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Performance of flow and heat transfer in vertical helical baffle condensers☆

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ABSTRACT

The feed water heaters in power plants are actually the condensers using turbine extracting steam to heat feed water. The vertical feed water heater occupies less area than the horizontal one and convenient to lift tube bundles out in maintenance. However, the lower heat transfer coefficient due to thick condensate film limits its application. A novel trisection helical baffled vertical condenser (feed water heater) is proposed with liquid dams and gaps for facilitating condensate drainage. The flow and condensation heat transfer characters of two vertical condensers with variable angled trisection helical baffles of both single-thread and dual-threads and a variable spanned segmental baffled one were numerically studied with Mixture model of Fluent software. The distributions of velocity, pressure, volume fraction of condensate, and local heat transfer coefficient in these heat exchangers were demonstrated. The simulation results show that the inclined baffles with liquid dam and drainage gaps could drain condensate effectively from tube bundle surfaces and prevent liquid film from entraining into vapor, and that the variable angled trisection helical baffled vertical condenser with dual-threads could greatly improve the condensation heat transfer coefficient up to 35.7% higher than that of the variable spanned segmental baffled one.

1. Introduction

The feed water heaters in power plants are actually the condensers that the steam extracted from turbines condenses at their shell-side for heating the tube-side feed water. The vertical feed water heaters occupy less area than the horizontal ones and the large steam turbine workshop has available lifting equipment as well as sufficient height for lifting tube bundles out vertically. However, the film thickness of condensate gathered on the vertical tube surface in a segmental baffled vertical shell-and-tube heat exchanger is thicker and the heat transfer coefficient is lower than those of the horizontal one, which limits the application of vertical feed water heaters. Nevertheless, the horizontal heater has higher heat transfer coefficient but occupy larger area, and also needs space for dragging the tube bundle out with comparable size as the heat exchanger. To solve this dilemma, promoting the vertical feed water heaters with enhanced condensation heat transfer technique can be a savage, as the features of simpler structure, less occupied area, and reduced maintenance time are quite attractive to the power generation industry as well as petroleum and chemical industry.

Steam vapor condensation takes place at the shell-side of the feed water heater, thus the key to improve the shell-side heat transfer coefficient of a vertical feed water heater is to reduce film thickness on outside of vertical tubes. Many experts proposed different enhanced heat transfer structures for vertical tube condensation. Gregorig [1] proposed a method to reinforce the heat transfer of laminar film condensation on vertical wall by applying a series of drainage plates along a longitudinal grooved tube. Thomas [2] studied film condensation heat transfer on vertical tubes by vertical wires. An et al. [3] studied condensation heat transfer enhancement in spiral grooved tube with staggered tube bundle. Hafiz and Adrian [4] and Zhu [5] theoretically studied respectively the condensation phenomenon on pin-fin tubes and a vertical fluted tube. Most of these enhanced technologies are utilizing surface tension effect to drive the condensate flow into the groove valley and then to reduce the film thickness on the protrusion surfaces of tubes. There are also some enhanced methods worth considering other than on vertical surfaces. Chang and Yeh [6] studied condensation heat transfer enhancement in horizontal elliptical tube. Cavallini et al. [7] and Hicham et al. [8] investigated respectively condensation heat transfer enhancement in minichannels and in a horizontal non-circular microchannel. Chen et al. [9] numerically studied condensation flow of the refrigerant FC-72 in a rectangular microchannel with a 1 mm hydraulic diameter using the volume of fluid model. Caruso et al. [10] experimentally studied film condensation in inclined tubes with non-condensable gases. Peng et al. [11] proposed an idea of condensing on series short channels and draining the condensate before the wet vapor enters the next channel, which takes advantage of the initial thin film part during condensation with higher heat transfer coefficient. Cvengros et al. [12] studied a divided condenser that drains the condensate from different heights.

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Nomenclature

Latin Symbols

E	Energy, kJ
F	Volume force, kN
G	Mass flow, kg/s, kg/h
h	Enthalpy, kJ/kg
k	Turbulence kinetic energy, m^2/s^2
$\dot{m}$	Mass flux, $kg/(m^2 \cdot s)$
p	Pressure, Pa
q	Specific heat, kJ/kg
S	Source term
T	Temperature, K
u	Velocity component in x direction, m/s
v	Velocity, velocity component in y direction, m/s
w	Velocity component in z direction, m/s

Greek symbols

α	Volume fraction
β	Incline angle, °
ε	Turbulence kinetic energy dissipation rate, m^2/s^3
ΔH	Condensation enthalpy, kJ
λ_{eff}	Effective thermal conductivity, $W/(m \cdot s)$
μ	Dynamic viscosity, $kg/(m \cdot s)$
ρ	Density, kg/m^3

Subscript

c	Condensate
E	Energy
i	Tube inside
k	Phase number
m	Mixture
M	Mass
o	Tube outside
v	Vapor
dr	Drift
sat	Saturation

Similar to the ideas of Gregorig [1], Cvengros et al. [12] and Peng et al. [11], Chen et al. [13] proposed a vertical trisection helical baffle condenser (VTHBC) scheme. The VTHBC is a modified shell-and-tube heat exchanger with circumferential overlap trisection helical baffles [14,15]. The inclined baffle plates not only support the tube bundle but also divide the tubes into short segments and scrape the condensate from tube surface successively. Also the downstream straight edge of each sector baffle plate is widened and bent to form a liquid dam to prevent the accumulated condensate from blown into the vapor stream. The draining gaps are set at the curved edge of each sector baffle plate to drain the condensate along the inner surface of the shell to the bottom, thus to enhance the condensation heat transfer coefficient of the vertical feed water heater.

Numerical simulation has become an effective tool in understanding the heat transfer performance of a heat exchanger design. Recently, CFD modeling has been gradually applied to two-phase flow cases [16–20]. The CFD numerical simulation method is adopted in this paper to study the flow and heat transfer features of vertical feed water heaters with trisection helical baffles and segmental baffles.

2. Condensation models

2.1. Geometry and meshing

As shown in Fig. 1, there are three vertical condenser schemes for calculation and comparison, each condenser consists of a tube bundle of 16 tubes and 3 rods with equilateral triangular layout and a cylindrical shell with inlets and outlets for steam/condensate and cooling water. In account to the decreasing trend of volumetric flow of vapor during condensation, the two helical baffled schemes both have three sections with baffle incline angles of 35°, 25°, and 15°, and the segment baffled scheme has also three sections of different spanned baffles, forming decreased cross-section area. Each helical baffle scheme has dual-thread helical baffles at first and second sections of incline angles of 35° and 25°, while its 15° incline angled third section remains single-thread. The detailed diagrams and structural parameters of these schemes are shown in Figs. 1, 2, Tables 1 and 2.

Gambit software is used in building and meshing the 3D models of vertical trisection helical baffle condensers and segmental baffle condenser. Considering the complexity of the geometry, unstructured grids are adopted in the meshing. Fig. 3 is a grid graph of VTHBC at the shell side. The mesh refinement was conducted at the boundary layer of tubes. With the consideration of both independence of the grid number and computational time consumption, the grid numbers around 2.5 million for both VTHBC and VSBC schemes were finally adopted.

2.2. Governing equations

Considering the velocity difference of vapor and liquid phase, the Mixture model of Fluent was selected for the two-phase flow models in simulation, which assumes the local equilibrium in the control volumes. By solving the continuity equation, momentum equation and energy equation of the mixed phases, the two-phase flow features of relative velocity, the liquid phase volume fraction and the heat transfer coefficient could be acquired. The conservation equations of continuity, momentum, and energy are as follows [14,15],

$$\nabla \cdot \left(\rho_k \vec{v}_k\right) = S_{Mk} \tag{1}$$

$$\nabla \cdot \left(\rho_m \vec{v}_m \vec{v}_m\right) = -\nabla p + \nabla \cdot \left[\mu_m \left(\nabla \vec{v}_m + \nabla \vec{v}_m^{\,T}\right)\right] + \rho_m \vec{g} + \vec{F} + \nabla \cdot \left(\sum_{k=1}^{2} \alpha_k \rho_k \vec{v}_{dr,k} \vec{v}_{dr,k}\right) \tag{2}$$

$$\nabla \cdot \sum_{k=1}^{2} \left[\alpha_k \vec{v}_k (\rho_k E_k + p)\right] = \nabla \cdot (\lambda_{eff} \nabla T) + S_{Em} \tag{3}$$

where, the subscript m stands for the mixture; k is the phase sequence number, and 1 for vapor and 2 for liquid; $\vec{F}$ is the volume force; α_k, $\vec{v}_{dr,k}$, ρ_k and $\lambda_{eff\,k}$ are respectively the volume fraction, the drift velocity, the density, and the effective thermal conductivity of phase k; ρ_m, μ_m, and $\vec{v}_m$ are, respectively, density, viscosity, and velocity of the mixture, which are expressed respectively as

$$\rho_m = \sum_{k=1}^{2} \alpha_k \rho_k, \ \mu_m = \sum_{k=1}^{2} \alpha_k \mu_k, \ \vec{v}_m = \frac{1}{\rho_m} \sum_{k=1}^{2} \alpha_k \rho_k \vec{v}_k \tag{4}$$

For incompressible phase, $E_k = h_k$, where h_k is the sensible enthalpy of phase k. For compressible phase,

$$E_k = h_k - \frac{p}{\rho_k} + \frac{v_k^2}{2} \tag{5}$$

For a vapor and condensate two-phase flow system, the vapor and condensate are set as the primary and secondary phases, respectively. The relative velocity $\vec{v}_{cv}$, which is also the slip velocity, and the drift velocity $\vec{v}_{dr,c}$ are defined respectively as

$$\vec{v}_{cv} = \vec{v}_c - \vec{v}_v \tag{6}$$