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Experimental and numerical analyses of finned cross flow heat exchangers efficiency under non-uniform gas inlet flow conditions

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Abstract The work deals with experimental and numerical thermodynamic analyses of cross-flow finned tube heat exchangers of the gas-liquid type. The aim of the work is to determine an impact of the gas non-uniform inlet on the heat exchangers performance. The measurements have been carried out on a special testing rig and own numerical code has been used for numerical simulations. Analysis of the experimental and numerical results has shown that the range of the non-uniform air inlet to the considered heat exchangers may be significant and it can significantly affect the heat exchanger efficiency.

Keywords: Cross flow heat exchangers; Experimental analysis; Numerical analysis; Non-uniform fluid flow

Nomenclature

c_p – specific heat capacity at constant pressure, J/(kg K)
$\dot{Q}$ – heat flow rate, W
t – temperature, °C
$\dot{V}$ – volumetric flow rate, m^3/s

Greek symbols

ρ – density, kg/m^3

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Subscripts

a	–	air
ex	–	experimental
in	–	inlet
num	–	numerical
out	–	outlet
w	–	water

1 Introduction

The plate exchangers (with the mixed flow) and the finned cross-flow heat exchangers, which core has the form of a bunch of pipes with flat plate ribs, are the most important types among the currently applied heat exchangers with extended surfaces. Small size, low weight and a high efficiency determine the strong position of such devices. A compact ribbed heat exchangers are commonly used in heat technology, refrigeration, air-conditioning and automotive industries.

The first investigations referred to the heat exchangers with non-uniform flow of fluids were performed at the Institute of Heat Technology of the Silesian University of Technology (ITT SUT) for gaseous mediums and they were only of a computational form [1]. Investigations of the gas-liquid type cross-flow heat exchanger have been conducted at the ITT SUT since a few years now to evaluate the influence of a non-uniform gas inlet on the exchanger operation [2]. A range and form of the air inflow non-uniformity have been determined on the special testing rig (see Fig.1 in the next section). Configuration of the measuring system of the testing rig allow to determine the air velocity and temperature distribution at the exchanger inlet and outlet. This testing rig, in its original arrangement, allowed only for "cold" experiments, it means without presence of the hot medium. Thus, the influence of the measured non-uniformity has been assessed by means of numerical simulations performed by the computer code called HEWES [2] – worked out for thermal analysis of the considered exchanger. In his work R. Piątek [2] concludes that the maldistribution of the air inlet may significantly influence the effectiveness of the heat exchanger.

A unique feature of the investigations realized at the ITT SUT is experimental consideration of the air flow non-uniformity. Similar heat exchangers have been investigated by D. Taler with co-workers [3,4] by means of physical experiments as well as numerical simulations. Very good compliance of experimental and numerical results has been achieved, but the problem of the non-uniform fluids flow is neglected and this fact simplified experimental measurements. Many researches considering the problem of unequal fluid flow have been realized only numerically. Authors of [5] have simulated the plate fin heat exchanger using the finite elements method and found out that the influence of the non-uniformity of the liquid flow may have significant meaning in some work regimes. A very significant drop of the heat exchanger efficiency has been also observed by authors of [6]. The contrary results have obtained by authors of [7]. Numerical simulations realized for a rotary heat exchanger in the first work and optimization procedure presented in the second one have not shown significant dependence on the fluids flow non-uniformity. Experimental analyses considering maldistributions of the fluids flow through the heat exchangers are very rare. The results presented for example in [8] and [9] indicate that the non-uniformity influences the efficiency of heat exchangers to a large extent, while authors of [10] concluded about minor effects of this phenomena.

The very short literature review of the selected positions shows that the problem of the non-uniform fluid inflow to the heat exchangers has been the subject of many computational and experimental investigations, but the results obtained are in some cases ambiguous. It refers first of all to the heat exchanger effectiveness. There is lack of complete investigations of the finned cross-flow heat exchangers of the gas-liquid type with non-uniform inflow of the fluids, especially of non-uniform inflow of the gas.

The first results obtained by the authors were presented in [11] and in [12]. These papers are however related to the mathematical model of the heat exchanger and refer to the problem of evaluating the heat transfer coefficient and the problem of water non-uniform inlet to the exchanger. In papers [13] and [14] there have been presented the selected results obtained by use of the modified test station for the case of typical car cooler.

The aim of the research undertaken in the framework of this project has been to experimentally and numerically investigate the influence of a non-uniform gas inflow on the performance of the heat exchanger under consideration.

2 Experimental investigations

2.1 The testing rig and measurement procedure

The testing rig consists of two main modules: the air supply module (see Fig. 1) and the hot water supply module (Fig. 2). The air supply module originally was a special experimental set-up constructed during realization of the project [2] for determination of a form and scope of the air inflow non-uniformity. The main element of the measuring system is a thermoanemometric sensor installed onto a measuring probe the movement of which is controlled by a computer. It allows to determine velocity and temperature fields of air at the exchanger inlet and outlet. The testing rig has been modernized and the hot water supply module was installed. The measuring system allows at the moment for acquisition of the following parameters: total air volumetric flow, water mass flow rate, inlet and outlet water temperatures, distribution of air velocity and temperature at the inlet and outlet of the heat exchanger.

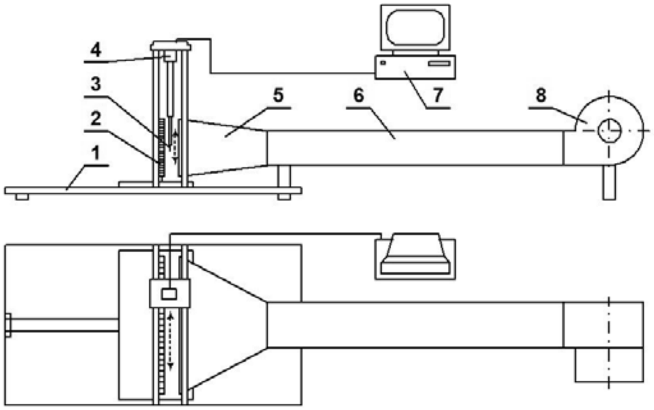

Figure 1. Testing rig — the air supply module (1 – support plate, 2 – heat exchanger, 3 – thermoanemometric sensor, 4 – measuring probe, 5 – diffuser, 6 – channel, 7 – control computer, 8 – fan).

The air temperature and velocity distribution measurement need the measuring task to be defined in the form of an input file for the program controlling the measuring probe work. The time constant of the measurement and the number of measurements realized in each node should be entered to the file. A regular measuring mesh of 196 nodes has been used in measurements. Such mesh divides the whole measuring cross section

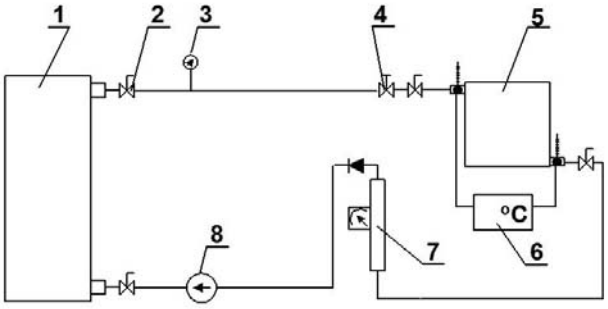

Figure 2. Testing rig — the hot water supply module (1 – electric heater, 2 – cut-off valve, 3 – manometer, 4 – control valve, 5 – heat exchanger, 6 – temperature measuring system, 7 – flow meter, 8 – pump).

onto identical rectangles in the middle of which the measuring nodes are localized. The measuring program has been started after a steady state conditions were achieved.

There were three heat exchangers investigated during realization of this work (see Fig. 3):

- HE-1 – typical car cooler (Skoda Favorit 135L) with the core made of 15 rows of cylindrical pipes finned with the plate fins (380 on each pipe),
- HE-2 – the cross-flow heat exchanger made by GEA Heat Exchangers Company with the core made of 10 rows of elliptical pipes finned with the plate fins (175 on each pipe),
- HE-3 - the cross-flow heat exchanger made by GEA Heat Exchangers Company with the core made of two bundles each of 10 rows of elliptical plate finned pipes (81 fins on each pipe in the first bundle and 140 fins on each pipe in the second bundle).

2.2 Selected experimental results

Selected results of the measurements are gathered in Tab. 1. The heat exchanger capacity can be determined as the heat flow rate transferred in the

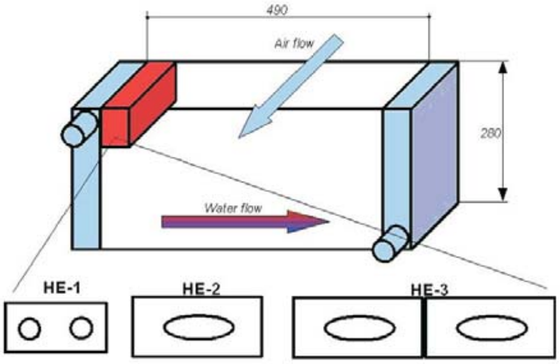

Figure 3. General sketch of the heat exchanger under consideration and the recurrent elements of three versions of the heat exchanger.

Table 1. Selected experimental results.

Measurement No.	$\dot{V}_w$ [dm^3/min]	$T_B^{(1)}$ [°C]	$t_{w,in}$ [°C]	$t_{w,out}$ [°C]	$\dot{Q}_w$ [kW]
HE-1/1	27.0	50	49.8	43.9	11.03
HE-1/2	27.0	70	68.9	56.7	22.61
HE-1/3	27.0	90	86.4	67.9	34.08
HE-1/4	27.0	50	49.7	44.5	9.72
HE-1/5	27.0	70	69.2	58.7	19.42
HE-1/6	27.0	90	88.0	72.2	29.11
HE-2/1	27.0	50	48.2	42.8	10.07
HE-2/2	27.0	70	69.6	62.0	14.08
HE-2/3	27.0	90	90.2	79.5	19.58
HE-2/4	27.0	50	48.0	45.6	4.48
HE-2/5	27.0	70	68.5	62.0	12.04
HE-2/6	27.0	90	89.8	79.0	19.76
HE-3/1	27.0	50	49.3	42.7	12.39
HE-3/2	27.0	70	69.1	59.8	17.31
HE-3/3	27.0	90	87.8	74.6	24.08
HE-3/4	27.0	50	50.1	47.1	5.51
HE-3/5	27.0	70	69.6	61.6	14.81
HE-3/6	27.0	90	88.7	75.4	24.30

(1): the temperature set at the electric boiler outlet

exchanger computed from the air and the water side. Obvious relationships describing the agents enthalpy decrease (increase) are as follow:

$$\dot{Q}_a = \dot{V}_a \rho_a c_{pa} \left(t_{a,out} - t_{a,in} \right) \ , \tag{1}$$

$$\dot{Q}_w = \dot{V}_w \rho_w c_{pw} \left(t_{w,in} - t_{w,out}\right) \; . \tag{2}$$

Air density has been calculated using the ideal gas law for the absolute pressure and the air average temperature at inlet to the exchanger. Density of water has been assumed according to thermodynamic tables for the outlet temperature. It should be noted here that the maximum uncertainty of determining the total heat flow rate from Eq. (2) does not exceed ±4%.

The air inlet velocity distributions are one of the most important experimental results (see Figs. 4, 5 and 6). One may observe that the scope and form of the non-uniform air inlet depends on the air flow rate. Additional analyses carried out in the framework of [15] have shown that the fan generates a non-uniform agent flow.

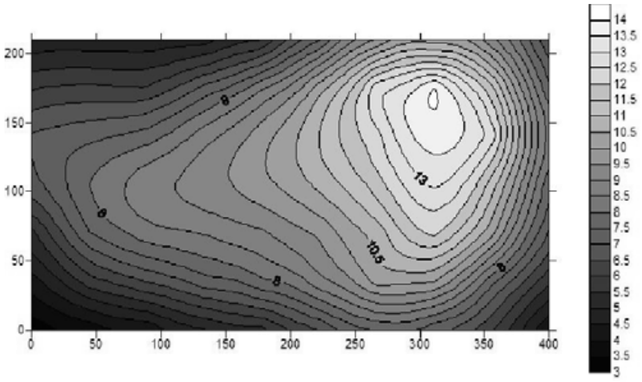

Figure 4. Distribution of air velocity at the inlet for the HE-1/1 measurement, m/s.

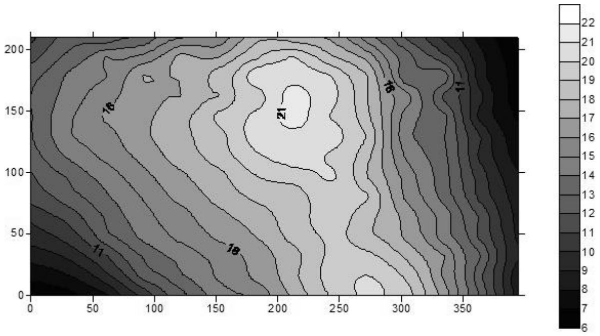

Figure 5. Distribution of air velocity at the inlet for the HE-2/1 measurement, m/s.

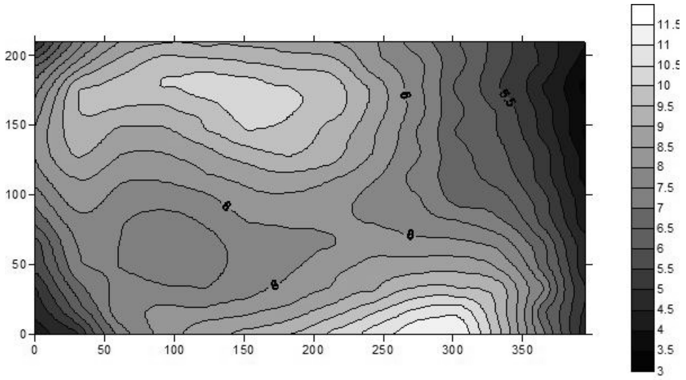

Figure 6. Distribution of air velocity at the inlet for the HE-3/4 measurement, m/s.

3 Numerical simulations

3.1 Mathematical model of the heat exchanger

The mathematical model of the considered heat exchanger was worked out taking into account the following simplifying assumptions (only most important ones): steady state conditions, one-dimensional flow of fluids, no internal heat sources, radiation is neglected, heat losses are neglected, heat flow is normal to the boundary, real fin is replaced with a round or a plate-elliptic fin of the same surface. It was also assumed that the air inflow is non-uniform and the water inflow may be non-uniform. An influence of temperature on thermal properties of the fluids was also taken into account.

The analysed real cross-flow heat exchanger has been replaced with a model rectangular heat exchanger. The model was then divided onto elementary fragments (Fig. 7). Each fragment represents a recurrent element of the real heat exchanger – a single tube with the fin.

The energy balance equations for each fragment constitute the mathematical basis of the model. The control volume method based model of heat transfer for the recurrent fragment of the heat exchanger has been worked out to calculate the average temperature of the fins and tube outer surface. The detailed description of the model and equations can be found in [2].

The parameters calculated with the model of the recurrent fragment are: outlet and average temperature of the water flowing in the pipe, average temperature of the air, average temperature of the rib and the pipe surface, average values of heat transfer coefficients at the gas side and heat flux

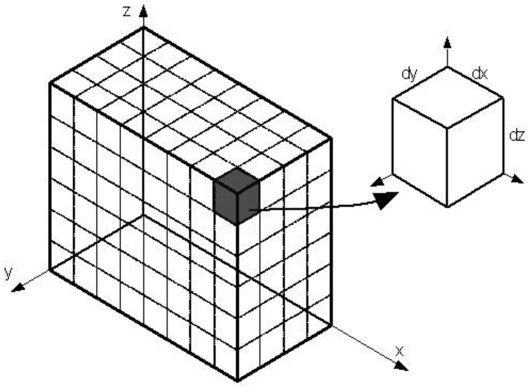

Figure 7. The model heat exchanger and the recurrent fragment.

transported in the recurrent fragment. The heat transfer coefficient from the hot water to the pipe has been computed from the Colburn's formula.

The heat transfer coefficient at the gas side may be determined on the way of numerical simulations for the numerical model (CFD) of the recurrent fragment of the considered heat exchanger [11,12] or may be computed from one of available Nusselt number correlations – namely the Kays-London, Norris-Spofford or Berman's correlation. The results presented here have been obtained using the mentioned above CFD model.

The calculation procedure for the whole exchanger model is iterative and it is repeated for all the recurrent fragments of the considered heat exchanger. First, the air temperature increase in the analysed fragment is assumed. Next, heat transfer coefficients for water and gas sides are calculated as well as the rib and pipe surface average temperature. The heat flux transported in the recurrent fragment is then computed and the accuracy criterion is checked. If the criterion is satisfied the procedure is realized for the next fragment. If the criterion is not fulfilled the described procedure is repeated for the given recurrent fragment till the demanded accuracy is achieved.

3.2 Computational results

Simulations were aimed at determination of the non-uniform air inlet impact on the heat exchangers efficiency and have been realized using the described earlier model and the computer code HEWES [2]. All these simulation have been performed applying the uniform air inflow to the exchanger. The uniform mass flow rate of air has been derived assuming that the total mass flow rate of air spreads equally on the all measuring fields. The selected results of computations are gathered in Tab. 2 and, as it was expected, they shown quite significant improvement of the efficiency of the heat exchanger. The efficiency growth raises with increasing the air flow rate and water inlet temperature.

Table 2. Selected computational results.

Measurement No.	Q_{num} [kW]	Q_{ex} [kW]	δ_Q [%]
HE-1/1	12.78	11.03	15.9
HE-1/2	26.44	22.61	16.9
HE-1/3	39.96	34.08	17.3
HE-1/4	11.11	9.72	14.3
HE-1/5	22.44	19.42	15.6
HE-1/6	33.48	29.11	15.0
HE-2/1	11.59	10.07	15.1
HE-2/2	16.24	14.08	15.4
HE-2/3	22.75	19.58	16.2
HE-2/4	5.10	4.48	13.9
HE-2/5	13.76	12.04	14.3
HE-2/6	22.68	19.76	14.8
HE-3/1	14.27	12.39	15.2
HE-3/2	20.02	17.31	15.6
HE-3/3	28.10	24.08	16.7
HE-3/4	6.28	5.51	14.0
HE-3/5	17.00	14.81	14.8
HE-3/6	28.14	24.30	15.8

The effects of actions undertaken for improving the exchanger efficiency have been evaluated by computation of the total heat flow rate relative change:

$$\delta_Q = \frac{\dot{Q}_{num} - \dot{Q}_{ex}}{\dot{Q}_{ex}} \; . \tag{3}$$

4 Conclusions

The experiments performed for three considered cross-flow heat exchangers have shown that the air inflow non-uniformity range may be significant and its form depends on the air volumetric flow rate in the considered configuration. The experimental data allowed for determination of the total heat flow rates transported between the fluids in heat exchangers.

The computational results, as it was expected, have shown significant growth in the heat flow rates for the exchanger with fully uniform air inflow. Two aspects should be taken into account while evaluating the numbers from Tab. 2: the measurements errors and the accuracy of the HEWES code. This last parameter has been assessed during the validation of the code (see [14] for more details) and the differences between numerical and experimental results may reach almost 11%. The numerical results are underestimated in all investigated cases [14]. Considering this one may conclude that the efficiency improvement may be not as significant as shown in Tab. 2.

At last, it should be noted that the impact of the non-uniform inlet of the fluids may be significant from the point of view of the heat exchanger efficiency. Results concerning the increase of the efficiency obtained in this work remain in the range achieved by the other researchers.

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