The evaluation of energy efficiency of convective heat transfer surfaces in tube bundles

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Abstract. When evaluating the effectiveness of the heat exchange surfaces in the main considered characteristics such as heat flow (Q, Watt), the power required for pumps (N, Watt), and surface area of heat transfer (F, m²). The most correct comparison provides a comparison "ceteris paribus". Carried out performance comparison "ceteris paribus" in-line and staggered configurations of bundles with a circular pipes can serve as a basis for the development of physical models of flow and heat transfer in tube bundles with tubes of other geometric shapes, considering intertubular stream with attached eddies. The effect of longitudinal and transverse steps of the pipes on the energy efficiency of different configurations would take into account by mean of physical relations between the structure of shell side flow with attached eddies and intensity of transfer processes of heat and momentum. With the aim of energy-efficient placement of tubes, such an approach opens up great opportunities for the synthesis of a plurality of tubular heat exchange surfaces, in particular, the layout of the twisted and in-line-diffuser type with a drop-shaped pipes.

1. Introduction

Heat exchangers with heat transfer surfaces in the form of tube bundles are widely used in many industries: energy, petrochemical, oil and gas processing, food industry, etc. Intensification of heat transfer is the main direction of improvement of heat exchange equipment. The choice of one or the other method of enhancement includes a study of the thermal and hydrodynamic characteristics, the analysis of which allows to choose the optimal ratio between working surfaces, the heat exchange efficiency and pressure loss. In the work connected with the creation of a new efficient heat exchange surfaces, the need for comparative evaluation occurs when one have received first data on heat transfer and resistance, to determine the ways for further research. Thus, the problem of comparative evaluation of the effectiveness of the heat exchange surfaces at the stage when the usual methods of feasibility analysis can not be used, is relevant.

For a quantitative estimation of various ways of an intensification of a convective heat transfer use concept of power efficiency of heat exchange surfaces which is presented by the power factor E - the relation of thermal flow Q, transferred by a surface, to power N spent for overcoming hydrodynamic resistance of heat-transfer fluid flow [1]
The value \( E \) is used for evaluation of thermalhydraulic efficiency of heat transfer surfaces. From two compared surfaces that will be more effective which power factor is more, as at equal hydraulic losses, i.e. at equal powers to heat-transfer fluid pumping, a surface can transfer more heat.

Using the data on heat exchange and resistance, for example, in the form of dependences \( St \) (Re) and \( \xi \) (Re), and also assuming that all thermal resistance is concentrated on the side of one of heat-transfer fluids, it is possible to present components of power factor as follows:

\[
Q = \alpha \cdot \Delta t \cdot F = St \cdot Re \cdot F \cdot \rho \cdot C_p \cdot \nu \cdot \Delta t / d_e
\]

\[
N = \Delta p \cdot f \cdot w = \xi \cdot \rho \cdot w^2 \cdot L / 2d_e = \xi \cdot Re^3 \cdot F \cdot \rho \cdot \nu^3 / 8d_e^3
\]

where \( \Delta t \) – a temperature pressure; \( St = \alpha / \rho C_p w \) – Stanton number; \( C_p \) – average isobaric thermal capacity; \( \rho \) – density; \( w \) – average velocity; \( Re = wd / \nu \) – Reynolds number (\( \nu \) – kinematic viscosity); \( d_e = 4l / F \) – equivalent diameter (\( f \) – cross-sectional area, \( L \) – length of a surface), \( \Delta p = \xi \rho w^2 L / 2d_e \).

The intensification of heat exchange by increasing of speed of heat-transfer fluid would be the most simple way to raise the power efficiency of heat exchange surfaces. However, from above relationships it follows that with growth of speed the power factor decreases, as the numerator grows approximately proportionally speeds, and a denominator (at a turbulent flow) – is proportional to the third extent of speed. Therefore by working out of new effective heat-exchange surfaces are aimed to increase intensity of heat exchange at losing growth of resistance. Thus, the correct estimation of various alternatives of the convective heat-exchange surfaces is of great importance, and final conclusions about efficiency of application of this or that way of an intensification under the same conditions can be gained only comparing various heat exchange surfaces on one of existing methods.

Various methods of comparison of heat exchange surfaces were offered and analysed by many researchers [2-8]. Many criteria of an estimation are offered: power, thermodynamic, economic. The usual method of technical-economical analysis is based on comparison of one of economic criteria, more often – the resulted annual charges. However this method can be used only for an estimation of such heat-exchange surfaces which are produced in lots and concerning which operating experience is saved up. Questions of efficiency of heat exchange surfaces is connected in present with active search of new ways of an intensification of heat exchange. From the latest works on the given subjects it is possible to gate out [9-11].

2. Method comparison "with other things being equal"

At a comparative estimate of efficiency of surfaces of heat exchange a thermal flow (Q, W), the power spent for the heat transfer fluid transportation (N, W), and a heat exchange surface area (F, m²) are considered as the cores characteristics. Thus the most correct requirements of comparison are provided with comparison "with other things being equal" [2]. Proceeding from it, we have following variants of comparison:

- On a thermal flow (at identical powers and the areas);
- On power (at identical thermal flows and the areas);
- On the area (at identical thermal flows and powers).

Values of three measure of efficiency are accordingly determined:

on thermal flow \( K_Q = \frac{Q_2}{Q_1} \), on power \( K_N = \frac{N_2}{N_1} \), on area \( K_F = \frac{F_2}{F_1} \).

Further consider the tubular heat exchange surfaces cross-flowed by a gas which are widely used in various heat exchanges. For cross-flowed tube bundles thermalhydraulics properties one have to make certain demand - configuration of a tube bundle should be energy effective.
For an estimate of energy efficiency of tube bundles we will use the sedate relations featuring a convective heat exchange and hydraulic resistance \( \text{Nu} = C \cdot \text{Re}^a \), \( \text{Eu} = A \cdot \text{Re}^b \) [12, 13]. Then for the relative quantities of a thermal flow \( K_Q \) and power \( K_N \) it is gained such expressions:

- for \( K_N = 1 \), \( K_F = 1 \)

\[
K_Q = \left( \frac{C_2}{C_1} \right) \left( \frac{A_1}{A_2} \right) \cdot \left( \frac{a_1 - 1}{a_2 - 1} \right)^{\frac{m_2}{k_2+3}} \cdot \text{Re}_1^{\left( \frac{k_1+3}{k_2+3} (m_1^{-m_2}) \right)}
\]

- for \( K_N = 1 \), \( K_F = 1 \)

\[
K_N = \left( \frac{A_2}{A_1} \right) \left( \frac{a_2 - 1}{a_1 - 1} \right) \cdot \left( \frac{C_1}{C_2} \right)^{\frac{k_2+3}{m_2}} \cdot \text{Re}_1^{(k_2+3)\left( \frac{m_1}{m_2} \right) - (k_1+3)}
\]

In formulas subscript «1» corresponds to a reference surface of heat exchange, and a index «2» - an investigated surface. Besides further following labels will be also used: \( a = S_1/d \) – a relative cross-flow spacing, \( b = S_2/d \) – a relative longitudinal spacing of tubes in bundle (figure 1). The surface with an index «2» is more effective than a surface with an index «1», if \( K_Q > 1 \), and \( K_N < 1 \). As follows from abovementioned formulas, values of \( K_Q \) and \( K_N \) depend on initial value of \( \text{Re}_1 \). Therefore results of comparison should be considered at certain range of a Reynolds number.

On the stated procedure comparison of energy efficiencies of surfaces of bundles of smooth tubes traditional (staggered and in-line) configurations "with other things being equal" is done.

![Figure 1](image-url)  
**Figure 1.** Traditional configurations of tube bundles: a) in-line; b) staggered

Hydrodynamics and heat transfer in such bundles are well explored [14]. Therefore, using the well-known data on heat transfer and hydraulic resistance, it is possible to gain reliable results of energy efficiency and, thus, to check up the developed procedure.

### 3. Results of comparison

For the base configuration the in-line tube bundle with the relative spacings \( a \times b = 2 \times 2 \) and outer tube diameter \( d = 31 \text{ mm} \) is considered with Reynolds number \( \text{Re}_d = 10^5 \). It was necessary that the areas of reference and compared surfaces of heat exchange are identical, i.e. \( F_1 = F_2 \).

The results of calculations of energy efficiency in the form of dependence on compactness of in-line bundle (the relation of the heat transfer area to volume) \( \beta = \frac{F}{V} = \frac{\pi}{a \cdot b \cdot d} \) (m²/m³) are presented at figure 2 (a, b), and for staggered – at figure 3 (a, b).
The comparison on the heat flow $Q$

The comparison on the spent power $N$

**Figure 2.** Energy efficiency of in-line tube bundles via compactness $\beta$. 
4. Results and discussion

The analysis of results of calculations shows that energy efficiency of in-line and staggered configurations increases with reduction of spacing of tubed in a bundle. The most effective are the closest bundles.

For staggered configurations it has been confirmed by experimental researches thermal-aerodynamics characteristics at small relative spacing, $a \times b = 1.051 \times 0.910$ and $a \times b = 1.027 \times 0.889$ [15, 16]. Convective heat exchange pinch on 27 % and aerodynamic drag decrease on 7 % has been obtained. The conducted investigations have shown possibility of pinch of energy efficiency of staggered bundles at the expense of reduction of the equivalent diameter of a flow section.

At reduction of a cross-flow spacing energy efficiency of both configurations increases. Influence of a longitudinal spacing is not so unequivocal. If for staggered configuration reduction of a
longitudinal spacing raises efficiency of a bundle for all values of a cross-flow spacing, for in-line configurations reduction of a longitudinal spacing leads to pinch of efficiency of bundles with increased cross-flow spacing (a=2,0), and at reduction of a cross-flow spacing influence of the longitudinal spacing weakens and at the most small cross-flow spacing (a<1,1) disappears. Thus influence of a longitudinal spacing on energetic characteristics at large cross-flow spacing for in-line configuration bigger than for the staggered.

Scoring on a thermal flow Kn>1 (at Kν=1 и Kτ=1) for in-line configuration is more essential at reduction of cross-flow spacing, than longitudinal. Scoring on power on propulsion Kn<1 is reached equally as at reduction cross-flow, and longitudinal spacing.

In-line configuration has restrictions on parameters Kν and KN. It is related to convective heat transfer decrease at reduction of longitudinal spacing that it is possible to explain formation of stagnant vortex fields (effect of "shadowing") between tubes of longitudinal rows.

The analysis of influence of cross-flow spacing on thermal-aerodynamics characteristics of bundles was conducted on the basis of known similarity equations [14, 17].

So for staggered bundles at an arrangement of tubes in vertexes of equal sides triangles the aerodynamic drag was analyzed on a relation for Euler number Eu ~ (a–1)0,25; for in-line bundles– on relation Eu ~ (a–1)0,36, and for squeezed in-line bundles on a relation Eu ~ (a–1)1,31. At the convective heat transfer analysis it was considered that for deep rows of tubes in-line bundles the convective heat transfere does not depend on tube’s spacing, and for the staggered was estimated on a relation for Nusselt number Nu ~ (a/b)0,2.

Using known advantages of in-line configuration or its updatings (considerably smaller aerodynamic drag in comparison with staggered configuration), it is possible to increment number of rows of tubes, maintaining aerodynamic drag the same. Such idea demands, of course, checkout on the basis of the analysis of energy efficiency. However in respect of statement of a problem of working out of highly effective tube’s surfaces of heat exchange, the initial position select has crucial importance.

5. Conclusion
Carried out performance comparison "ceteris paribus" in-line and staggered configurations of bundles with a circular tubes can serve as a basis for the development of physical models of flow and heat transfer in tube bundles with tubes of other geometric shapes, considering intertubular stream with attached eddies that were partly confirmed in works [18–20].

The effect of longitudinal and cross-flow spacing of the tubes on the energy efficiency of different configurations would take into account by mean of physical relations between the structure of shell side flow with attached eddies and intensity of transfer processes of heat and momentum. With the aim of energy-efficient placement of tubes, such an approach opens up great opportunities for the synthesis of a plurality of tubular heat exchange surfaces, in particular, the layout of the twisted and in-line-diffuser type with a drop-shaped tubes.

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