Revista Română de Inginerie Civilă, Volumul 4 (2013), Numărul 2

Comparative Analysis of Experimental vs. Theoretical Thermal Performances of a Single Phase Flow Minichannel Heat Exchange

Analiza comparativă, experimental vs. teoretic a performanțelor termice ale unui schimbător cu minicanale, la curgerea monofazică

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Abstract: *This paper presents experimental results concerning thermal performances of a mini-channel heat exchanger (hydraulic diameter of 1.54mm) of domestic make, in single phase water flow. Such heat exchangers may be used in air handling units in air-conditioning systems, as an alternative to classical heat exchangers represented by fan coils. The heat exchanger was tested in an experimental set-up, of loop type and worked as an air heater. By design, the experimental stand allowed testing under steady state conditions for both air and water inlet parameters. Heating capacity of the minichannel heat exchanger on both air and water side has been measured. The experimental results have been validated by heating capacity calculations on both working fluids side showing relative errors of approximately 8% on average. The final part of the paper presents a comparative analysis of experimental and theoretical values of the convective heat transfer coefficient on water side, based on different recommended correlations for single phase flow minichannels. The study concluded that Peng et al. and Adams et al., (1994) correlations covered best the experimental results.*

Key words: minichannels, heat transfer, single phase flow and equipment

Rezumat: *Lucrarea prezintă rezultate experimentale referitoare la performanțele termice ale unui schimbător de căldură cu minicanale (cu diametrul hidraulic de 1.54mm), de construcție autohtonă, la circulația monofazică a agentului de lucru. Acest tip de schimbător poate echipa centrale de tratare a aerului utilizate în cadrul unor sisteme de climatizare şi reprezintă o alternativă la schimbătoarele de căldură de tip clasic, reprezentate de ventiloconvectoare. Schimbătorul de căldură a fost testat într-un stand experimental de tip circuit închis, cu rol de baterie de încălzire a aerului. Prin construcția sa, standul experimental permite atingerea şi păstrarea unui regim cvasistaționar de transfer de căldură pe partea apei şi aerului. S-a determinat experimental puterea termică a schimbătorului atât pe partea aerului cât şi pe partea apei. Rezultatele experimentale au fost validate prin calculul puterii termice pentru ambii agenți de lucru, iar erorile relative au fost de aproximativ 8%. În ultima sa parte lucrarea prezintă o analiză comparativă între valorile teoretice şi cele experimentale ale coeficientului convectiv de transfer de căldură pe partea apei. Pentru calculul teoretic al*

acestui coeficient s-au folosit câteva ecuații criteriale recomandate pentru curgerea monofazică în minicanale.Studiul a condus la concluzia că cea mai bună concordanță între rezultatele teoretice şi cele experimentale au fost obținute la utilizarea corelațiilor ecuațiile criteriale Peng et al. şi Adams et al.

Cuvinte cheie: minicanale, transfer de căldură, curgere monofazică, echipament.

1. Introduction

Compact minichannel heat exchangers represent an alternative of great interest to the classical heat exchangers. The reason lies in their indisputable advantages regarding high thermal performances, associated to a high compactness and low material consumption. The heat exchanger subjected to testing, is made of aluminum of finned type. It has been tested as air heater using warm water as primary fluid. The experimental results, consisting of thermal performance lead to convective heat transfer coefficient on the water side, flowing inside the minichannel. This paper presents a comparative analysis of experimental and theoretical values of this coefficient.

2. Experimental stand

The schematic layout of the experimental stand is presented in Figure 1. The configuration of the tubes, in cross section is presented in Figure 2. Its geometrical configuration is described by the following parameters: number of tubes: 35; number of minichannels / tube: 6; minichannel hydraulic diameter:1,54 mm; number of passages 2; number of tubes / passage: 18 and 17 respectively. Aluminum plate fins of 0.2 mm thickness are welded in between the tubes. The overall dimensions of the heat exchanger are: length: 0,645m; height: 0,342m; width: 0,018m.

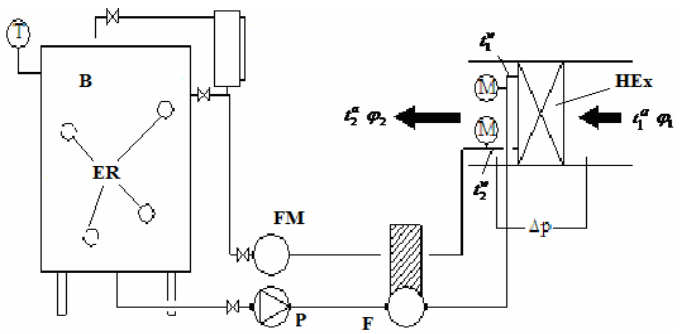

Fig. 1 Schematic layout of the experimental stand

Legend: B – boiler; ER – electrical resistances; P – water pump; F – filter; FM – water flow meter; T – thermostat; HEx - minichannel heat exchanger; M – pressure gauges on water inlet and outlet; t_1^a, t_2^a - air temperature; φ_1 φ_2, – inlet / outlet air relative humidity; t_1^w, t_2^w - water inlet / outlet temperature.

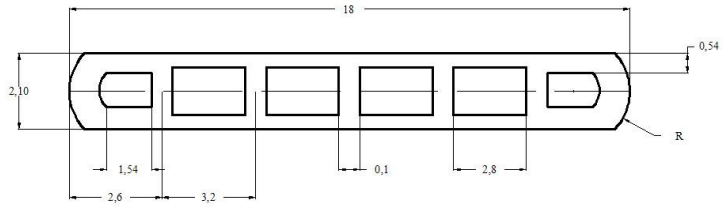

Fig. 2 Tube configuration

The tested minichannel heat exchanger is placed inside the lower horizontal branch of an air loop - made of insolating panels of 20mm thickness of sandwich type, exteriorly covered with thin aluminum sheet. The cross section of the loop is 0,5 x 0,5 m. Inside the same lower horizontal branch there is the air handling unit whose role is to maintain at approximately constant values the air parameters at the air heater inlet. In order to do this, the air handling unit consists of an air cooler and a fan of 3000m^3/h. The air cooler is fed with cooled water from the existing water management system of the laboratory.

3. Methodology

In order to experimentally establish the heating capacity on the water side the following parameters have been measured: water inlet and outlet temperature, water flow rate and microchannel wall temperature. The heating capacity on the water side, $\dot{Q}_w$, was computed based on experimental values, according to the following equation:

$$\dot{Q}_w = \dot{m}_w \cdot c_{p,w} \cdot \Delta t_w \text{, [W]} \tag{1}$$

Every heating capacity measurement on water side presented a deviation of maximum 4% thus complying with currently valid standard provisions. This is why the heating capacity on water side has been further considered to calculate the convection heat transfer coefficient. It was then validated by the measured heating capacity on the air side. In order to experimentally determine the latter, the following parameters have been measured: dry bulb and wet bulb temperature of moist air and relative humidity both at the inlet and outlet of the heat exchanger. Based on these values the inlet – outlet enthalpy change of the humid air was computed. The heating capacity on the air side, $\dot{Q}_a$, was computed based on experimental values, according to the following equation:

$$\dot{Q}_a = \dot{m}_a \cdot (h_2 - h_1) \text{, [W]} \tag{2}$$

The water to air deviation in the energy balance of 8% was considered by the authors to be an acceptable one, as it provided the validation for the heating capacities on water side. The convection heat transfer coefficient on the water side was computed based on experimental values, according to the following equation:

$$\alpha_{\exp} = \frac{\dot{Q}_w}{S_i \cdot (T_{m,w} - T_{wall})}, \text{ [W/(m}^2\text{ K)]}, \tag{3}$$

where: $T_{m,w}$ - mean water temperature; T_{wall} - mean tube wall temperature.

The theoretical convection heat transfer coefficient on the water side was computed based on the following correlations recommended for turbulent flow:

- Peng et al. (1994):

$$Nu = 0{,}072\left(\frac{D_h}{W_0}\right)^{1{,}15} \cdot [1 - 2{,}421 \cdot (Z - 0{,}5)] \cdot \mathrm{Re}^{0{,}8} \cdot \mathrm{Pr}^{\frac{1}{3}} \tag{4}$$

where: D_h - hydraulic diameter of the microchannel; W- width; H- height; W_c – microchannel spacing; Z- dimensionless variable accounting for the side ratio of the microchannel. The optimum Z value leading to a maximum heat transfer is 0,5.

- Wu & Little (1998): $Nu = 0{,}00222 \cdot \mathrm{Re}^{1{,}09} \cdot \mathrm{Pr}^{0{,}4}$ (5)

- Adams et al. (1998): $Nu = (Nu)gn \cdot [1 + F]$ (6)

where: - F – coefficient raging within 0,6-1,75, for Reynolds numbers of approximately 3000-20000 and diameters of 0,76 mm and 1,09 mm;

- $(Nu)gn$ – Nusselt number given by Gnielinski correlation (1976):

$$(Nu)gn = \frac{f}{8}[\mathrm{Re} - 1000] \cdot \frac{\mathrm{Pr}}{\left[1 + 12{,}7 \cdot \left(f/8\right)^{1/2} \cdot (\mathrm{Pr} - 1)^{2/3}\right]} \tag{7}$$

where: f – friction factor, computed with the Filonenco relation [1954]:

$$f = [1{,}82 \log(\mathrm{Re}) - 1{,}64]^{-2} \tag{8}$$

Relation (6) is valid for Re = 2600…23000 and Pr = 1,53…6,43.

- Dittus & Boelter (1930), which is currently used for single phase inside conventional size tubes:

$$\alpha = \frac{\lambda}{D_h} \cdot 0{,}023 \cdot (\mathrm{Re})^{0{,}8} \cdot (\mathrm{Pr})^{0{,}4}, \text{ [W/(m}^2\text{K)]} \tag{9}$$

As Figure 1 shows, the warm water flow rate has been measured by an electronic flow meter (Danfoss); the air flow rate was measured based on the air velocity measurements (flow rate velocity method), complying with currently valid standards. Air velocity was measured in 25 places within the cross section of the loop as shown in Figure 3.

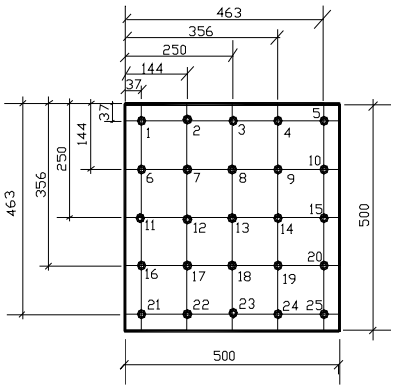

Fig. 3 Positions of the air velocity measurements

The air flow rate has been measured with an Ahlborn hot wire anemometer, of 3% accuracy. The warm water temperature has been measured at the air heater inlet and outlet. Two cross sectional grids have been used in order to measure the air temperature at the air heater inlet and outlet, each of them with 5 measuring points. The temperature has been thus calculated as an averaged value.

The tube wall temperature was measured in 5 locations on the collector height, 5 locations on the distributor height and 3 locations on the heat exchanger height, equally distanced from the collector and distributor. All the temperature measurements (water, air and wall) have used type K thermocouples, ± 0.25°C accuracy.

The experiments have been carried out under quasi steady state conditions that is for constant inlet water temperature, controlled by the boiler thermostat. It was considered as such the regime characterized by maximum ±5% variation of the measured parameters, along 10 consecutive readings, at 10 minutes apart. Reliable measured data provided under these conditions have been further used in theoretical calculations based on averaged values of the measured parameters.

4. Rezults

4.1. Experimental results

The measured parameters on water side are shown in Table 1.

Table 1.

Measured parameters on the water side

#	Water temperature, [°C]			Mean wall temperature, [°C]	Water flow rate, [kg/s]
	inlet	outlet	mean		
1	53.00	43.30	48.2	47.1	0.295
2	54.50	45.30	49.9	48.8	0.290
3	52.20	40.90	46.6	45.4	0.294
4	52.50	41.60	47.1	46.00	0.292
5	52.00	42.90	47.5	46.4	0.293
6	55.20	44.00	49.6	48.5	0.292
7	53.50	43.10	48.3	47.2	0.294
8	55.00	44.90	50.0	48.9	0.295
9	54.20	43.80	49.0	47.9	0.293

Based on the measured parameters on both water and air side heating capacities and deviations have been computed. They are all shown in Table 2 and Figures 4 and 5.

Table 2.

Heating capacity based on the experimental measurements

#	Heating capacity on air side, [W]	Heating capacity on water side, [W]	Heating capacity deviation, [%]
1	13035	11967	0.08
2	12929	11158	0.14
3	14307	13870	0.03
4	14095	13310	0.06
5	12113	11150	0.08
6	14852	13677	0.08
7	14536	12787	0.12
8	14009	12460	0.11
9	13377	12743	0.05

The above data show that generally, that is in most cases, the heating capacity measured on the water side is close to the heating capacity measured on the air side. The largest difference is of 14%. Nonetheless, the experimental heating capacity on the water side may be regarded as a reliable one as measuring of air parameters implies difficult experimental conditions to be met in order to obtain highly accurate results.

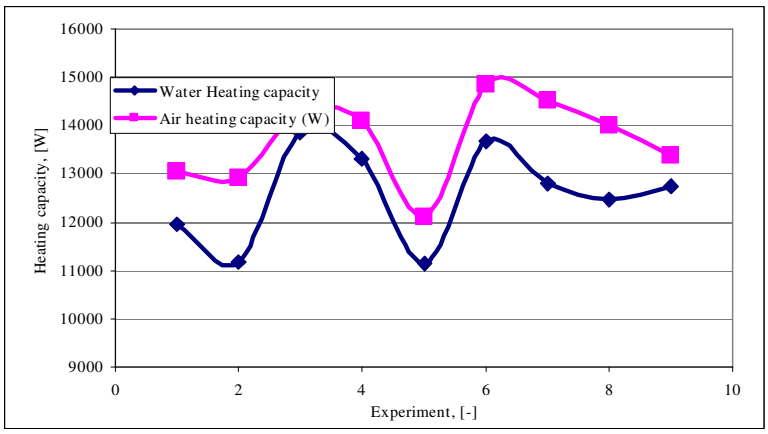

Fig. 4 Heating capacity on air and water side

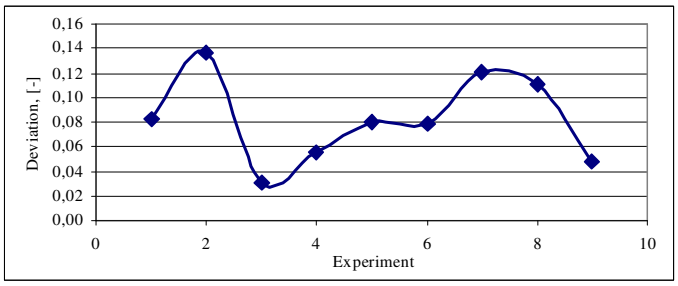

Fig. 5 Deviation of the heating capacity on air and water side

Based on the experimental water side heating capacity data the mean convective heat transfer coefficient has been assessed according to equation (3). The experimental values of the mean convective heat transfer coefficient are exhibited in Table 3.

Table 3.

Mean convective heat transfer coefficient

#	Heating capacity on the water side, [W]	Water temperature, [°C]			Average wall temp*. [°C]	Temp*. difference between water and wall surface [°C]	Experimental convective heat transfer coefficient, [W/(m^2 K)]
		inlet	outlet	Aver-age			
1	11967	53.0	43.3	48.2	47.1	1.1	11774
2	11158	54.5	45.3	49.9	48.8	1.1	10479
3	13870	52.2	40.9	46.6	45.4	1.2	12459
4	13310	52.5	41.6	47.1	46.0	1.1	13096
5	11150	52.0	42.9	47.5	46.4	1.1	10971
6	13677	55.2	44.0	49.6	48.5	1.1	12844
7	12787	53.5	43.1	48.3	47.2	1.1	12009
8	12460	55.0	44.9	50.0	48.9	1.1	12259
9	12743	54.2	43.8	49.0	47.9	1.1	11968

temp*. - temperature

4.2. Comparative analysis of experimental and theoretical values of the convective heat transfer coefficient on the water side

Experimental values of the convective heat transfer coefficient, α, on the water side, shown in Table 3, have been compared to theoretical values, calculated with the correlations: (4) - Peng et al. (1994), (5) – Wu & Little (1998), (6) - Adams et al. (1998), and (9) - Dittus & Boelter (1930). The theoretical α, values have taken into account the water passages inside the heat exchanger (2 in all), and at the same time the number of tubes on each passage (18/17). Table 4 shows theoretical values of the convective heat transfer coefficient, α.

Table 4.

Theoretical α values

#	Convective heat transfer coefficient, α, [W/(m^{2}K)]			
	Peng et al. (1994)	Wu & Little (1998)	Adams et al. (1998)	Dittus & Boelter (1930)
1	12827	8635	12190	8958
2	12109	8007	11634	8438
3	12251	8112	10658	8579
4	12225	8094	10792	8555
5	12326	8185	11113	8619
6	13886	9601	14440	9677
7	12331	8210	13624	8537
8	13007	8819	13908	9036
9	12665	8508	12732	8819

Figure 6 shows the experimental and theoretical α values and Figure 7 shows the deviation between the experimental and theoretical values.

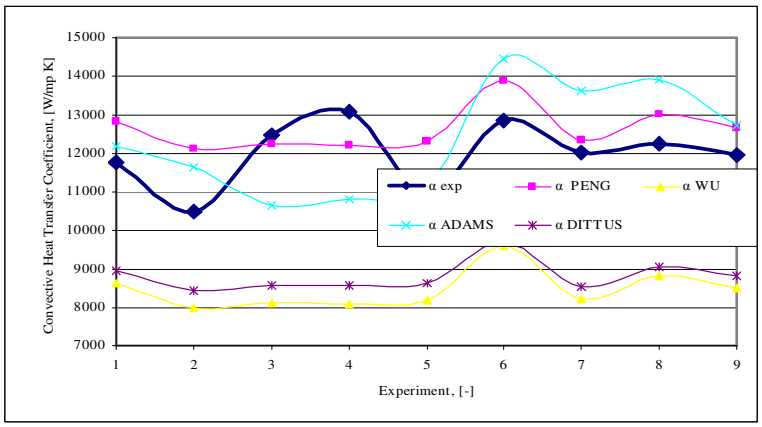

Fig. 6 Experimental and theoretical α values

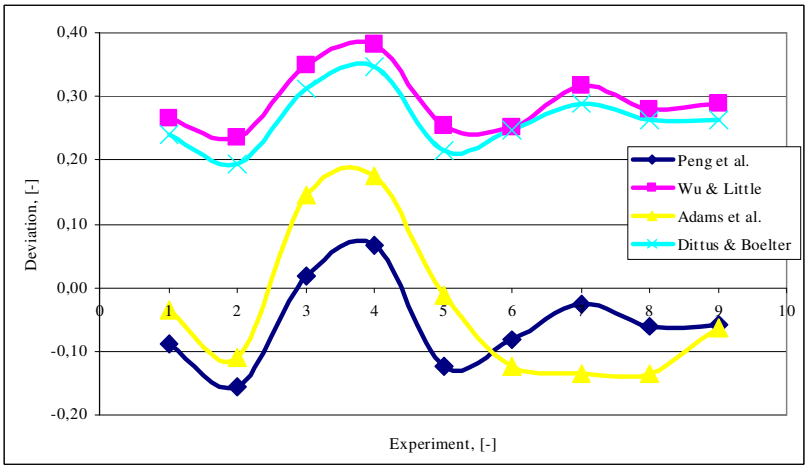

Fig. 7 Deviation of experimental from theoretical values

Figures 6 and 7 show that experimental values are best covered by the theoretical values from Peng et al. and Adams et al. correlations. The experimental to theoretical deviations range from 2% to 16% for Peng et al. correlation, under the assumption of maximum heat transfer in turbulent flow, implying $Z = 0{,}5$. Deviations range from 1% to 18% for Adams et al. correlation, under the assumption of intensified heat transfer in turbulent flow, implying $F = 1{,}6$.

At the same time, Figures 6 and 7 show that experimental values are very different from the theoretical ones, so that the deviation rages from 24% to 38% for Wu & Little correlation and from 19% to 35% for Dittus & Boelter correlation.

5. Conclusions

The experimental research conducted on the minichannel heat exchanger in order to determine which correlations covered best the experimental results lead to the following conclusions:

- data listed in Table 4 and graphically presented in Figures 6 and 7 show that theoretical to experimental deviation of the convective heat transfer coefficient cover a rather large scattering area;

- two of the correlations used, that is Wu & Little and Dittus & Boelter, lead to theoretical values that are close to each other, but present large deviations from the experimental results: 29% on average for Wu & Little and 26% on average for Dittus & Boelter;

- the other two correlation used, that is Peng et al. and Adams et al. also lead to theoretical values that are close to each other; the mean deviation of experimental values from Peng et al. is -5,3% and from Adams et al. is 3%.

In the authors' view, the explanation for which correlations of Peng et al. and Adams et al. cover best the experimental results lies in adopting the following computation assumptions:

- maximum heat transfer inside the minichanels that implied $F = 1,6$;
- selection of F value, within the range of 0,6…1,75, as indicated by Adams et al. for Reynolds numbers of approximately 3000 to 20000.

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