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Evaluating Jet Pump Turbulizers in Double Tube Heat Exchangers: A Preliminary CFD Study

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ABSTRACT

This paper presents a preliminary study of a novel type of turbulizers based on the jet pump working principle and designed to intensify heat transfer in double-tube heat exchangers. To determine the potential for their use, Computational Fluid Dynamics (CFD) and realizable k-ε turbulence model were applied to four turbulizers differing in the angles and diameters of the nozzle and diffuser. During studies, a number of parameters were determined, including: (i) Euler number Eu, (ii) overall heat transfer coefficient k_T and (iii) reduced ratio of heat duty to mechanical power ε_r; and then compared with data available for strip turbulizers. The tests yielded the following maximum values of: Eu – 1.5 times higher, k_T – 2 times higher, and ε_r – 27 times higher than those obtained by classical strip turbulizers, which indicates the high potential of presented jet pump turbulizers and justifies their further development.

1. INTRODUCTION

Nowadays, Computational Fluid Dynamics (CFD) is increasingly being used to study heat transfer intensification and optimise the performance of heat exchangers. In contrast to experimental measurements, the use of CFD provides insight into the essence of heat transfer in an exchanger by visualising the flows and knowing the properties of the media in each cell of the computational grid, without disturbing the flow with the presence of sensors. The degree to which CFD is used varies strongly from study to study: from visualising velocity fields and flow direction in the apparatus to show the mechanism of turbulizers [1] by modelling a section of the exchanger wall with turbulent inserts in 2D geometry [2] to obtaining results simulating the operation of a complete shell and tube heat exchanger in 3D geometry [3]. Thanks to the high computing power of personal computers and the spread of commercial software such as Ansys Fluent, simulating the operation of a heat exchanger has become a less costly and less labour-intensive alternative to making and testing prototypes [4].

Heat transfer processes are present in almost every plant in the chemical industry. Typically, thermal apparatuses are characterised by a large surface area to maximise heat transfer between the media. The increasing cost of construction materials and the pressure to minimise environmental impact have prompted an increase in exchanger efficiency. This could be archived achieved mainly by increasing the turbulence of the flow and, consequently, the intensity of heat transfer from the exchanger wall to the fluid. The easiest way to do this is to increase the flow velocity through the exchanger. Such a solution is not preferred because of the higher power requirements for the pumps and electrical energy costs. An alternative solution is to use passive turbulence promoters called turbulizers. Turbulizers come mainly in two forms: surface modifications of the exchanger and inserters [5, 6]. Regarding surface modifications, many researchers studied the many shapes of grooves and ribs [7-11]. The main disadvantage of surface modification method is the problematic cleaning of the exchanger equipped with the above-mentioned devices. This problem/difficulty is solved by the second class of turbulizers: Easy to disassemble turbulizer inserts. The characteristics of twisted tape turbulizers have been well studied [12-14] as classic turbulizers. Many authors have described their modifications [3, 15, 16] and completely different types of turbulizing inserts. The effects of Białecki rings were studied by Dziak and Ratajczak [17], nodular turbulizers by Charun [1], rotating turbulizers by Fodemski and Staniszewski [18], vortex generating turbulizers by Saraç and Bali [19]. In the course of the above mentioned studies, an increase in the achievable Nusselt number of several tens to more than one hundred per cent was obtained compared to exchangers without turbulizers, especially for low values of the *Re*. On the other hand, the use of some turbulizers increased pressure drops by up to 50 or 100 times. Consequently, the use of turbulizers led to a reduction in the size and cost of heat exchangers while requiring more efficient and more expensive pumps.

At the same time, in the chemical industry, jet pumps are successfully used for pumping corrosive fluids, dispersing suspensions and emulsions and as components of mixers. Their hydrodynamics has been described in full, among others, by Karambirov and Chebaevskii [20] and Winoto et al. [21]. The operating principle of a typical jet pump is shown in

Figure 1. The high-pressure working fluid enters through the main inlet (A). In the nozzle (B), the decreasing diameter forces the flow to accelerate and form a jet (C), which simultaneously reduces its static pressure. The moment the pressure drops below the pressure on the side inlet (D), the pressure difference starts to push the secondary fluid through the side inlet and into the mixing chamber. There, the high velocity difference between both fluids creates a high sheer stress and turbulences which start to mix both fluids. The mixing continues inside the diffusor (E), where the diameter gradually increases to decrease the fluid's velocity and recover some of its original static pressure. The mixed fluid flows through the jet pump outlet (F). Many researchers focus on the optimisation of the jet pumps for specific applications and operating conditions [22, 23] or determining the influence of individual geometry features [24, 25], but their studies concerned typical applications of jet pumps – as pumps or ejectors.

The authors therefore set out to test whether the use of a novel jet-pump-like structure, composed of a nozzle and diffusor parts placed inside the heat exchanger pipe will reproduce suction and mixing behaviour typical for jet pumps. The use of this new type of turbulizers, called jet pump turbulizers or ejector turbulizers, could result in increased turbulent flow and jet mixing accelerating the heat transfer deep into the flowing stream, and consequently noticeably increases the amount of heat exchanged. In this study, Computational Fluid Dynamics (CFD) was used to investigate the performance of jet pump turbulizers, compared to classical strip turbulizers in a double tube heat exchanger, whose experimental data of pressure drops and heat transfer coefficients are available in the literature.

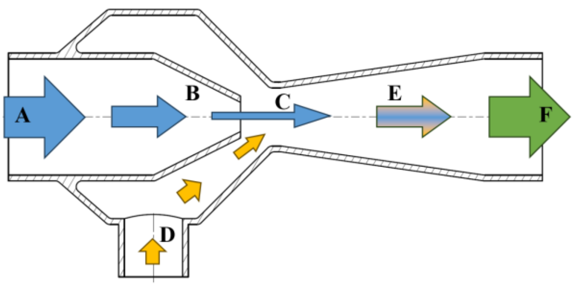


Figure 1. Schematics of typical jet pump with the main flows marked

2. METHODOLOGY

The study simulated the placement of the tested jet pump turbulizers in the inner tube of a countercurrent flow, laboratory-type double tube heat exchanger with water as the cold (outer tube) and hot (inner tube) medium. In the first part of the study, the focus was on the hydrodynamics by modelling the flow of constant-temperature water only in the inner pipe. The operation of the entire exchanger was simulated in the second stage of the heat transfer study. In order to evaluate the performance of the proposed turbulizers against existing solutions, the results were compared to those of the performance of strip turbulizers, which are one of the simplest and most commonly used designs. For this purpose, use was made of available experimental data previously obtained at the Wrocław University of Science and Technology by Wiśniewska [26] during her research on her master's thesis, for the same exchanger containing strip turbulizers (shown in Figure 2).

Figure 2. Strip turbulizers used by Wiśniewska [26]

2.1 Measurement strategy

The authors selected three main parameters that provide cross-sectional information on the performance of turbulizers and allow them to be compared independently of the measurement method. These are: Euler number Eu, the overall heat transfer coefficient k_T and the reduced ratio of heat duty to mechanical power of the exchanger ε_r. In order to compare the results from this study to the work of Wiśniewska [26] the authors used identical simplifications and formulae in calculating the ratios. The values of Eu were calculated by means of the pressure drop over the entire length of the exchanger, the velocity at the inlet and the density at medium temperature:

$$Eu = \frac{\Delta p}{0.5 \cdot \rho_{\left(\frac{T_{in}+T_{out}}{2}\right)} \cdot v_{in}^2} \tag{1}$$

while the overall heat transfer coefficient was determined by simplification to a flat wall:

$$k_T = \frac{\dot{Q}}{A \cdot \Delta T_m} = \frac{2 \cdot \dot{Q}}{\pi \cdot (d_2 + d_3) \cdot L \cdot \Delta T_m} \tag{2}$$

The ratio of the heat exchanger's thermal power to the mechanical power needed to pump the medium (Eq. (3)) was used as an indicator to intuitively compare the benefits of the tested solution against other turbulizers [1, 18]. In this case, the reduced form ε_r is much more convenient (Eq. (4)):

$$\varepsilon = \frac{\dot{Q}}{P} = \frac{4 \cdot \dot{Q}}{\Delta p \cdot v_{in} \cdot \pi \cdot d_3^2} \tag{3}$$

$$\varepsilon_r = \frac{\varepsilon}{\varepsilon_0} \tag{4}$$

where, ε_0 is the value ε estimated for a given exchanger operating under identical conditions (same temperatures and flows at inlets), but without turbulizers. These estimates were made by interpolating experimental data. For a more accurate description of the jet pump behaviour, ejector mass flow ratio M [21, 27]:

$$M = \dot{m}_{su} / \dot{m}_{no} \tag{5}$$

and leakage coefficient U:

$$U = \frac{\dot{m}_{in} - \dot{m}_{dif}}{\dot{m}_{in}} \tag{6}$$

was also calculated.

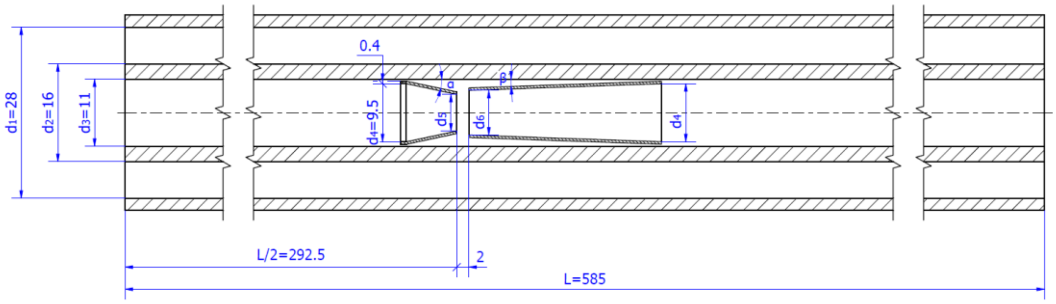


Figure 3. Schematic of the modelled exchanger (dimentions in mm)

2.2 Device geometry and numerical mesh

The modelled exchanger with an effective length of L=585 mm consists of a 16×2.5 mm inner tube and a 32×2 mm outer tube, made of brass. In the inner tube of the exchanger, the tested jet turbulizer was placed in such a way that the outlet of its nozzle was located exactly halfway along the exchanger (Figure 3). The influence of the basic dimensions of the jet was investigated by simulating four variants of the turbulizer with diameters and angles collected in Table 1.

Table 1. Geometrical configurations of the tested jet pump turbulizers

No.	d_5 [mm]	d_6 [mm]	α [°]	β [°]
1.	4.5	6.0	12	2
2.	6.0	7.3	12	2
3.	4.5	6.0	17.5	4
4.	6.0	7.3	17.5	4

For use in simulations, the exchanger geometry, including the jet pump, was simplified to the axisymmetric 2D case. Only the flow through the inner pipe was modelled on the hydrodynamics of the turbulizers tested. The numerical mesh had a variable cell size depending on the exchanger region, ranging from 300 μm in the free space of the pipe to 30 μm at the outlet of the jet nozzle (a total number of cells for hydrodynamic calculations was from 150,411 to 297,576, for thermal calculations – from 203,323 to 340,201 depending on the jet pump). In addition, a 7-layer boundary grid was used to increase the resolution of the calculations on the walls. During grid validation, it was confirmed that there was no effect of its further refinement on the results (for a grid that was twice as dense, the pressure drop differed by 0.0661% and the ejection coefficient by 0.687%).

2.3 CFD simulation parameters

Simulations were carried out using the Ansys CFD package. A realizable k-ε model was adopted to describe the turbulence, as confirmed in literature to give good results with modelling of both jet pumps [22, 23] and double tube heat exchangers equipped with the turbulizers [1, 3]. Simulations were performed as steady-state, with pressure-based solver, Semi-Implicit Method for Pressure Linked Equations (SIMPLE) pressure-velocity coupling scheme, least squares cell based gradient discretization and second order "upwind" properties discretization along with hybrid initialization. All other settings, including model calibration constants and relaxation factors, were left as default values. The temperature dependence of density, viscosity, thermal conductivity and specific heat of water was approximated by polynomials based on experimental data available in the literature [28] while the properties of brass were assumed to be constants due to insufficient data for approximation [29]. The boundary condition velocity inlet was used at the inlet of the device, whereas the boundary condition pressure outlet was used at the outlet of the apparatus. In hydrodynamic studies, inlet velocities v_{in} were 0.112÷0.775m/s and medium temperatures T_{in} 12÷50°C. In heat transfer tests, the velocities and temperatures were, respectively: for the cold medium (in the inner pipe) $v_{in,c}$=0.054÷0.727m/s and $T_{in,c}$=8.7÷17.1°C, for a hot medium (in the outer pipe) $v_{in,h}$=0.187÷0.196m/s and $T_{in,h}$=30÷60°C.

3. DISCUSSION OF THE RESULTS

3.1 Hydrodynamic of the turbulizers

The jet pump turbulizers tested were expected to suck the fluid from the tube wall layer and mix it with the flow core, significantly reducing the occurrence of the laminar boundary layer and its heat transfer resistance in the section downstream of the turbulizer. This behaviour is confirmed and shown in Figure 4. It also shows the rapid acceleration of the flow in the gap between the nozzle and the exchanger wall, which further disrupts the laminar boundary layer.

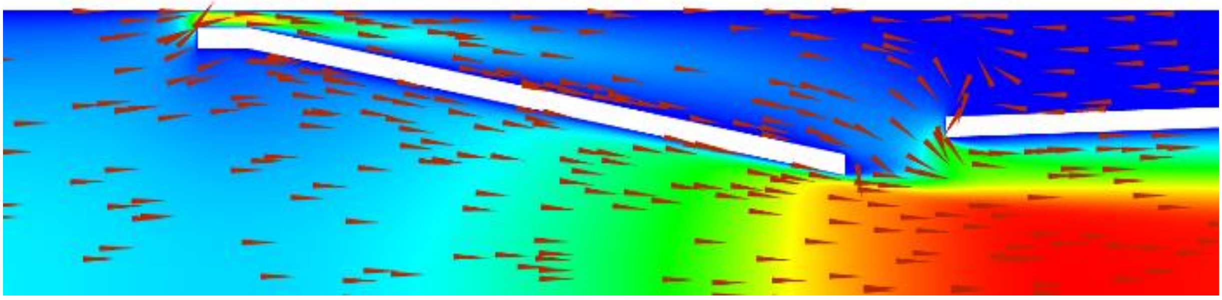
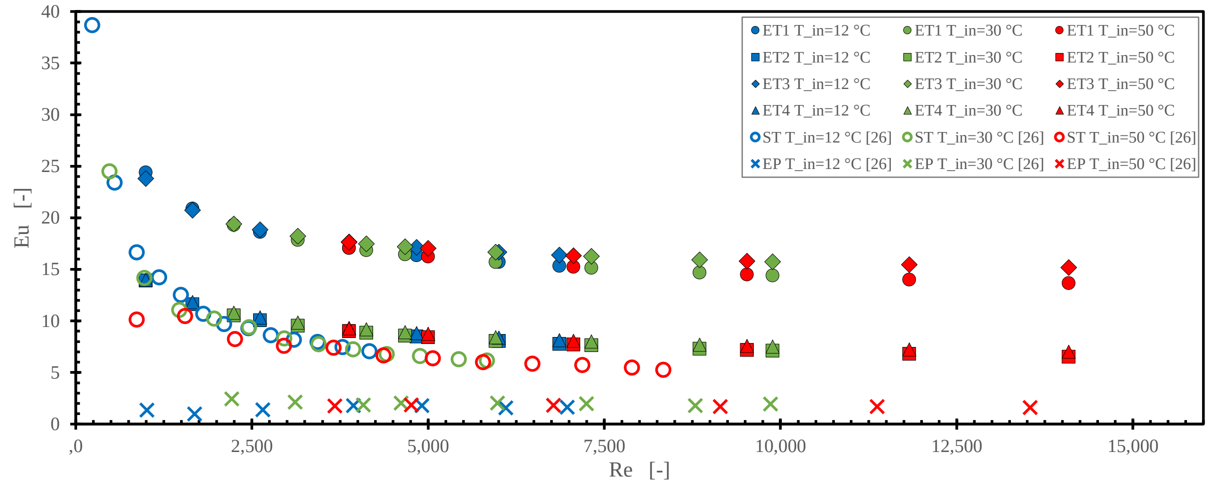
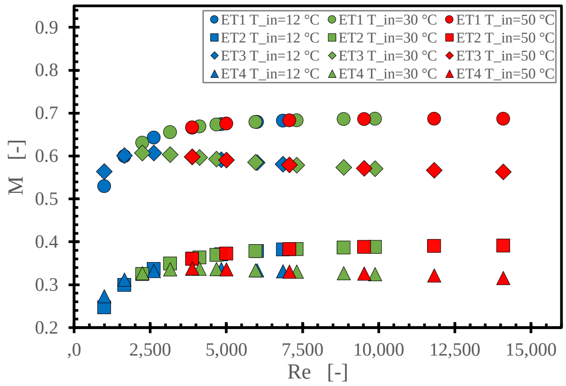


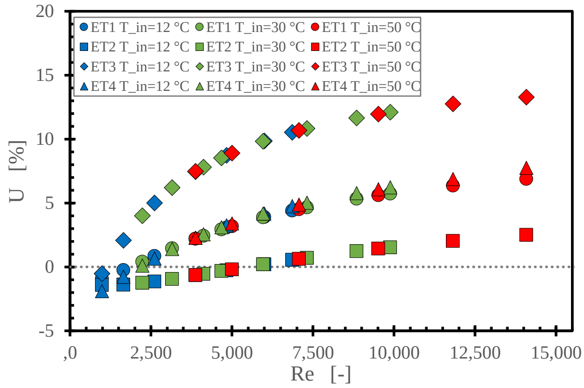
Figure 4. Close up on the generated suction flow (velocity vectors, in m/s)



(a) Euler number



(b) Ejector mass flow ratio



(c) Leakage coefficient

Figure 5. Jet pump turbulizer operating parameters obtained from simulations for different water temperatures at the inlet to the inner pipe

In order to compare the pressure drops caused by the individual turbulizers, the Euler number calculated according to Eq. (1) was applied. The values are shown in Figure 5(a). For jet pump turbulizers and strip-type turbulizers, a drop in values is evident in the Eu in the laminar range and their stabilisation in the transient and turbulent ranges. Tested geometries No. 2 and 4 (with larger nozzle diameters) show a similar results to strip turbulizers, larger by a maximum up to 1.5 times only. Variants No. 1 and 3 (with smaller diameters) obtained values Eu higher by approximately 10 over the entire range tested. This suggests that the nozzle diameter has a decisive influence on the induced flow resistance: reducing the nozzle diameter significantly increases the flow resistance and, consequently, Eu and pressure drops in the heat exchanger. On the other hand, different nozzle and diffuser angles caused minimal change to Eu value, at least for the tested geometries. For future study, the jet pump turbulizers with even a larger nozzle diameter will be used to minimalize pressure drop caused in heat exchangers. The exchanger without turbulizers shows a relatively constant Eu value of approx. 2 over the entire range of flow intensities, confirming that the turbulizers are the main source of flow energy losses and pressure drops.

The nozzle diameter also reveals a strong influence on the ejection coefficient M presented in Figure 5b. Again, higher values over the entire range Re are shown by variants with smaller nozzle diameters. The probable cause of this is the higher velocity and lower static pressure of the jet flowing out of the nozzle with a smaller diameter. This means a higher pressure difference between the front of the turbulizer and the nozzle-diffuser gap, which forces a larger flow through the nozzle-wall gap and into the diffuser, just like the "secondary flow" in a typical jet pump. For Re>2,300, the influence of nozzle and diffuser angles is apparent: jets with large angles show a reversal of the trend and a gradual decrease in the ejection coefficient with increasing flow rate. The probable cause of this is the separation of the flow from the nozzle wall with larger angles during higher flow rates, which disturbs and inhibits the flow of the sucked stream.

Figure 5(c) shows the values for the leakage rate. This determines how much of the flow bypasses the diffuser by flowing outwards. For all geometries tested, it shows an increasing trend with increasing Re. The highest values (up to 13.3%) were obtained for jet pump No.3. This means that for a small diameter and large nozzle and diffuser's angle the flow through the turbulizer deviates most from the operation of a typical injector. The probable explanation for this high percentage of water flowing outside the diffuser might be a combination of high resistance to the flow through a small diameter nozzle (which directs more flow through the nozzle-wall gap) and smaller suction generated by turbulizers with higher nozzle and diffusor angles. This behaviour is undesirable for jet pump turbulizers because fluid that bypasses the diffuser is not transported inside the flow core and is not mixed with it, with a limited amount of heat transported by convection from the region near the wall of the exchanger to the flow core. The lowest leakage rate values (not exceeding 2.5%) were obtained for jet pump No.2 (which with a large diameter nozzle and small angles is the opposite of geometry No.3). The negative U values for turbulizer No.2 up to Re approx. 5,000 indicate that part of the flow is turned back outside the diffuser and sucked in again, which was one of the expectations for the operation of jet pump turbulizers. This recirculation is profitable from a heat transfer point of view because it extends the contact time of the fluid with the exchanger wall.

3.2 Heat transfer

The study showed a significant increase in the heat transfer coefficient values achieved in the heat exchanger by using jet pump turbulizers for all geometries tested. Figure 6. presents how all simulated jet pump turbulizers show a rapid increase in value k_T with increasing turbulent flow up to a maximum of 3,075W/(m^2·K) for geometry No. 4 (with large nozzle diameter and angles) to 3,250 W/(m^2·K) for geometry No. 3. (with small nozzle diameter and large angles). This means values 1.7÷3 times higher than the empty pipe and 1÷2 times higher than the strip turbulizers tested by Wiśniewska [26], with the difference between various jet pump turbulizers being practically insignificant compared to the difference between jet pump turbulizers and strip turbulizers.

3.3 Comparison factor ε_r

The values for the reduced thermal-to-mechanical power ratio are shown in Figure 7. The downward trend is as expected, as it was also observed for rotating [18] and nodular turbulizers [1]. All jet pump turbulizers tested show significantly higher values ε_r than strip turbulizers, with geometries No. 2 and 4 showing the highest values. Both of these turbulizers have large nozzle diameters and low Eu values, which translates into lower pressure drops and lower mechanical power needed to pumping fluid through the exchanger. The values of angles do not make a noticeable difference in this parameter. They show ε_r>1 for flow rates up to Re=4,500, suggesting that it is in this range that their use may be most economical – they will increase the heat exchanger's thermal output the most in relation to increasing the energy input for pumping the medium.

For the lowest water flow velocity, all jet pump turbulizers achieve the highest value of the reduced thermal-to-mechanical power ratio. At the same time, hot medium temperature shows a higher impact on ε_r that differs in jet pump turbulizers geometry. However, the high uncertainties from ε_0 interpolation makes it impossible to draw binding conclusions about this flow region.

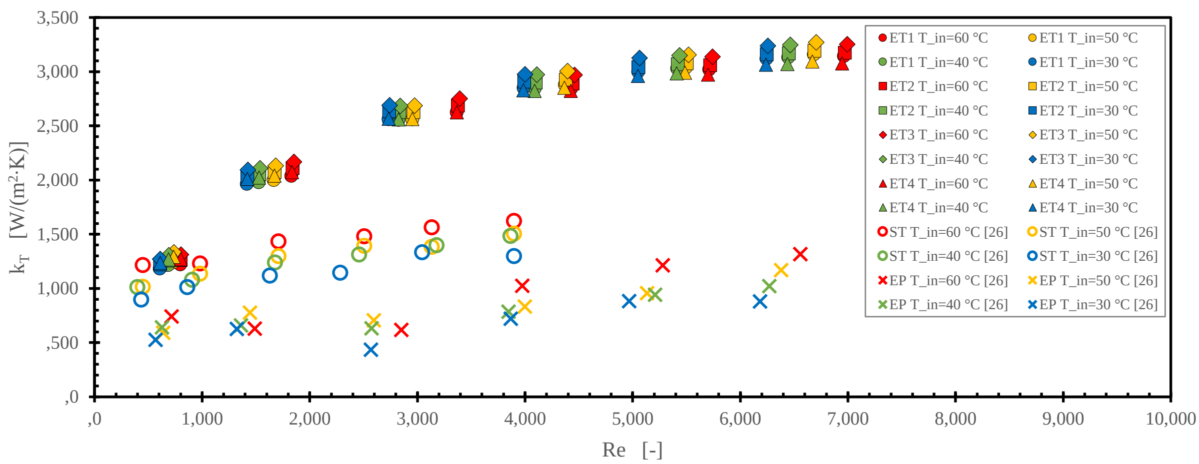


Figure 6. Comparison of heat transfer coefficients

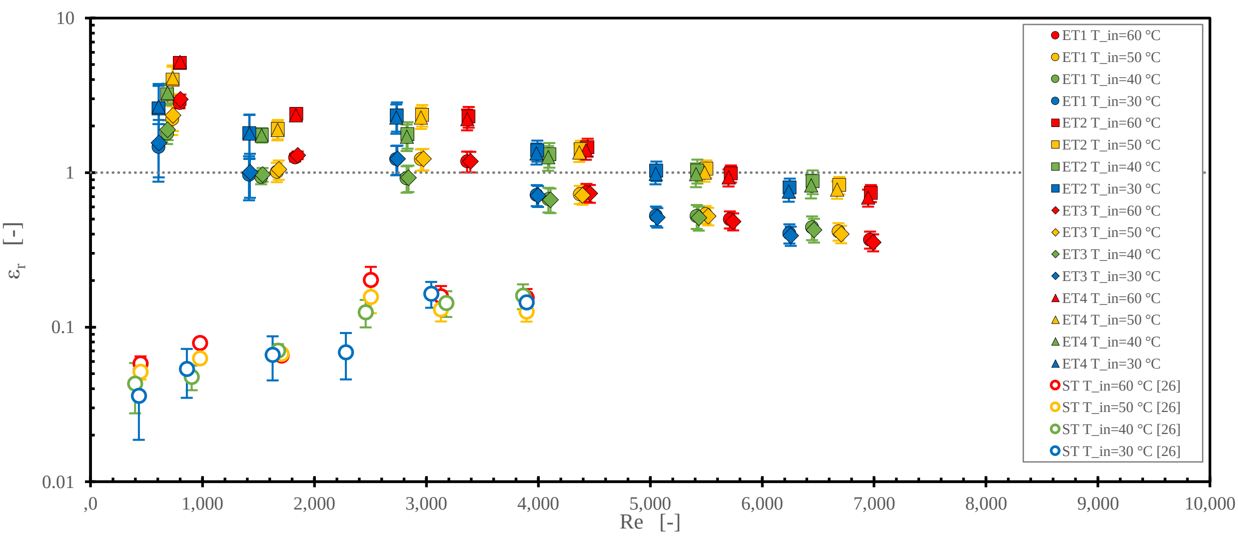


Figure 7. Comparison of reduced ratio of heat duty to mechanical power

4. CONCLUSIONS

This paper deals with the idea of jet pump like construction of turbulizers never before described in literature. CFD simulations made it possible to test the performance of the proposed new turbulizer design. The existence of suction and mixing of the fluid from the laminar boundary layer into the jet was confirmed, with a consequent significant reduction in the heat transfer resistance in the area beyond the turbulizer. Partial flow reversal was also demonstrated, increasing mixing and temperature uniformity, as well as rapid flow in the gap between the nozzle and the exchanger wall, further disrupting the boundary layer.

The preliminary study found that the proposed jet pump turbulizers show considerable potential for use in double tube heat exchangers. CFD simulations suggest achieving 5÷10 times higher Euler number values Eu and 1.7÷3 times higher values of overall heat transfer coefficient k_T with the use of jet pump turbulizers compared to an empty heat exchanger, as well as 1.2÷4 times higher Euler number values and 1÷2 times higher values of overall heat transfer coefficient against strip turbulizers. Of the geometry variants considered, especially the one with a small nozzle and diffuser inclination angle and a large nozzle diameter (geometry No.2) promises the most favourable increase in heat load of the exchanger in relation to the increase in pressure losses, especially at low flow rates of Re<4,500, as indicated by the value ε_r>1.

Preliminary numerical research has yielded promising results for the proposed jet pump turbulizers, which justifies further work on their development. Construction of the experimental installation is planned to confirm the results and validate the simulation model for further studies on determining the optimal jet turbulizer geometry.

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NOMENCLATURE

Latin letters

A	Area of heat transfer [m^2]
d_i	i-th diameter [m]
k_T	Overall heat transfer coefficient [W/(m^2·K)]
L	Effective length of exchanger [m]
M	Ejector mass flow ratio [-]
$\dot{m}$	Mass flow [kg/s]
P	Mechanical power [W]
Δp	Total pressure drop [Pa]
$\dot{Q}$	Heat flow [W]
T	Temperature [K]
U	Leakage coefficient [-]
v	Flow velocity [m/s]

Greek letters

α	Nozzle angle [°]
β	Diffuser angle [°]
ε	Ratio of heat duty to mechanical power [-]
ε_r	ε value divided by the empty exchanger case with the same load [-]
ρ	Density [kg/m^3]

Dimensionless criteria numbers

Eu	Euler number [-]
Re	Reynolds number [-]

Subscripts

c	Cold medium
dif	Diffuser flow
h	Hot medium
in	Inlet
m	Log mean
no	Nozzle flow
out	Outlet
su	Suction flow
(T)	Properties at given temperature

Abbreviations

EP	Empty pipe
ET	Ejector (jet pump) turbulizer
ST	Strip turbulizer