

Heat Transfer Intensification with Fluids' Flowing in Laminar Regime

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Abstract: In the paper the effect was studied of the presence of helical elements, having the role of turbulence promoters, on heat transfer intensification in coaxial tube heat exchangers, with fluids flowing in laminar regime. The quantitative determination of heat transfer intensification in the presence of turbulence promoters placed inside the device inner tube, was performed by calculating, from the energy balance data, the total heat transfer coefficients and reporting the values obtained to the total coefficient values calculated in the absence of promoters.

Keywords: heat transfer, Kenics mixer, laminar flow

1. Introduction

As the values of heat transfer coefficient by convection α are insufficiently high, the heat transfer represents a limiting factor for the performance of heat exchangers. For this reason, any technical measure is useful, which is functionally and economically justified, which is able to improve the level of such performance. In this context, from among the methods that enhance convection, studied and partially implemented in practice, the following can be mentioned: increasing fluid velocity, vibration/pulsation of the fluid or of the heat exchanger surface, the use of electrostatic or electromagnetic fields, injection and absorption of the boundary layer, the use of additives or turbulence promoters (uniform or discrete artificial roughness, turbulence generators) [1,2].

Turbulence promoters cause an additional local turbulence which leads to the transfer speed increase. The heat transfer can be enhanced by modifying the surface state (introduction of artificial roughness - channels, ribs, granular). These modify the boundary layer flow, which leads to the increase of superficial heat transfer and of friction coefficient as well. Also, promoters are used for the fluid movement that create an artificial turbulence throughout its mass. The elements are mounted inside pipes and can be plane spirals, turbulence pads, conical surfaces, twisted metallic bands [2,3].

In all Kenics Series Static Mixers, a patented helical mixing element directs the flow of material radially toward the pipe walls and back to the center. Additional velocity reversal and flow division results from combining alternating right- and left-hand elements, thus increasing mixing efficiency. All material is continuously and completely mixed, eliminating radial gradients in temperature, velocity and material composition. Numerous independent studies have shown that Kenics Static Mixers maximize mixing efficiency without the wasted energy and material blockage typically found in more restrictive motionless mixers. The static mixer is the best choice for demanding mixing applications, such as those involving fluids with extreme viscosity or volume ratios. The static

mixer utilizes cross-stream mixing and flow splitting to achieve very rapid blending. Each element is approximately one pipe diameter in length and consists of multiple intersecting blades, which generate fluid layers as the mixture flows downstream. Each blade features concave construction offering better cross-stream flow than flat blades for superior mixing per unit length in tough, high viscosity ratio applications. Sheets of low viscosity additives are driven along the trough of each blade and abruptly sheared by strong cross-stream velocity gradients as they pass around the upstream surface.

The Kenics Heat Exchanger consists of a continuous string of static mixer elements within each heat exchanger tube. Fluid flow is directed radially toward the pipe walls and back to the element, regardless of velocity. Additionally, momentum reversal and flow division also contributes to the mixing efficiency. All processed material is continuously and completely intermixed to eliminate radial gradients in temperature, velocity and material composition. As a result, Kenics Heat Exchangers provide predictable, controlled mixing, and the most efficient form of thermal transfer available today [4-7].

2. Experimental

The heat exchanger used is made of copper tubes with the following dimensions: the inner pipe diameter $d = 22 \times 1 \text{ mm}$, the outer pipe diameter $D = 35 \times 1 \text{ mm}$, the tubes length $L = 0.75 \text{ m}$.

On the outer surface of the inner pipe a copper coil was wrapped (80 turns) to improve the turbulence intensification of the cold fluid which flows through the intertubular space and whose temperatures are measured at entry and exit of the space, respectively (Fig. 1).

The hot fluid flows through the inner tube, provided with thermometers at ends and with helical elements with the role of turbulence promoters inside.

The helical elements (26 pieces) were made of aluminum sheet with dimension $l \times L \times \delta = 20 \times 32 \times 1.5 \text{ mm}$, 13 twisted to left and 13 twisted to right (Fig. 2).

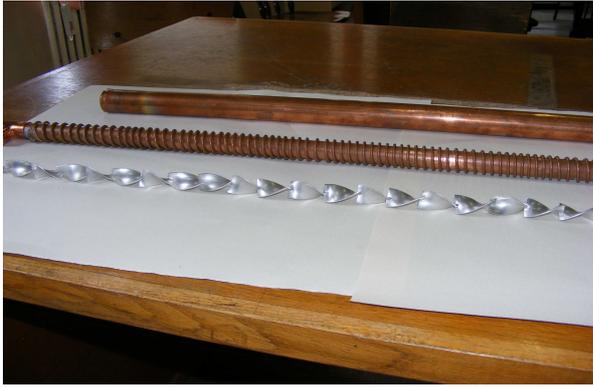


Figure 1. The constructive elements of the heat exchanger



Figure 2. Helical elements as turbulence promoters

The heat exchanger is vertically positioned. Hot water comes from a thermostat, its output temperatures from the thermostat being 60°C and 70°C, respectively. Such hot working fluid flow rates were chosen as the flow be laminar ($Re < 2300$). Cold water comes from the water supply system and was adjusted to a constant flow rate. The situations considered in the experimental determinations have included the parallel and counter flow, respectively, for: a) inner pipe without helical elements; b) inner pipe with 26 elements; c) inner pipe with 12 elements.

TABLE 1. Experimental data for parallel flow, without turbulence promoters

$t_{1h}, ^\circ\text{C}$	$V_{HW} \cdot 10^5, \text{m}^3 \cdot \text{s}^{-1}$	Re	Q, W	$K, \text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$	$\alpha_{HW}, \text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$	$\Delta t_{med}, ^\circ\text{C}$
60	0.972	1081.6	529.2	423.0	787.0	25.3
	1.167	1318.4	537.3	402.4	905.5	27.0
	1.389	1595.4	639.7	466.0	1003.0	27.75
70	0.972	1205.0	712.3	466.1	782.8	30.9
	1.167	1467.0	757.1	475.4	956.8	32.2
	1.389	1773.3	814.1	506.5	1042.0	32.5

TABLE 2. Experimental data for counter-flow, without turbulence promoters

$t_{1h}, ^\circ\text{C}$	$V_{HW} \cdot 10^5, \text{m}^3 \cdot \text{s}^{-1}$	Re	Q, W	$K, \text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$	$\alpha_{HW}, \text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$	$\Delta t_{med}, ^\circ\text{C}$
60	0.972	1063.0	610.6	532.2	767.0	23.2
	1.167	1307.4	659.4	544.2	919.5	24.5
	1.389	1568.6	756.0	613.9	1019.0	24.9
70	0.972	1243.7	895.5	564.1	838.3	32.1
	1.167	1490.5	928.0	590.0	950.0	31.8
	1.389	1737.7	959.5	623.8	1054.4	31.1

3. Results and Discussion

The total heat transfer coefficient is determined from the energy balance equation:

$$Q_1 = Q_2 + Q_3 = Q_4 \quad (1)$$

$$Q_1 = m_{HW} \cdot c_{water} \cdot (\Delta t)_{HW} \quad (2)$$

$$Q_2 = m_{CW} \cdot c_{water} \cdot (\Delta t)_{CW} \quad (3)$$

$$Q_4 = K \cdot A \cdot \Delta t_{med} \cdot \tau \quad (4)$$

where: Q_1 - heat delivered by the hot fluid, W; Q_2 - heat received by the cold fluid, W; Q_3 - the lost heat, W; Q_4 - the transmitted heat, W; m_{HW}, m_{CW} - hot and cold water flow rates, respectively, $\text{kg} \cdot \text{s}^{-1}$; c_{water} - specific heat of water, $\text{J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$; $\Delta t_{HW}, \Delta t_{CW}$ - the temperature differences for hot and cold water, respectively, at the heat exchanger extremities, $^\circ\text{C}$; Δt_{med} - the average temperature difference between the hot fluid and the cold one, $^\circ\text{C}$; K - the total heat transfer coefficient, $\text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$; τ - time, s. The heat transfer area A was calculated with the relationship $A = \pi \cdot d_{med} \cdot L = 49.5 \cdot 10^{-3} \text{m}^2$, where d_{med} is the average diameter of the inner pipe.

Tables 1 and 2 list the values of Reynolds criteria (Re), delivered heat Q and total heat transfer coefficient K , calculated from experimental data, for parallel and counter-flow, respectively, at different flow rates for hot water and different temperatures of it, at the exit of thermostat (in the absence of turbulence promoters). Also, the tables include the values of the partial heat transfer coefficient for the hot fluid (α_{HW}), analytically calculated in the case of heat transfer in laminar regime through vertical pipes, when the forced movement direction is reversed to the free convection ($Gr \cdot Pr > 5 \cdot 10^5$), Gr and Pr being Grashof and Prandtl criteria [8, 9].

Similarly, the delivered heat Q' and total heat transfer coefficient K' from the experimental data and energy balance were determined for the presence of 26 turbulence promoters in the flow space of hot water, and Q'' , K'' respectively, for 12 promoters. For the case of parallel flow, for a thermostat output temperature for hot water of 70°C, the variation of delivered heats and total heat transfer coefficients, respectively, against the number of turbulence promoters placed on the hot fluid path, is shown in Fig. 3 and 4.

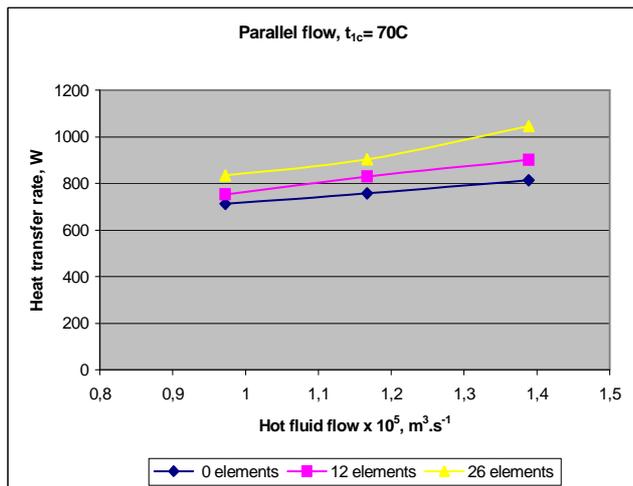


Figure 3. The delivered heat evolution against the hot fluid flow rate and the presence of turbulence promoters

It is observed that the presence of turbulence promoters leads to heat transfer improvement from the hot

fluid to the cold one, the delivered heat and the total heat transfer coefficient increase as the number of promoters' increases.

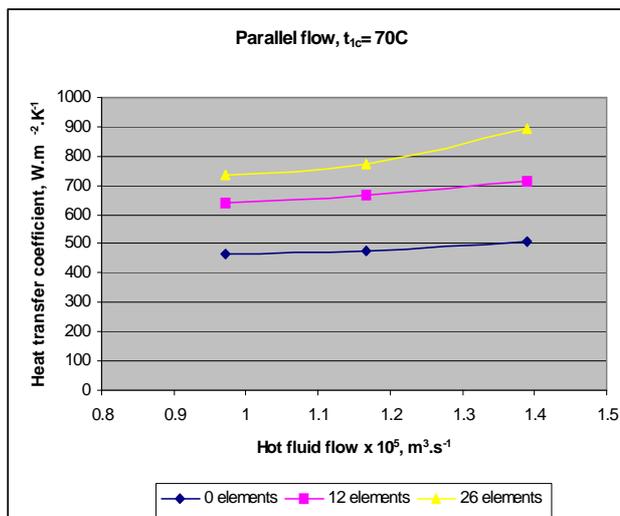


Figure 4. The total heat transfer coefficient evolution against the hot fluid flow rate and the presence of turbulence promoters

In Tables 3 and 4 the heat transfer improvement is quantitatively presented depending on the number of turbulence promoters, in the case of parallel flow and counter-flow of the two fluids.

It is observed that, in the same operating conditions, for the counter flow situation, the values of delivered heat and total heat transfer coefficient are higher compared with the case of parallel flow.

TABLE 3. Heat transfer intensification with turbulence promoters (26 pieces)

$t_{1h}, ^\circ C$	$V_{HW} \cdot 10^5, m^3 \cdot s^{-1}$	Q'/Q		K'/K	
		Parallel flow	Counter-flow	Parallel flow	Counter-flow
60	0.972	1.31	1.46	1.88	1.89
	1.167	1.36	1.52	1.75	2.00
	1.389	1.36	1.50	1.85	1.99
70	0.972	1.17	1.23	1.58	1.65
	1.167	1.19	1.34	1.63	1.75
	1.389	1.28	1.42	1.77	1.79

TABLE 4. Heat transfer intensification with turbulence promoters (12 pieces)

$t_{1h}, ^\circ C$	$V_{HW} \cdot 10^5, m^3 \cdot s^{-1}$	Q''/Q		K''/K	
		Parallel flow	Counter-flow	Parallel flow	Counter-flow
60	0.972	1.11	1.13	1.57	1.47
	1.167	1.18	1.18	1.54	1.52
	1.389	1.09	1.15	1.46	1.47
70	0.972	1.06	1.07	1.38	1.49
	1.167	1.10	1.13	1.41	1.54
	1.389	1.11	1.18	1.41	1.54

In Fig. 5 the combined influence is shown of the number of turbulence promoters placed on the hot fluid path and of its flow rate increase upon the heat transfer intensification. The heat transfer improvement is quantified through the increase of the total heat transfer coefficient.

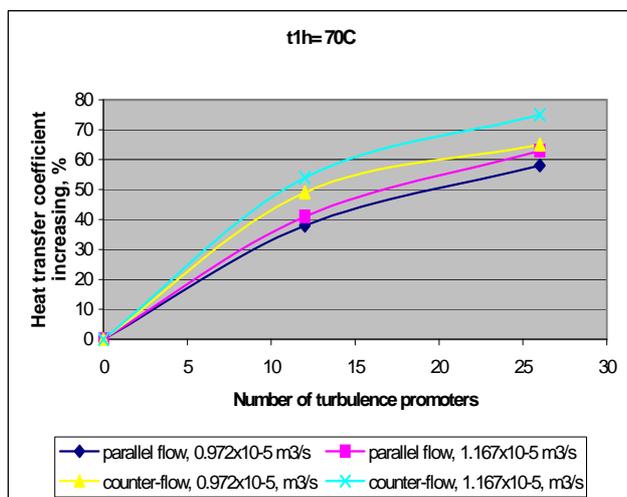


Figure 5. The increase of total heat transfer coefficient in the presence of turbulence promoters

4. Conclusions

Based on the information from literature on the role of turbulence promoters on the intensification of heat transfer with fluids flowing in laminar regime, a coaxial tube heat exchanger was designed and made, with metallic elements inside the inner pipe that caused a more efficient "mixing" of the hot fluid.

The heat delivered by the hot fluid increases with the growth of its flow rate and its output temperature from the thermostat.

In the same conditions of temperature and flow rate, the heat delivered in the case of counter-flow is higher than in parallel flow.

The partial heat transfer coefficient for the hot fluid increases with the fluid flow regime intensification and has higher values in counter flow. The determination of the partial heat transfer coefficient was done only in the case of the absence of turbulence promoters in the hot fluid flow space (the inner pipe);

The presence of turbulence promoters leads to an improvement of the heat transfer. In similar operating conditions (flow rate of the hot fluid and its output temperature from thermostat, respectively) the values of the heat transfer coefficient increase with the increase of the number of promoters.

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