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Thermal–hydraulic performance of a novel shell-and-tube oil cooler with multi-fields synergy analysis

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ABSTRACT

In the present paper, a novel shell-and-tube heat exchanger is proposed for the application of oil cooler. It is numerically investigated compared to a rod baffles shell-and-tube heat exchanger using the commercial software FLUENT 6.3 and GAMBIT 2.3. The results of heat transfer, flow characteristics, and comprehensive performance are analyzed for both tube-side and shell-side with verifications of correlations and experimental apparatus. For tube-side, the novel heat exchanger demonstrates evidently excellent overall performance; while for shell-side, the novel heat exchanger illustrates slightly lower comprehensive performance than the rod baffles one. The path lines, pressure field, and temperature field are analyzed and the multi-fields synergy principle is adopted to evaluate the synergy extent between velocity, temperature, and pressure fields.

1. Introduction

Shell-and-tube heat exchangers (STHXs) are widely used in the petro-chemical industry, manufacturing industry, food preservation, electrical power production and energy conservation systems, due to their structural simplicity, relatively low cost and design adaptability. According to Master and co-workers, they account for more than 35–40% of the heat exchangers used in global heat transfer processes [1]. The conventional heat exchangers with segmental baffles (STHXsSB) are one of the most commonly used exchangers in the practical application. However, they have the disadvantages of high pumping power, fouling problems in the dead zones, and induction vibration of tube bundles [2]. Therefore, it is of great significance to propose new heat exchangers in order to overcome the above-mentioned drawbacks.

Lots of novel structures [3–29] have been suggested to enhance heat transfer, reduce power consumption and increase cost-effectiveness for the past decades. Among the those new heat exchangers, the main concept is altering the shell-side flow from zigzag pattern to longitudinal or helical pattern to avoid the impact of tube bundles and reduce the relaminarization and recirculation flow. As a result, this flow pattern variation increases heat transfer area, compresses heat exchanger, and improves cost-efficiency. Although the open literature is replete of multifarious novel heat exchangers, it is difficult to apply one heat exchanger for all fields since each design contains certain disadvantages. Therefore investigating new heat transfer enhancement techniques and proposing novel design to increase thermal–hydraulic performance are still in demands.

Experiments can provide highly reliable measurements of thermal–hydraulic performance; however, experiments can be extremely expensive and time-consuming compared to computational fluid dynamics (CFD). For very complex flows, such as those prevailing in the rod-baffle shell-and-tube heat exchanger, selecting an appropriate modeling approach can be difficult. There are complex tradeoffs between accuracy and computational expense. For example, a heat exchanger with 500 heat transfer tubes and 10 baffles requires at least 150 million computational cells to resolve the geometry. So far there are four main modeling approaches used for numerical simulations: the unit model [30,31,20], the periodic model [32,33], the porous model [34–37,21] and the whole model [38–40]. Recently Yang et al. [41] summarized the four modeling approaches, conducted a comparison of four different models on numerical accuracy, grid system size, computational period, and restriction, and provided an approach on selecting the most appropriate model for the practical situation.

So far, the performance evaluation criteria (*PEC*) [42,43], efficiency evaluation criterion (*EEC*) [44,45], and multi-fields synergy principle [46–53] have been proposed to evaluate performance and effectiveness. They all have been successfully utilized to analyze thermal–hydraulic performance. Through a wide literature survey, it is noticed that little work about its application on the whole model simulation of a shell-and-tube heat exchanger have

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Nomenclature

A_h	hydraulic area (m^2)
A	heat transfer area (m^2)
c_p	specific heat capacity (kJ kg^{-1} K^{-1})
$C_{1\varepsilon}$	empirical constant (–)
$C_{2\varepsilon}$	empirical constant (–)
C_μ	empirical constant (–)
D_i	tube inner diameter (m)
D_o	tube outer diameter (m)
D_s	shell inner diameter (m)
D_h	hydraulic diameter (m)
f	friction coefficient (–)
G_k	producing item of k by average velocity gradient (kg m^{-1} s^{-3})
h	heat transfer coefficient (W m^{-2} K^{-1})
i, j, k	component on x, y, z coordinates (–)
k	turbulent kinetic energy (m^2 s^{-2})
L	baffle pitch (mm)
L_0	baffle distance from head (mm)
L_t	tube length (m)
n	tube quantity (–)
Nu	Nusselt number (–)
PEC	performance evaluation criteria (–)
P_h	hydraulic length (m)
p_s	pitch of the helix curve (mm)
p	pressure (Pa)
ΔP	pressure drop (Pa)
Q	heat transfer power (W)
Re	Reynolds number (–)
S	eccentric distance (mm)
T	temperature (K)
u	inlet average velocity (m s^{-1})
U	flow velocity of fluid (m s^{-1})
r, θ, h	cylindrical coordinates (–)

Greek symbols

ρ	fluid density (kg m^{-3})
λ	thermal conductivity (W m^{-1} K^{-1})
μ	dynamic viscosity (kg m^{-1} s^{-1})
ε	turbulent dissipation rate (m^2 s^{-3})
σ_k	Prandtl numbers corresponded to k (–)
σ_ε	Prandtl numbers corresponded to ε (–)
β	synergy angle (°)
θ	synergy angle (°)

Subscripts

ave	average value
e	enhanced heat exchanger
p	primary heat exchanger
w	wall

been reported in open literature since these procedures consume too much computational resources. In this paper, a novel oil cooler is proposed to provide an alternative solution for industrial designers. 3-D numerical simulations of the heat exchanger for both tube-side and shell-side are developed. The thermal–hydraulic performances of tube-side and shell-side are investigated and *PEC* is used to analyze the results. The present work also extends the application of multi-fields synergy principle on the whole model simulation of shell-and-tube heat exchanger, thus filling the gap in open literature.

2. Model formulation

2.1. Geometric introduction

Recently our research group invented a novel heat transfer tube called *eccentric spiral tube* as shown in Fig. 1. Each cross section of the tube is a circle. The centerline of tube is a helix curve and its equation in cylindrical coordinate system is expressed as follows:

$$\begin{cases} r(t) = \left\{ \begin{array}{ll} \sin t \cdot s & 2 \cdot (k-1) \cdot \pi \leqslant t \leqslant \frac{2k\pi}{4} \\ s & \frac{2k\pi}{4} \leqslant t \leqslant \frac{6k\pi}{4} \\ -\sin t \cdot s & \frac{6k\pi}{4} \leqslant t \leqslant 2k\pi \end{array} \right\} k \in Z \\ \theta(t) = t \\ h(t) = p_s \cdot t \end{cases} \tag{1}$$

where r, θ, h are the coordinates in cylindrical coordinate system; s stand for the eccentric distance of the tube centerline; p_s stands for the pitch of the helix curve; D_o and D_i represent the outer and inner diameter of tube cross-section, respectively. In the present work, D_o and D_i is set as 16 and 14 mm, p is set as 40 mm, s is set as 2.5 mm.

Motivated from the rod baffle heat exchanger [6–9] and twisted tubes heat exchanger [10–14], our research group proposed a novel shell-and-tube heat exchanger as shown in Figs. 2 and 3. The original plain tubes in the conventional STHXsSB are replaced by this novel heat transfer tubes. This alignment results in heat transfer tubes contacting at many points along the length of tube in bundle. So all tubes are tightly braced and there exists no tube movement during working condition. The spiral tubes are assembled into such a bundle that there is no need to install any baffles (segmental, helical, orifice, rod, trefoil-hole, or flower baffles) or supporting parts (ring) between each tube in the heat exchanger. Therefore, it is expected that this shell-and-tube heat exchanger with spiral tubes (STHXsST) has the advantages of higher thermal–hydraulic performance, higher thermal effectiveness, tube bundle vibration elimination, and lower fouling due to its unique structure in both tube-side and shell-side. As one of the most outstanding inventions in the field of shell-and-tube heat exchanger, the shell-and-tube heat exchanger with rod baffles (STHXsRB), rather than STHXsSB, is taken as the reference group in order to demonstrate the novelty and improvement of the new oil cooler. The optimized geometric parameters [54,55] are adopted in the present work as presented in Table 1. For STHXsRB and STHXsST, all geometric parameters including shell diameter, shell length, tube number, inlet and out-

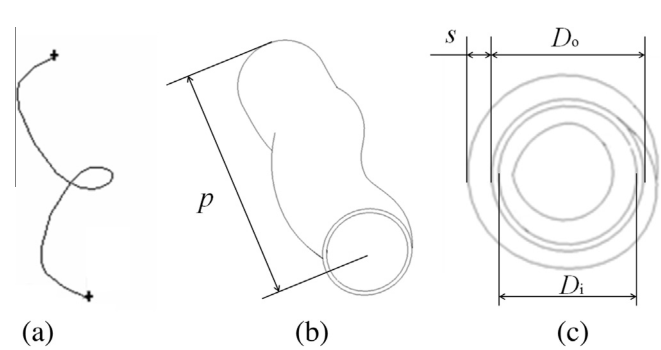

Fig. 1. Schematic diagram of the eccentric spiral tube: (a) the tube centerline; (b) the spiral tube; (c) the front view.