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Thermal and economic investigation of straight and U-bend double tube heat exchanger with coiled wire turbulator

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Abstract Experimental research has been carried out for four individual heat exchanger constructions, i.e., plain double tube, turbulized double tube, plain U-bend and U-bend with turbulator. Tests were made for the water-water system. The study covered a wide measuring range, i.e., Re = 800–9000 – on the shell side, for a constant cold water temperature of 9 °C and hot water of 50 °C. The heat exchangers were made from copper tubes with external diameter of 10 mm and 18 mm respectively and wall thickness of 1 mm. The helicoidal vortex generator was made from brass wire with a diameter of 2.4 mm, coil diameter of 13 mm and pitch of 11 mm. For these geometries, the values of pressure drop, heat flux and heat transfer coefficient were determined. Wire coil turbulator increased the heat transfer coefficient (HTC) over 100% and pressure drop up by 100%. The comparison of heat transfer efficiency was performed based on the number of transfer units-effectivenes (NTU-ε) method. The modified construction achieved a similar efficiency. Economic analysis of wire coil turbulator was made to validate its use in the system. It showed that a coiled wire turbulator can greatly decrease the investment cost of the double tube heat exchanger while maintaining transferred heat at a constant level.

Keywords: NTU – number of transfer units; Heat exchanger; Heat transfer coefficient; Energy efficiency

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Nomenclature

A	–	heat transfer area, m^2
c	–	thermal capacity, W/K
C	–	cost, USD
c_p	–	specific heat, kJ/kgK
D	–	diameter, m
d	–	diameter of the inner tube, m
ΔP	–	pressure drop, Pa
g	–	gravitational acceleration, m/s^2
G	–	mass flux, kg/m^2s
Gr	–	Grashof number
HTC	–	heat transfer coefficient
k	–	unit cost, USD/MWh
L	–	length of annulus tube, m
LMTD	–	log. mean temp. difference, K
$\dot{m}$	–	mass flow rate, kg/s
NTU	–	number of transfer units
Nu	–	Nusselt number
P	–	Power, W
Pr	–	Prandtl number
$\dot{Q}$	–	Heat flux, W
Ra	–	Rayleigh number
Re	–	Reynolds number
T	–	temperature, K
ΔT	–	temperature difference, K
U	–	total heat transfer coefficient, W/m^2K
$\dot{V}$	–	volumetric flow rate, m^3/s

Greek symbols

α	–	heat transfer coefficient, W/m^2K
β	–	thermal expansion coefficient, 1/K
ϵ	–	heat transfer effectiveness
η	–	pump efficiency
λ	–	thermal conductivity, W/m K
μ	–	dynamic viscosity, Pa s
ν	–	kinematic viscosity, m^2/s
ρ	–	density, kg/m^3

Subscripts

a	–	annulus
air	–	ambient air
av	–	average

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c	–	cold
cw	–	coiled wire
contr	–	contraction
el	–	electric
exp	–	expansion,
frict	–	frictional
h	–	hot
i	–	inner
in	–	inlet
inv	–	investment
is	–	insulation
loss	–	loss to ambient
o	–	outer
oper	–	operational
out	–	outlet
sh	–	shell
tot	–	total
w	–	wall

1 Introduction

In recent years, investigations aimed at increasing heat transfer efficiency are of primary importance [1]. Reducing the energy requirement for energy-intensive systems and also maximization of energy utilization are common subjects of research [2]. Regardless of strong progress in numerous fields of engineering, simple constructions, such as double pipe, U-bend or coil heat exchangers are widespread. The growing, negative industry impact on the environment strongly accelerates the search for new solutions in the field of heat transfer engineering [3,4]. It should be noted that in all types of heat exchangers heat and fluid flow can be complicated and difficult to predict by means of literature correlations [5–7].

One of the most frequently used types of heat transfer apparatus is still U-bend heat exchanger [8]. In particular, this design is popular in process engineering and refrigeration systems [9–12]. This is due to both the simplicity of the design and the relatively good efficiency, but above all their reliability [13]. However, this construction does not belong to solutions with a high compactness index [14]. Also, compared to other constructions such as plate-fin and shell-and-tube heat exchangers, they are less effective [15]. Despite that fact, there are still many works in open literature on heat transfer enhancement in the double tube [16] or U-bend heat exchangers [17]. Most of the concepts are utilizing a variety of turbulizer inserts. These passive techniques are beneficial compared to the active techniques because

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their manufacturing process is simple and can be effortlessly employed in the existing heat exchanger. Insertion of swirl flow device enhances the convective heat transfer by making a swirl in the bulk flow and disrupting the boundary layer at the tube surface due to repeated changes in the surface geometry. Popular methods of increasing heat transfer efficiency are usually based on the increase of the heat transfer surface, what in turn, often causes more frequent need for device servicing. Attempting to reduce the size of the heat transfer apparatus by use of mini and microchannel technology complicates the process of repairs. Unfortunately, many of today's high-performance heat recovery solutions are characterized by high failure rates [18].

Cost-effectiveness leads research towards new technologies of heat transfer enhancement in heat exchangers design. One of the methods is to reduce the investment cost by replacement of conventional metallic heat exchangers with plastic components [19]. Significant lowering of operational and possibly capital costs can also be made by a real-time optimization of the heat-exchanger working conditions [20]. Additionally, low-cost modification of well-established technologies can reduce the negative impact on climate [21].

Review by Sheikholeslami *et al.* describes passive techniques that are commonly used in simple heat exchangers [22]. Authors conclude that wire coil give a better overall performance when considered is the pressure drop penalty. Also, research by Liu and Sakr was focused on passive heat transfer enhancement techniques [23]. The authors noted that the twisted tape inserts offer better performance in laminar than turbulent flow regime. Other passive techniques such as ribs, conical nozzle, and conical ring, etc., were described as more efficient in the turbulent flow regime. In both reviews, most of the analyzed works were performed in straight tubes.

Yadav studied the influence of turbulators on heat transfer and pressure drop of a U-bend double pipe heat exchanger [24]. During the experiments, the swirling flow was introduced by using half-length twisted tape located inside the inner tube of the heat exchanger. The results obtained from the modified heat exchanger were compared with those of plain one. The author emphasized that the 40% increase of heat transfer coefficient is found to be strongly influenced by tape-induced swirl or vortex motion. Also, Raj and Reddy pursued heat transfer enhancement using the same type of turbulators in the U-bend double tube heat exchanger [25]. Authors used inserts in the inner pipe of counterflow U-bend double pipe heat exchanger

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for creating a swirl flow and thus enhancing the heat transfer. Twisted tapes of different twist ratios and insert without any twist were employed to enhance the heat transfer. For a mass flow rate of 8 L/min, the enhancement in heat transfer was above 55% compared with that of the plain tube while pressure drop was only 20% higher.

As can be seen from literature review there are many studies concerned with heat transfer enhancement techniques in case of double pipe heat exchangers. However not many studies concerned at augmentation of heat transfer in case of U-bend double pipe heat exchanger (HX). What is more important, most of the existing works were concerned at heat transfer enhancement at inner pipe not annulus side of heat exchanger. Thus, the key task set by the authors of this paper was to increase the heat transfer by using simple inserts with a negligible effect of such element on heat exchanger durability, and cost.

2 Experimental rig

In order to verify initial assumptions, the experimental study of the final geometry of the U-bend heat exchanger was preceded by an analysis of the straight double pipe heat exchanger made from identical materials, as presented in Fig. 1.

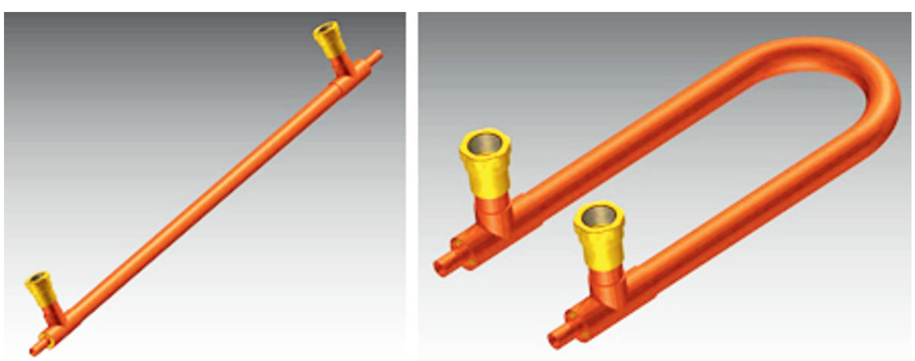

Figure 1: View of the designed double pipe straight and U-bend heat exchangers.

The heat exchangers were made of copper pipes with the diameters as follows: inner pipe diameter $d_i = 10$ mm, outer pipe diameter $d_o = 18$ mm and wall thickness 1 mm, see Fig. 2. The total length of the exchanger, in each configuration, was constant $L = 530$ mm. The helical turbulator

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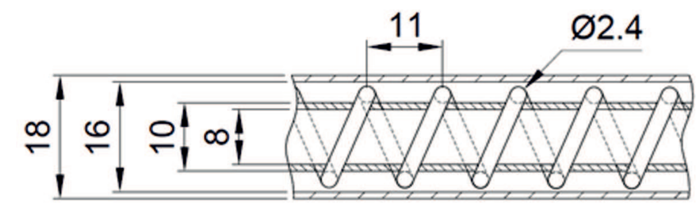

Figure 2: Turbulator dimensions and cross-section of the apparatus with a view of the helical turbulent element.

is made of brass wire with a diameter of 2.4 mm and a pitch of 11 mm, see Fig. 3. Moreover, turbulator was located in the outer pipe of the heat exchanger. Scheme of the experimental rig was shown in Fig. 4.

Figure 3: Pictures of the manufactured double pipe straight and U-bend heat exchangers with helicoidal turbulator.

Structure of the test stand allows to calculate the energy balance of the heat exchangers in water - water configuration. The K-type thermocouple in the first class of accuracy with 0.5 mm diameter (with insulated weld) was used to measure temperatures. Thermocouples were connected to the CHY510 meter made in the ITS-90 standard. Pressure drop was measured using a differential pressure transducer with range 0–300 kPa and 0.25 accuracy class made by Peltron. The volumetric flow rate of water was measured on the shell side of the heat exchanger with a ROL 16 rotameter in class 2.5 and on the inner tube side with a Meterc water meter (T30/90). Table 1 presents the systematic errors of measurement.

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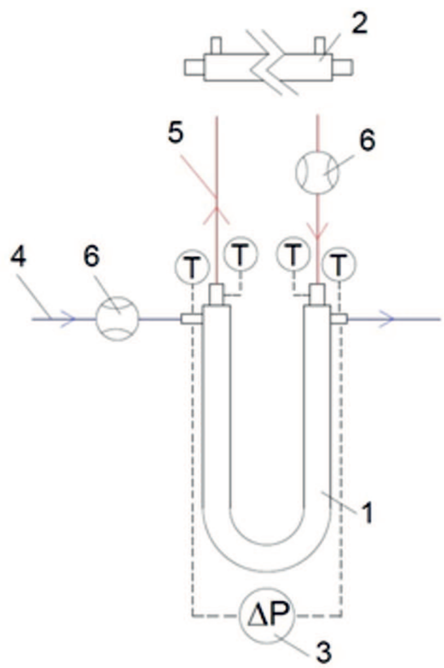

Figure 4: The test rig schematic: 1 – U-bend heat exchanger, 2 – straight heat exchanger, 3 – differential pressure transducer, 4 – cold water circuit, 5 – hot water circuit, 6 – flow meter, T – thermocouples.

Table 1: Uncertainty of the essential parameters.

Parameter	Measuring range	Systematic error
T	283–333 K	termocouple class +/- 1.5 K CHY510 meter error +/-0.3 K maximum error = +/-2.8 K
$\dot{m}_c$	0.00887–0.094 kg/s	maximum error = 2.5%
$\dot{m}_h$	0.094 kg/s	maximum error = 2.5%
ΔP	0–200 kPa	maximum error = 0.5 kPa
Re	827–18423	maximum error = 5.7%
$\dot{Q}$	100–3000 W	5.4–8.1%
U	115–614 W/m^{2}K	7.91–10.6%

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To estimate heat loss to ambient the infrared camera was used. Based on data from infrared photos the average temperature difference can be calculated as logarithmic mean temperature difference (between outer insulation temperature $T_{sh,w}$ and ambient T_a), according to

$$\mathrm{LMTD} = \frac{(T'_{sh,w} - T_a) - (T''_{sh,w} - T_a)}{\ln\left(\frac{T'_{sh,w} - T_a}{T''_{sh,w} - T_a}\right)} , \tag{1}$$

where the prime and double prime symbol denoting inlet and outlet, and subscript *sh,w* refer to the shell side wall, respectively.

Heat transfer coefficient for air side was calculated by using experimental correlation for the vertical cylinder:

$$\mathrm{Nu}_{air} = \left[0.6 + \frac{0.387\mathrm{Ra}^{\frac{1}{6}}}{\left[1 + \left(\frac{0.559}{\mathrm{Pr}}\right)^{\frac{9}{16}}\right]^{\frac{8}{27}}}\right]^2 , \tag{2}$$

$$\mathrm{Ra} = \mathrm{GrPr} = \mathrm{Pr}\frac{\beta g d^3}{\nu^2}\mathrm{LMTD} , \tag{3}$$

$$\alpha_{air} = \frac{\mathrm{Nu}\lambda_{air}}{L} , \tag{4}$$

$$\dot{Q}_{loss} = \mathrm{LMTD}\alpha_{air}\pi d_{is} L , \tag{5}$$

where Nu, Ra, Gr, and Pr are the Nusselt, Rayleigh, Grashof, and Prandtl numbers respectively, β is a thermal expansion coefficient, ν is a kinematic viscosity, g is a gravitational constant, d is a characteristic dimension, α_{air} is a heat transfer coefficient to ambient and λ_{air} is the thermal conductivity of air.

The heat exchanger was insulated with 0.1 m thick layer of polyurethane foam. The heat loss was calculated assuming cylinder shape of the heat exchanger, following Eqs. (2)–(5). It turned out that total heat losses were not larger than 1% of the transferred heat, therefore it could be neglected in energy balance.

3 Data reduction

The transferred heat was calculated as the product of the water mass flow rate, temperature difference (inlet-outlet) and water specific heat, for hot

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and cold water respectively:

$$\dot{Q}_h{=}\dot{m}_h c_p\left(T_{h,in}{-}T_{h,out}\right)\ , \tag{6}$$

$$\dot{Q}_c{=}\dot{m}_c c_p\left(T_{c,out}{-}T_{c,in}\right)\ . \tag{7}$$

The number of transfer units was calculated as

$$\mathrm{NTU} = \frac{UA}{c_{min}}\,, \tag{8}$$

where A is the heat transfer area and the minimum value of the heat capacity was based on hot and cold fluid capacity

$$c_{min}{=}f\left(c_h, c_c\right). \tag{9}$$

Respectively the overall heat transfer coefficient, U, was calculated with the aid of following equation:

$$U{=}\frac{\dot{Q}_{av}}{\left[\dfrac{(T_{h,in}{-}T_{c,out}){-}(T_{h,out}{-}T_{c,in})}{\dfrac{T_{h,in}{-}T_{c,out}}{T_{h,out}{-}T_{c,in}}}\right]}\,, \tag{10}$$

where

$$\dot{Q} = \dot{Q}_c \tag{11}$$

Heat transfer effectiveness, ε, was calculated from following eqations:

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}}\,, \tag{12}$$

$$\dot{Q}_{max} = c_{min}\triangle T_{max}\ \ , \tag{13}$$

$$\triangle T_{max} = T_{h,in}{-}T_{c,out}\,, \tag{14}$$

$$\dot{Q}_c = \dot{Q}_h\,, \tag{15}$$

where $\dot{Q}_{max}$ is a maximum possible heat transfer in the tested heat exchanger.

In shell side flow, Reynolds number is calculated as the mass flow rate through hydraulic diameter, which is given by

$$\mathrm{Re}_a{=}\frac{G_a D_e}{\mu}\ , \tag{16}$$

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where $G_a = \frac{\dot{m}_c}{A_a}$ is the mass flux, and A_a is annuli cross-sectional area. Hydraulic diameter, μ is the kinematic viscosity, and D_e, depends on the inner tube diameter and inner shell diameter

$$D_e = \frac{D^2 - d^2}{d} \tag{17}$$

and is diminished by the wire cross-sectional area when turbulator is inserted in the annuli. The measured pressure drop is the sum of friction pressure drop, expansion and contraction losses in the headers at both ends of the test section

$$\triangle P = \triangle P_{frict} + \triangle P_{exp} + \triangle P_{contr}\,. \tag{18}$$

The calculation procedure was adopted from the literature [11,12,26]. What is also important, based on experimental pressure drop the theoretical pumping power was calculated

$$N = \frac{\triangle P \dot{V}}{\eta}\ , \tag{19}$$

where $\dot{V}$ is a volumetric flow of water at annulus side and η is a pump efficiency (assumed as 100%). Uncertainty of key parameters was calculated based on [27] and presented in Tab. 1.

4 Results

Because wire turbulator inserted into the annuli reduces the flow cross-sectional area, the resulting Reynolds number obtained for same mass flow rate will be affected, as depicted in Fig. 5. Because both straight and U-bend type HX have same dimensions only two typical characteristics are presented. As can be seen in Fig. 6 the turbulator highly affects the pressure drop in tested configurations. The curvature of the HX has a lesser effect. As expected, the pressure drop increase accelerates with higher flow rates.

Conducted experiments show that both straight and U-bend double pipe heat exchangers with turbulator, are characterized by higher heat transfer rates, see Fig. 7. This results directly in higher values of the heat transfer coefficient (HTC). At the same time, these constructions have significantly higher pumping power ratio values (higher flow resistance values),

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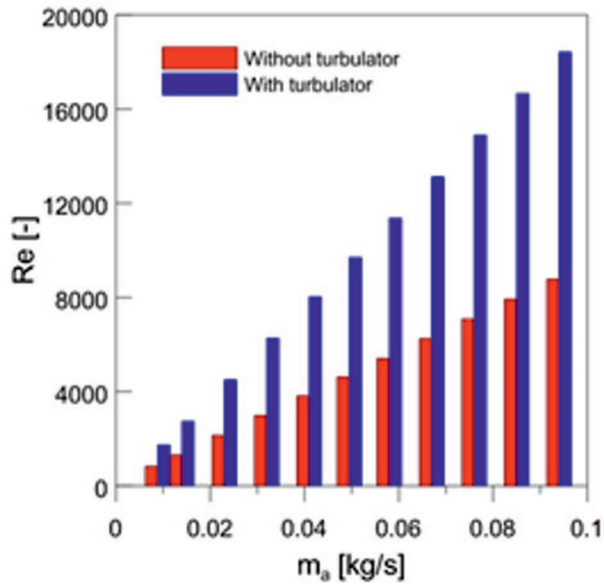

Figure 5: Reynolds number as a function of water mass flow rate.

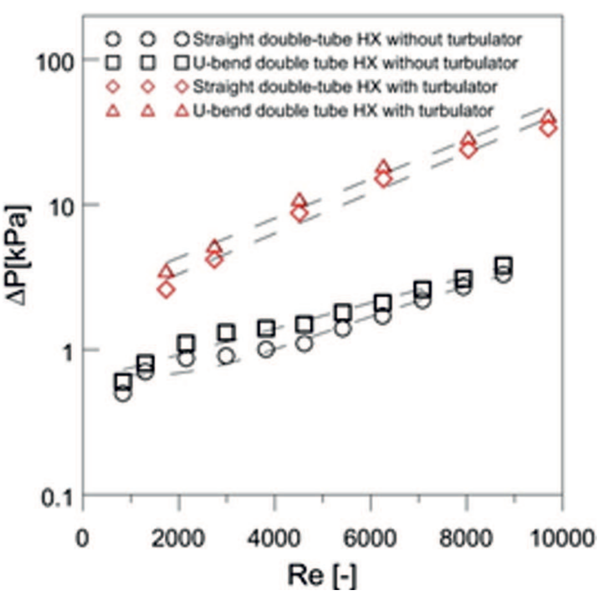

Figure 6: Pressure drop as a function of Reynolds number.

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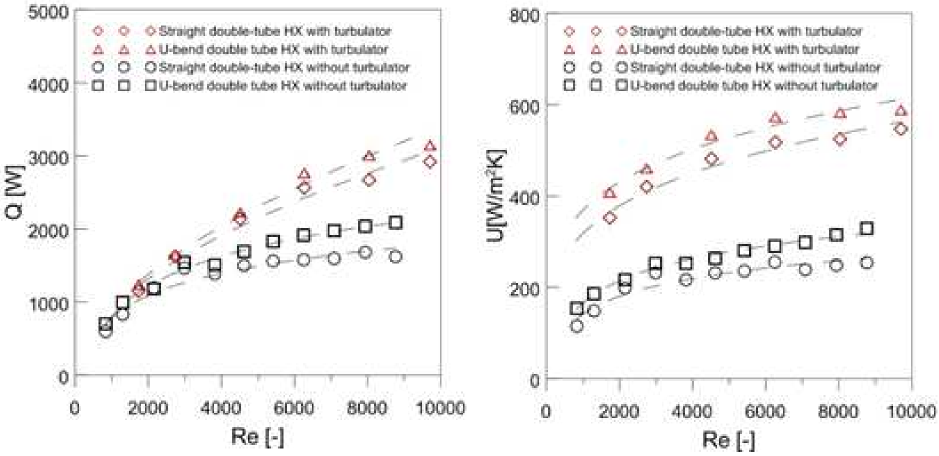

Figure 7: Received heat flux (right) and the heat transfer coefficient (left) as a function of Reynolds number.

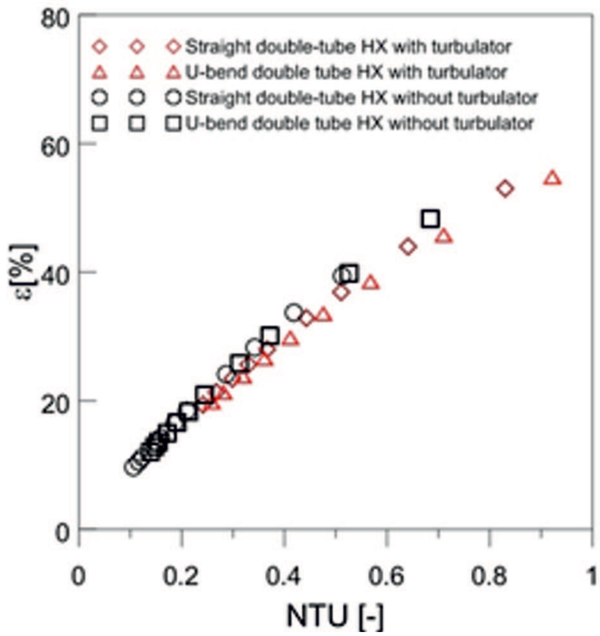

Figure 8: The dependence of heat transfer efficiency as a function of the NTU at left heat exchangers without turbulators and at right heat exchangers with turbulators.

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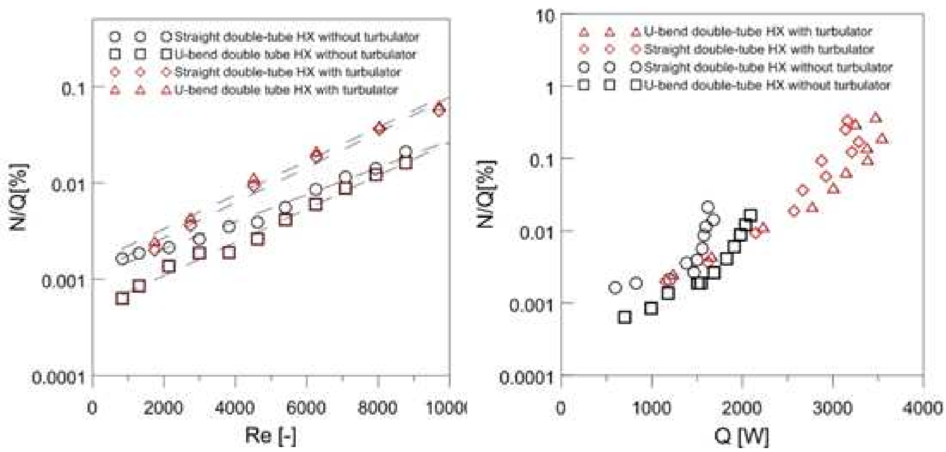

Figure 9: Comparative parameters for tested heat exchangers: pumping power ratio to the transferred heat as a function of Reynolds number (at the left), pumping power to the transferred heat ratio as a function of transferred heat(on the right).

Fig. 9. As expected, due to the presence of centrifugal force, U-bend double tube heat exchanger made in a smooth configuration compared to the classic straight double pipe construction is characterized by higher values of heat transfer coefficient. As discussed above this translates into the amount of the transferred heat. Nevertheless, modified constructions offer more than twice the thermal power compared to plain solutions. However ε-NTU analysis (Fig. 8) shows that all of the designs presented are of similar efficiency. Heat exchangers with turbulization, for the same thermal-flow parameters, are characterized by higher NTU values, which ultimately results in a maximum efficiency of 75% for straight double pipe configuration and nearly 80% for the U-bend type. Analysis of Fig. 9 shows directly the influence of heat exchanger modifications. As can be easily seen, the plain constructions quickly reach an optimal heat transfer rate, after which it can be increased at a very high cost of pumping power. Still, this increase is limited. This effect has a lower impact on U-bend configuration due to the centrifugal force effect that is increasing with fluids velocity. The similar effect is also visible in case of heat exchanger constructions with turbulator, as a change in the inclination of the data points.

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5 Economic analysis

Direct comparison of the investment cost is conducted in the simplified form adopted from literature [28]. The annualized total cost consists of the investment costs, i.e., materials manufacturing, and the operating cost, which combines with the electricity cost for the circulating pumps, as

$$C_{tot} = C_{inv} + C_{oper}. \tag{20}$$

The investment costs in USD for plain heat exchanger can be calculated as

$$C_{inv} = \frac{1722.7 {A_{hx}}^{0.679}}{1.22} \ [\text{USD}] . \tag{21}$$

Operation costs are calculated by following equation:

$$C_{oper} = \left[k_{el} \tau \frac{\triangle P_h m_h}{\eta \rho_h} \right] , \tag{22}$$

where the operating cost, k_{el}, τ, and η are the electricity in Poland unit price (180 USD/MWh), operation hours and pump internal efficiency (0.65), respectively. Also ρ_h is the density and m_h is the mass of the pumped medium. The manufacturing cost of the coiled insert was treated as an additional material cost

$$C_{inv} = \frac{1722.7 {A_{hx}}^{0.679}}{1.22} + \frac{1722.7 {A_{cw}}^{0.679}}{1.22} \ [\text{USD}] \ . \tag{23}$$

In the case of the U-bend heat exchanger the additional manufacturing cost due to forming was estimated as 2% of the material costs, resulting in

$$C_{inv} = 1.02 \frac{1722.7 {A_{hx}}^{0.679}}{1.22} \ [\text{USD}] \tag{24}$$

for plain heat exchanger and

$$C_{inv} = 1.02 \frac{1722.7 {A_{hx}}^{0.679}}{1.22} + \frac{1722.7 {A_{cw}}^{0.679}}{1.22} \ [\text{USD}] \tag{25}$$

for heat exchanger with wire insert.

Direct comparison of total costs is presented in Fig. 10. Calculations were made for a mass flow rate of 0.32 kg/s. Because of heat transfer enhancement total heat transfer area of the enhanced heat exchangers is smaller resulting in lower initial investment costs. Therefore data is presented in relation to heat transferred heat, Q. Similar pumping power requirements of heat exchangers result in lowest expenditures, for U-bend heat exchanger with wire insert.

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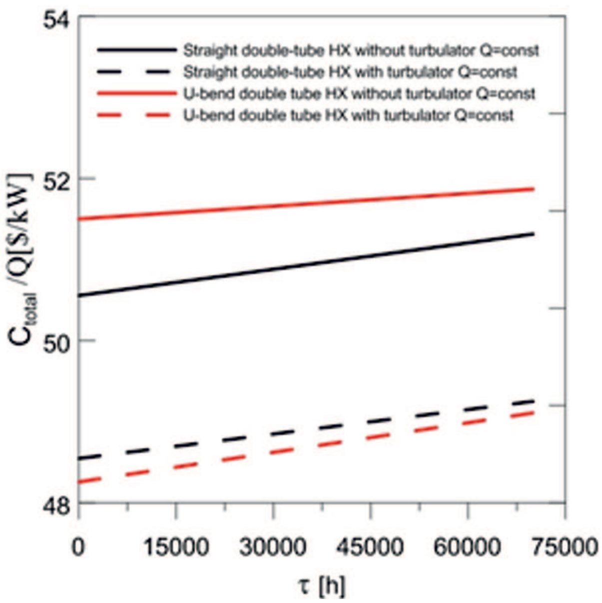

Figure 10: Comparison of total investment costs for investigated heat exchangers, for identical transferred heat.

6 Conclusions

Presented results show that centrifugal force (induced by the curvature of the U-bend HX) has a lesser effect in improving the energy efficiency of the thermal apparatus, than a properly selected vortex generator operating in the boundary layer region. Moreover, this modification can greatly reduce initial investment costs according to presented analysis. Investment savings are expected to be beneficial during the expected HX lifespan. Fact that the simplest construction equipped with a turbulization element has comparable efficiency, showing that it is not always worthwhile to aim for the maximum complication of the geometry of the heat exchanger. It is possible to use a turbulization with a higher number of coils or made of wire with a larger cross-section. Wire with expanded, corrugated, ribbed, etc. surface can also be used. However, this does not end the possibility to improve the energy efficiency of such apparatus. It is also possible to put the turbulization element in the area of the 'flow core' or use of the developed heat exchange surface. To enhance the effect of the centrifugal force, the U-bend heat exchanger can be modified to coil form, what unlocks the

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whole spectrum of new design possibilities.

In the author's opinion, further work is needed towards the development of 'simple construction' of heat exchangers based on a relatively slightly invasive methodology involving the use of turbulence inserts. Thus provides the relative flexibility of the apparatus (easy cleaning, low cost, easy adaptation for various flow conditions).

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