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EXPERIMENTAL STUDIES OF HEAT EXCHANGE AND AERODYNAMICS OF THE EFFICIENT METHODS FOR HEAT EXCHANGE INTENSIFICATION

The paper estimates heat exchange intensification and pressure losses from the application of turbulizing inserts with round and rectangular openings in the round channel of gas-water heat exchanger. Experimental studies have been performed. Dependencies for intensified heat exchange calculation are proposed.

Keywords: *intensification, pressure losses, Nusselt number, heat exchange.*

Introduction

Systems with microfluidic elements (impact jets) are rather efficient for cooling systems. Under optimal conditions jet blasting provides 3 – 5 times growth of heat transfer intensity as compared with longitudinal flow over the surface with relatively low power consumption. These systems are used for cooling gas turbine components and in other engineering fields. Investigation of the flow structure in such systems have shown the existence of double-vortex structures, which are generated in jet mixing layers in the form of vortex cords, and then are destroyed due to their instability and take a sinusoidal shape. As it is shown in the works of Dyban, all this facilitates heat transfer intensification [1].

Application of such method for designing various heat transfer elements, particularly, for heat transfer intensification in heat exchangers of low-power water boilers operating on natural gas or biogas, is an important current problem.

The aim of this work is experimental study of the efficiency of original heat transfer intensification methods – application of inserts with round openings and rectangular slots.

Research results

The authors have verified the efficiency of inserts of original design, patented by them [2]. Two series of experiments on the investigation of efficiency of the inserts with round openings and rectangular slots have been carried out (Fig. 1).

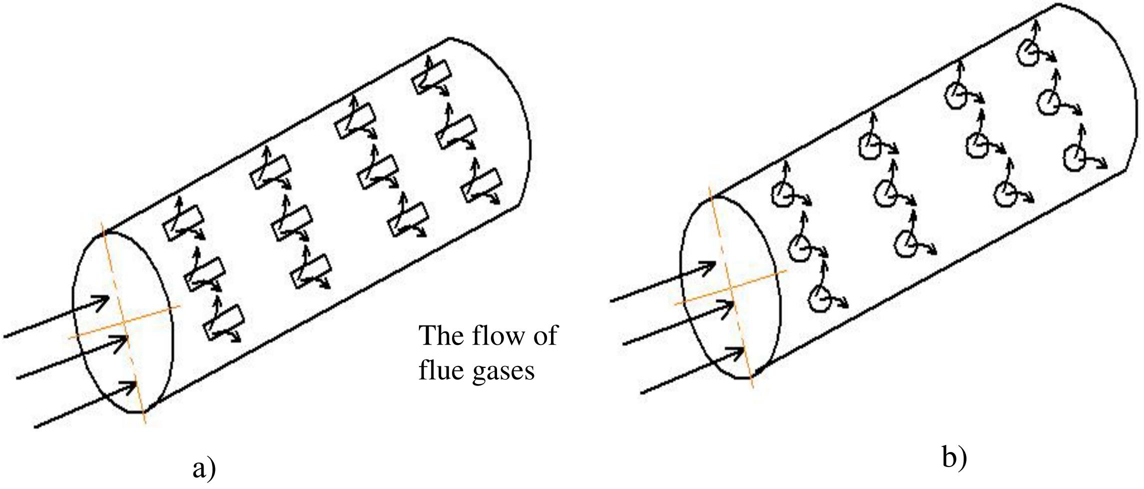
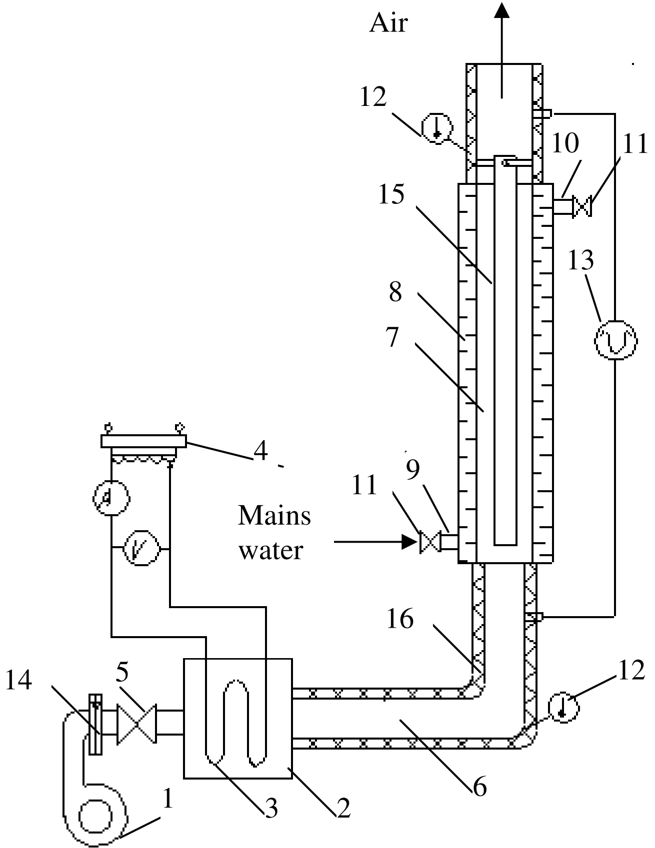


Fig. 1. Heat transfer intensifiers with rectangular slots (a) and round openings (b)

Test stand

Installation, principal circuit of which is shown in Fig. 2, includes the following main components: air heater and heat exchange element. Air is supplied to the air heater by fan 1. Air heater consists from air heating chamber 2 and electric heater 3. Voltage is supplied to the heater and regulated by means of power unit 4 ЛИПС-68, the load is controlled by laboratory ammeter Э59 and voltmeter Щ4313. Air flow rate is regulated by ball valve 5, mounted in front of the air heater, and is measured by rotameters PM4 and PM0,4. Air, heated to the required temperature in the air heater, is supplied to the heat exchange element by means of air blower 6, made from the pipe DN 32. The heat exchange element includes vertical round air-passing channel 7 with inner diameter of 42 mm and water jacket 8 with equivalent water-passage diameter of 11 mm. Heat exchange intensifiers are fixed in vertical channel 7. Mains cold water is supplied to the water jacket through lower nozzle 9 and, after heating, exits through upper nozzle10. Mains water flow rate is regulated by valve 11 and is measured by means of a measuring tank. Air temperature at the input and output of the heat exchange element is measured by thermometers 12 with scale division value of 0.1 °C, which are installed in front and behind the heat exchange element. Air pressure losses in the heat exchange element are measured with MCM manometer 13 with measuring range of 5...1960 Pa. To reduce air temperature drop in the air ducts due to the heat loss into the environment, they are covered with insulation layer 16 with the thickness of 9 mm and thermal conductivity coefficient $\lambda = 0{,}038$ W / (m × K).

Tests were conducted in the following parameter variation range: temperature difference 22.8 ... 80 ° C; air flow rate 0.26 ... 12 m^3 / h; mains water flow rate 0.02 ... 0.16 kg / s; inlet air temperature 75 – 135 ° C; outlet air temperature – 67 ° C; Reynolds number Re = 300 – 5500 .

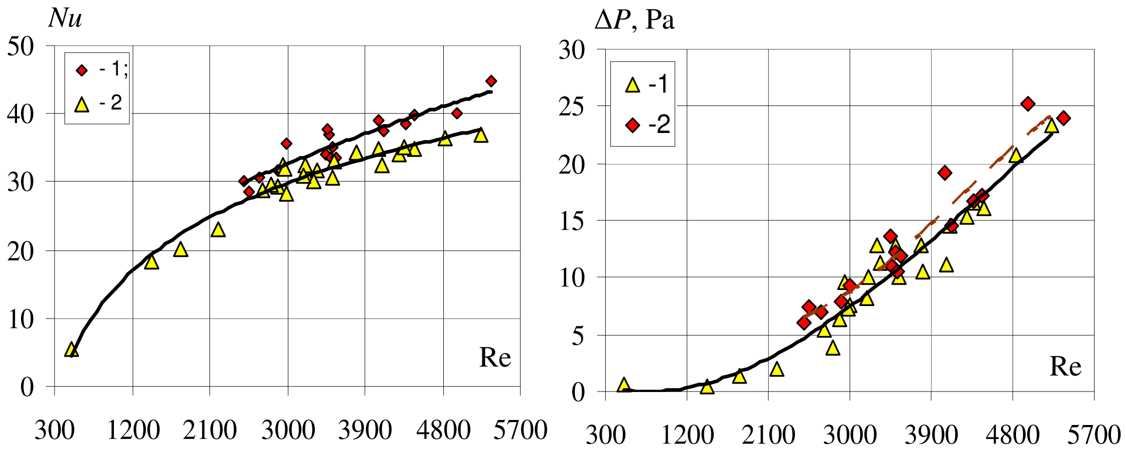


1 – fan;
2 – air heating chamber;
3 – electric heater;
4 –power unit;
5 – ball valve;
6 –air duct;
7 – heat exchange element;
8 – water jacket;
9 – lower nozzle;
10 – upper nozzle;
11 – mains water flow rate regulation valve
12 – thermometer;
13 – micromanometer;
14 – rotameter;
15 – heat exchange intensifier;
16 – heat insulation

Fig. 2. Test stand circuit

Two series of experiments for studying the efficiency of inserts with round openings and rectangular slots were conducted (Fig. 3, Fig. 4). Inserts are mounted in the upper portion of the heat transfer pipe. Ratio of the length of the insert with rectangular slot to the duct length is L_{ins} / L_{duct} =

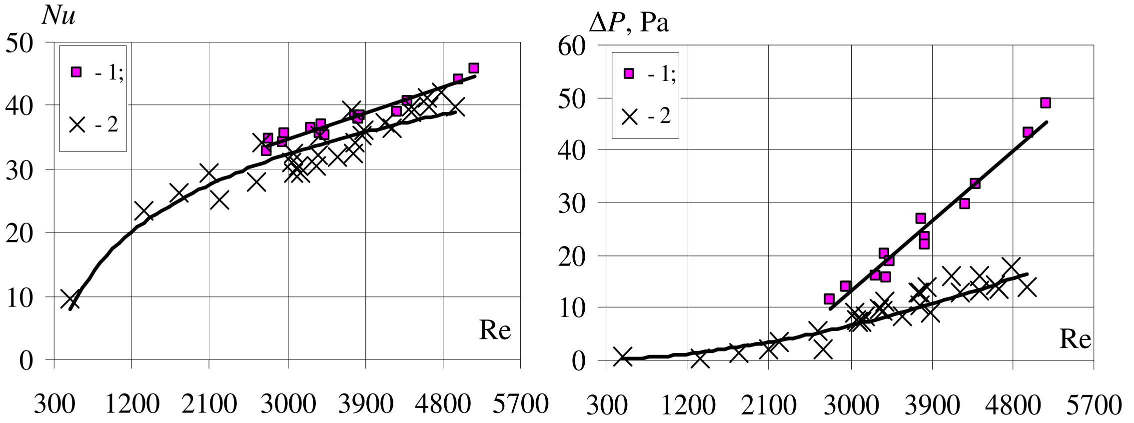
0. 4; for the insert with round openings this indicator is $L_{ins} / L_{duct} = 0.59$.



a) b)

Fig. 3. Dependence of Nusselt number (a) and pressure losses in the heat exchange element (b) on Reynolds number for inserts with round openings:
1 - d = 8 mm; 2 - d = 9 mm

The number of round openings in the experiment was 24. Inserts with smaller diameter of the openings d = 8 mm are more efficient in terms of heat exchange (Fig. 3). When cross-section of the openings was increased 3 – 5 times 1. 27 times (from d = 8mm to d = 9 mm), pressure losses decreased 1.07 – 1.2 times and Nusselt number value Nu – 1.09 – 1.18 times.

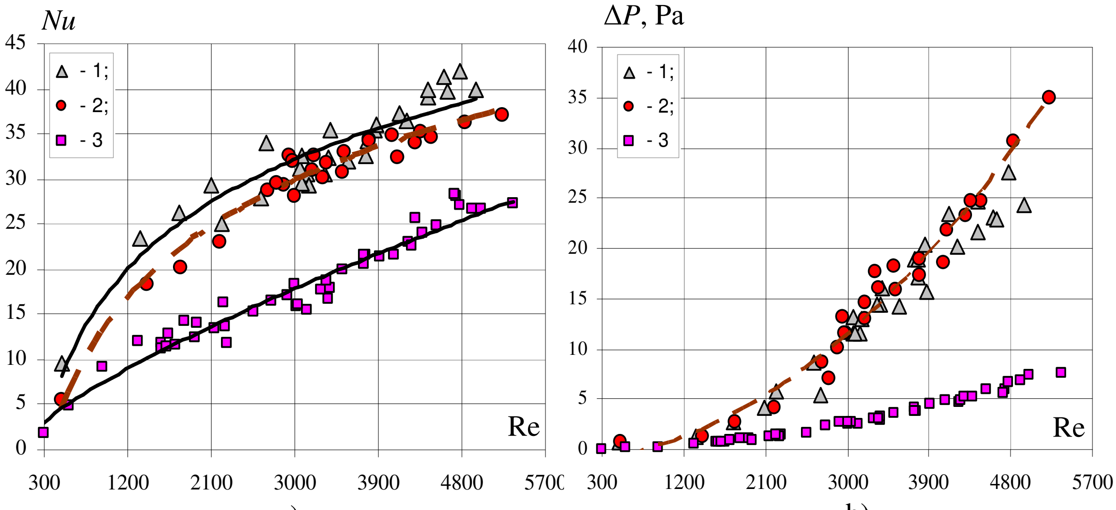


a) b)

Fig. 4. Dependence of Nusselt number (a) and pressure losses in the heat exchange element (b) on Reynolds number for inserts with rectangular slots: 1 - s × h = 1 × 37 mm;
2 - s × h = 2 × 50 mm

Inserts with rectangular slots have the following parameters: upper and lower portions have 4 rectangular slots each with the size 1 × 37 mm and total area 2. 98 × 10^{-4} m^2. In the second series of the experiments size of the slots was increased to 2 × 50 mm. In the second series of the experiments Nusselt number decreased 1. 04 – 1. 12 times while pressure losses decreased 2 – 2. 6 times (Fig. 4).

Fig. 5 presents the effect of heat transfer intensification and resistance growth for the pipes with insert, having rectangular slots 2 × 50 mm, and insert with openings d = 9 mm as compared with a smooth pipe.



a) b)

Fig. 5. Heat exchange intensity (a) and pressure losses (b) for the smooth pipe and the pipe with intensifier depending on Reynolds number: 1 – pipe with insert with slots 2×50 mm; 2 – with insert with openings $d = 9$ mm; 3 – smooth pipe

As it is evident from Fig. 5, inserts with slots 2 × 50 mm and openings $d = 9$ mm provide almost the same heat transfer intensification. In the laminar mode inserts with slots increase heat transfer 1.98 – 2.11 times as compared with the smooth pipe with 2.77 – 6.7 times resistance increase, while in the transient mode heat transfer intensity increased 1.49 – 2.11 times with 3.26 – 3.8 times resistance increase respectively. For inserts with openings 1.12 – 1.685 times Nusselt number growth is observed in laminar region with 2.77 – 7.8 times resistance increase, while for transient mode heat transfer intensity increases 1.37 – 1.685 times with 2.77 – 4.57 times resistance increase respectively. Experimental data of studying the efficiency of the insert having original design with round openings are described by empirical dependence with determination coefficient R = 0,92. For the inserts with rectangular slots the following dependence was obtained:

$$Nu = 0{,}407 \cdot Re^{0{,}56} \cdot Pr^{0{,}43} . \quad (1)$$

For inserts with rectangular slots with determination coefficient R = 0. 93 the obtained dependence is given by

$$Nu = 0{,}586 \cdot Re^{0{,}52} \cdot Pr^{0{,}43} . \quad (2)$$

In dependencies (1) – (2) the key size is the pipe diameter, the key temperature is average temperature of the flow and the key speed – average flow rate without taking into account blockage with the intensifier. Thus, application of the inserts with round openings and rectangular slots is an effective method for heat exchange intensification. This effect is explained by jet outflow of the air from openings (slots), which results in intensive turbulence of the flow in the “air – wall” boundary layer of the heat exchanger.

In high-temperature heat exchangers heat transfer will be also increased due to emission from the inserts. They are expedient to be used in hot water boilers of small capacity, especially in those with forced draft.

Conclusions

Investigation of heat transfer intensification using models of heat exchangers makes it possible to obtain, with small material expenses, reliable initial data for the development of more efficient designs of heat exchanging apparatus. The paper studies thermal hydrodynamic efficiency of heat exchange intensifiers in laminar and transient modes of heat carrier motion, which is characteristic for low-power (up to 100 KW) water boilers. Inserts with rectangular slots and round openings provide efficient heat transfer intensification with significant growth of hydraulic resistance. In laminar mode inserts with slots provide 1.98 – 2.11 times heat transfer increase as compared with a smooth pipe which provides 2.77 – 6.7 times growth of resistance. In the transient mode 1.49 – 2.11 times heat transfer intensification is observed with 3.26 – 3.8 times resistance growth respectively. For inserts with openings Nusselt number in the laminar region increases 1. 12 – 1.685 times as compared with a smooth pipe with resistance growth by 2.77 – 7.8 times, while for transient mode 1.37 – 1.685 times growth of heat transfer intensity is observed with resistance increased 2.77 – 4.57 times respectively. Therefore, it is possible to use such inserts in boilers with forced draft. If such inserts are used in high-temperature heat exchangers (fire-tube boilers with forced draft), heat transfer intensification due to emission could be expected.

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