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NUMERICAL STUDY OF FRICTION FACTOR AND HEAT TRANSFER CHARACTERISTICS FOR SINGLE-PHASE TURBULENT FLOW IN TUBES WITH HELICAL MICRO-FINS

The paper presents a numerical study on the heat transfer and pressure drop, related to flow in pipes with helical micro-fins. For all tested geometries, one applied a constant wall heat flux and fully developed 3D turbulent flow conditions. The influence of the angle of micro fins (referred to the tube axis) on thermal-flow characteristics were tested. The value of this angle was varied – with a step of 10 degrees – from 0 to 90 degrees (representing grooves parallel and perpendicular to the axis, respectively). Before numerical investigation, the pipe with helical angle of 30 degree was tested on an experimental stand. The results obtained from experiment and numerical simulations were compared and presented on the charts. As an effect of the numerical simulations, the friction factor f and Nusselt number characteristics was determined for the range of Re=$10^4 \div 1.6 \times 10^6$. The analysis of the results showed high, irregular influence of the helical angle on thermal characteristics and pressure drops. Additionally, the ratios: f/f_{plain}, Nu/Nu$_{plain}$ and efficiency indexes (Nu/Nu$_{plain}$)/(f/f_{plain}) as a function of the Reynolds number for every helical angle were shown on the charts.

NOMENCLATURE

A – cross section area [m^2]

D – outside diameter [m]

d – root diameter [m]

d_t – inside diameter [m]

d_h – hydraulic diameter [m]

e – roughness height for plain pipe

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f	–	friction factor
f_{plain}	–	friction factor for plain pipe
H	–	grooves height [m]
k	–	relative roughness for plain pipe
L	–	pipe length [m]
n	–	number of grooves
Nu	–	Nusselt number
Nu_{plain}	–	Nusselt number for plain pipe
p	–	pitch [m]
P	–	wetted perimeter [m]
Δp	–	pressure drop [Pa]
Re	–	Reynolds number
Tf	–	wall thickness [m]
u_{av}	–	average velocity [m/s]
α	–	apex angle [deg]
β	–	helical angle [deg]
ρ	–	fluid density [kg/m^3]

1. Introduction

The purpose of the heat transfer enhancement in heat exchangers is to reduce their cost and size, or to increase the heat efficiency – for a given dimension. One way to enhance the heat transfer in heat exchanger ducts is to increase the heat transfer area or the flow turbulization. In pipes with notched helical micro grooves on the wall, these two methods are applicable, aiming at the heat transfer augmentation.

In the available literature, one can find articles concerning similar geometries, but with a different test method and range of the parameters. Ravigururajan and Bergles [1] have studied several types of duct roughness – transverse ribs, helical ribs, wire coils (and their profiles) in different geometrical configuration. For geometry with helical ribs, they tested the influence of the following parameters on heat transfer enhancement and pressure drop: pitch, ribs height and helical angle, in the range of Re number from 7000 to 40000. They tested only a few helical angles (30.2, 32.2, 40.6 and 82.12 deg) with different pitch and rib heights. They found that, for micro-ribs with varying angle and pitch, the analogy method does not give too accurate correlations. Wang and Chiou [2] also studied micro-fin tubes, but for a small range of helical angles 18÷25 deg and 2000÷40000 Reynolds number.

The characteristics curves of $f(Re)$ and Nu(Re) have weak similarity to the ones presented in this paper – they differ quantitatively. Wen-Tao et al. [3] conducted a study on friction factor and heat transfer in turbulent flow for 16 internally grooved tubes. On the basis of measurements, they tested an extension of Gnielinski equation for Nu number. Siddique and Alhazmy [4] tested only one tube with 18 deg helix angle, at Re number from 4000 to 20000, grooves number of 50, and twice greater the rate of groove height to root diameter. They observed a variation in the friction factor at an increasing of Re number: firstly there was a decreasing trend, then about Re=10000 the factor started to increase, and subsequently it started to decrease again. It is similar to the characteristics of 40 deg helix angle presented in this paper in Fig. 8, but its characteristic is shifted towards smaller Re numbers. Four micro-fin tubes with varying Re (3000–40000) with regular, triangular shape of the ribs were investigated by Han and Lee [5]. They observed that, as far as the efficiency index is concerned, the tubes with smaller helical angle and higher relative roughness appears to have a better heat transfer performance than the tubes with smaller relative roughness and larger helical angle.

As one of the few, Dong et all [6] tested 4 spirally-corrugated tubes, under a wide range of Re number varying from 6000 up to the 93000 and relative roughness $H/d \approx 0.025$ (very similar to that presented in this paper). The authors also observed that, for the studied helical angles of 78÷ 82°, the efficiency index over a wide range of Re number was not uniform, and had its maximum at the value of Re being in the range 40000÷50000. Naphon et al. [7] conducted a study on several tubes with one constant helical angle equal to 45 degree and varying relative roughness and pitch of helical ribs. These results showed that for all pipes having Re number in the range of 7000÷25000 the Nu number ratio ($\mathrm{Nu}/\mathrm{Nu}_{plain}$) decreased, and friction factor ratio (f/f_{plain}) increased when the Re number rised.

Generally, most of the papers available in literature describe the results obtained from experiment. These data were burdened with some errors, which are due to the nature of these investigations, e.g. measurement of average values or uncertainty of obtaining a fully developed flow in the tested pipes. In good numerical investigations, these defects do not exist. If we have a long tube with some periodic section, it is sensible to model only one, repeatable part of the pipe and set special boundary conditions to enforce calculation of fully development flows for velocity and temperature distributions. In this way, one can be obtain very precise results.

2. Formulation of the problem

The aim of this paper is to present results and characteristics for tubes with different helical angles of micro-fins – using numerical simulations.

As a basis geometry, we assumed an industrial tube (made by KME Germany AG & Co. KG and labelled “TECTUBE®_fin 12736CV50/65D” with helix angle β = 30 deg). Dimensions of this tube are presented in Fig. 1. The assumption made for this research was to maintain a constant groove number and groove dimensions in the cross-section. According to this approach, the pitch of micro-fins changes in all geometries.

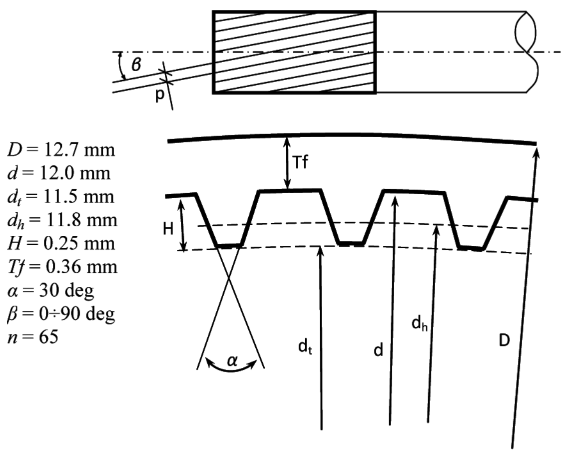

Fig. 1. Geometrical dimensions of tested micro-fin tube

Fig. 1 shows the geometrical dimensions and groove details of the investigated tube, presented in this paper. For the numerical study, we took into account 9 kinds of geometries: two extreme cases: grooves parallel ($\beta = 0°$) and perpendicular ($\beta = 90\,^{o}$) to the tube axis, and 6 with helical angle of micro-fins between 10 and 70 deg (with a 10 deg step). The tube with β = 80 deg was omitted in calculations because the micro-fins width was too big to create grooves - the fins would overlap each other. For β = 90 deg, the distance between micro ribs was the same as for the pipe with β = 0 deg. 3D view of the analyzed tubes with the appropriate micro-fin helix angles is shown in Fig. 2.

Numerical simulation allowed us to simplify the computational domain and to focus on the one, repeatable, periodical and symmetrical section of the duct. Thus, the created domain fulfils all conditions and can represent the entire tube with an appropriate geometry (Fig. 3). These assumptions give a significant benefit in calculation time, at the same time making it possible to achieve more accurate results by applying a fine mesh.

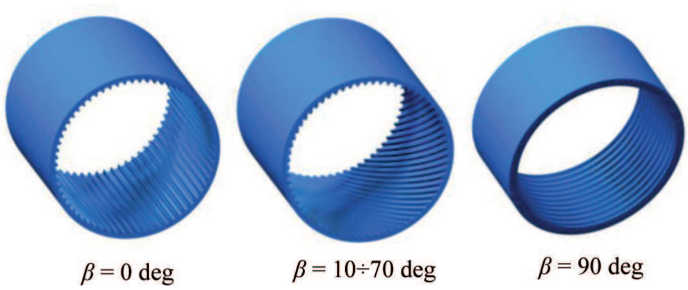

Fig. 2. Tubes with micro-fins

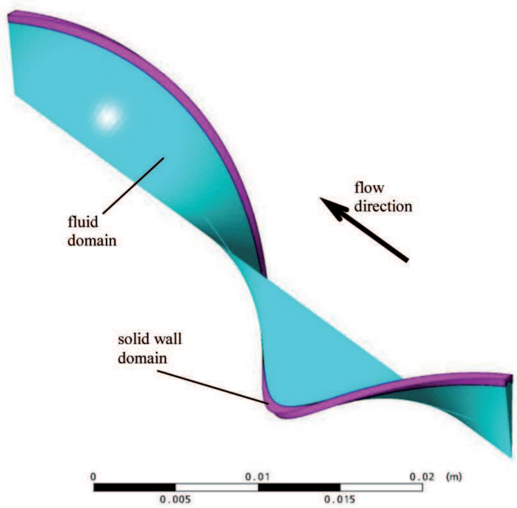

Fig. 3. Elementary and periodic section of the one micro-rib (helix angle β = 40 deg) used as a computational domain

For the presented geometry (elementary, periodic and repeatable section of the tube – see Fig. 3), the fully developed fluid flow and heat transfer was set. As the fluid, we used water at 310 K average temperature. To obtain fully developed flow, the fluid flow was forced by pressure gradient, and boundary conditions on the inlet and the outlet of the domain were set as a translational periodicity. The heat flux per unit length for the inner wall was constant for all cases – despite the fact that inner area of the heat transfer for every β angle was different. (This effect was achieved by appropriate matching of the heat flux for varying heat transfer area).

It is worth to mention some difficulties encountered during numerical investigation. One of them was a complicated procedure of running the fully developed flow with pressure gradient. It required setting the translational periodicity interface on the inlet and the outlet as a boundary condition, instead of "normal" inlet-outlet. In addition, a sensible convergence of calculation was more difficult to obtain in this case then for common settings. Nevertheless, all problems have been overcome in a reasonable time.

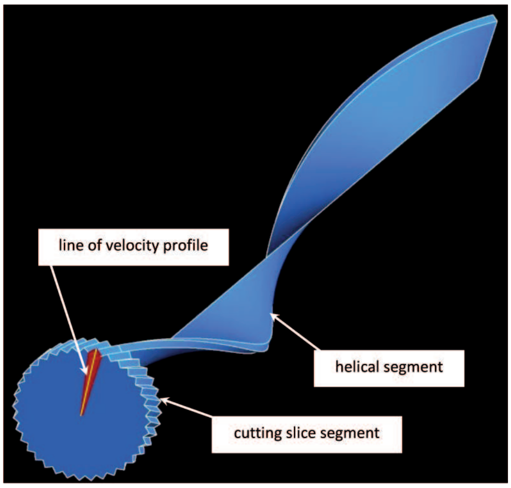

Fig. 4. Two tested geometries: helical and cutting slice segment

Before calculations, this kind of domain shape was numerically tested and compared to a full-circle micro-fin tube applying the same radial and longitudinal mesh resolution (Fig. 4). The simulation results of these two cases were identical, so the “one micro-fin domain” could be applicable for computation. The comparison of velocity profiles on the same line for two domains is presented in Fig. 5. As it can be seen, the two velocity profiles overlap – it means, they are identical.

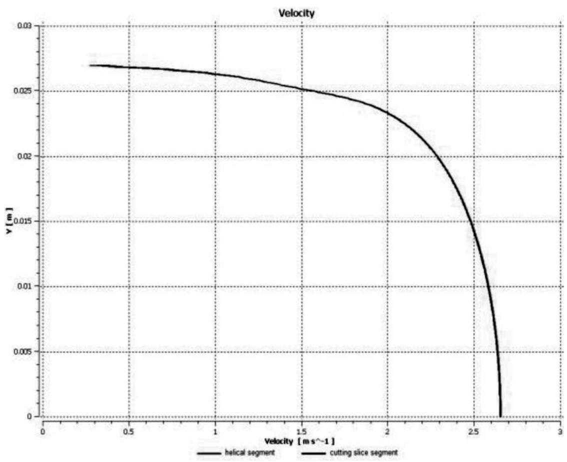

Fig. 5. Total velocity profiles on the line (Fig. 4) for two tested domains

3. Experimental validation

For one of the presented geometries, we carried out validation and verification of the computational code. An industrial tube with β = 30 deg, labelled "TECTUBE®_fin 12736CV50/65D" and made by KME Germany AG & Co. KG, was tested on a special experimental stand. In Fig. 6, there are shown results of the numerical and experimental study. Details of this procedure are presented in [8].

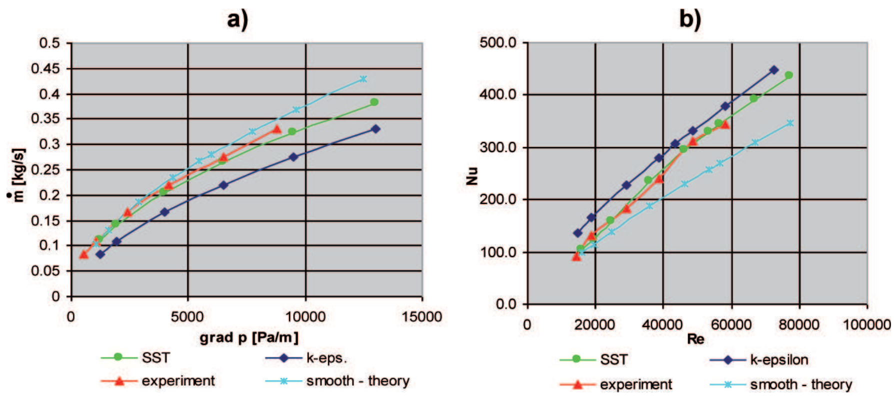

Fig. 6. Comparison between results of simulation and experiment for smooth tube, SST and k-ε turbulence model: a) mass flow vs. grad p, b) Nu vs. Re [8]

In this procedure, the two most popular turbulence models were also verified – the classic k-ε, and SST k-ω. It is clearly visible that the best-suited data and good agreement with experimental values, for both pressure drop and heat transfer, were obtained using the SST k-ω turbulence model.

The SST k-ω (Shear Stress Transport) turbulence model is one of the most popular models used in many CFD applications. Its main attribute is the ability to solve the viscous sublayer by applying the k-ω model close to the wall, and by using standard k-ε model in the core region. The "switching" between the two models is controlled by an embedded, special blending function [9,10]. The correct use of SST model requires several grid points inside this sublayer to maintain the dimensionless distance $y^+<2$ in the whole computational domain [11,12]. In the results presented here, the maximum value of y^+ did not exceed 1 in any of the geometries.

4. Results and discussion

For micro-fin tubes, one of the crucial matters is defining their hydraulic diameter. Classical definition, calculated from equation (1), yields incorrect results for friction factor, Fig. 7.

$$d_h = \frac{4 \cdot A}{P} \tag{1}$$

while the friction factor can be calculated from Darcy-Weisbach equation:

$$f = \frac{2 \cdot \Delta p \cdot d_h}{\rho \cdot u_{av}^2 \cdot L} \tag{2}$$

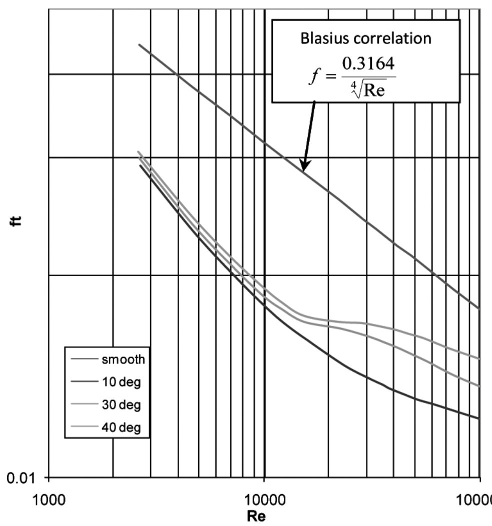

Fig. 7. Friction factor for smooth pipe and three sample tubes with helical micro-fins. Hydraulic diameter calculated from (1)

The analyzed micro-fin tubes have a wetted perimeter $P \approx 60$ mm, while in the plain tube of 12mm diameter this value is equal to $P = 36$ mm. This difference translates onto a hydraulic diameter of grooved pipes, which is equal to: $d_h = 7.3$ mm. Taking this hydraulic diameter for further calculations, we obtained sample characteristics of friction factor of the studied geometries (Fig. 7). It is worth noticing that in this approach the investigated geometries have a lower friction factor than a smooth pipe! Obviously, these results are not proper because in this case, on every micro-fin tube, there would be a lower pressure drop than on a smooth one. Many researchers (e.g. Brognaux et.al. [13]) observed this problem and Webb and Scott [14] formulated a concept of the equivalent diameter, i.e. the diameter, which would exist if all the micro-fins were melted down and the material came back to the tube wall.

Unfortunately, for the analyzed cases where we applied the assumption of a constant cross sectional area of the tube, the above solution would not be correct because every tested tube had different area of heat transfer (related

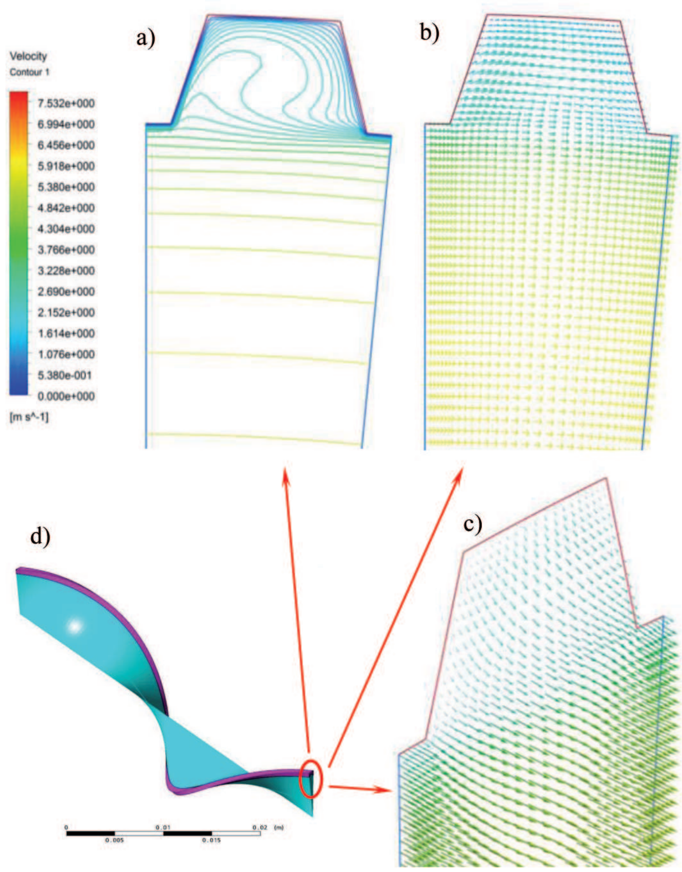

Fig. 8. View of the inlet velocity field: a) velocity contours, b) velocity vectors in orthogonal projection, c) velocity vectors in perspective view, d) computational domain

with dimensions of grooves) and each pipe would also have a different equivalent diameter. Therefore, for the studied micro-finned tubes, we calculated a diameter equal to that of a smooth pipe – with the same cross-sectional area. Thus, the hydraulic diameter for the considered cases was d_h =11.8 mm (while for a plain tube with the same cross-sectional area it was d =12 mm).

The helical micro-fins on the tube wall (apart of 0 and 90 deg cases) cause a swirling flow, especially in the hydraulic laminar layer between the grooves. Their shape also affects the flow in turbulent core outside the laminar and transition layer – it is also swirled but to a smaller degree than near the wall. This phenomenon is visible in Fig. 8-b). The pictures in Fig. 8a) and b) show views perpendicular to the pipe axis illustrating the velocity contours – Fig. a), and the rotational velocity component – Fig. b). In the tube with micro-fins parallel to the axis (β = 0 deg), the vortex is not produced because the grooves don't break the laminar layer. The grooves transverse to the flow direction (β = 90 deg), produce vortexes in the laminar layer, but the flow is axially-symmetric – there appears neither swirl stream, nor rotational velocity component.

4.1 Friction factor

The friction factor for the studied geometry was calculated from equation (2). The flow was enforced by a known pressure gradient ($\mathbf{grad}p = \Delta p/L$), and average velocity was the result of numerical solution. In Fig. 9, the friction factor for all helical angles is shown. Additionally, there is presented a smooth tube friction factor calculated from the Blasius (f =0.3164×$\mathrm{Re}^{-0.25}$), as a level of reference. The studied tubes had a regular, constant relative roughness, calculated as a ratio of the groove height to the root diameter (H/d =0.02 – see Fig. 1). In the analyzed geometries, only shape of this roughness (helix angle) wes changed, and the groove height remained constant.

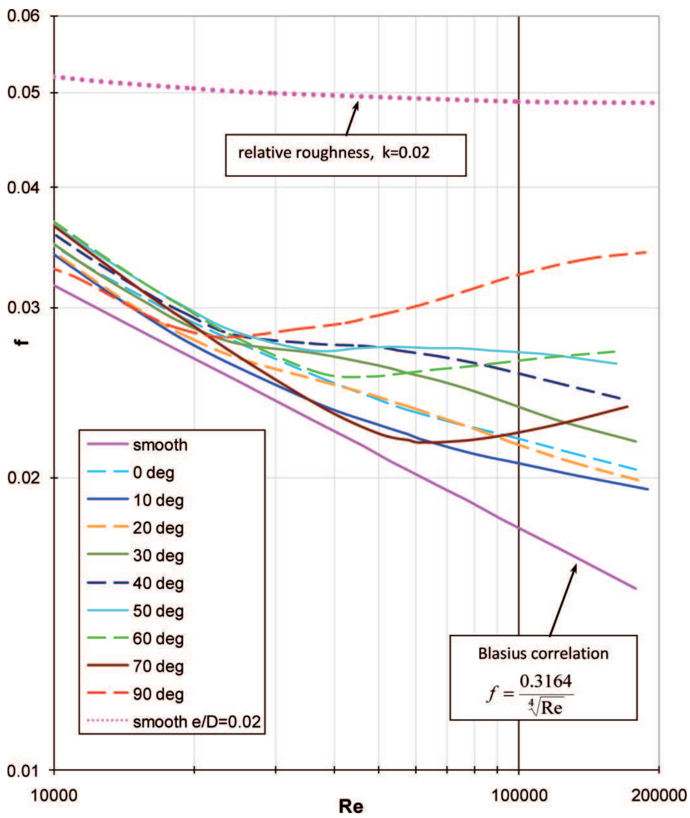

Fig. 9. Friction factor vs. Re number for all helical angle of micro-fins. The smooth pipe (Blasius formula) is shown as a level of reference

Friction factor for irregular roughness of plain pipes is given on the Moody's diagram, or can be calculated e.g. from the empirical formula of Swamee and Jain [15]:

$$f = \frac{0.25}{\left(\log\left(\frac{\varepsilon}{3.7 \cdot D} + \frac{5.74}{Re^{0.9}}\right)\right)^2} \tag{3}$$

The curve (k =0.02) shown in Fig. 9 was calculated from (3). Despite the fact that the analyzed tubes fulfil the conditions of relative roughness, it is really impossible to calculate their friction factor from equation (3). Fig. 9 illustrates a total dissimilarity between the curves obtained from numerical simulations and those calculated theoretically. Therefore, one of the main conclusions following from the conducted numerical studies is that there exists a great influence of micro fins shape on the friction factor. The same fact was noticed by Wang et all [2]. For rough pipes, the turbulent friction factor depends on the roughness height, type, shape and other types of geometrical parameters.

Analyzing the chart in Fig. 9, one can find a strong dependency of micro-fin helical angle on friction factors. In the range of Re number from 10000 to 20000÷25000, all the curves are rather regular and have the form of straight lines on the logarithmic graph. Above this range, these functions change their character and it is not possible to find any regularity. The friction factor for tubes with micro-fin angles from 10 to 40 deg decreases unevenly and the geometries from 50 to 90 deg have a minimal friction factor in a certain range of Re number. This is clearly visible in Fig. 11 – a), where the ratio of friction factors (of tested tubes to the smooth one f/f_{plain}) vs. Re number is presented. Therefore, it is difficult to find any simple fitting function, which would express the variability of helical angles in a mathematical formula. For that reason, the friction factor can be approximated as exactly as possible (separately for every helical micro-fin angles) just by a third-order exponential decay function (4). The variable parameters used in this formula are given in Table 1.

$$f = y_0 + A_1 \cdot \exp\left(\frac{Re}{t_1}\right) + A_2 \cdot \exp\left(\frac{Re}{t_2}\right) + A_3 \cdot \exp\left(\frac{Re}{t_3}\right) \tag{4}$$

4.2 Heat transfer

For the plane tube, the heat transfer can be calculated from commonly known formula proposed by Dittus-Boelter:

$$Nu = 0.023 \cdot Re^{0.8} \cdot \mathrm{Pr}^{0.4} \tag{5}$$

Table 1.

Fitting function parameters for equation (4)

	helical angle of micro fins								
	0 deg	**10 deg**	**20 deg**	**30 deg**	**40 deg**	**50 deg**	**60 deg**	**70 deg**	**90 deg**
y_0	0.0175	0.0189	0.0189	0.0208	0.0230	0.0143	0.0271	0.0237	-195.4
A_1	0.0069	0.0379	-0.0302	-0.0088	-0.0509	0.0210	0.0353	0.0444	-108.2
t_1	216766.9	3623.3	5145.2	29895.2	44221.1	6607.6	12905.4	3131.6	10794200
A_2	0.0349	0.0149	0.0739	0.0444	0.0534	0.0204	-0.0087	0.1565	303.6
t_2	3858.9	14854.9	5145.2	5623.2	52565.0	6616.1	38227.2	29384.8	30599400
A_3	0.0133	0.0058	0.0103	0.0151	0.0431	0.0134	14.0866	-0.1362	0.0323
t_3	20516.3	85716.3	75466.8	64774.9	6582.3	1565010	614.4	33005.7	6524.2

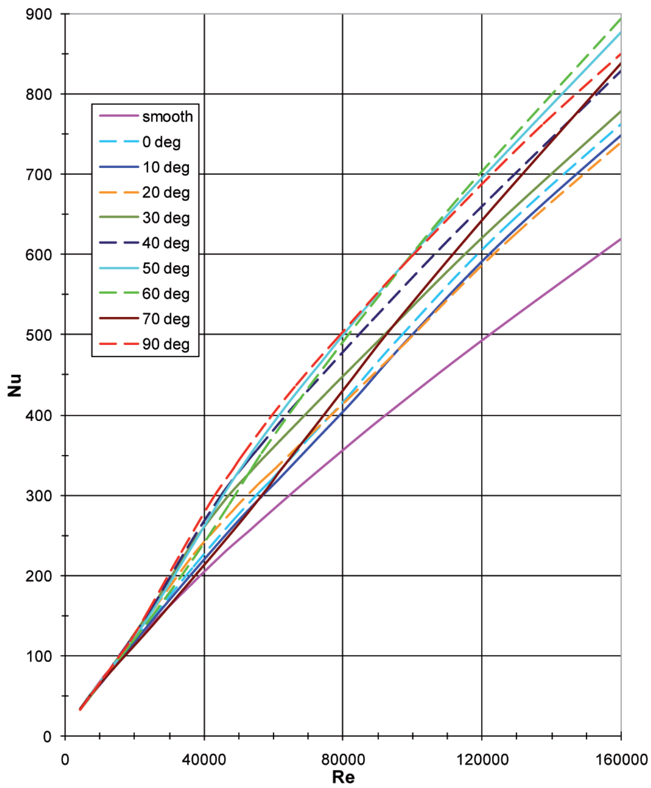

Fig. 10. Nu number vs. Re number for all helical angles of micro-fins and smooth pipe (Dittus-Boelter formula) – as a level of reference

In Fig. 10, there is shown one curve for smooth pipe calculated from (5), treated as a reference level. One can observe that Nu number for all tubes increases as Re number increases, more steeply than for a plain tube.

Fig. 11b presents a better view on this phenomenon, where the ratios of Nu number of tested tubes to Nu number of the plain pipe (Nu/Nu_{plain}) are shown. The rate of enhancement of the heat transfer of helical micro-finned tubes referred to the plane one is different, but for all tested range of Re number its values are between 1 and 1.45.

Generally, the presented results show completely irregular pattern of Nu numbers for various fluid flow rates. In the range of Re=10000÷ 80000, the best heat transfer augmentation was obtained for the tube with 90 deg helix angle, whereas in the range of 80000÷160000 of Re number the greatest values of heat transfer were achieved by the tubes with helical angles of 50 and 60 deg. These heat transfer characteristics demonstrate a significant influence of swirls in laminar boundary layer, which depend, among other things, on the micro-fin helix angle as well as the Re number. There is no apparent relationship representing the course of the presented curves, Nu vs. Re, (for different helical angles), however, it is possible to find some mathematical functions that fit for them.

The simplest fitting function, which can approximate all the characteristics, is the power function giving by equation (6):

$$Nu = A \cdot Re^{B} \cdot \mathrm{Pr}^{0.4} \tag{6}$$

where A and B are variable coefficients, and Pr value is assumed as for water. The parameters of equation (6) for every helical angle are presented in Table 2.

Table 2.

Fitting function coefficients for equation (6)

	helical angle of micro fins								
	0 deg	**10 deg**	**20 deg**	**30 deg**	**40 deg**	**50 deg**	**60 deg**	**70 deg**	**90 deg**
A	0.012	0.011	0.018	0.022	0.019	0.013	0.007	0.004	0.024
B	0.873	0.879	0.834	0.825	0.843	0.881	0.928	0.970	0.824

Inaccuracy of the theoretical expression for Nu numbers (according to eq. (6), and for Re = 10000÷40000) in relation to the numerical results, is enclosed in the range between -20% and +20%, approximately. For higher Re numbers, this inaccuracy decreases and can be estimated as ±10% in the range of Re >>40000.

4.3 Efficiency index

An enhancement of heat transfer in ducts is always connected with an increase in pressure drop. These two parameters can be compared as a ratio

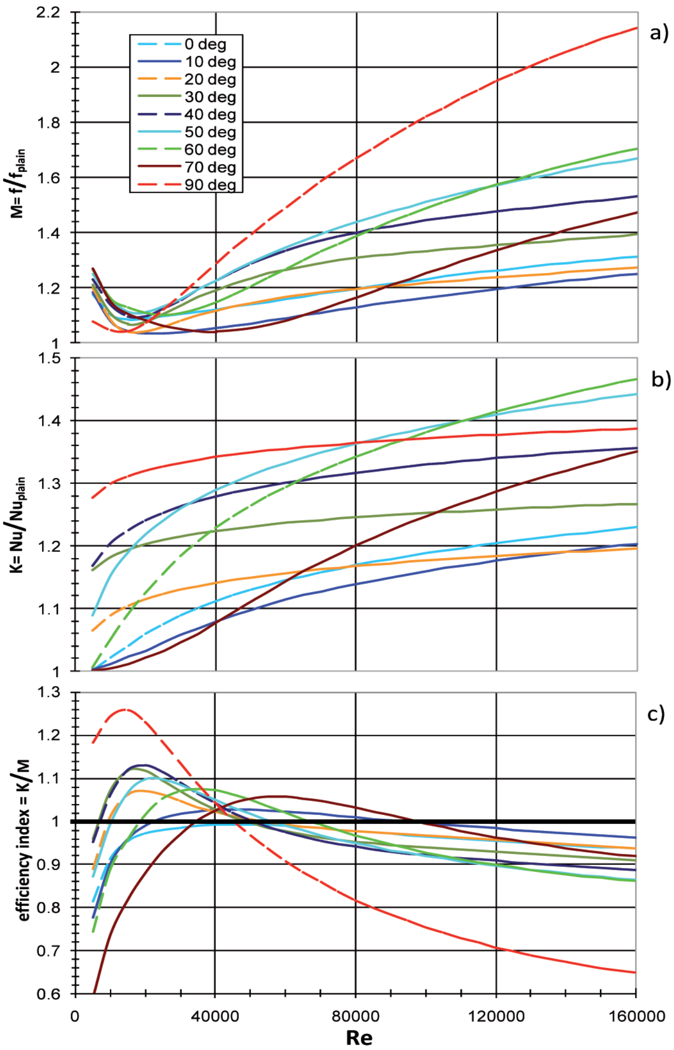

Fig. 11. Variation of the: friction factor ratio a), Nu number ratio b), efficiency index c) vs. Re number of the tested tubes

of the heat transfer increase to the increase of pressure drop, referred to that to the plain pipe. The "efficiency index" is one of the parameter used for this evaluation and is defined by:

$$\eta = \frac{Nu/Nu_{plain}}{f/f_{plain}} \tag{7}$$

This index for every helical angle is shown in Fig. 11 - c. Analyzing this plots, we can observe that the most beneficial range of Re numbers is between 10000 and 40000 where this index reaches maximum for most of the investigated tubes. The tubes with micro-finned helical angles of 60, 70 and

10 deg have a positive value of efficiency index (greater than 1) for the range of Re numbers 19000÷57000, 30000÷96000 and 20000÷93000 respectively. It follows that effective use of such tubes strongly depends on the range of Re number. With Re number values above 96000, for all tubes, this index has a value lower than 1. This "negative" value might indicate that at higher Re numbers the role of helical micro-fins in increasing the pressure drop is more significant than in increasing the heat transfer. Note that one of the tubes with 0 deg helical angle (grooves parallel to the pipe axis) in the whole range of Re number values has the efficiency index lower than 1. According to this evaluation method (efficiency index) of the heat-flow ducts, the use helical micro-fin tubes with Re numbers in the higher range brings no benefit – it is better to apply a smooth pipe.

The efficiency index (or the performance ratio) is one of the ways of evaluation which allows for optimisation of heat exchangers ducts. More complex optimisation technique of heat-flow ducts is the Entropy Generation Minimisation (EGM) method, which is based on "seeking" of minimum of entropy generated in two, irreversible opposed processes: heat transfer and pressure drop. This method, applied for the micro-fins geometries, is presented in [8,12].

5. Conclusions

1) The friction factor and Nusselt number for different angles of micro-finned tubes can be appropriately predicted in the range of Re number from 10000 to 160000 by applying relationships (4) and (6), using coefficients given in Table 1 and 2.

2) Despite the fact that the helical micro-fins and a rough smooth pipe have the same relative roughness (H/d =0.02), it is impossible to calculate the friction factor characteristics of micro-fin tubes from the equation for rough smooth pipe (3). The friction factor strongly depends on the helical angle of micro-fins.

3) The effect of Re number on the efficiency index was investigated for all tested tubes at different Re numbers. For the tubes with 90, 50, 40, 30 and 20 deg of helical angle, this ratio is higher than 1 in the narrow range of Re numbers, about 10000÷40000. The tube with a 70 deg helix angle has a wider range of Re numbers where this index is greater than 1, from about 30000 to 96000. The efficiency index for one of the pipes (with helical angle of 0 deg) does not exceed 1 in the whole range of Re numbers, which means that such a pipe is less efficient than the a smooth tube.

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Numeryczne badanie charakterystyk cieplnych i współczynników tarcia turbulentnych przepływów jednofazowych w rurach z mikro-ożebrowaniem helikalnym

S t r e s z c z e n i e

W tym artykule przedstawiono badania wymiany ciepła i spadku ciśnienia rur z mikro-ożebrowaniem śrubowym z wykorzystaniem symulacji numerycznych. Dla wszystkich badanych geometrii założono stały strumień ciepła na ściance oraz w pełni rozwinięty, turbulentny przepływ 3D. Badano wpływ kąta mikro-ożebrowania ścianki (odniesionego do osi rury) na charakterystyki cieplno-przepływowe takiego kanału. Wartość wspomnianego kąta zmieniano – co 10° – w zakresie od 0 do 90° (odpowiada on rowkom: równoległym i prostopadłym do osi rury). Przed wykonaniem symulacji numerycznych, zbadano doświadczalnie rurę z 30° kątem mikro-ożebrowania i porównano tak otrzymane rezultaty. Z symulacji numerycznych uzyskano charakterystyki współczynnika tarcia i liczby Nusselta w zakresie liczb Reynoldsa $10^4 \div 1.6 \times 10^6$. Ponadto, na wykresach zostały pokazane współczynniki (f/f_{plain}) oraz (Nu/Nu_{plain}) a także wskaźnik efektywności $(Nu/Nu_{plain})/(f/f_{plain})$ w funkcji liczby Reynoldsa dla każdego kąta mikro-ożebrowania. Analiza wyników wykazuje duży i nierównomierny wpływ kąta mikro-ożebrowania zarówno na spadki ciśnienia jak i na charakterystyki cieplne.