baffle cuts are aligned vertically and have several duties: to support tubes during assembly and operation and help avoid vibration from flow induced eddies, to lead the shell side working fluid back and forth across the tube bundle; to furnish effective velocity and heat transfer rates; also, they allow dirt particles setting out of the shell side fluid to be removed.

Heat transfer correlations for flow through tube banks considering a flow normal to the long axes of a set of tubes arranged in a geometrical array, lead, according to Holman (3), to the Nusselt number:

$$Nu = \frac{h_0 d_0}{k} = C\, Re^n\ Pr^{1/3} \tag{2}$$

$$Re = \frac{d_0 V_{max} \rho}{\mu} \tag{3}$$

In Reynolds number, d_0 is outside tube diameter, V_{max} is the maximum velocity of the fluid through the tube bank, which is assessed after evaluation of the cross-flow area; k, ρ and μ are the thermal conductivity, density and viscosity of the shell-side fluid, properties determined for the arithmetic average temperature of the fluid between the two end temperatures. In equation (2), "Pr" is Prandtl number of the shell-side fluid and exponent "n" and constant "C" can be found from tables, from dependency with pitch to outside diameter ratio.

The heat exchanger effectiveness is given bellow:

$$\varepsilon = \frac{\sqrt{2}}{\sqrt{2} + coth(NTU / \sqrt{2})} \tag{4}$$

In addition, the number of heat transfer units is found by using:

$$NTU = \frac{UA_0}{C_{min}} \tag{5}$$

The minimum heat capacity rate (C_{min}) is written in relation to the mass flow rate and specific heat capacity of the tube side fluid:

$$C_{min} = m_i c_{pi}, \tag{6}$$

Total tube outside heat transfer area (A_0) is described by the formula bellow, containing given tube length (L), outside tube diameter (d_0) and number of tubes (N_t):

$$A_0 = \pi L d_0 N_t \tag{7}$$

$$N_t = CTP \frac{\pi D_S^2}{4 A_i} \tag{8}$$

$$D_S = 0,637 \sqrt{\frac{CL}{CTP}} \left[\frac{A_0 (PR)^2 d_0}{L} \right]^{1/2} \tag{9}$$

where:

CL – tube layout constant,

CTP – tube count calculation constant related with the incomplete coverage of the shell diameter with tubes,
D_S – shell diameter,
PR – tube pitch ratio.

2.1. Pressure losses

2.1.1. Shell-side pressure losses, ΔP_0

Can be calculated using an equation involving Fanning function for flow on the shell side (f), the mass velocity on the shell side (f), the mass velocity on the shell side (V_S), the inside diameter of the shell (D_S), the number of baffles (N_B), the density of the shell side fluid (ρ_S), the equivalent diameter (D_e) and the viscosity of the shell side fluid (μ_S), given by Peters at al. (4):

$$\Delta P_0 = \frac{2 f V_S^2 D_S (N_B + 1)}{\rho_S D_e \left(\dfrac{\mu}{\mu_S} \right)} \tag{10}$$

with:

$$f = exp\left(0{,}576 - 0{,}19 \ln Re_S\right) \tag{11}$$

$$V_S = \frac{m}{S_m}$$

$$D_e = \frac{4\left(C_p S_n^2 - \dfrac{\pi d_0^2}{4} \right)}{\pi d_0} \tag{12}$$

$$Re_S = \frac{D_e V_S}{\mu_S} \tag{13}$$

where:
m – mass flow rate of the fluid,
S_m – cross flow area measured close to the central symmetry plate of the shell containing its axis,
C_p – constant depending on pitch tube layout (for square pitch C_p=1, for triangular pitch C_p= 0,86),
d_0 – outside diameter of tubes,
S_n – pitch (center–to–center distance) of tube assembly.

2.1.2. Tube-side pressure losses, ΔP_i

First, it is required the calculation of friction factor for flow through the tubes from the Reynolds number and the relative roughness with the application of the viscosity correction. Thus:

$$\Delta P_i = f_{cor} \frac{L}{d_i} \left(\frac{1}{2} \rho_i V \right) \cdot N_p \tag{14}$$

Above:
L – length of tubes,
d_i – tube inside diameter,
ρ_i – density of tube-side fluid,
V – average flow velocity through one tube,
N_p – number of passes.

It will be introduced an additional pressure drop, ΔP_r - the return pressure loss, related to the pressure drop associated with fluid entry into the tube bundle, fluid leaving the bundle and fluid flouring around bends.

So:

$$\Delta P_r = 4 N_p \left(\frac{G^2}{2\rho} \right) \tag{15}$$

where:
G – ratio between mass flow rate and total flow area available per pass.

2.2. Second law analysis

In all types of heat transfer process, thermodynamic irreversibility is evaluated in terms of entropy generation.

The main causes of irreversibility occurring in heat exchangers are: heat transfer between the flows, pressure losses due to fluid friction and energy dissipation to the environment (5).

The losses associated to heat transfer across a finite temperature and to pressure drop due to friction can be minimized by the help of entropy generation minimization, in order to achieve optimum heat exchanger design (6). Second law analysis is one of the strongest tools used to improve the performance of thermal process.

It is used to investigate irreversibility due to heat transfer in terms of entropy generation rate. Below are given the entropy generation rates caused by the finite temperature difference $\left(S_{gen\Delta T}\right)$ and by the pressure losses $\left(S_{gen\Delta P}\right)$ (7).

$$S_{gen\Delta T} = (mc_p)_i \, ln\frac{T_{to}}{T_{ti}} + (mc_p)_o \, ln\frac{T_{So}}{T_{Si}} \tag{16}$$

In equation (16) subscripts "i" and "o" refer to tube side fluid and shell side fluid, while T_{to}, T_{ti} are tube side outlet temperature and tube side inlet temperature, so as T_{So}, T_{Si} are shell side outlet temperature and shell side inlet temperature.

$$S_{gen\Delta P} = m_i \frac{\Delta P_i}{\rho_i} \frac{ln(T_{to}/T_{ti})}{T_{to} - T_{ti}} + m_o \frac{\Delta P_o}{\rho_o} \frac{ln(T_{So}/T_{Si})}{T_{So} - T_{Si}} \tag{17}$$

As a result, the total entropy generation rate in the heat exchanger is the sum:

$$S_{gen} = S_{gen\Delta T} + S_{gen\Delta P} \tag{18}$$

It is difficult to use the total entropy generation rate in the assessment of the performance of the heat exchanger, being more appropriate the evaluation of the irreversibility of heat exchangers by the help of the dimensionless form of the entropy generation rate (8). Its expression is known as the modified entropy generation number:

$$N_{SH} = \frac{S_{gen} T_{Si}}{Q} \tag{19}$$

3. Results

Thermal optimization of a system starts with the identification of the objective function, next step being the minimization of this objective function, which in our case is the modified entropy generation number.

Are given inlet temperatures for tube side fluid (85°C) and shell side fluid (30°C), and also outlet temperatures for tube side fluid (55°C) and shell side fluid (44°C); mass flow rates for tube side fluid and shell side fluid are 6 kg/s, respectively 9 kg/s.

Figures 1 and 2 show the dependency between the modified entropy generation number and the effectiveness of the heat exchanger; respectively number of tubes.

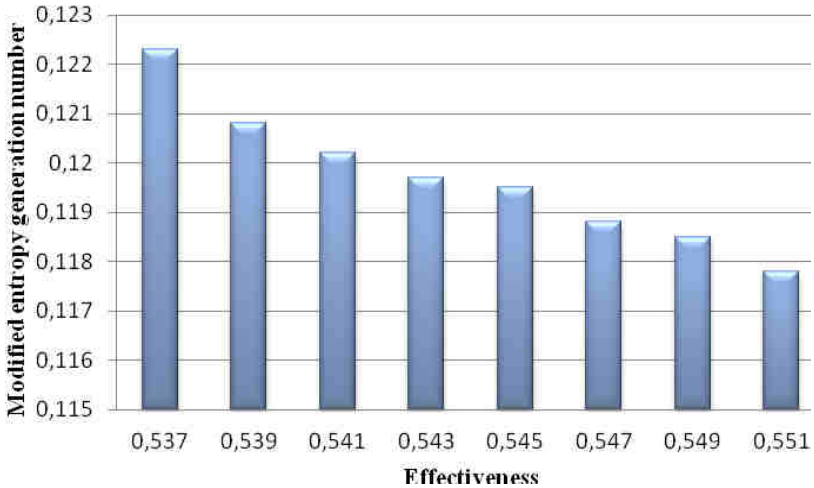

Figure 1. Relationship between modified entropy generation number and effectiveness

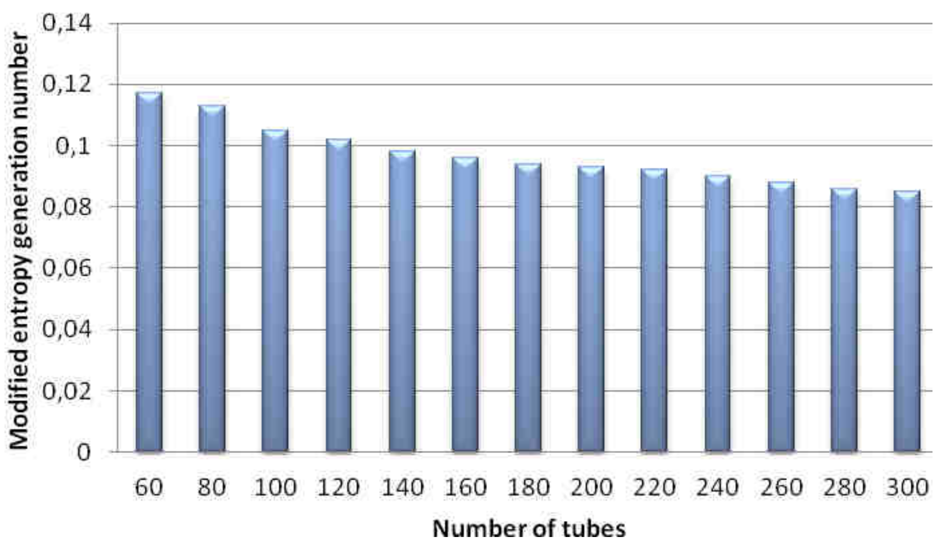

Figure 2. Effect of number of tubes on modified entropy generation number

From Figures 3 and 4, one can see effects of pressure losses on number of tubes.

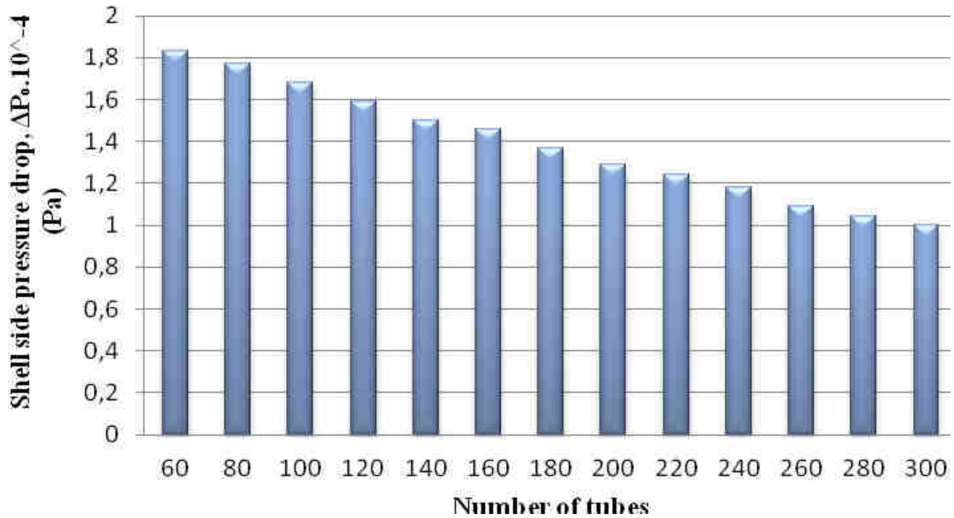

Figure 3. Effect of number of tubes on shell side pressure drop

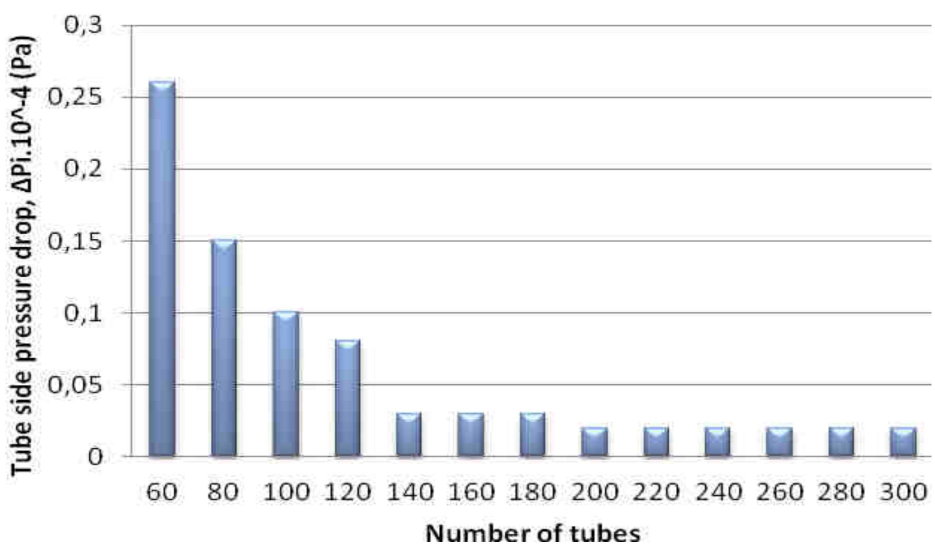

Figure 4. Effect of number of tubes on tube side pressure drop

4. Discussion and conclusion

Representations show that modified entropy generation number decreases together with the increase of effectiveness and of number of tubes. Both types of pressure drop decrease with the increase of tubes number.

The variation is visible in the case of shell side pressure drop, while tube side pressure drop presents a neglectable variation, when tubes are more than 140.

The entropy generation of any thermal system provides a useful measure of the extent of irreversibility.

Optimization of shell and tube heat exchangers based on the assessment of irreversibilities expressed by pressure drops (losses) and difference between cold and hot streams was the goal of this paper.

Entropy generation minimization approach was implemented by the study of the relationships between modified entropy generation number and effectiveness but also number of tubes and pressure drops.

The modified entropy generation number was the objective function in this optimization.

The smaller the modified entropy generation number, the better the performance of the heat exchanger would be.

Low values for the modified entropy generation number were found for high values of effectiveness and high number of tubes; an increase in tubes number led to a decrease of pressure drops, being seen that tube side pressure drop is not significant in comparison with shell side pressure drop.

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