Evaluation of the Ratio of Heat Exchanging Tube Finning on Heat Exchanger Efficiency

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Abstract. For the purpose of thermal and hydraulic and aerodynamic testing of the heat exchanger tube bundles, various full-scale tube specimens with different finning ratio have been proposed. Formulas that allow to evaluate the effect of the ratio of heat exchanging tube finning on the heat exchanger efficiency are presented.

Moisture separator-reheaters are heat exchangers where heat-exchanging processes are accompanied by load and temperature variations caused by the action of variable forces in the tube and shell sides, both in the transverse and axial directions. A combination of variations in force factors, existing or new gaps between the mating surfaces, where the sediment penetrates, either result in crevice corrosion followed by depressurization of the heat-exchanging circuits, or in a rapid loss of tube fastening tightness and strength [1]. For this purpose, the development of new designs of heat-exchanging cartridges using advanced materials is a critical task to ensure the strength and reliability of power plants.

Up to date, a lot of works are focused on the study of the heat-exchanging tube bundle strength subject to the specifics of the manufacture and assembly processes. Tube-to-tubesheet fixation methods, evaluation of the residual stress-strain state of fixing points and their quality are widely covered in [2-7]. However, the use of finned heat-exchanging tubes of various designs requires new thermal and hydraulic and aerodynamic studies to determine the optimal dimensions and number of fins, as well as the choice of materials that ensure the up-state of heat-exchanger for the specified service life of all the equipment.

The development of theoretical models that describe the heat and hydrodynamics of the tube bundles, and integral-differential equations are extremely difficult to treat mathematically [8]. Therefore, the development of experimental research methods and adequate models for thermal and hydraulic and aerodynamic testing is a critical task to ensure the tube bundle strength.

For this purpose, a number of research and development problems should be solved. The first task is to develop the methods for simulation of coolant flow in the tube and shell sides based on the coolant choice that adequately implement the thermal and hydraulic processes. The methods for simulating testing should be applicable for typical tube bundle designs. The heat-exchanging tubes with different arrangement of fins (longitudinal and transverse) are taken as models. The tube finning ratio is taken as an evaluation criterion.

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The second task is to develop the testing bench design, selecting the process equipment and measuring instruments for the process parameters to ensure the thermal and hydraulic conditions specified.

This approach enables to evaluate the influence of the tube finning ratio on the thermal and hydraulic and aerodynamic characteristics of the heat exchangers, and benchmark the heat-exchanging tube bundle strength.

The following approach is proposed for the first problem.

The heat-exchanging tubes may be bare, longitudinally or transversely finned. Some tube designs are shown in Fig. 1. These designs are characterized by a different heat exchange surface, which provides the equipment heating capacity required at the final temperatures and pressures specified.

Fig. 1. Heat-exchanging tube designs: a – bare tube; b – longitudinally finned tube; c – transversely finned tube.

For the purpose of study, we select tube specimens with typical geometric parameters that include: inner diameter, wall thickness and tube length, considered being equal for all the designs.

F heat exchange surface may be determined as follows [9]

$$F = \frac{Q}{k \Delta t_{cp}}; \Delta t_{cp} = \psi \Delta t_{lg}, \quad (1)$$

where Q is heat exchanger heating capacity under rated operation conditions; Δt_{cp} is average temperature drop; k is heat transfer coefficient; ψ is correction factor; Δt_{lg} is logarithmic temperature drop.

The heat exchanger performance depends to a large extent on the routing of coolant relative flows and the features of the hydrodynamics and heat transfer processes. The heat

transfer in tube bundles will be different under equal hydrodynamic conditions. The formulas for determining the heat exchange surface of the tube bundle, in relation to the designs specified, Fig. 1, will be as follows.

For bare tubes it is

$$H_1 = n\pi D l. \quad (2)$$

For finned tubes it is

$$H_{2,3} = n(H_p + H_T). \quad (3)$$

Formulas (2) and (3) have the following symbols: $H_{1,2,3}$ are heat exchange surface of tube bundle; n is number of tubes in tube bundle; D is bearing tube diameter; l is tube length; H_p is fin surface; H_T is unfinned surface.

The fin surface is determined geometrically subject to all the surfaces involved in heat transfer.

For longitudinally finned tube

$$H_{p2} = 2zh_p\delta_p. \quad (4)$$

For transversely finned tube

$$H_{p3} = z \left[\frac{\pi}{2} (D_p^2 - D^2) + \pi D_p \delta_p \right]. \quad (5)$$

For formulas (4) and (5) z is number of fins; D_p is diameter over fin top; δ_p is fin thickness; h_p is longitudinal fin width.

The unfinned surface is calculated as follows.

For longitudinally finned tube

$$H_{T2} = \pi(Dl - z\delta_p l_p). \quad (6)$$

For transversely finned tube

$$H_{T3} = \pi(Dl - zD_p \delta_p). \quad (7)$$

where l_p is longitudinal fin width.

The finning ratio will be evaluated using finning coefficient, as the rate of the fin total surface to the conventional surface of the bearing tube as follows

$$\varphi = \frac{H_p}{H_0} = \frac{H_p}{\pi D l}. \quad (8)$$

where H_0 is bearing tube conventional surface.

Thus, the bare tube will be considered as a structure with a zero finning ratio.

The unfinned bearing surface will be expressed in terms of the bearing tube conventional surface

$$\frac{H_T}{H_0} = 1 - \varphi = 1 - \frac{H_p}{\pi D l}. \quad (9)$$

Design value H of the heat exchange surface is determined as follows

$$H = H_e k_s, \quad (10)$$

where H_e is estimated heat exchange surface; k_s is safety factor, $k_s = 1.1 \div 1.2$.

The design heat exchange surface for the tube bundle that consists of an equal number of the tubes with different finning ratio is a constant value, other things being equal. It characterizes the heat exchanger performance.

Then subject to (8) it is true that

$$H = k_s (\varphi H_0 + H_T), \quad (11)$$

Hence, it becomes possible to experimentally benchmark the influence of heat exchanging tube finning ratio.

Thermal calculation and heat transfer parameters are determined according to the recommendations of [9]. Given that, we set the thermal and hydraulic testing conditions.

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