Design and experimental-based thermal characterization of heat exchangers made of wavy tubes

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Abstract:

The main objective of presented study was to study thermal characteristics of a classic concentric-tube and a multi-tube counter-flow heat exchanger (HE) that was used in a local ventilation device. Since the need for providing an acceptable level of working and living conditions dictates, among other demands, an appropriate air quality in buildings, our team designed a local ventilation device that used natural energy sources. The inner and outer tubes of the presented HE had a wavy surface, with the wave height of 3 mm. Each wave was oriented in the transversal direction, with respect to fluid flow. In order to determine the thermal characteristics, an experiment has been performed. With the Wilson-plot method the average convective heat-transfer coefficients on the inner (warm air) and outer (cold air) sides of the inner tube were separately determined. All experimentally obtained values were also compared with average convective heat-transfer coefficients for a smooth straight tube. The results reveal that the heat transfer coefficients on the inner and the outer side of the inner wavy tube of both types of HE are in most cases similar or even lower, compared to heat transfer coefficients in a smooth tube. However, the heat transfer rate in the case of wavy tubes was noticeably higher due to the increased heat-transfer surface area for both types of HE.

Keywords:

Heat exchanger, Wilson-plot method, convective heat-transfer coefficient, wavy tube.

1. Introduction

Long-term global warming trends among other factors tend to increase cooling energy demands in order to provide an acceptable level of comfort [1]. That fact is in contrary to worldwide priorities to reduce energy use and consequently to reduce greenhouse gas emissions. While heating systems are nowadays well developed and energy efficient, cooling systems on the other hand are often liable to a bad design and lower efficiency due to the economic restrictions [1, 2]. Different and randomly chosen cooling systems from various manufacturers are used, which are often inefficient and harmful to environment [3, 4]. As a consequence the use of electrical energy, which is reputed as the form of energy with the highest quality, is rapidly increasing for cooling purposes [5]. The solution to above problems is possible by direct use of energy efficient building utility systems and natural energy sources for heating and cooling purposes.

The main feature of passive heating and cooling processes is that energy for heating and cooling of air is not used [6]. It is used only for air supply into the desired area or facility. Those processes exploit natural physical properties of air at different temperatures without the use of mechanical devices. Exchange of air is thereby a result of pressure differences between the interior and the surroundings [7]. The use of passive heating and cooling processes enables the reduction of energy consumption for ventilation and cooling purposes, as well as to recuperate the heat (cold) of the internal exhaust air. In addition to use of natural energy sources there is also the possibility of increasing the efficiency of heat transfer in a local ventilation devices, which both contribute to efficient and sustainable energy based HVAC systems.

The two most significant ways to enhance the heat transfer are known. This can be done by increasing the heat transfer area or by altering the flow conditions with geometrical changes on the surface of the tube, which affects the average convective heat-transfer coefficient [8-11]. Unlike the heat transfer in concentric-tube heat exchangers (HE) made of smooth tubes, the heat transfer in concentric-tube HE made of wavy tubes is less researched.

In this article we present a study on thermal characteristics of a classic concentric-tube and a multi-tube counter-flow HE that was used in a local ventilation device. This applies natural energy sources and enables heat transfer between the exhaust and fresh outside air. The main feature of the presented HE regards the wavy surface of the inner and outer tubes. In order to evaluate the thermal characteristics we separately determined Nusselt numbers on both sides of the inner tube(s) (inside the inner tube and in annulus) using the Wilson-plot method, and compared those values with Nusselt numbers for a smooth tube.

2. Convective heat-transfer coefficients determination

2.1. The Wilson-plot method

The original Wilson-plot method, which was developed in 1915, is based on the separation of the total thermal resistance into the one-side thermal resistance and the remaining heat resistance [12]. It enabled evaluation of the film heat-transfer coefficient in shell and tube condensers. Later, the Wilson-plot method was upgraded by some researches. Those modifications made the proposed method suitable for the evaluation the convective heat-transfer coefficient in HE with single phase fluid flow [13]. The modified Wilson-plot method enable the calculation of the convective heat-transfer coefficient from measured data in circular tubes. For example we can use the well-known correlation, proposed by Dittus – Boelter:

$$Nu = C \cdot Re^{n} \cdot Pr^{0.4} \tag{1}$$

where the Prandtl number is known. The process of constant C and exponent n determination is described in detail in [14]. A modified Wilson-plot method was used to obtain the Nusselt numbers from experimental results of two types of HE made of wavy tubes.

2.1. Empirical correlations for a smooth tube

- Laminar fluid flow regime

In case of fully developed laminar velocity profile the average Nusselt number for a circular tube with the length L and the diameter d for a fixed wall temperature was calculated using correlation [15]:

$$Nu = 3.66 \tag{2}$$

which is valid for:

$$Pe \cdot d/L < 10^{2}$$

- Turbulent fluid flow regime

In case of fully developed turbulent velocity profile in a circular tube with the length L and the diameter d, the Dittus – Boelter correlation was used [15]:

$$Nu = 0.023 \cdot Re^{4/5} \cdot Pr^{m} \tag{3}$$

which is valid for:

$$30 < L/d < 120$$

$$7 \cdot 10^3 < Re < 3 \cdot 10^5$$
$$0.5 < Pr < 120$$

❖ Transitional fluid flow regime

In case of Reynolds numbers, which belong to a transitional fluid flow regime, the Petukhov – Kirillov correlation, modified by Gnielinski was used [15]:

$$Nu = \frac{\frac{f}{2} \cdot (Re - 1000) \cdot Pr}{1 + 12.7 \cdot \frac{f}{2}^{1/2} \cdot (Pr^{2/3} - 1)} \tag{4}$$

$$f = (1.58 \cdot \ln Re - 3.28)^{-2} \tag{5}$$

which is valid for:

$$2300 < Re < 10^4$$
$$0.5 < Pr < 200$$

Another correlation for Nusselt number calculation in case of transitional fluid flow regime in a circular tube with the length L and the diameter d was proposed by Hausen [16]:

$$Nu = 0.037 \cdot \left(Re^{3/4} - 180\right) \cdot Pr^{0.42} \cdot \left[1 + \left(\frac{d}{L}\right)^{2/3}\right] \tag{6}$$

which is valid for:

$$\frac{L}{d} \approx 10$$
$$2300 < Re < 10^5$$
$$0.5 < Pr < 500$$

3. Experimental setup

The local ventilation device, used in our research, was performing with two types of HE, which are shown in Fig. 1 and Table 1:

- **Type 1:** A classic concentric-tube HE made of outer tube with diameter d_o = 130 mm and a single inner tube with diameter $d_{i,c}$ = 80 mm.
- **Type 2:** A multi-tube HE made of outer tube with diameter d_o = 130 mm and 7 inner tubes with diameter $d_{i,m}$ = 30 mm. All inner tubes were interlaced among each other.

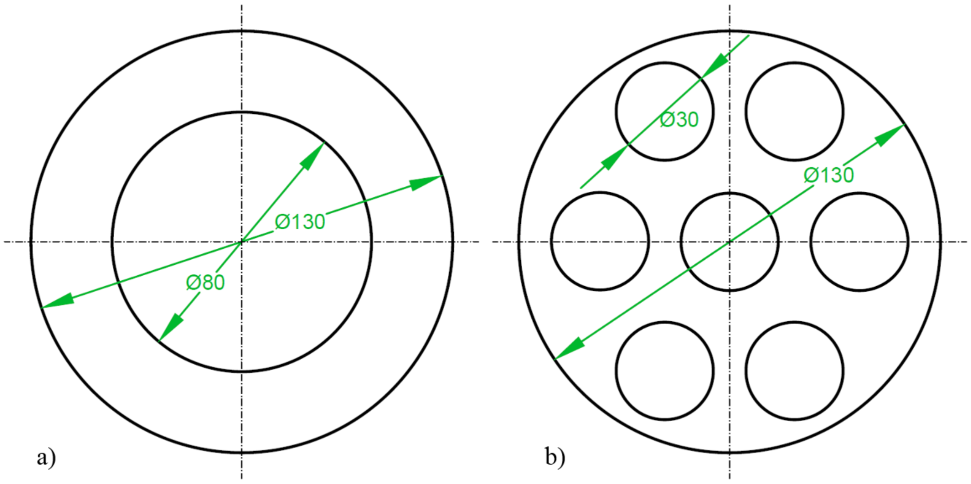

Fig. 1. Cross-section view of the classic concentric-tube HE (a) and multi-tube HE (b).

The inner and outer tubes of the presented HE had a wavy surface, with a wave height of 3 mm. Each wave was perpendicular with respect to fluid flow. An example of a wavy tube, used in both types of the presented HE is shown in Fig. 2.

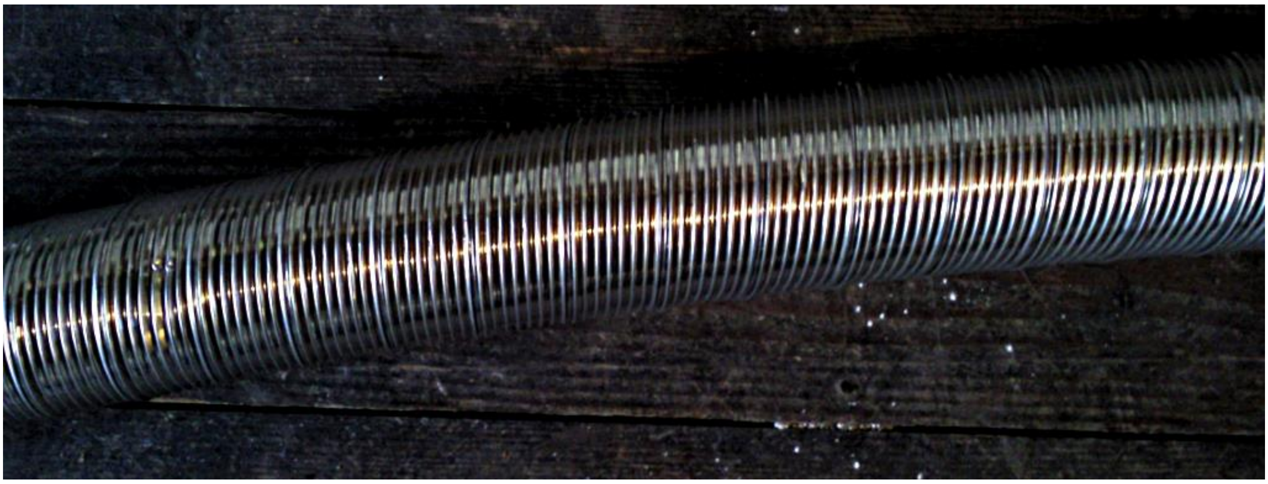

Fig. 2. An example of a wavy tube, used in a local ventilation device.

Such geometry of the tubes, used in our experiment, yielded to an increased heat-transfer surface. The heat-transfer surface was corrected with a waviness ratio. All geometric characteristics of both types of HE are presented in Table 1. Heat transfer mechanism in both HE was due to the forced convection, which was achieved by two energy-efficient fans with the electrical power of 5 W at 12 V voltage. To maintain the desired temperature of an internal exhaust air the PID-controller-based heating unit with nominal heating power of 400 W was used. Thermal insulation was custom-made with built slots for all elements of ventilation device. The material of thermal insulation was compact Styrofoam.

Table 1: Geometric characteristics of both types of HE

Type	Number of inner tubes [/]	Outer tube diameter [mm]	Inner tube diameter [mm]	Heat-transfer area (smooth tube) [m^2]	Waviness ratio [/]	Actual heat-transfer area [m^2]	Wave height [mm]
1	1	130	80	0.89	1.875	1.67	3
2	7	130	30	0.89	2.609	6.09	3

The experimental setup is shown in Fig. 3. In case of both types of HE the air temperatures were measured at the inlet and outlet on hot ($T_{1,in}$ and $T_{1,out}$) and cold ($T_{2,in}$ and $T_{2, out}$) sides of the HE. Four thermocouples of type T: Cu-CuNi were used at each measuring point. The calibration procedure was performed by using a thermostatically controlled bath. As a reference thermometer, the resistance temperature detector (RTD) was used. The air flow was measured indirectly by measuring the speed of air with a vane anemometer with a diameter of 80 mm. It was calibrated with a sharp-edged orifice.

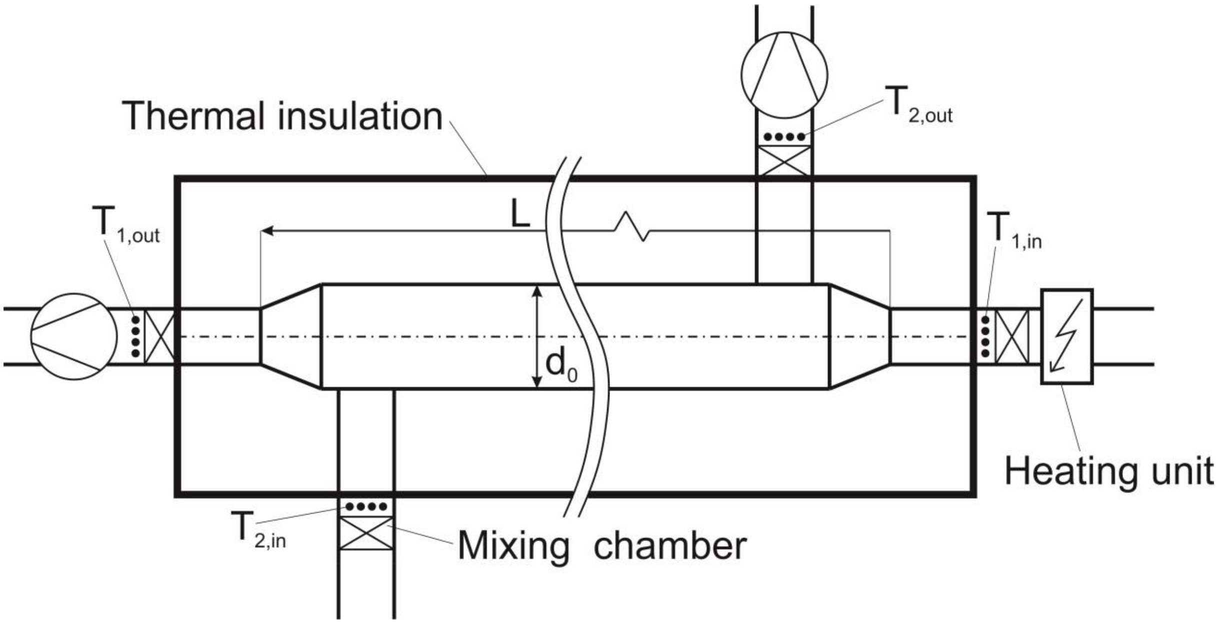

Fig. 3. The experimental setup scheme. L = 3.54 m, d_o = 130 mm.

4. Results

For both types of HE the Nusselt numbers in inner and outer side of the inner wavy tube(s) were determined using the Wilson-plot method. During the experiments the warmer air was in inner tube and the air flow range was from 10 to 40 m^3/h. For both types of HE two series of experiments were done. In each series the air flow was varied on one side of HE, while the air flow and the mean temperature were held constant on the other side.

4.1. Classic concentric-tube heat exchanger

- **Air flow variation on cold side**

Measurement results for different air flow rates on cold side of the Type 1 HE are presented in Fig. 4. It can be noticed that the values of Reynolds number are between 2000 and 4000, which indicates the transitional fluid flow regime. Empirical correlation for a smooth tube 2 (blue line), which is valid for a transitional fluid flow regime shows the best agreement with the measured Nusselt numbers in annulus (red line). Values of Nusselt numbers for a smooth tube 1 (purple line) and 3 (green line) show bigger deviation from the measured Nusselt numbers (red line). All measured results were also evaluated as if heat heat-transfer surface area was smooth, which is depicted as the black line. In this case the Nusselt numbers are significantly higher compared to the Nusselt numbers in annulus.

- **Air flow variation on hot side**

Measurement results for different air flow rates on hot side of the Type 1 HE are presented in Fig. 5. Values of Reynolds number are between 3000 and 8000. Values of Nusselt numbers, calculated from all three empirical correlations for a smooth tube 1 (turbulent flow regime), 2 and 3 (both transitional flow regime) are bigger compared to the measured Nusselt numbers (red line). All measured results were also evaluated as if heat heat-transfer surface area was smooth, which shows the black line. In this case the Nusselt numbers are even more significantly high compared to the Nusselt numbers inside the inner wavy tube.

Note: Smooth tube 1 – equation 3, Smooth tube 2 – equation 4, Smooth tube 3 – equation 6.

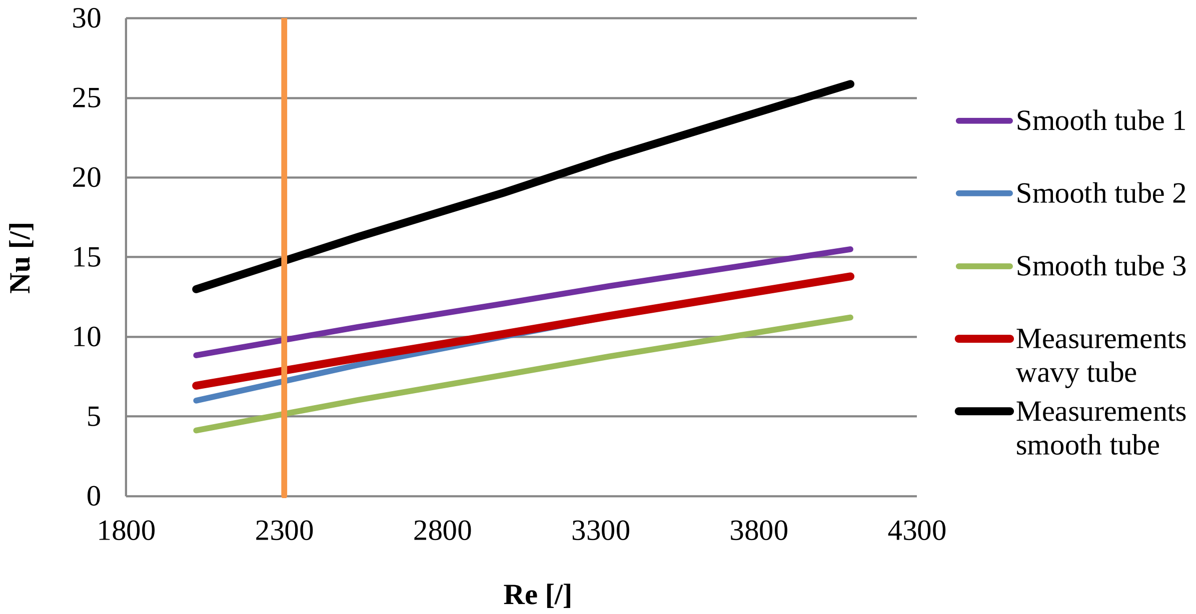

Fig. 4. Measured and calculated Nusselt numbers versus Reynolds number in annulus for wavy and smooth tubes of Type 1 HE. The orange vertical line represents transition from laminar to transitional fluid flow regime.

Note: Smooth tube 1 – equation 3, Smooth tube 2 – equation 4, Smooth tube 3 – equation 6.

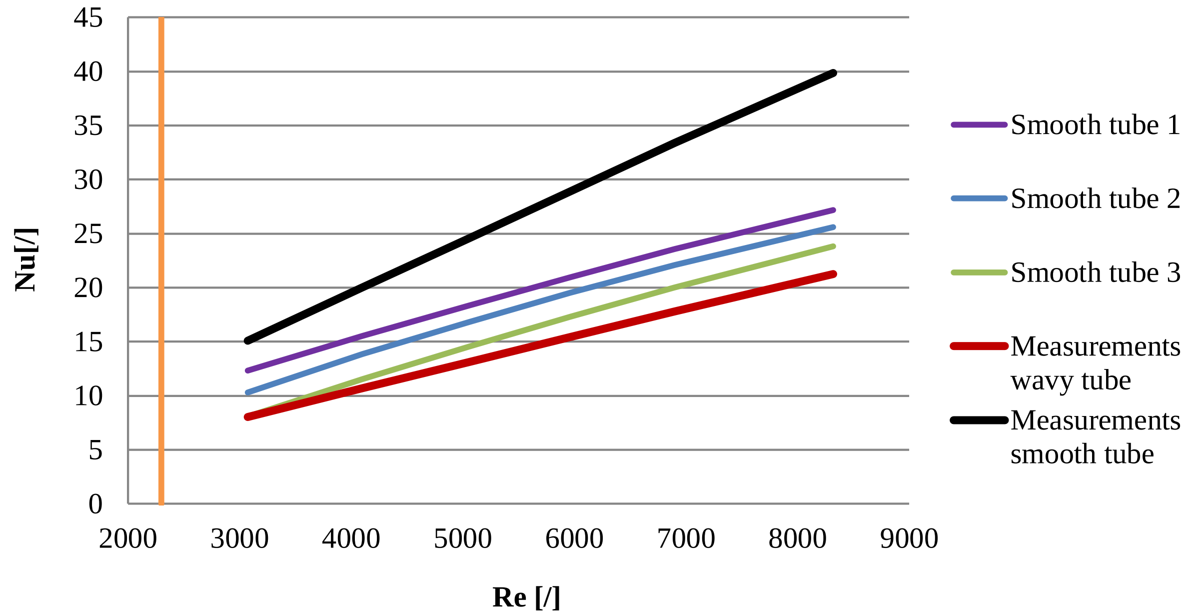

Fig. 5. Measured and calculated Nusselt numbers versus Reynolds number in the inner tube for wavy and smooth tubes of Type 1 HE. The orange vertical line represents transition from laminar to transitional fluid flow regime.

4.2. Multi-tube heat exchanger

- **Air flow variation on cold side**

Measurement results for different air flow rates on cold side of the Type 2 HE are presented in Fig. 6. It can be noticed that the values of Reynolds number are between 800 and 2400, which indicates the laminar fluid flow regime. Nusselt numbers, based on empirical correlation for a laminar fluid flow regime for a smooth tube 4 (grey horizontal line line) are independent of Reynolds number. The green line represents Nusselt numbers in case of a transitional fluid flow regime for smooth tube 3. The slope of the green line is much steeper compared to the slope of the red line, which represents Nusselt numbers in annulus. All measured results were also evaluated as if heat heat-transfer surface area was smooth, which is depicted as the black line. In this case the Nusselt numbers are significantly higher compared to the Nusselt numbers in annulus.

- **Air flow variation on hot side**

Measurement results for different air flow rates on hot side of the Type 2 HE are presented in Fig. 7. Values of Reynolds number, which are between 1200 and 3200, indicate the laminar as well as transitional fluid flow regime. Looking at the Nusselt numbers in case of Reynolds number above 2300, calculated from empirical correlation for a smooth tube 3 (transitional flow regime – green line) it can be seen that those values are in best agreement with the measured Nusselt numbers inside inner wavy tube(s) (red line). Similar for Reynolds number values of around 1600 the Nusselt numbers, calculated from empirical correlation for a smooth tube 2 (transitional flow regime – blue line) are in best agreement with the red line. Below Reynolds number of 1600 only Nusselt numbers calculated for laminar fluid flow show agreement with the red line. The purple line, which represents an empirical correlation for a smooth tube 1 (turbulent flow) deviates the most from the red line. All measured results were also evaluated as if heat heat-transfer surface area was smooth, which is depicted as the black line. In this case the Nusselt numbers are also significantly higher compared to the Nusselt numbers inside the inner wavy tube.

Based on presented results it can be concluded that for both Type 1 and Type 2 HE, heat transfer coefficients on inner and outer sides of the inner wavy tube(s), compared to a smooth tube, are similar or in some cases even lower. Apparently, the geometry of the used wavy tubes did not alter the air flow regime in such way that would increase heat transfer coefficients. However, when comparing measured Nusselt numbers for wavy tubes with measured Nusselt numbers for smooth tubes, the heat transfer rate in case of wavy tubes was noticeably higher due to the increased heat-transfer surface area.

Note: Smooth tube 3 – equation 6, Smooth tube 4 – equation 2.

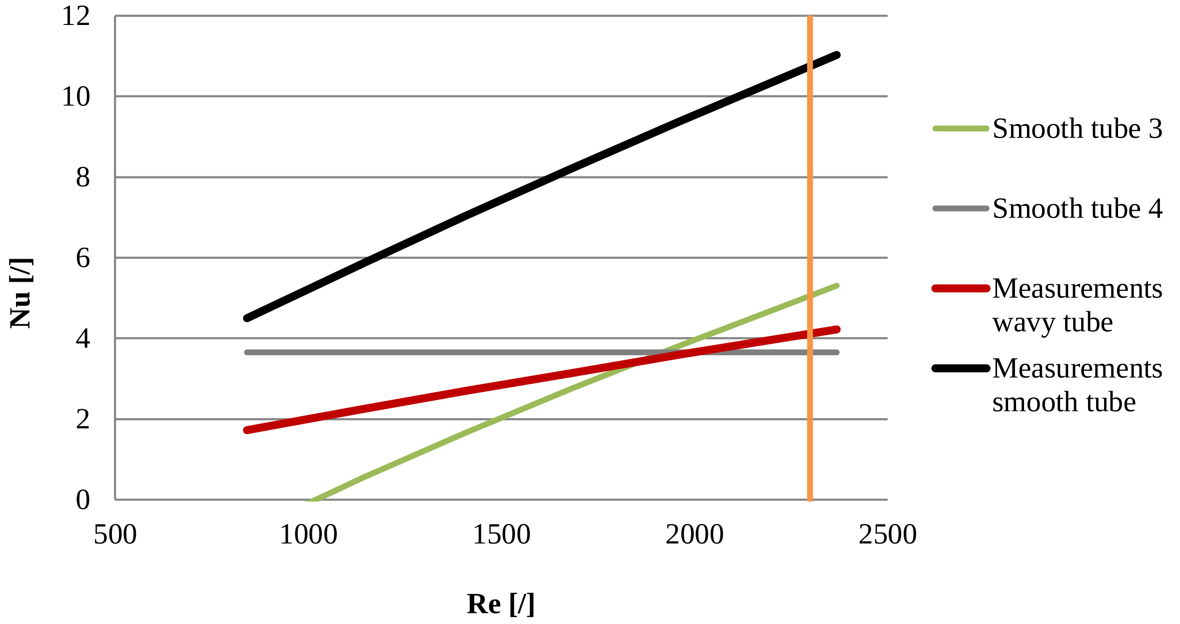

Fig. 6. Measured and calculated Nusselt numbers versus Reynolds number in annulus for wavy and smooth tubes of Type 2 HE. The orange vertical line represents transition from laminar to transitional fluid flow regime.

Note: Smooth tube 1 – equation 3, Smooth tube 2 – equation 4, Smooth tube 3 – equation 6, Smooth tube 4 – equation 2.

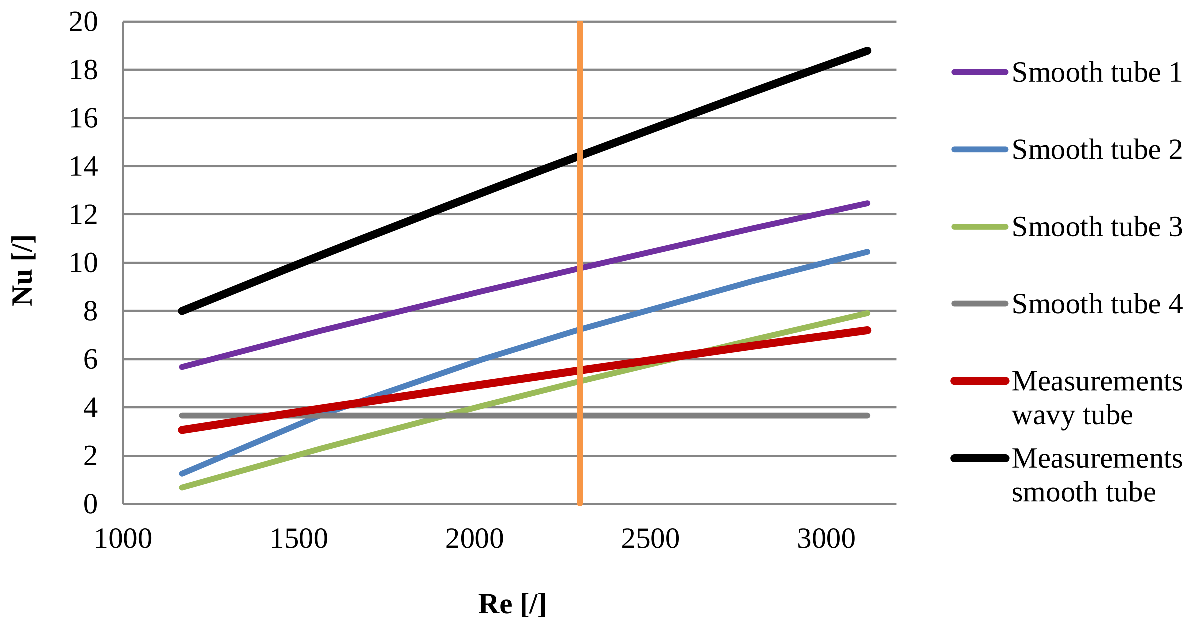

Fig. 7. Measured and calculated Nusselt numbers versus Reynolds number in the inner tube for wavy and smooth tubes of Type 1 HE. The orange vertical line represents transition from laminar to transitional fluid flow regime.

5. Conclusions

In this paper, we present a research study of the thermal characteristics of a classic concentric-tube and a multi-tube counter-flow heat exchanger (HE) that was used in a local ventilation device. The purpose of this device was ability of active natural heating or cooling of building.

In order to achieve this we used the modified Wilson-plot method, which is based on the separation of the total thermal resistance into the internal thermal resistance and the remaining heat resistance. This method allowed us to calculate the Nusselt numbers for a turbulent flow in circular tubes using the well-known correlation, proposed by Dittus – Boelter. All experimentally determined values were also compared with the Nusselt numbers, based on empirical correlations for a smooth straight tube found in the literature.

The experimental setup was built on the basis of concentric-tube HE, which had two types of internal tube: Type 1 had single wavy tube of diameter 80 mm, while Type 2 had 7 wavy tubes, where each of these had a diameter of 30 mm. Forced convection was achieved using two energy-efficient fans. To control and maintain the desired temperature the PID-controller-based heating unit with nominal heating power of 400 W was used. Air temperatures at four measuring points were measured using Cu-CuNi thermocouples, while the air flow was measured indirectly with a vane anemometer.

The results showed that, compared to a smooth tube, the heat transfer coefficients on inner and outer sides of the inner wavy tube of both types of HE are similar or in some cases even lower. However, the heat transfer rate in case of wavy tubes was noticeably higher due to the increased heat-transfer surface area for both types of HE.

Nomenclature

C	constant, /
d	diameter, m
f	friction factor, /
L	length, m
Nu	Nusselt number, /
Pe	Peclet number, /
Pr	Prandtl number, /
Re	Reynolds number, /
T	temperature, K

Subscripts and superscripts

1	hot
2	cold
c	classic
i	inner
in	input
m	multi-tube, exponent
n	exponent
o	outer
out	output

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