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Numerical investigation on the multi-objective optimization of a shell-and-tube heat exchanger with helical baffles

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ABSTRACT

An improved method combining numerical simulation with multi-objective genetic algorithm (MOGA) was applied to study the flow and heat transfer characteristics of shell-and-tube heat exchanger with helical baffles (STHXsHB). It overcomes the dependence on empirical correlations. The helix angle and overlapped degree of helical baffles were chosen as optimization parameters, while the overall heat transfer coefficient K and pressure drop ΔP of STHXsHB were optimized by MOGA. The results showed that both overall heat transfer coefficient K and pressure drop ΔP varied adversely with the helix angles. The pressure drop ΔP was favorably affected by the overlapped degrees. The overall heat transfer coefficient K did not vary significantly with the overlapped degree. Three optimum configurations were obtained by the MOGA to maximize the overall heat transfer coefficient K and minimize the shell-side pressure drop ΔP. Compared with the original heat exchanger, the overall heat transfer coefficient K increased averagely by 28.3%, while the average pressure drop reduced averagely by 19.37%.

1. Introduction

Shell-and-tube heat exchangers (STHXs) are widely used in energy and chemical process industries because of their relatively simple construction and the multi-purpose application possibilities in a very large temperature and pressure range [1–3]. Baffles are one of the most important parts of STHXs. They force the shell-side fluid to flow across the tubes to ensure high heat transfer rates and provide support for tube bundle. The most commonly used baffles, called segmental baffles, cause the shell-side fluid to flow in a zigzag manner across the tube bundle, improving the heat transfer with a large pressure drop penalty [4–5]. Studies of shell-side flow patterns on the conventional segmental baffled revealed complicated conditions with dead zones, localized plug or mixed flow, bypass or recirculated current [6–9]. Significant effort has been devoted in recent years to improve the baffle for higher heat transfer coefficients, reduced fouling factor and less pressure drop [10–12]. The STHX with helical baffles (STHXsHB) proposed by Lutcha and Nemcansky effectively improved the performance on the shell side [13].

Compared with the conventional shell-and-tube heat exchangers, heat exchangers with helical baffles have some advantages such as increased heat transfer coefficient, reduced fouling and pressure drop [14–15]. Furthermore, the shell-side spiral flow causes the radial velocity gradient, which thins the boundary layer and increases the heat transfer coefficient. Since helical baffles have significant advantages over conventional segmental baffles, many handbooks covering the influence factors on the fluid flow and heat transfer performance of the helical baffle heat exchanger are available [16–18]. It was found that helix angle and overlapped degree of the helical baffles are important structural parameters that affect both pressure drop and heat transfer in shell side of STHXsHB. Many experimental and numerical investigations on the structure parameters of heat exchanger with helical baffles have been performed in the past. Lutcha and Nemcansky [13] investigated the flow pattern and heat transfer of tubular heat exchangers with helical baffles. They found that the Nusselt number of the helical baffled heat exchanger significantly depended on the helix angles. At small helix angles, the flow of working fluid resembled a cross-flow. In the range of 25°–40°, the heat transfer coefficient increased with helix angles. The Nusselt number dropped rapidly when helix angle was higher than 40°. Wang [19] measured the flow field in shell-and-tube heat exchangers with helical baffles using laser Doppler anemometry. He pointed out that the optimum helix angle depended on the Reynolds number of the shell-side working fluid. Cao et al. [20] investigated the effects of the overlapped degree on the shell-side performance of

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Nomenclature

A	heat transfer area, m^2
d_0	external diameter of tubes, mm
d_i	internal diameter of tubes, mm
D_e	equivalent diameter, mm
D_i	inner diameter of the shell-side, mm
e	overlapped degree
Nu	Nusselt number
ΔP	pressure drop, Pa
Pr	Prandtl number
G_k	the generation of turbulence kinetic energy due to the mean velocity gradients U velocity, m/s
Re	Reynolds number
Q	heat transfer rate, kJ/s
R_i	tube-side fouling resistance, m^2 K/W
R_o	tube-side fouling resistance, m^2 K/W
K	overall heat transfer coefficient, W/(m^2 K)
k	turbulent kinetic energy
Δt_m	log-mean temperature difference, K
S_z	Shell side minimum cross-sectional area of distribution, m^2

Greek symbols

β	helix angle, °
ε	turbulent energy dissipation
μ	dynamic viscosity, Pa s
ρ	density, kg/m^3
λ	thermal conductivity, W/(m K)
δ	tube thickness, mm

Subscripts

in	inlet
out	outlet
o	oil
w	water

STHXsHB. It was concluded that under the same helical pitch, the shell-side heat transfer coefficient and pressure drop decreases with the overlapped degree. Yang [21] compared the shell-side heat transfer coefficient and pressure drop of the heat exchanger at the helix angles of 12°, 18°, 30° and 40°. The results showed that the performance of the heat exchanger is the best under the condition of 18° helix angle. However, the above research on structural optimization of STHXsHB mainly concentrated on one certain parameter. There were few attentions payed on the study of coupled multi-parameters relationship. Therefore, it is difficult to accurately obtain the optimum structure on the basis of extreme value theory.

Over the last years, based on the empirical correlations, most of them are Bell–Delaware method, STHXs with segmental baffles (STHXsSB) is optimized using genetic algorithm (GA) by many researchers. Rajasekaran and Kannadasan [22] used the GA to achieve the minimum surface area of STHEsSB with given constrained condition. Other objectives like minimum entropy generation [23] and minimum total costs [24–26] of STHEsSB were also investigated. Meanwhile, several investigators used multi-objective genetic algorithm (MOGA) to optimize the multi-objective of STHEsSB, including the total annualized cost and the amount of cooling water [27], the effectiveness as well as exchanger weight or pressure loss [28], the effectiveness and total cost [29] and heat transfer rate and total cost [30], etc. Recently, Wang et al. [31] investigated the effects of configuration parameters on flow and heat transfer characteristics of spiral-wound heat exchanger (SWHE) based on Multi-Objective Genetic Algorithm. Lee and Kim [32] applied a multi-objective genetic algorithm with surrogate modeling technique to maximize heat transfer and minimize pressure drop in a printed circuit heat exchanger. Peng and Ling [33] and Xie et al. [34,35] used the GA to achieve minimum total weight or total annual cost of Plate-fin heat exchanges with given constrained condition.

The application of MOGA in optimization of STHXsSB has shown the effectiveness and robust. Unfortunately, this method needs to rely on the empirical correlations. Therefore, so far, few researches on optimization of STHEsHB have been published because there is no an acknowledged empirical correlation for designing STHXsHB. To overcome the dependence on empirical correlations, the method combining numerical simulation and multi-objective genetic algorithm (MOGA) was firstly applied to optimize the overall heat transfer coefficient and pressure drop of STHXsHB in this paper. The helix angle and baffle overlap degree were considered as optimization parameters. In addition, according to the parametric simulation results, the optimum configurations of helical baffles by means of MOGA were obtained.

2. Mathematical modeling

2.1. Physical geometry and mesh generation

The configuration with four elliptical sector-shaped plates joined in succession is generated and each pitch is made up of four ellipse baffles, whose projected angles to the normal cross-section of the heat exchanger is 90°. The heat exchanger is 2420 mm in length and the shell-side inner diameter is 250 mm. The tube arrangement is square with the tube diameter of 19 mm and the tube spacing is 25 mm. To simplify numerical simulation but with a reasonable accuracy in the model, several assumptions are made as follows: Firstly, the thickness of the baffles is neglected to reduce the number of grids. Secondly, the gaps between tubes and baffles and those between baffles and the shell are neglected. Thirdly, the heat loss is ignored for the well-insulated. Fourthly, the properties of the shell-side fluid is considered constant [36]. Conduction oil and water are shell-side and tube-side fluids, respectively. The helix angle and overlapped degree are depicted briefly in Fig. 1 where β refers to the helix angle and the overlapped degree is presented by $e = \frac{l}{L}$.

The computational domain is meshed by unstructured grids, which are generated by ICEM CFD. In order to ensure the accuracy of the numerical results, the grid adaptive process and grid independence tests are conducted. To accurately simulate the flow in the near-wall region, partial encryption is made near the pipe wall. Eventually the grid numbers vary from 8,935,624 to 9,985,987 according to different helix angles and overlapped degrees.

2.2. Governing equations and numerical method

The renormalization group (RNG) k-ε turbulence model which is applicable to high strain rate and larger degree of streamline curvature flow is adopted to account for the turbulent behavior of both shell-side fluid and tube-side fluid. It is noted that one of its obvious advantages is its higher prediction precision of the spiral flow [37–38]. The governing equations can be expressed as follows:

Continuity

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0 \tag{1}$$

Momentum:

$$\frac{\partial}{\partial x_i}(\rho u_i u_k) = \frac{\partial}{\partial x_i}\left(\mu \frac{\partial u_k}{\partial x_i}\right) - \frac{\partial p}{\partial x_i} \tag{2}$$