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Numerical study of the thermo-hydraulic characteristics in a circular tube with ball turbulators. Part 2: Heat transfer

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ABSTRACT

This paper is the second part of the article on the same title (the first one is devoted to flow resistance). It presents the results of numerical studies of heat transfer in a circular tube with a turbulising ball insert and is focused on an analysis of the influence of its geometry (i.e., ball diameters and longitudinal distance) on the heat transfer intensification in the turbulent flow. The investigations were conducted for a range of $Re = 10{,}000 \div 300{,}000$, different diameters of the balls ($Db = 7$, 10, 13, 16 and 19 mm) and various longitudinal distances between them ($L = 20$, 24, 28, 32, 36, 40, 48, 60 and 85 mm). It was observed that for the tested range of ball diameters and their longitudinal distance, there was an analytical formula that allowed one to express the *Nu* number by a relationship in the form $Nu = C \cdot Re^D \cdot Pr^{0.4}$, where the constants *C* and *D* were defined as a function of two variables: the ball dimensionless diameter ($X = Db/Dp$) and the dimensionless longitudinal distance ($Y = L/Dp$). These constants are closely related to each other and can be calculated analytically for the entire range of *X* and *Y* using a formula of the fourth order surface polynomial (15 coefficients of this polynomial for both *C* and *D* constants are presented in this paper). For the tested geometries, an increase in the intensity of heat exchange by a few times was obtained depending on the ball spacing and diameter (with respect to the tube without an insert). The thermal characteristics *Nu*(*Re*) of inserts for all these parameters and geometric configurations are presented on diagrams.

1. Introduction

Heat transfer processes are inherent phenomena in all branches of industry, such as petroleum, chemical or electricity generation. Improvement in the thermal efficiency of such processes can be achieved, for example, by reducing the dimensions of heat exchangers or energy savings. Since heat transfer in one channel of the heat exchanger reflects the overall efficiency of the entire device, it is usually sufficient to test a single tube (channel) in order to determine the efficiency of the whole heat exchanger. This approach is often used in numerical studies, where the computational model is additionally simplified by separation of its repeating element (being a periodic computational domain) from the test channel. It allows for a proper definition of obvious and simple boundary conditions of this element, as because of its periodicity and geometric symmetry, the velocity field has also the same qualities [1–4].

There are many ways to enhance heat transfer, e.g., by an application of various types of inserts, twisted tapes, wire coils or internal ribs of different shapes. All of these methods can be classified as the so-called passive techniques, which means, that they do not require any additional external energy (in order to improve heat transfer). On the other hand, they cause an increase in flow resistance, resulting in a rise of the power requirement for pumping the fluid.

The tested insert ball is not a geometrically complex flow turbulator, which can be easily applied to both new and existing channels of heat exchangers. Unfortunately, the lack of investigations on this type of inserts in the available literature makes a direct reference to this geometry impossible (i.e., without performing the adequate, presented in this paper analysis). In view of the insert simplicity and possibility of applications it (to the existing heat exchangers or another heat channels), the analytical dependences given as a functions of $f(Re,X,Y)$ and $Nu(Re,X,Y)$ can be helpful to calculate an appropriate flow-heat parameters in the tubular channel.

The most similar geometrically turbulising insert was studied by Charun [5]. He investigated heat transfer and a pressure drop

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Nomenclature

$b_1, \ldots, b_{15}$	polynomial coefficients
C	coefficient of equation $Nu = C \cdot Re^D \cdot Pr^{0.4}$
D	coefficient of equation $Nu = C \cdot Re^D \cdot Pr^{0.4}$
Db	ball diameter (m)
Dp	pipe diameter (m)
f	friction factor
g	gravity factor (m/s^2)
h	heat transfer coefficient (W/m^2 K)
i	enthalpy (m^2/s^2)
i_{tot}	total enthalpy (m^2/s^2)
L	longitudinal distance between balls (m)
Nu	Nusselt number
p	static pressure (Pa)
q_l	wall heat flux on the unit length (W/m)
q_w	wall heat flux on the inside surface of the pipe (W/m^2)
q_{Vol}	volumetric heat flux (W/m^3)
Re	Reynolds number
T_b	bulk temperature (K)
T_w	wall inside temperature (K)
u	average velocity (m/s)
$\overline{U}$	vector of velocity (m/s)
X	diameter ratio (Db/Dp)
y	distance to the nearest wall (m)
Y	longitudinal distance ratio (L/Dp)
y^+	non-dimensional wall distance
Δn	distance between the first and second grid points of the wall (m)
Δy	distance from the wall (m)
λ	thermal conductivity (W/m K)
δ	relative error
μ	dynamic viscosity (Pa s)
μ_t	eddy viscosity (Pa s)
ν	kinematic viscosity (m^2/s)
ν_t	kinematic eddy viscosity (m^2/s)
ρ	density (kg/m^3)
τ_w	wall shear stress (Pa)

Indexes

s	smooth pipe
loc	local

in a vertical tube with a ball insert placed in the axis. He examined four types of inserts with one variable parameter – different ball diameters in the range of $Re = 3000 \div 30{,}000$. In all cases the balls were adjacent to each other, creating a cascade (so, there was no second parameter to be examined in that study – a longitudinal distance between the balls). He observed the largest Nu numbers, but also the greatest flow resistance for the biggest diameter of the ball. Balls with the smallest diameter gave a low pressure drop, but also a slight intensification of heat transfer. This trend is also consistent with the investigations presented in this paper.

Eiamsa-ard and Promvonge [6] studied turbulising inserts with a similar layout of disturbing elements, i.e., axially symmetric structures, but in the shape of diamonds in the end-to-end contact with each other. Variable parameters were: length of a single element and an inclination angle of the diamond wall. The largest Nu number was observed for shortest elements and with largest inclination angles.

Kongkaitpaiboon et al. [8] investigated experimentally an insert consisting of circular rings fixed on the tube wall. As in the present paper, he analyzed two geometric variables: an inner ring diameter and the distance between them (three diameters and three longitudinal distances) in a small range of Re numbers ($4000 \div 20{,}000$). He found in his study that the best heat transfer occurred at the smallest inner diameter of the rings and the smallest distance between them.

There are more and more numerous works in the literature based on numerical studies. Of special interest are cases using the *SST k–ω* turbulence model, confirming a good agreement of this model with the experimental investigations. Eiamsa-ard [3] examined numerically a twisted tape in the tube and he observed an improved accuracy of calculations when using the *SST k–ω* model in comparison to other models of the *k–ε* family. Tang and Zhu [7] studied heat transfer and flow resistance in a narrow, rectangular channel with discrete embossed ribs. They also found that the *SST k–ω* turbulence model was more suitable for thermal–hydraulic calculations in channels than the *k–ε* model. Also Di Liberto and Ciofalo [17] in their study on thermal–hydraulic characteristics in a curved pipe observed that the *SST k–ω* model gave the best compatibility with the experimental results.

In general, on the basis of the literature survey, we can conclude that the majority of investigations is related to spiral inserts (twisted tapes) placed in the pipe. On the other hand, however, there are few studies of inserts with turbulising elements arranged in the axis of the channel, such as shown in this paper.

2. Numerical simulations

2.1. Modelling of turbulence – governing equations

Numerical calculations of heat transfer and frictional resistance were performed using the ANSYS-CFX 12.1 code. During the calculations, the basic equations of conservation of mass (1), momentum (2) and energy (3), which have the form:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \overline{U}) = 0 \tag{1}$$

$$\frac{\partial (\rho \overline{U})}{\partial t} + \nabla \cdot (\rho \overline{U} \times \overline{U}) = -\nabla p + \nabla \cdot \mu_e \left(\nabla \overline{U} + (\nabla \overline{U})^T - 2/3\, \delta \nabla \cdot \overline{U} \right) + \left(\rho - \rho_{ref} \right) g \tag{2}$$

$$\frac{\partial (i_{tot} \rho)}{\partial t} = \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \overline{U} i_{tot}) = \nabla \cdot (\lambda \nabla T) + \nabla \cdot (\overline{U} \tau_w) + S_E \tag{3}$$

where the term $\nabla \cdot (\overline{U}\tau_w)$ represents the work of viscous forces, S_E is a term of energy sources, and the total enthalpy i_{tot} is expressed as: $i + 1/2\,\overline{U}^2$ [14], are solved.

For the calculations, the *SST k–ω* turbulence model was used. In contrast to the standard *k–ε* model, it is recommended for thermal–hydraulic calculations, inter alia to cases where there is a separation of the boundary layer from the wall (and such a nature of the flow occurs in the tested inserts). This eddy-viscosity model is a combination of the Wilcox *k–ω* model, which calculates the flow in the boundary layer, with the standard *k–ε* model, which analyzes the flow in the turbulent core region. The *SST* model contains two transport equations, which are related to two variables – the turbulent kinetic energy k (4) and the turbulent frequency ω (5):

$$\frac{\partial (\rho k)}{\partial t} + \frac{\partial}{\partial x_j}(\rho U_j k) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + P_k - \beta' \rho k \omega + P_{kb} \tag{4}$$