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Prediction of a cross-flow heat exchanger performance in media flow maldistribution conditions

The paper deals with numerical thermodynamic analyses of cross-flow finned tube heat exchangers of the gas-liquid type. The authors postulate that some improvement of the heat exchanger performance may be achieved by applying a special ribbing structure – fitted to certain media flow conditions. First, the measurements have been carried out in order to determine the air inflow non-uniformity and next an own computer code HEWES has been used for numerical simulations in order to evaluate the impact of the measured non-uniformity on the exchangers' efficiency for the heat exchangers with unified ribbing structure. The numerical simulations have been repeated for heat exchangers with special ribbing structures – determined on the basis of experimental results. Results confirm the hypothesis – some increase in the total heat flow rate may be observed for the considered heat exchanger.

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
ρ	–	mass density, kg/m^3

Subscripts

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

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1 Introduction

Non-uniform flow of media through a cross-flow heat exchanger significantly complicates thermal-hydraulic analysis of such heat exchangers. This problem has been the subject of number of investigations in the past, but the results are somewhat unambigous. Some authors concluded about major impact of this issue on the exchangers' efficiency, other say that the effect of non-uniform flow of the agents is negligible. Many researches considering the problem of unequal agent flow have been realized only numerically. Authors of [1] 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 [2]. The opposite results have obtained authors of [3]. 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 agents flow non-uniformity. Experimental analyses considering maldistributions of the agents flow through the heat exchangers are very rare. The results presented for example in [4] and [5] indicate that the non-uniformity influences the efficiency of the heat exchangers to a large extent, while authors of [6] concluded about minor effects of this phenomena. Berryman and Russell have studied flow maldistribution across tube bundles in air-cooled heat exchangers. Their experimental results have detected thermal degradation up to 4% [7], which is much less than in previously cited works.

Investigations of the gas-liquid type cross-flow heat exchanger have been conducted at the Institute of Thermal Technology of the Silesian University of Technology (ITT SUT) since a few years to evaluate an influence of a non-uniform gas inlet on the exchanger functioning. Analyses presented in [8] deals with a typical ribbed cross-flow water cooler and consist of experimental determination of a range and form of the air inflow non-uniformity and numerically (the in-door computer code HEWES) assessed influence of the flow maldistribution on the heat exchanger's efficiency. The author concludes that the maldistribution of the air inlet may significantly influence the effectiveness of the heat exchanger. These results have been next confirmed by experiments realized on the modified version of testing station [9] and simulations done using the validated version of the HEWES code [10].

The following questions have emerged after analysis the experience gained so far: how the own results compare with other researches outcomes (especially while taking into account ambiguity of these results) and if are these results repeatable? The whole analytical procedure (experiments and numerical simulations) has been repeated for two new cross-flow heat exchangers with different ribbing structure in order to answer these questions. This analysis allowed to prove that the flow maldistribution has an important meaning for the heat exchanger performance [11,12].

Several hypotheses of improving the heat exchanger performance have been checked out so far (see [10,13]). The obvious hypothesis assumes that making the air flow more uniform increases the total heat flow rates transferred in the heat exchanger. However, in many cases a heat exchanger user can not change the media flow conditions (due to different limitations). The authors postulate that in such cases some improvement may be achieved by installing a heat exchanger with special ribbing structure – the ribbing density should be fitted to certain media flow conditions. The ribbing structure could be only designed properly if one knows the media flow structure (velocity field at the exchanger's inlet). The air inflow conditions to the heat exchanger under consideration have been determined experimentally on a special testing station. Then simulations have been accomplished using the abovementioned computer code in order to predict performance of the specially designed heat exchanger.

2 Experimental investigations

2.1 Testing station and measurement procedure

The test station 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 test station 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 which shifting is controlled by a computer. It allows to determine velocity and temperature fields of the air at the exchanger inlet and outlet. The test station has been modernized and the hot water supply module was installed. The measuring system allows for acquisition of the following parameters at the moment: total air volumetric flow, the water mass flow rate, inlet and outlet water temperature, distribution of the air velocity and temperature at the inlet and outlet of the heat exchanger.

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 in the file. A regular measuring net of 196 nodes has been used for measurements. Such net divides

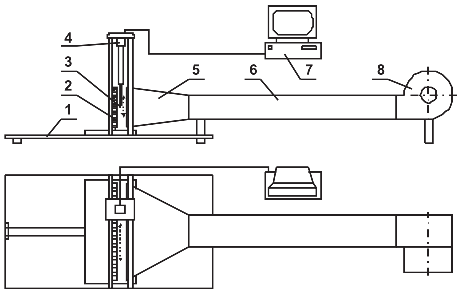

Figure 1. Testing station – the air supply module (1 – support plate, 2 – heat exchanger, 3 – thermoanemometric sensor, 4 – measuring probe, 5 – diffuser, 6 – channel, 7 – control computer, 8 – fan).

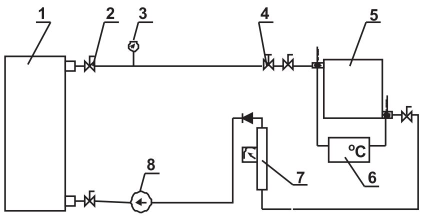

Figure 2. Testing station – the hot water supply module (1 – electric heater, 2 – cut-out valve, 3 – manometer, 4 – control valve, 5 – heat exchanger, 6 – temperature measuring system, 7 – flow meter, 8 – pump).

the whole measuring cross section 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.

2.2 Analyzed heat exchangers

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

- HE-2 – the cross-flow heat exchanger made by GEA Heat Exchangers Company with the core made of 10 rows of elliptical type FE280 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 (FE60 type pipes in the first bundle – 81 fins on each pipe; FE70 type pipes in the second bundle – 140 fins on each pipe).

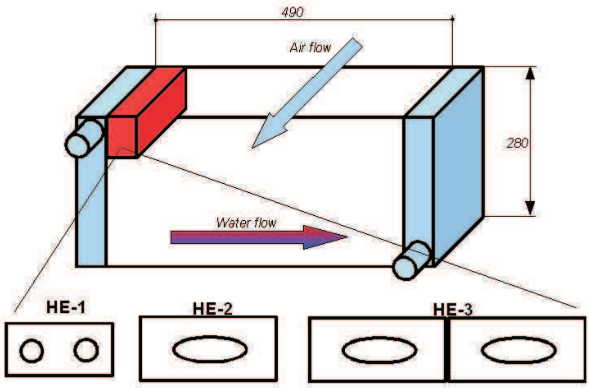

Figure 3. General sketch of the heat exchangers under consideration and the recurrent elements of three versions of the heat exchangers (HE-1 is the unit initially investigated in [8]).

2.3 Selected experimental results

The heat exchanger capacity can be determined as the heat flow rate transferred in the exchanger computed from the air and the water side. Obvious relationships describing the agents enthalpy decrease (increase) are as follow:

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

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

The air density has been calculated using the ideal gas law for the absolute pressure and the air average temperature at the inlet to the exchanger. The density of water has been assumed according to thermodynamic tables for the outlet temperature.

Selected results of the measurements are gathered in Tab. 1. The air inlet velocity distributions are one of the most important experimental results (see Fig. 4). These results decided of the design of the ribbing structure for the heat exchangers investigated next numerically.

3 Numerical 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): steady state conditions, one-dimensional agents flow, no internal heat sources, radiation is neglected, heat losses are neglected, heat flow is normal to the boundary, real

Table 1. Selected experimental results.

Measurement No.	$\dot{V}_w$	$t_B^{(1)}$	$t_{w,in}$	$t_{w,out}$	$\dot{Q}_w$
	[dm^3/min]	[^{o}C]	[^{o}C]	[^{o}C]	[kW]
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

$^{(1)}$temperature set at the electric boiler outlet

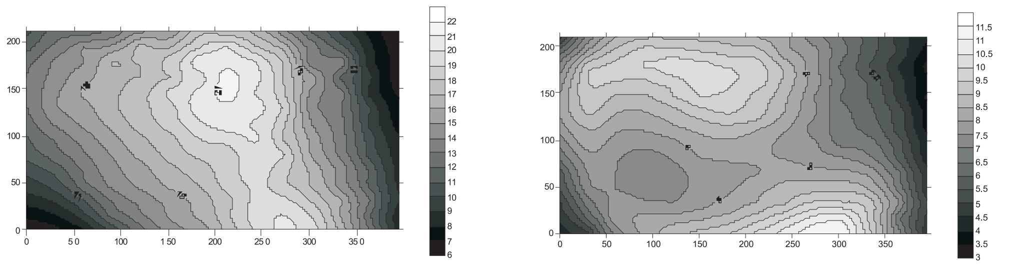

Figure 4. Distribution of the air velocity at the inlet for the HE-2/1 (on left) and HE-3/4 (on right) measurements, m/s.

rib is replaced with a round or a plate-elliptic rib 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 agents was taken into account too.

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. 5). Each fragment represents a recurrent element of the real heat exchanger - a single tube with the fin.

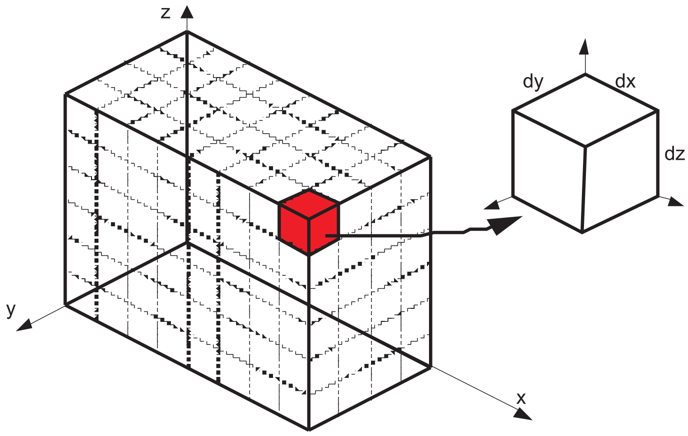

Figure 5. The model heat exchanger and the recurrent fragment.

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 [8].

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 the heat transfer coefficients at the gas side and the heat flux transported in the recurrent fragment. The heat transfer coefficient from the hot water to the pipe has been computed from Colburn's formula. The heat transfer coefficient at the gas side may be determined on the way of the numerical simulations for a numerical model of the recurrent fragment of the considered heat exchanger or may be computed from one of available Nusselt number correlations.

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, the heat transfer coefficients for the water and the 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 then repeated for the given recurrent fragment till the demanded accuracy is achieved.

4 Numerical simulations

4.1 The hypothesis

The obvious hypothesis assumes that making the air flow more uniform increases the total heat flow rates transferred in the heat exchanger. This hypothesis has been partially confirmed by experiments [10], as well as the hypothesis assuming that modelling of the air flow can lead to some improvement also [13]. However, in many cases a heat exchanger user can not change the media flow conditions (due to different limitations). The authors postulate that in such cases some improvement may be achieved by installing a heat exchanger with special ribbing structure – the ribbing density should be fitted to certain media flow conditions. The general idea of this hypothesis is that extending the heat transfer surface in the area where a medium flow is more intensive would also intensify the heat transport.

The ribbing structure could be only designed properly if one knows the media flow structure (velocity field at the exchanger's inlet). These data have been determined experimentally, as described in Section 2. Some variations in the air non-uniformity form have been observed, depending on the fan capacity, but averaging the results the ribbing structures as shown in Tab. 2 have been proposed for two considered heat exchangers.

4.2 Results of simulations

The knowledge of the standard heat exchanger performance is necessary in order to evaluate the abovementioned hypothesis. The relevance data are presented in the last column of Tab. 1. The relative change of the total heat flow rates transported in the heat exchanger may be used as a criterion for evaluating the hypothesis. This can be calculated as follow:

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

Moreover, it is also important what is the difference between the observed relative change of the total heat flux with the change caused by making the air flow uniform. The ordinary differences in total heat flow rates ΔQ have been also computed. So, the first simulations have been realized for the heat exchangers described in Subsection 2.2 and they were aimed in 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. All these

Table 2. Structures of the proposed heat exchangers.

Pipe No. (from top to bottom)	HE-1m	HE-2m	
		1^{st} row	2^{nd} row
1	FE60	FE60	FE70
2	FE70	FE70	FE70
3	FE280	FE70	FE70
4	FE280	FE280	FE70
5	FE280	FE280	FE70
6	FE70	FE280	FE70
7	FE70	FE70	FE70
8	FE70	FE70	FE70
9	FE60	FE60	FE70
10	FE60	FE60	FE70
FE280 – pipe length 490 mm, 175 plate ribs every 2.8 mm			
FE60 – pipe length 490 mm, 81 plate ribs every 6 mm			
FE70 – pipe length 490 mm, 140 plate ribs every 3.5 mm			

simulation have been performed applying the uniform air inflow to the exchanger. The uniform mass flow rate of the air has been derived assuming that the total mass flow rate of the air spreads equally on the all measuring fields.

The selected results of computations are gathered in Tab. 3 and, as it was expected, they shown quite significant improvement of the efficiency of the heat exchanger with the uniform air flow. The efficiency growth raises with increasing the air flow rate and water inlet temperature. The modifications of the ribbing structure are also predicted to bring a positive effect – the average growth in the total heat flow rates is about 10%, which can be named as important value.

5 Conclusions

The hypothesis assuming that a specially designed heat exchanger, which ribbing structure fits the certain media flow conditions may perform better than a standard one has been numerically confirmed. A more detailed analysis of this hypothesis, including experiments, is planned in the nearest future.

The computational results, as it was expected, have shown significant growth in the heat flow rates for the exchanger with fully uniform air inflow. The av-

Table 3. Selected computational results.

Measurement No.	Real heat exchanger	Heat exchanger with uniform air flow		Heat exchanger with modified ribbing structure		Uniform modified
	Q_{ex} [kW]	Q_{num} [kW]	δ_Q [%]	Q_{num} [kW]	δ_Q [%]	ΔQ
HE-2/1	10.07	11.59	15.1	11.26	11.8	0.33
HE-2/2	14.08	16.24	15.4	15.36	9.1	0.89
HE-2/3	19.58	22.75	16.2	21.03	7.5	1.71
HE-2/4	4.48	5.10	13.9	4.89	9.3	0.21
HE-2/5	12.04	13.76	14.3	12.83	6.6	0.93
HE-2/6	19.76	22.68	14.8	21.89	10.8	0.79
HE-3/1	12.39	14.27	15.2	13.58	9.6	0.70
HE-3/2	17.31	20.02	15.6	18.97	9.5	1.06
HE-3/3	24.08	28.10	16.7	27.26	13.2	0.85
HE-3/4	5.51	6.28	14.0	5.99	8.8	0.29
HE-3/5	14.81	17.00	14.8	16.02	8.2	0.99
HE-3/6	24.30	28.14	15.8	26.91	10.7	1.23

erage drop in the heat exchanger efficiency, as the effect of the non-uniform air inflow, is of the order of 15%, which is a significant value. Two factors have to be considered while evaluating these results: measuring errors and uncertainties of numerical results. The maximum measuring error is 4%, and the differences between experimental and numerical results may reach up to 11%, but the computational outcomes are always underestimated.

Considering all abovementioned facts the following conclusions may be withdrawn:

- a heat exchanger with specially designed ribbing structure may be an answer in the cases where the media flow maldistribution can not be mitigated (application of such solution needs the economic evaluation of course),
- making the flow uniform, if possible, is more efficient way for improving a heat exchanger performance.

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Prognozowanie efektywności pracy krzyżowoprądowego wymiennika ciepła w warunkach niejednorodnego przepływu czynników

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

Praca dotyczy termodynamicznej analizy krzyżowoprądowego, ożebrowanego wymiennika ciepła typu gaz-ciecz. Autorzy weryfikowali hipotezę zakładającą możliwość uzyskania poprawy efektywności działania urządzenia w wyniku zastosowania specjalnej, dostosowanej do warunków przepływu czynników, struktury ożebrowania. W pierwszej kolejności wykonane zostały pomiary określające zakres i postać niejednorodnego dopływu powietrza do analizowanego wymiennika ciepła, a następnie wykorzystano własny kod komputerowy HEWES do oceny wpływu tej niejednorodności na wydajność cieplną wymiennika. Obliczenia te dotyczyły wymiennika ciepła z jednorodnym ożebrowaniem. Symulacje numeryczne zostały następnie powtórzone dla przypadku wymiennika ze specjalną strukturą ożebrowania, która została dobrana dla wyznaczonych eksperymentalnie warunków dopływu powietrza. Uzyskane rezultaty potwierdzają postawioną na wstępie hipotezę – widoczny jest pewien wzrost wartości strumienia ciepła przekazywanego w wymienniku.