EXPERIMENTAL DETERMINATION OF THE HEAT TRANSFER COEFFICIENT IN INTERNALLY RIFLED TUBES

by

Slawomir GRADZIEL[a*], Karol MAJEWSKI[b], and Marek MAJDAK[a]

[a]Institute of Thermal Power Engineering, Cracow University of Technology, Cracow, Poland
[b]EthosEnergy, Wroclaw, Poland

Original scientific paper
https://doi.org/10.2298/TSCI19S4163G

Development of the heat transfer surfaces on the tube inside makes it very difficult or even impossible to determine the heat transfer coefficient analytically. This paper presents the experimental determination of the coefficient in an internally rifled tube with spiral ribs. The tests are carried out on a laboratory stand constructed at the Institute of Thermal Power Engineering of the Cracow University of Technology. The tube under analysis has found application in a supercritical circulating fluidized bed boiler. The heat transfer coefficient local values are determined for the Reynolds numbers included in the range of ~6000 to ~50000 and for three ranges of the heating elements power. As the medium flows through internally rifled tubes with spiral ribs, the heat transfer process gets intensified compared to similar processes taking place in smooth tubes. Based on the obtained experimental data, a correlation is developed enabling determination of the dimensionless Chilton-Colburn j factor. The equation form is selected so that a comparison with existing results of tests performed on rifled tubes can be made. Comparing the Nusselt number values calculated based on the developed correlation with those obtained using other correlations described in the literature, it can be observed that the criterial number is about twice higher. The research results confirm the thesis that the element internal geometry has a substantial impact on the heat transfer process.

Key words: *heat transfer coefficient, helically internally ribbed tubes, supercritical power boiler*

Introduction

The heat transfer phenomenon occurs in some applications, from those ensuring the safe operating temperature of electronic equipment to complex production processes. For this reason, there are ongoing works on the development and optimization of methods of the heat transfer intensification. The methods include 13 ways divided into two groups: active and passive [1, 2]. Each modification of the basic heat transfer surface involves a rise in the working medium flow resistance, and active heat transfer systems require additional supplies of energy. This paper describes the impact of the development of a heat transfer surface by using helical ribs on the heat transfer coefficient.

[*] Corresponding author, e-mail: gradziel@mech.pk.edu.pl

Carnavos [3] was one of the first to carry out comprehensive works on tubes with internal ribs. He presented the testing results obtained during the cooling of air. The analysis concerned flow through 21 tubes. The tubes inner diameters ranged from 3.18 to 23.8 mm. Inside the analyzed tubes there were 5 to 41 ribs with a different geometry. In most tubes they were arranged longitudinally (rib helical angle = 0°), in 6 cases, the helical angle varied from 2.5° to 20°. Using the experimental data, the author related the geometrical dimensions and the helical angle of the ribs to the heat transfer coefficient. Carnavos [3] also estimated that using tubes with internal ribs could reduce the number of tubes needed in the exchanger by a few dozen percents. Another advantage observed in the process was the potential 12-66% rise in the power of existing heat exchangers without having to increase the medium mass flux.

More works were performed by Webb *et al*. [4]. Seven rifled tubes with internal helical ribs were tested. Their inner diameter was 15.54 mm. The tests were carried out using water for Prandtl numbers from the range of 5.08-6.29. The tubes had from 18 to 45 ribs with the rib helical angle from 25° to 45° and the rib height from 0.33 mm to 0.55 mm. Compared to rough tubes, the difference between the obtained results ranged from 1.4 to 5.4%. The measurements made it possible to obtain correlations enabling determination of the friction factor and the Chilton-Colburn *j* factor for single-phase flows. Knowing the *j* factor and the definition of the Stanton and the Nusselt criterial numbers, it is possible to find the heat transfer coefficient [5]. The authors recommended the developed formulae as correct for commercially applied rifled tubes with diameters similar to that of the tubes used during the experiments.

Similar tests were performed by Zdaniuk's *et al*. [6]. Measurements were made during the water flow through tubes with internal helical ribs. The rib helical angle was included in the range of 25° to 48°, and the number of ribs in the section was from 10 to 45. The ratio between the rib height and the tube diameter totaled from 0.0199 to 0.0327. All the tubes used in the tests had the outer diameter of 18.8 mm and the wall thickness of 0.7 mm. The rib height was from 0.31 to 0.51 mm [6]. The obtained measuring data were compared with the Dittus-Boelter equations for the convective heat transfer. The results were also used to develop the heat transfer coefficient formulae [7].

Pan *et al*. [8] present the results of testing the water flow through vertical rifled tubes. The experiment was performed at pressure and heat flux values ranging from 12 to 30 MPa and from 133 to 719 kW/m^2, respectively. The tested tube inner diameter was 20 mm. It had four 0.92 mm-high ribs arranged with the pitch of 19 mm and with the helical angle of 39.5°. The results included the wall temperatures at the fluid different pressure and the local heat transfer coefficients for different values of the heat flux.

Zhu *et al*. [9] dealt with the phenomenon of self-compensation in the case of the flow of water through parallel rifled tubes with a hydraulic diameter of 15.24 mm. The paper described the testing full procedure and the phenomena arising in the water flow through tubes heated with a different heat flux. The results were obtained for different values of pressure and of the heat flux used to heat the analyzed tubes.

Li *et al*. in [10] and Yang *et al*. [11] presented numerical calculations of the heat transfer coefficient in vertical rifled tubes for supercritical CO_2. The analysis covered the impact of changes in the rib helical angle on the wall temperature and the flowing fluid velocity fields. In both cases, the tested tube had an inner diameter smaller than 20 mm.

Xu *et al*. [12] presented experimental testing aiming to determine the heat transfer coefficient for Therminol 55 in an internally ribbed tube. The tube inner and outer diameters are 14.2 and 28.8 mm, respectively. The rib height e = 0.85 and there are four ribs with the helical angle of 54°.

Zhang *et al*. [13] carried out tests of a vertical electrically-heated tube with upward flow. The heated test section was 2510 mm long. The tube geometrical dimensions were as follows: the tube outer diameter – 33.40 mm, the tube inner diameter – 20.62 mm, the rib height – 1.25 mm, the number of ribs – 6, and the rib helical angle – 50^o. The testing parameters in the electrically-heated test section were as follows: pressure – (22.5-28) MPa, mass flux – (400-1000) $kg/(m^2s)$, and heat flux – (300-700) kW/m^2. A new correlation was developed for the analyzed tube that enables the determination of the heat transfer coefficient of supercritical water, which depends on the load and the flowing fluid mass flux.

A rifled tube with the outer and inner diameters of 31.8 and 20.2 mm, respectively, was analyzed in [14]. Four ribs were considered with the rib height of 1.24 mm and the helical angle of 50^o. The tests were performed on a stand with subcritical water as the working medium. A new correlation was developed that enables the determination of the supercritical water heat transfer coefficient.

The aforementioned publications presented experimental testing of tubes with internal spiral ribs. The testing was mainly related to the heat transfer. The experiments were performed for different fluids and tubes with different geometries. The diameter values applied in studies suggest that most of the analyzed tubes had an inner diameter smaller than 20 mm. Such tubes can be used in the design of industrial heat exchangers, or they make up a part of cooling or air-conditioning installations. First tests were carried out using air. Later on, water under different pressure or cooling agents were used as the working medium.

Tubes with internal helical ribs are also applied in large industrial installations. One example is the use of rifled tubes to make the evaporator or the evaporator elements in power boilers [15, 16]. Rifled tubes are suitable for heating surfaces which are most exposed to the impact of big thermal loads. They are also good for zones where film boiling may occur. Spiral ribs induce the fluid helical flow and a centrifugal force that throws the fluid droplets onto the wall. Due to that, a water film is kept on the tube surface. A water layer on the heat transfer surface prevents a boiling crisis and thereby protects the evaporator tubes from exceeding the applied steel limit temperatures. These advantages were used by the Siemens company to develop a steam boiler based on the Benson solution [17, 18].

One of the manufacturers of boilers using rifled tubes that should be mentioned is the Foster Wheeler company [19]. Based on the Benson boiler, its engineers designed module circulating fluidized bed (CFB) boilers that can be used for different fuels and a wide range of pressure values. A CFB supercritical boiler with the electric power of 460 MW was built at the Lagisza power plant in Poland. The thermally most loaded surfaces of the boiler – the X-walls – are made of tubes with internal helical ribs.

Summing up, the main task of tubes with internal spiral ribs is to force a change in the fluid flow structure, which becomes helical. This makes it possible to limit or shift the rise in the tube temperature due to a boiling crisis. For this reason, heat exchangers or their elements which are the most exposed to the impact of high heat fluxes are frequently made of tubes with internal spiral ribs.

The presented analysis proves that very few experimental tests are performed on internally rifled tubes applied in industrial installations such as power boilers. The analyses of the heat transfer in the boiler evaporator made of rifled tubes have so far been carried out using correlations developed for tubes installed in compact heat exchangers with different fluids and smaller diameters.

This paper presents experimental testing carried out on a newly-built stand equipped with a tube with internal helical ribs which finds application in the evaporator of a supercritical CFB boiler. The analyzed tube inner diameter is almost 2.5 times bigger compared to the tubes presented in the literature. Based on the obtained experimental data, a correlation is developed enabling determination of the dimensionless Chilton-Colburn j factor. The equation general form is selected so that a comparison with existing results of tests performed on rifled tubes can be made. A comparison is also made between the Nusselt number values calculated based on the developed correlation and those obtained using other correlations described in the literature.

Heat transfer coefficient in tubes with internal helical ribs

The criterial number that defines the convective heat transfer intensity is the Nusselt number expressed:

$$\mathrm{Nu} = \frac{\alpha l}{k} \tag{1}$$

where α [Wm^{-2}K^{-1}] is the heat transfer coefficient, l [m] – the characteristic dimension, and k [Wm^{-1}K^{-1}] – the heat conductivity coefficient.

Rearranging the eq. (1), the correlation defining the heat transfer coefficient can be obtained. For tubes with internal helical ribs, the Nusselt number determination is rather troublesome. In many works, the heat transfer coefficient is found using the dimensionless Chilton-Colburn j factor defined [5]:

$$j = \mathrm{St}\,\mathrm{Pr}^{2/3} \tag{2}$$

The Stanton number has the following form:

$$\mathrm{St} = \frac{\mathrm{Nu}}{\mathrm{Pr}\,\mathrm{Re}} \tag{3}$$

where Pr is the Prandtl number and Re – the Reynolds number.

Using eqs. (2) and (3), the following form of the relation describing the heat transfer coefficient is obtained:

$$\alpha = j c_p G \mathrm{Pr}^{-2/3} \tag{4}$$

where c_p [Jkg^{-1}K^{-1}] is the specific heat and G [kgm^{-2}s^{-1}] – the mass flux.

Many publications point to a correlation that relates the j factor to characteristic dimensions of internally ribbed tubes. The correlations are based on the Reynolds number, the number of ribs in a given cross-section, the rib height-to-inner diameter ratio, and the rib helical angle.

The following relations that make it possible to determine the dimensionless Chilton-Colburn j factor are presented in [6, 7, 20]:

$$j = 0.029\mathrm{Re}^{-0.347} N^{0.253} \left(\frac{e}{d_i}\right)^{0.0877} \beta^{0.362} \tag{5}$$

$$j = 0.0206\mathrm{Re}^{-0.219} N^{0.220} \left(\frac{e}{d_i}\right)^{0.486} \beta^{0.544} \tag{6}$$

where N [–] is the number of ribs, e [mm] – the rib height, d_i [mm] – inner diameter (ribs omitted), and β [°] – rib helical angle.

Equation (5) is developed based on the studies presented in [6], and the coefficients which occur in the formula are calculated using the least squares method. In the case of eq. (6), the coefficients are selected based on calculations using neural networks [7]. Zdaniuk *et al*. [20] created a correlation using five simple groups of parameters that describe the geometry, the number and the helical angle of ribs. The equations presented in the mentioned works are compared to the results obtained by other research teams.

A correlation similar in form to eqs. (5) and (6) is developed and presented in [4]:

$$j = 0.00933\mathrm{Re}^{-0.181} N^{0.285} \left(\frac{e}{d_i}\right)^{0.323} \beta^{0.505} \tag{7}$$

The authors observed that the histories of results obtained for tested rifled tubes with low ribs were similar to rough tubes, and the difference was about 5.5%.

Test stand for experimental determination of the heat transfer coefficient in rifled tubes

The heat transfer coefficient was determined on a test stand recently constructed at the Institute of Thermal Power Engineering of the Cracow University of Technology, Cracow, Poland.

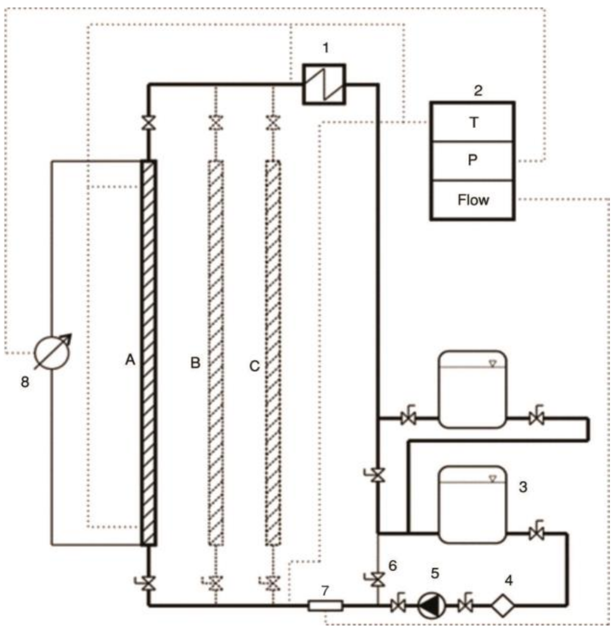

Figure 1. Diagram of the test stand: *A – rifled tube used to determine the heat transfer coefficient, B, C, – other tested objects, 1 – cooler, 2 – data acquisition system (T – temperature and P – pressure), 3 – circulating water containers, 4 – contamination filter, 5 – circulation pump, 6 – bypass; 7 – flowmeter, 8 – heating elements power control system*

Diagrams illustrating different test stands used for such experimental testing are presented in [6, 8, 9, 21, 22]. The object of the experimental testing is a tube with internal helical ribs which finds application in a supercritical CFB boiler. The analyzed tube cross-section is presented in fig. 2.

The markings of the tested element geometrical dimensions are shown in fig. 3, and the characteristic quantities are listed in tab. 1 [23-26].

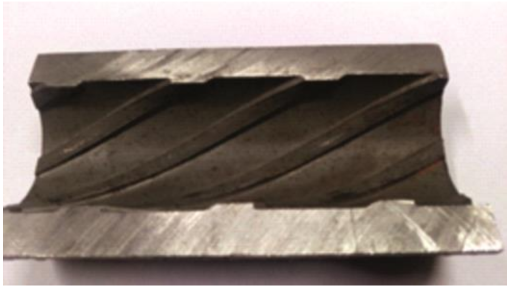

Figure 2. Cross-section of the tested tube with internal helical ribs

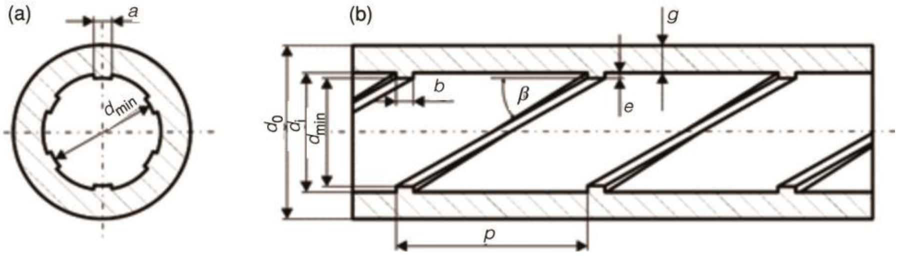

Figure 3. Geometrical dimensions of rifled tubes; (a) transverse section and (b) longitudinal section; *a – rib width at the base, b – rib average width, d_o – outer diameter, d_i – inner diameter (with no ribs), d_{min} – minimum diameter with ribs, e – rib height, g – wall thickness, p – rib pitch, β – rib helical angle*

Table 1. Characteristic dimensions of the experimentally tested tube with internal helical ribs

Characteristic dimension	Value
Outer diameter, d_o	50.8 mm
Inner diameter (without ribs), d_i	34.9 mm
Minimum diameter, d_{min}	32.9 mm
Wall thickness, g	7.95 mm
Rib height, e	1 mm
Pitch, p	30 mm
Rib width at the base, a	5 mm
Rib average width, b	4.5 mm
Rib helical angle, β	30°
Number of ribs in the cross-section, N	6

A 5-metre long tube was used for the experiments. The tube length was selected to ensure a sufficient entrance length for the fluid-flow. The entrance length totaled 1.5 m, which is more than 40-fold of the inner diameter. The heat transfer coefficient is determined in a heated 3-metre long vertical section. The thermocouples measuring the temperature close to the wall inner surface are located along the tube in 7 measuring points which are 0.5 m from each other. Each measuring point is made of 5 thermocouples arranged uniformly on the tube perimeter. This is related to the fact that ribs disturb the temperature field in the wall. By installing five thermocouples, a good approximation of the mean temperature on the inner surface can be achieved. The temperature field is mapped well because the local temperature is measured not only in the rib zone or on the surface between the ribs, but in a randomly select-

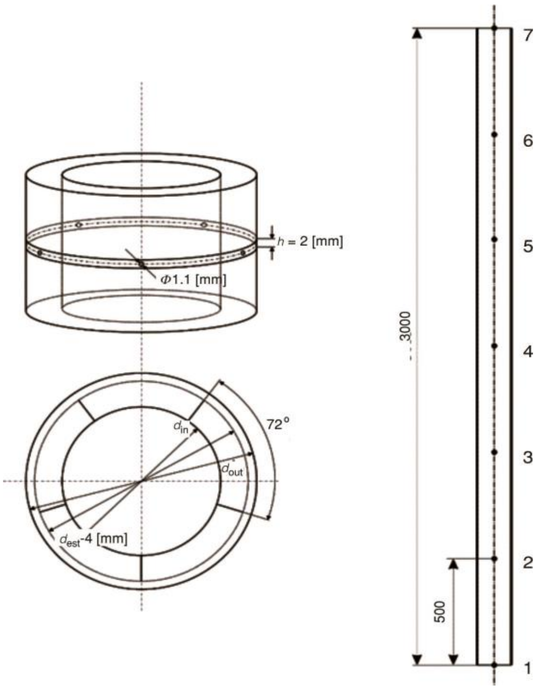

Figure 4. The arrangement of measuring points and detail illustrating the location of thermocouples on the tube perimeter

ed location. The arrangement of the measuring points and the exact location of the thermocouples are shown in fig. 4.

The wall temperature was measured using jacket thermocouples with a diameter of 1 mm. The diameter of the holes where the sensors were placed was 1.1.mm. The thermocouples are located as close to the tube inner surface as possible. In order to reduce heat losses to the environment through the thermocouple jacket and to increase the temperature measurement accuracy, and also to make it possible to carry out the thermocouples in one point, a groove was cut on the entire perimeter to hold the sensors. The groove is 2 mm deep and 2 mm wide, fig. 4.

Band heaters were put in between the measuring points. Each heater was almost 24 cm high. This means that there were two heaters in between the measuring points and their location is almost level with the edge of the groove used to carry out the thermocouples. The entire area was insulated against the effect of high temperatures.

Determination of the heat transfer coefficient

The heat transfer coefficient was calculated using 37 temperature measurements (35 close to the inner wall surface and two water temperature measuring points), and measurements of the water volume flow rate and momentary power of the heaters. Using the measured quantities, the local heat transfer coefficient values were found in the following relation:

$$\alpha = \frac{\dot{q}}{T_\mathrm{w} - T_\mathrm{b}} \tag{8}$$

where $\dot{q}$ [Wm^{-2}] is the heat flux, T_w [K] – the temperature on the wall inner surface, and T_b [K] – the mass-averaged temperature of the flowing fluid.

The mean heat flux value was determined on the outer surface:

$$\dot{q}_\mathrm{out} = \frac{Q}{A_o} = \frac{Q}{\pi d_o L} \tag{9}$$

where Q [W] is the heaters power, A_o [m^2] – the tube outer surface area, L [m] – the heated section length, and d_o [m] – the outer diameter.

The mean heat flux calculated using eq. (9) had to be related to the inner surface area, and it is in this very form that it is used in eq. (8). Apart from the heat flux value, in order to find the heat transfer coefficient, the temperature difference between the fluid and the wall has to be known. The fluid temperature was measured at the heated section inlet and outlet. So that relation (8) could be used, the water temperature was interpolated for the wall temperature measuring point. The fluid temperature approximation is possible assuming that the mean heat flux value on the outer surface is constant over the heated section length. For this reason, the power of the heaters in both heating sections was constant throughout the experiments. The wall temperature was measured using the measuring points presented in fig. 4.

Experiment

The measuring series were carried out for three set levels of the heaters power. The total power of the heating elements during the testing totaled about 6, 7, and 8 kW, respectively. The power range of the heating elements was selected so that the biggest possible increment in the flowing water temperature should be obtained. Some oscillations in the achieved power of the heaters were observed during the testing, which was related to variations in the power network voltage. Mean values were adopted for the calculations. The powers of the heating elements and the values of the heat flux obtained during individual measuring series are listed in tab. 2.

Table 2. Mean power of heaters and obtained values of heat flux on the tested tube outer and inner surfaces

Mean power of heaters during individual measuring series, [W]	Mean heat flux on the outer surface, [Wm^{-2}]	Mean heat flux on the inner surface, [Wm^{-2}]
7989	16686	24287
7093	14814	21563
6004	12540	18253

The measuring data were archived in time intervals of 2 seconds. The analysis was conducted for selected periods where the difference in the water temperature measured between the outlet and the inlet was constant for at least ten measuring periods. For the power of the heaters of about 8 kW, 68 periods were selected, for the power close to 7 kW, there were 77 periods, and for the power of about 6 kW, 60 measuring periods were taken. The local value of the heat transfer coefficient was determined in points of the wall temperature measurement close to the tube inner surface. The selected measuring data made it possible to obtain over 1400 values of the heat transfer coefficient in total. The final analysis was performed for selected values obtained in measuring points 2-6, fig. 4. This was related to the fact that in the outermost locations the measuring point for the temperature on the wall inner surface was heated on one side only. Due to that, the heat transfer coefficient values in points 1 and 7 reached up to 10 times the mean value obtained in the other measuring points. The deviations from the mean values arise because the denominator of the applied eq. (9) includes the temperature difference between the fluid and the wall. In the case of the outermost points, the difference was much smaller compared to the other points under analysis. Assuming a constant heat flux along the entire length of the tested section, this produced very high and, at the same time, incorrect values of the heat transfer coefficient. For this reason, only 1025 of the heat transfer coefficient calculated values were finally analyzed. The heat transfer coefficient local values for the performed measurements are presented in figs. 5-7: fig. 5 – for the power of the heaters of about 8 kW, fig. 6 – for the power of the heaters of about 7 kW, and fig. 7 – for the power of the heaters of about 6 kW.

In many publications on the phenomena considered in this paper, the obtained data are used in the form of the Chilton-Colburn analogy. Using eqs. (2)-(4) and the data obtained from the experiments, the Chilton-Colburn j factor was calculated. The factor values obtained for all the used powers of the heaters are presented in one chart, fig. 8. The experimental corr-

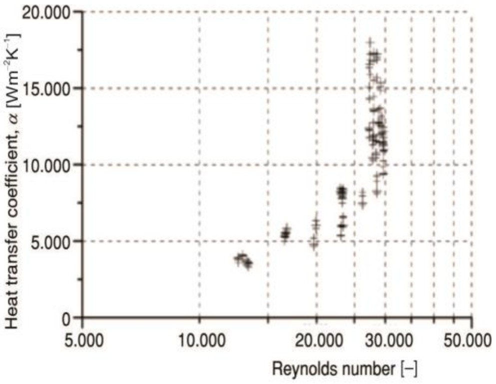

Figure 5. Heat transfer coefficient experimental values (power of heaters – 8 kW)

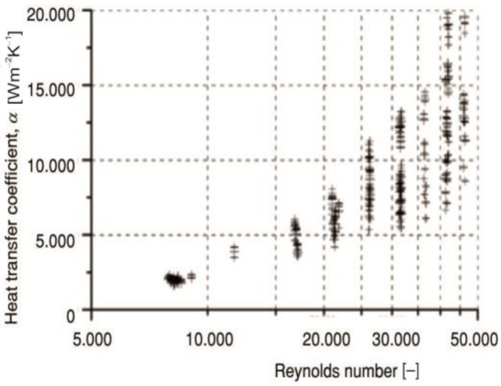

Figure 6. Heat transfer coefficient experimental values (power of heaters –7 kW)

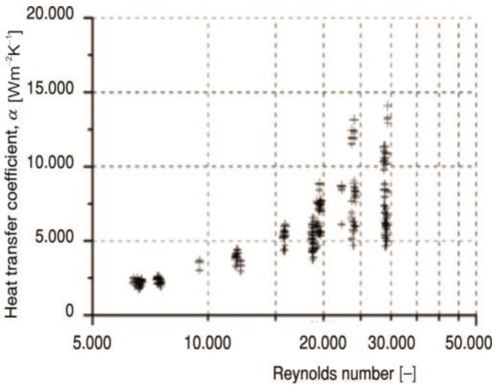

Figure 7. Heat transfer coefficient experimental values (power of heaters – 6 kW)

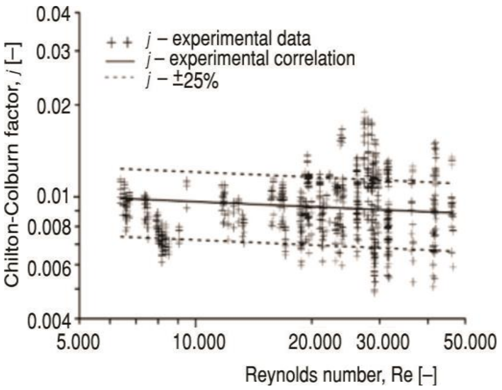

Figure 8. Chilton-Colburn *j* factor, experimental data and the proposed function history

correlation was also developed. The equation form is selected such that enables comparison with eqs. (5)-(7), which were developed for tubes with internal spiral ribs with other diameters and rib geometries. The correlation developed based on the experimental data takes the following form:

$$j = 0.010\mathrm{Re}^{-0.055} N^{0.010} \left(\frac{e}{d_i}\right)^{0.323} \beta^{0.505} \tag{10}$$

where $\beta = 30^{\circ}$ and $e/d_i = 1/34.9$.

The developed correlation for the heat transfer coefficient determination is not universal and it concerns a tube with the geometry used during the testing. Further studies are needed to develop a universal correlation for tubes with a similar inner diameter classed as tubes with micro-ribs.

The values obtained from eq. (10) are presented in fig. 8 with a 25% confidence interval. In literature, the confidence interval is often up to 30%. This is justified because the heat transfer determination process is difficult and the obtained values are often local and momentary. Analyzing the values produced by eq. (10) at the assumed confidence interval, it can be seen that for Reynolds numbers of Re < 25000, most of the calculated values are included in the assumed range.

The determined Chilton-Colburn j factor was compared with the literature results. The values developed using correlation (10), and eqs. (5)-(7) are presented in fig. 9. The application range of functions (5)-(7) covers the Reynolds number values from 20000 to 60000. The lower limit of the application range of eqs. (5)-(7) is represented by the vertical line.

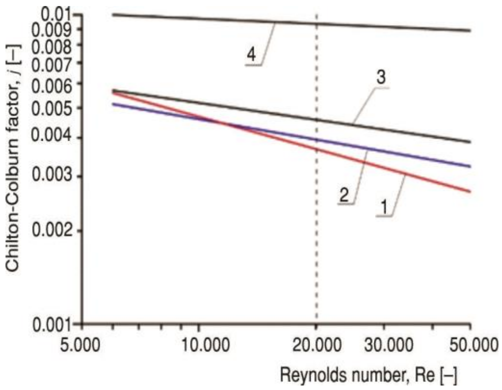

Figure 9. Comparison between histories of the *j* factor calculated using respective equations; *1 – eq. (5), 2 – eq. (6), 3 – eq. (7), and 4 – eq. (10)*

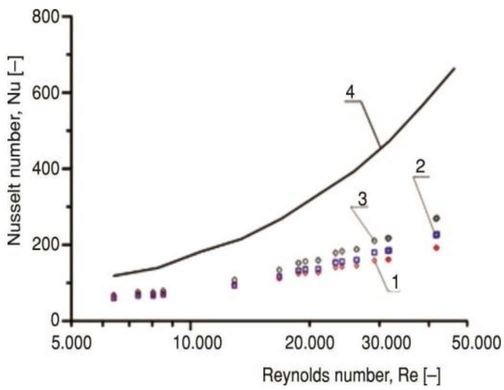

Figure 10. Comparison between the Nusselt number values obtained using selected correlations; *1 – eq. (5), 2 – eq. (6), 3 – eq. (7), and 4 – eq. (10)*

The functions compared in fig. 9 are extended below the analyzed range. If Re < 20000, the presented correlations may not map real quantities. The developed function (10) was constructed based on testing with Reynolds numbers from the range of about 6000 to 50000. Analyzing the histories of the compared functions, it can be seen that the Chilton-Colburn j factor calculated using relation (10) takes twice bigger values than the results obtained from the other equations. The differences may be an effect of *e. g.* the fact that the correlations were established based on the testing of tubes with about a twice smaller diameter and with a bigger number of ribs than the object tested within the present works.

The calculated values of the Chilton-Colburn j factor were used to determine the Nusselt number values. A graphical comparison between selected correlations is presented in fig. 10.

Conclusions

This paper presents the experimental determination of the heat transfer coefficient in an internally rifled tube with spiral ribs. The tests were carried out on a laboratory stand recently constructed at the Institute of Thermal Power Engineering of the Cracow University of Technology. The tested object was a tube that has found application in a supercritical CFB boiler.

Based on the obtained experimental data, a correlation was developed enabling determination of the dimensionless Chilton-Colburn j factor. The equation general form is selected so that a comparison with existing results of tests performed on rifled tubes can be made. The developed correlation is not universal and it concerns a tube with the geometry used during the testing, where $\beta = 30^{\circ}$ and $e/d_i = 1/34.9$. Further studies on similar tubes are needed to develop a universal correlation for tubes with a similar inner diameter, classed as tubes with micro-ribs. Comparing the Nusselt numbers calculated based on the correlations developed for flows through tubes with internal spiral ribs with the inner diameter of about 18 mm, an approximately two-fold increase in this criterial number can be noticed.

The heat transfer coefficient values calculated based on the measuring data display a substantial convergence, which can be observed in figs. 5-7. For Reynolds numbers higher than 25000, many of the determined values of the heat transfer coefficient are located beyond the assumed confidence interval of ±25%. This is the effect of the limitations related to the maximum power of the heating elements and of the fact that the tested tube outer diameter is about 2.5 times bigger than in the studies presented in the literature. During the performed experimental tests, the maximum heat flux on the outer surface totals about 16.7 kW/m^2, which makes it possible to reach on the tube inner surface value of about 24.3 kW/m^2. In some research works, the heat flux obtained on the tube outer surface exceeded 250 kW/m^2. However, this was the case for tubes with a much smaller outer diameter compared to the tube being the object of the experiment.

References

[1] Webb, R., Kim, N.-H., *Principles of Enhanced Heat Transfer*, Taylor & Francis, Oxford, UK, 2005
[2] Bergles, A., *et al*., *Heat Transfer Enhancement of Heat Exchangers*, Kluwer Academic Publishers, Boston, Mass., USA, 1998
[3] Carnavos, T., Cooling of Air in Turbulent Flow with Internally Finned Tubes, *Heat Transfer Engineering*, *1* (1979), 2, pp. 41-46
[4] Webb, R., *et al*., Heat Transfer and Friction Characteristics of Internal Helical-Rib Roughness, *Journal of Heat Transfer*, *122* (2000), 1, pp. 134-142
[5] Hewitt, G., *et al*., *Process Heat Transfer*, Begell House: CRC Press, Boca raton, Fla., USA, 1994
[6] Zdaniuk, G., *et al*., Experimental Determination of Heat Transfer and Friction in Helically-Finned Tubes, *Experimental Thermal and Fluid Science*, *32* (2008), 3, pp. 761-775
[7] Zdaniuk, G., *et al*., Correlation Heat Transfer and Friction in helically-Finned Tubes Using Artificial Neural Networks, *Heat and Mass Transfer*, *50* (2007), 23, pp. 4713-4723
[8] Pan, J., *et al*., Experimental Investigation on Heat Transfer Characteristic of Low Mass Flux Rifled Tube with Upward Flow, *International Journal of Heat and Mass Transfer*, *54* (2011), 13-14, pp. 2952-2961
[9] Zhu, X., *et al*., Self-Compensating Characteristic of Steam-Water Mixture at Low Mass Velocity in Vertical Upward Parallel Internally Ribbed Tubes, *Applied Thermal Engineering*, *30* (2010), 16, pp. 2370-2377
[10] Li, Z., *et al*., Improved Gas Heaters for Supercritical CO_2 Rankine Cycles: Considerations on Forced and Mixed Convection Heat Transfer Enhancement, *Appl Energy*, *178* (2016), Sept., pp. 126-141
[11] Yang, Z., *et al*., Numerical Study on the Heat Transfer Enhancement of Supercritical CO_2 in Vertical Ribbed Tubes, *Applied Thermal Engineering*, *145* (2018), Dec., pp. 705-715
[12] Xu, W., *et al*., Experimental and Numerical Investigation on Heat Transfer of Therminol heat Transfer Fluid in an Internally Four-Head Ribbed Tube, *Int. J. of Thermal Sciences*, *116* (2017), June, pp. 32-44
[13] Zhang, Q., *et al*., Experimental Study on Heat Transfer to the Supercritical Water Upward Flow in a Vertical Tube with Internal Helical Ribs, *International Journal of Heat and Mass Transfer*, 89 (2015), Oct., pp. 1044-1053
[14] Shen, Z., *et al*., Heat Transfer Characteristics of Water Flowing in a Vertical Upward Rifled Tube with Low Mass Flux, *Exp Therm Fluid Sci*, *70* (2016), Jan., pp. 341-353
[15] Taler, J., *Thermal and Flow Processes in Large Power Boilers. Modelling and Monitoring* (in Polish), Wydawnictwo Naukowe PWN, Warszawa, 2011
[16] Gradziel, S., Modern Design Solutions of Power Boilers (in Polish), *Piece Przemysłowe i Kotły*, VII-VIII, 2012, pp. 24-30
[17] ***, Siemens Power Generation, BENSON Boiler. Economic, flexible, innovative, Siemens Power Generation, Erlagen, Germany, 2001.
[18] Franke, J., Kral, R., Benson Boiler – Best Choice. Excellent Performance Features Make the Benson Boiler the Most Widely Used Type of Once-Through Boiler, *Siemens Power Journal Online*, Oct., 2001
[19] Foster Wheeler, Available: http://www.amecfw.com/.

[20] Zdaniuk, G., *et al*., Linear Correlation of Heat Transfer and Friction in helically-Finned Tubes Using Five Simple Groups of Parameters, *Heat and Mass Transfer*, *51* (2008), 13-14, pp. 3548-3555
[21] Cheng, L., Chen, T., Study of Vapour Liquid Two-Phase Frictional Pressure Drop in a Vertical Heated Spirally Internally Ribbed Tube, *Chemical Engineering Science*, *62* (2007), 3, pp. 783-792
[22] Yang, D., *et al*., Experimental Investigation on Heat Transfer and Frictional Characteristics of Vertical Upward Rifled Tube in Supercritical CFB Boiler, *Experimental Thermal and Fluid Science*, *2* (2011), 35, pp. 291-300
[23] Majewski, K., Concept of a Measurement and Test Station for Determining Linear Pressure Drop and Heat Transfer Coefficient of Internally Ribbed Tubes, *Journal of Power Technologies*, *93* (2013), 5, pp. 340-346.
[24] Gradziel, S., Majewski, K., Simulation of Thermal and Flow Phenomena in Smooth and Internally Rifled Tubes, *Journal Heat Transfer Engineering*, *39* (2018), 13-14, pp. 1243-1250.
[25] Majewski, K., Gradziel, S., CFD Simulations of Heat Transfer in Internally Ribbed Tubes, *Chemical and Process Engineering*, *37* (2016) 2, pp. 251-260
[26] Gradziel, S., Majewski, K., Simulation of Temperature Distribution and Heat Transfer Coefficient in Internally Ribbed Tubes, *Procedia Engineering*, *157* (2016), Aug., pp. 44-49

Paper submitted: January 5, 2019
Paper revised: February 22, 2019
Paper accepted: February 26, 2019