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# The development of long-range heat transfer surfaces for marine diesel engine charge air coolers

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**Abstract**

Charge air cooling is essential for the efficient operation of marine diesel engines. This work presents the results of research on the characteristics of long-range heat transfer surfaces for marine diesel engines. Elliptical and flat-oval tubes were considered. This study was carried out using mathematical models that consisted of the equations for energy conservation, motion, continuity, and state. The RSM turbulence model was used to close the system of equations. To solve the resulting system of equations, the RANS approach was used, which was implemented in the software package Code Saturne with a free license and the SimScale cloud service. The mathematical model was verified by comparing the model results with the experimental results obtained using a prototype heat-exchange surface of a charge air cooler at a test bench. The discrepancy between the theoretical and experimental heat transfer coefficient $\alpha$ was ≤ 8.3%. An estimate of the compactness of smooth elliptical and flat-oval tube banks compared with round ones was carried out. A 19.6% increase in compactness was obtained for elliptical tubes and 17.5% for flat-oval tubes. Based on the profiled finning surfaces, it is possible to improve their thermohydraulic characteristics by up to 40% when using them together with elliptical tubes compared with round ones and up to 26% when using them with flat-oval tubes.

## Introduction

Modern marine power engineering is based on diesel engines, which are used both as the main engines and drive motors for diesel-driven alternators (Significant Ships, 1991–2017). Modern marine diesel engines are combined and cannot operate without gas turbine charging. Increasing the density of the air charge in the compressor, followed by its effective cooling in the charge air cooler, helps increase the effective engine power, reduce specific fuel consumption, and also reduce harmful emissions in the exhaust gases.

The pressure ratio of the charge air in the modern turbochargers of marine diesel engines reaches 4–5, which increases the charge air temperature to

220–260°C. Since the temperature of the charge air entering the engine cylinders should be in the range of 30–40°C, it is necessary to cool it, which is carried out using charge air coolers. The heat transfer surfaces of such coolers are formed mainly from finned circular tubes, which makes it difficult to create compact heat exchangers (Bazhan, 1981). The advantage of tubular heat exchangers is their simple design and reliability; however, an increase in engine power increases the weight and size of coolers, which makes it difficult to create compact power plants.

It has been experimentally shown (Bazhan, 1981) that a 10°C decrease in temperature in the charge air cooler helps reduce the specific fuel consumption of the engine by 2 g/(kW·h). Pre-estimate calculations carried out by the authors of the operating cycle of a MAN B&W S60ME engine showed that by increasing the thermal efficiency of the cooler, the fuel consumption can be decreased by 1.4–1.6 g/(kW·h), and an additional decrease in consumption by 0.3–0.4 g/(kW·h) can be achieved by reducing the resistance of the heat exchanger; thus, it is necessary to consider the heat transfer methods and their intensification, in which an increase in heat transfer exceeds the increase in hydrodynamic resistance required to achieve it.

The formation of heat transfer surfaces based on circular tubes with discrete roughness (Gaus & Savicheva, 2020), which is performed by rolling various configurations, has been proposed. The proposed heat transfer surfaces require additional research to determine how their heat transfer characteristics depend on the rolling geometry.

For additional cooling, evaporative cooling was proposed (Somwanshi & Sarkar, 2020), but this method is difficult to use in transport power plants. The results of mathematical and physical modeling of a hybrid air cooler were presented.

For use in charge air coolers, smooth-tube and finned heat exchangers made of a material with a high thermal conductivity coefficient have been investigated (ACT, 2020; Kelvion, 2020). In such heat exchangers, an increase in resistance exceeds the increased heat transfer during heat transfer intensification, which makes it difficult to create compact heat exchangers.

Packets of flat-oval pipes with dimples have been proposed for use as heat transfer surfaces made of non-circular pipes (Kondratyuk, Pis'mennyi & Terekh, 2015; Khalatov, Kovalenko & Meyris, 2017); however, in these investigations, there was no comparison of the compactness characteristics with other types of pipes with non-circular cross-sections.

As a profiled finned surface (Kuntysh et al., 2012), it was proposed to use a tube with a circular cross-section with *L*-shaped fins with dimples; however, the proposed system of dimples, applied along the entire fin, is extremely low-tech, which will complicate the production of such fins. In addition, there are no recommendations for selecting dimple parameters depending on the conditions of use.

Dolphin Company (Dolphin Manufacturing LLC, 2020) offers heat exchangers with pipes with round and flat-plane cross-sections, which are finned with solid ribs for use as charge air coolers; however, the reduction of edge resistance as the main source of hydrodynamic resistance was not considered.

To estimate the compactness of the heat transfer surface (Wong, 1977), the coefficient of geometric compactness is used:

$$K_{\text{geom}} = \frac{F}{V} \tag{1}$$

where $F$ and $V$ are the area and volume of the heat transfer surface (m$^2$ and m$^3$), respectively.

In the open literature, many parameters have been proposed for estimating the heat transfer efficiency. For the flow conditions in tube banks, the Reynolds analogy factor is (Khalatov, 2005):

$$\text{FAR}_{\alpha} = \frac{\dfrac{Nu}{Nu_0}}{\dfrac{f}{f_0}} \tag{2}$$

where $Nu$ is the Nusselt number, $f$ is the drag coefficient, the index 0 corresponds to a cylindrical channel on both sides (as the most thermohydraulically investigated). A modified Reynolds analogy factor (Kuznetsov, 2020) can also be used

$$\text{FAR}_{k} = \frac{\dfrac{k}{k_0}}{\sum \dfrac{Eu}{Eu_0}} \tag{3}$$

where $k$ is the overall heat transfer coefficient (W/(m$^2$·K)), $Eu$ is the Euler number, index 0 corresponds to a cylindrical channel on both sides (as the most thermohydraulically investigated).

The analysis showed that, for use as heat transfer surfaces in recuperative charge air coolers, banks of smooth and profiled pipes with round sections and finned pipes are recommended, in which the increase in resistance exceeds the increase in heat transfer. The main source of resistance of the heat transfer surface is the ribs, which makes it difficult to create compact heat exchangers. The use of finned pipes with a flat-plane cross-section is recommended, but methods for reducing the resistance of the ribs are not considered.

## Methodology

The aim of this work was to estimate the thermal and hydraulic efficiency of using finned-profiled heat transfer surfaces with non-circular cross-section tubes for designing compact charge air coolers for marine diesel engines.

For this purpose, it is necessary to solve the following problems:

- To justify and verify the mathematical model for researching long-range heat transfer surfaces of charge air coolers.
- To make a comparative estimation of the compactness of profiled smooth elliptical and flat-oval banks of tubes compared with round-section tubes.
- To conduct a comparative estimation of the thermohydraulic efficiency of profiled elliptical and flat-oval tube banks compared with round-section tubes.

The *object of the research* is the processes of heat exchange in surface charge air coolers.

The *subject of the research* is the geometric parameters of surface compactness and their relation to heat exchange and the hydrodynamic parameters of the heat conversion processes occurring in charge air coolers.

*The research method* is the geometric modeling of the elements' location of the heat-transfer surface in the tube bank, and mathematical and physical modeling of heat transfer processes on heat transfer surfaces.

## Simulation model

Mathematical modeling of the processes was carried out based on the numerical solution of the equations of conservation of energy, motion, continuity, and state. To close the system of equations based on the recommendations of (Bystrov et al., 2005), the RSM turbulence model was used. To solve the resulting system, the RANS approach was used, which was implemented in a software package with a free license, Code Saturne (Code_Saturne, 2020), and the SimScale cloud service (SimScale, 2020).

The mathematical model was verified by comparing the results of test modeling with the results of testing a prototype of the heat exchange surface of the charge air cooler (CAC) at the specialized test bench of the Department of Internal Combustion Engines, Plants and Technical Exploitation of the Admiral Makarov National University of Shipbuilding.

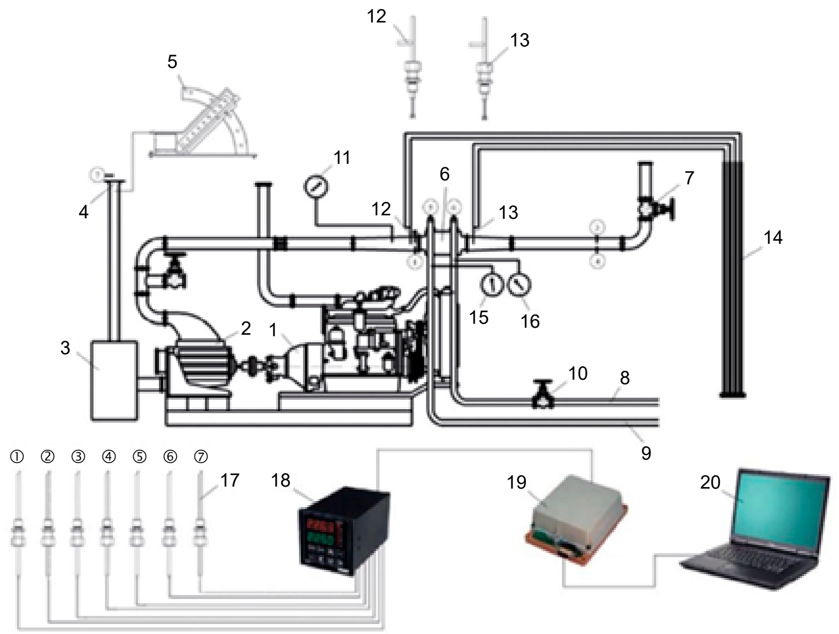


**Figure 1. A schematic of the test bench: 1 – diesel engine; 2 – rotary-vane compressor; 3 – cistern; 4 – flow restrictor; 5 – micromanometer MMH-240; 6 – experimental CAC; 7 - orifice valve; 8 – inlet pipe of the water to CAC; 9 – outlet pipe of the water from CAC; 10 – slide-valve for the flow control through CAC; 11, 15, 16 – model manometers; 12, 13 – static-pressure tubes; 14 – block water piezometer tubes; 17 – Chromel-Copel thermocouples; 18 – eight-channel temperature monitoring device "OVEN"; 19 – interface adapter RS232 "OVEN" AC-2; 20 – PC**

The test bench scheme is shown in Figure 1. The source of compressed air at the test bench was a rotary vane compressor (2), which was driven by a direct drive from a diesel engine (1) SMD-19. The capabilities of the stand made it possible to obtain air with an absolute pressure of 20–160 kPa and a flow rate of 0.1 to 0.5 kg/s, while the maximum air temperature ahead of the front of the experimental CAC was 110–130°C.

The scheme of the heat transfer surface is shown in Figure 2. The heat-transfer surface was an aligned bank of flat-oval tubes with continuous finning by flat transverse plates. The fins had a transverse pimple in the form of a triangular protrusion (Figure 2).

The heat transfer surface had the following geometric parameters: height of the tube cross-section $d_w$ = 3.8 mm; distance between the tubes in the cross-row $S_1$ = 10 mm; step between the cross-rows of tubes $S_2$ = 23 mm; the largest dimension of the tube cross-section $S_3$ = 17 mm; step between the ribs $S_4$ = 2.05 mm; thickness of the fin plate $\delta_{pl}$ = 0.1 mm; beam height $H_P$ = 110 mm; beam length $L_P$ = 184 mm; beam width $B_P$ = 100 mm.

Based on the test results, the criterion equation was obtained:

$$Nu = 0.4613 \cdot \mathrm{Re}^{0.4727} \tag{4}$$

The geometric dimensions of the calculated geometric model used to verify the mathematical model were the same as those of the prototype. The mathematical model verification results are presented in Figure 3.

The discrepancy between the theoretical and experimental heat transfer coefficients α did not exceed 8.3%, which makes it possible to use the obtained mathematical model to further investigate the heat-hydraulic characteristics of heat exchange surfaces.

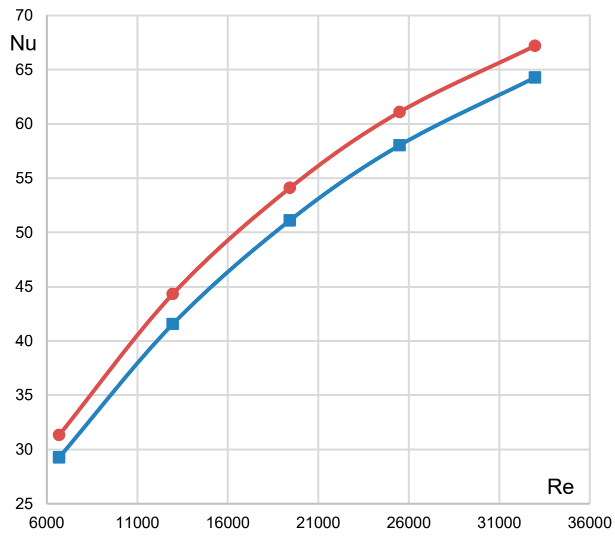


**Figure 3. The results of mathematical model verification: ■ – values obtained from the equation (4); ● – simulation results**

## Results

To form heat transfer surfaces, the concept of "good" and "bad" streamlined surfaces obtained from classical hydromechanics was used. Tubes with flat-oval and elliptical cross-sections were considered as "well"-streamlined surfaces, which were compared with a "poorly"-streamlined surface – tubes with a circular cross-section.

The heat transfer surface area of the cooler can be represented as the sum of the surfaces of the non-finned and finned parts. Since smooth tubes are the actual basis for the formation of a heat-transfer surface, the compactness of surfaces consisting of smooth tubes is considered. In this consideration, the same surface area and the minimum possible flow area between the tubes were accepted (Figure 4).

Under the accepted conditions, the modified Reynolds analogy factor (2) – the parameter describing the efficiency of the overall heat transfer – $\mathrm{FAR}_k$

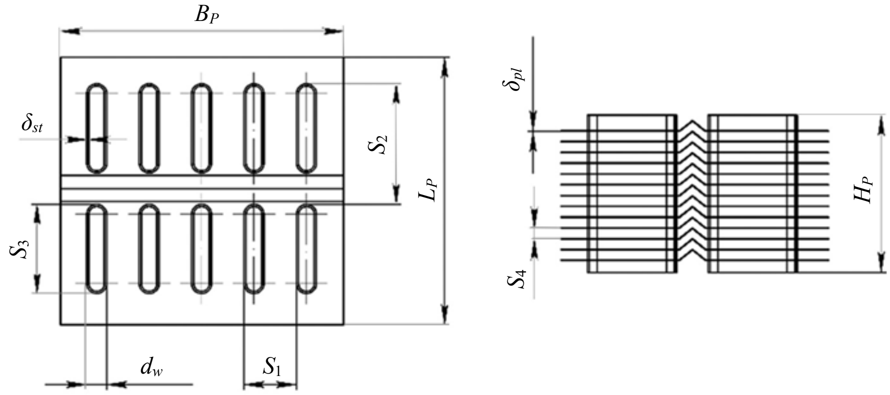


**Figure 2. The scheme of the experimental heat transfer surface element**

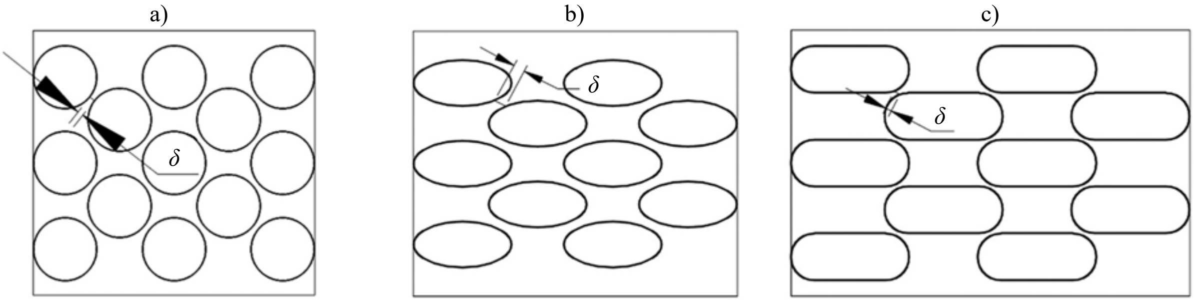


**Figure 4. The smallest possible flow area between tubes in bundles: a) round tubes; b) elliptical tubes; c) flat-oval tubes**

– will be equal to: for round tubes = 1; elliptical tubes = 1.67; flat-oval tubes = 1.35. The geometric compactness parameter will be: for round tubes = 0.094; elliptical tubes = 0.117; flat-oval tubes = 0.114.

The improvement of the finned portion of the heat transfer surface was carried out in the direction of the outstripping growth of heat transfer over the resistance growth. The use of dimple systems provides an effective way to improve this direction (Khalatov, 2005). When the flow passes along an edge with a dimple, a vortex-like structure, such as a natural vortex "tornado", is formed in it. This structure, leaving the dimple, carries the energy of the vortex into the external flow, which prevents flow separation while reducing the hydrodynamic resistance of the fins and the entire surface. According to data (Khalatov, 2005), depending on the size of the dimples, the negative (compared with the atmospheric) static pressure zone ranges from 30 to 80% of the dimple length. The maximum positive static pressure value is located near the trailing edge of the recess, after which it drops sharply due to flow separation, before becoming negative again. This allows for an increase in heat transfer over an increase in resistance.

The location of the dimples is determined as follows. We represent the fin in the form of a plate and determine the change along the length of the plate in the thickness of the laminar and turbulent boundary layers and the heat transfer coefficient. The following equations were used to calculate:

- The thickness of the laminar boundary layer

$$\delta_l(x) = \frac{4.64 \cdot x}{\sqrt{\mathrm{Re}_x}}$$

- The thickness of the turbulent boundary layer

$$\delta_t(x) = \frac{0.37 \cdot x}{\sqrt[5]{\mathrm{Re}_x}}$$

- The heat transfer coefficient

$$Nu_x = 0.0255 \cdot \mathrm{Re}_x^{0.8}$$

These equations were based on recommendations by (Wong, 1977).

The flow parameters – the air temperature $t_f$ =150°C and speed $w_f$ = 15 m/s – were taken from (Bazhan, 1981). The calculation results are shown in Figure 5.

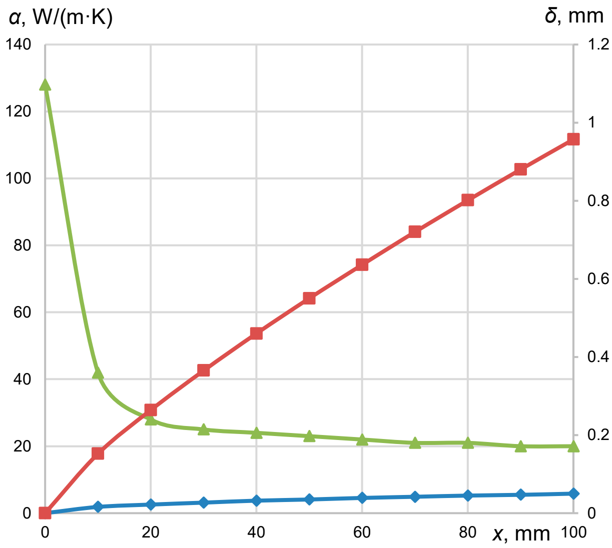


**Figure 5. Changes in the characteristics of the boundary layer when flowing around the plate: ▲ – heat transfer coefficient; ♦ and ■ – the thicknesses of the laminar and turbulent boundary layers respectively**

Analysis of the diagrams shows that the most significant change in the heat transfer coefficient and thickness of the laminar boundary layer occurs in the section from the beginning of the plate to $x$ = 10 mm. At a distance of $x$ = 10–20 mm, the decrease in the heat transfer coefficient stabilized and remained practically constant starting from the middle of the plate; therefore, the dimple layouts in Figure 6 were used.

Figure 6a shows a variant of the location of the dimple systems on a bank of flat-oval ribbed tubes, similar to that used in the mathematical model verification. The dimples are spaced 10 mm apart. Based on the geometrical dimensions of the model, the parameters of the dimples were taken according to

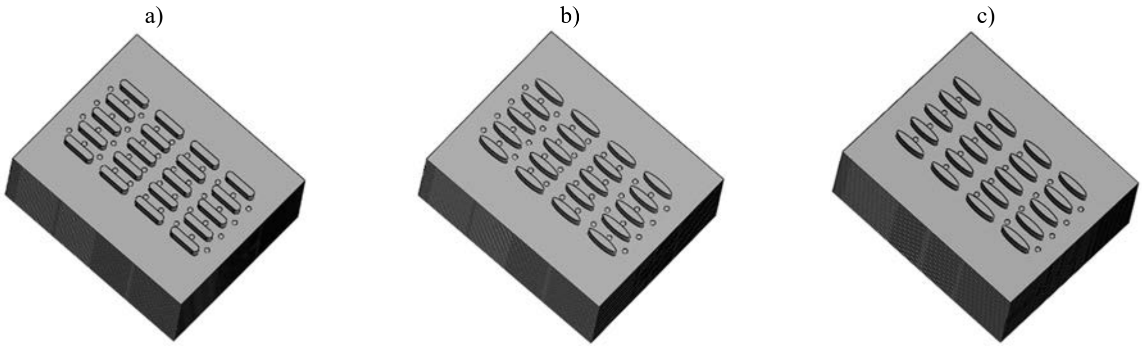


**Figure 6. Layout schemes for dimples on the fins of profiled surfaces**

the recommendations of (Khalatov, 2005). Figure 6b and 6c show the considered options with elliptical tubes. The hydraulic diameter and layout of the elliptical tubes were similar to that of the flat-oval layout. The differences lie in the step of the dimples – in Figure 6b – 10 mm, and 6c – 20 mm.

The calculation results are presented in Figure 7 as the dependence of the Reynolds analogy factor $FAR_{\alpha}$ on Re.

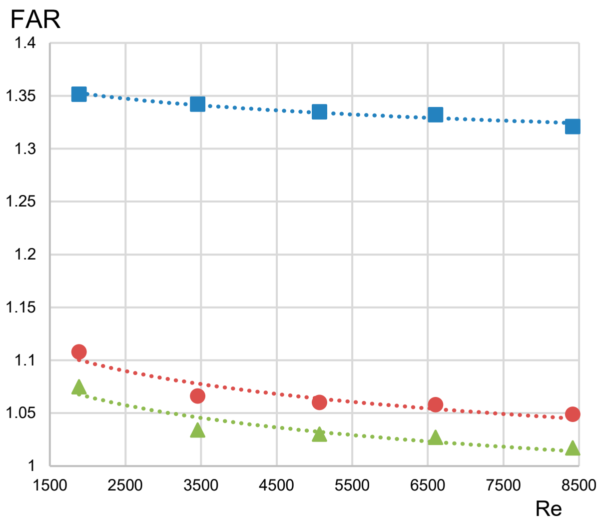


**Figure 7. The efficiency of using the profiled heat exchange surfaces: ■ – $FAR_{\alpha 1}$; ● – $FAR_{\alpha 2}$; ▲ – $FAR_{\alpha 3}$**

The results were calculated according to dependence (2) and are presented as follows: $FAR_{\alpha 1}$ – the efficiency of using smooth elliptical pipes compared with flat-oval pipes; $FAR_{\alpha 2}$ – the efficiency of using flat-oval tubes with dimples on the fins (Figure 6a) compared with tubes with a smooth fin; $FAR_{\alpha 3}$ – the efficiency of using elliptical tubes with dimples on the fins (Figure 6b) compared with tubes with smooth fins. The efficiency of variant 6c was ±3.2%, similar to variant 6b; therefore, it is not shown on the diagram.

A feature of the obtained results is the use of tubes with non-circular cross-sections to form a heat transfer plate-finned surface for charge air coolers. At the same time, due to the better hydrodynamic flow around the surface, the bank efficiency with smooth elliptical tubes was 31–35% higher than those with flat-oval ones; however, the use of dimple systems showed better efficiency on a flat-oval bank; thus, the final choice of the surface and layout of the dimple systems should be based on the desired design and technological production capabilities.

Further results will be aimed at investigating the effectiveness of various shapes and geometrical sizes of dimples in the conditions under consideration.

The obtained results can be used not only to create a heat exchange surface for charge air coolers for marine diesel engines, but also for other transport engines – locomotives, automobiles, as well as stationary diesel power stations – since they can reduce the weight and size of the coolers. In addition, the heat exchange surface investigated by the authors can also be used to create radiators operating in the cooling systems of internal combustion engines in various marine infrastructure power plants.

## Conclusions

The mathematical model used to investigate the thermohydraulic characteristics of charge air coolers for marine diesel engines has been determined. It was verified using the experimental data obtained at a test bench, which confirmed the possibility of its further use.

A comparative estimate of the compactness of smooth elliptical and flat-oval tube bundles compared with round ones was carried out. The increase in compactness was 19.6% for elliptical tubes and 17.5% for flat oval tubes.

It has been established that based on the profiled finning surfaces, it is possible to improve the thermohydraulic characteristics by up to 40% when used together with elliptical tubes instead of round ones and by up to 26% when using flat-oval tubes.

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