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Experimental study of the heat transfer for a tube bundle in a transversally flowing air

BY HORIA NECULA, ELENA NINETA GHIZDEANU, GEORGE DARIE
Faculty of Power Engineering
University "Politehnica "of Bucharest
Splaiul Independentei 313, Sector 6 060042 Bucharest, Romania

ROMANIA

Abstract. - The simplest form of cross flow heat exchanger may be regarded as a series of identical heat transfer surfaces in a transverse stream that each has an influence on, and is in turn influenced by its neighbour. Therefore, in order to obtain a prediction for the heat transfer rate to or from a bundle of surfaces in cross flow it is usual to initially consider a single surface in isolation as a basis for correlation. In one of the most common arrangements, heat is transferred between a fluid flowing through a bundle of tubes and another fluid flowing transversely over the outside of the tubes. The main goal of this study is the experimental determination of the convective heat transfer coefficient transferred between a fluid, which is the air, flowing through a bundle of tubes in a transversally flowing air in a staggered arrangement and the comparing with the theoretical correlations.

Keywords: heat exchanger, bundle of tubes, staggered arrangement, convective heat transfer coefficient.

Nomenclature:

d_h	hydraulic diameter [m]
n	coefficient
Nu	Nusselt Number, $Nu = \frac{\alpha \cdot d_h}{\lambda}$
m	coefficient
Pr	Prandtl Number, $\Pr = \frac{\nu}{a}$
Re	Reynolds Number, $\mathrm{Re} = \frac{w \cdot d_h}{\nu}$
s	pitch [m]
w	air velocity in the minim section

Greek symbols

α	convective heat transfer coefficient $[W \cdot m^{-2} \cdot K^{-1}]$
ε	correction coefficient function of the number of tube rows crossed by the fluid
ν	air cinematic viscosity $[m^2 \cdot s^{-1}]$

Subscripts symbols

s – surface
N – row number
L – longitudinal
T – transversal

1. INTRODUCTION

The heat exchangers are thermal equipment present in almost all industrial sectors, playing an essential role in many processes and systems. Increasing the efficiency of this equipment determines functioning conditions and performance of technological assemblies.

Heat transfer to or from a bundle of tubes in cross flow is relevant to numerous industrial applications such as steam generation in a boiler or air cooling in the coil of an air conditioner.

The overall heat transfer coefficient for a cross flow heat exchanger is made up of three components: the surface heat transfer coefficient for the fluid flowing through the tubes, the thermal conductivity and thickness of the tube material and the surface heat transfer coefficient for the fluid flowing over the external surface of the tubes. For a better dimensioning of these devices there is necessary a better evaluation of the convective heat transfer coefficients [3].

The characteristics of the heat exchanger can be established wither directly by experimental measurements or by numerical simulations.
The experimental measurements are needed in order to develop new the new heat exchanger designs, and for the establishment of the optimal operational parameters [6].

In the case of a gas, which in our case is air, flowing throughout a bundle of tubes, the assessment of the effective heat transfer coefficient is very important seeing that in general there are the lowest convective heat transfer coefficients, which influence the effective global heat transfer coefficients.

Making a piece of equipment as compact as possible for obtaining a heat transfer rate as big as possible is the main concern in the research activity of heat exchangers.

2. EXPERIMENTAL DEVICE

Our experimental device presented in figure 1 consists in the air duct, which is vertically mounted glass reinforced plastic duct with bell mouth intake at its upper end. The fan is mounted on an epoxy coated welded steel frame. Air duct is directly mounted on the frame and fan intake.

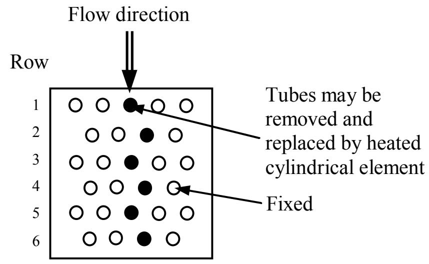

Figure 1. Experimental setup

The active element is a thick cylinder electrically heated. The maximum temperature reached by the active element is 100 $^{\circ}$C. Extreme ends are insulated to reduce errors due to wall effects. Integral thermocouple senses surface temperature.

The clear plastic plate with 27 fixed plastic tubes of 16 mm nominal diameter arranged in a staggered pitch with the longitudinal pitch s_L = 27.5 mm, and the transverse pitch s_T = 32 mm. Near the centre of each row is a dummy tube that may be removed and replaced with the active element.

All electronic instrumentation and control is housed in a plastic coated steel console which consists of the digital electronic thermometer with 0.1 resolution, which indicates element surface temperature and, via a biased switch, the duct air temperature and an analogue voltmeter indicating the voltage across the active element heater. The maximum voltage is 70 V.

The pressure is measured by 2 duct mounted inclined manometer recording intake depression one with the range from 0 to 70

mmH_2O and the second one from 0 to 30 mmH_2O.

On this experimental device it can be made the following experiments:

- Steady state determination of the mean surface heat transfer coefficient for tubes in the 1st, 2nd 3rd, 4th, 5th and 6th rows of a cross flow heat exchanger;
- Determination of the mean surface heat transfer coefficient for cross flow heat exchangers with one to six rows;
- Deduction of the relationship between Nusselt, Reynolds and Prandtl numbers for each of the six tube rows.

The experimental study was realized with the device presented above. The active element was introduced into the top open hole in the tube plate and plugs the lead into the element console. The other five remaining dummy tubes are left in position in the lower holes. The duct pressure is connected of the lower manometer and the fan is set up to the lower position which corresponds with the fan closed in order to obtain a low velocity The device is started up there is adjusted the heater control by increasing the voltage to give an indicated element surface of approximately 95 ºC. When stable conditions are occurred indicated by a constant active element surface temperature are recorded the active element surface temperature, the air temperature, the pressure and the voltage. After that the iris damper of the fan exhaust is adjusted in order to increase the air velocity. By iris dumper of the fan exhaust variation the velocity range is setup between 5 m/s and 12 m/s. In this new case the heater control is increased to give approximately the original active element surface temperature. Again, when stable there are recorded the same variables. The procedure is repeated by increasing the air velocities up to the maximum, which corresponds of an iris damper fully open. After that the element is cooled and it is placed in the second row hole and the dummy tube from this hole is placed in the first row hole. The whole experiment is repeated for a similar range of air velocity. The entire procedure is repeated with the active element in the row 3, 4, 5 and 6.

3.THEORETICAL CORRELATION

Turbulent flow conditions do not lend themselves to simple theoretical analysis and therefore alternative methods are required in order to evaluate surface heat transfer coefficients for general flow conditions. One of these methods is to apply the principle of dynamic similarity, which can be summarized by writing [3]:

$$Nu = f(\mathrm{Re}, \mathrm{Pr}) \tag{1}$$

Using the dimensional analyzing the general relation (1) results in the following relation:

$$Nu = C \cdot \mathrm{Re}^m \cdot \mathrm{Pr}^n \tag{2}$$

The tube position within the bundle adds further variable to the general turbulent flow equation (2) and therefore this has the form:

$$Nu = C \cdot \mathrm{Re}^m \cdot \mathrm{Pr}^n \cdot \varepsilon_N \tag{3}$$

The values of the constant C and m are function on geometric parameters and tube – bundle arrangements. The values of constant n are fluctuating through 0.3; 0.33; and 0.36. ε_N represents a correction coefficient function of the position of tube rows in the bundle [3].

Alternatively, there are presented the most used correlations applicable described in the table 1.

Table 1 – The most used correlation for *Nu* in a staggered arrangement

Correlation	Applicability conditions	Observations
Isachenko [1] $Nu = 0.41\,\mathrm{Re}^{0.6}\,\mathrm{Pr}^{1/3}\left(\frac{\mathrm{Pr}}{\mathrm{Pr}_s}\right)^{0.25}\left(\frac{s_T}{s_L}\right)^{1/6}\varepsilon_N$	$10^3 < \mathrm{Re} < 10^5$; $0{,}7 < \mathrm{Pr} < 500$; $0{,}25 < \mathrm{Pr}/\mathrm{Pr}_s < 4$.	$N_L = 1, \varepsilon_N = \varepsilon_1 = 0.6$; $N_L = 2, \varepsilon_N = \varepsilon_2 = 0.7$; $N_L \geq 3, \varepsilon_N = 1$.
Kays [2] $Nu = 0.33\,\mathrm{Re}^{0.6}\,\mathrm{Pr}^{0.3}\,\varepsilon_N$	$\mathrm{Re} \geq 6 \cdot 10^3$; $0{,}7 < \mathrm{Pr} < 300$.	$\varepsilon_1 = 0.68; \varepsilon_2 = 0.75; \varepsilon_3 = 0.83$; $\varepsilon_4 = 0.89; \varepsilon_5 = 0.92; \varepsilon_6 = 0.95$; $\varepsilon_7 = 0.97; \varepsilon_8 = 0.98; \varepsilon_9 = 0.99$; $\varepsilon_{10} = 1.0$.
Miheev [3] $Nu = 0.4\,\mathrm{Re}^{0.6}\,\mathrm{Pr}^{0.36}\left(\frac{\mathrm{Pr}}{\mathrm{Pr}_s}\right)^{0.25}\varepsilon_N$	$\mathrm{Re} > 10^3$	$N_L = 1, \varepsilon_N = \varepsilon_1 = 0.6$; $N_L = 2, \varepsilon_N = \varepsilon_2 = 0.7$; $N_L \geq 3, \varepsilon_N = 1$.
Zhukauskas [4] $Nu = 0.35\left(\frac{s_T}{s_L}\right)^{0.2}\mathrm{Re}^{0.6}\,\mathrm{Pr}^{0.36}\left(\frac{\mathrm{Pr}}{\mathrm{Pr}_s}\right)^{0.25}\varepsilon_N$	$10^3 < \mathrm{Re} < 2 \cdot 10^5$; $s_T/s_L < 2$.	$\varepsilon_1 = 0.64; \varepsilon_2 = 0.76; \varepsilon_3 = 0.84$; $\varepsilon_4 = 0.89; \varepsilon_5 = 0.92; \varepsilon_7 = 0.95$; $\varepsilon_{10} = 0.97; \varepsilon_{13} = 0.98; \varepsilon_{16} = 0.99$.
Grimison [5] $Nu = C\,\mathrm{Re}^m\,\mathrm{Pr}^{1/3}\,\varepsilon_N$	$2 \cdot 10^3 < \mathrm{Re} < 4 \cdot 10^4$; $\mathrm{Pr} \geq 0{,}7$; $1.25 \leq s_T/D \leq 3$; $0.6 \leq s_L/D \leq 3$; $C, m = f\left(\frac{s_T}{D}, \frac{s_L}{D}\right)$; In our case *C*=0.465 and *m* =0.563.	$\varepsilon_1 = 0.68; \varepsilon_2 = 0.75; \varepsilon_3 = 0.83$; $\varepsilon_4 = 0.89; \varepsilon_5 = 0.92; \varepsilon_6 = 0.95$; $\varepsilon_7 = 0.97; \varepsilon_8 = 0.98; \varepsilon_9 = 0.99$.

4. EXPERIMENTAL STUDY

The determination of the correlation for the mean convective heat transfer coefficient for the tubes forming a cross flow heat exchanger is carried out experimentally using the device presented above. The measurements were made for each row of the tube bundle, for different air velocities, which implied different Reynolds numbers.

Using the correlation described in the table 1 and the experimental data there are obtained the results presented in the figures 2 – 6 for a tube bundle in a staggered arrangement.

In the figure 2 there is presented the variation of *Nu* with *Re* for each row from 1 to 6.

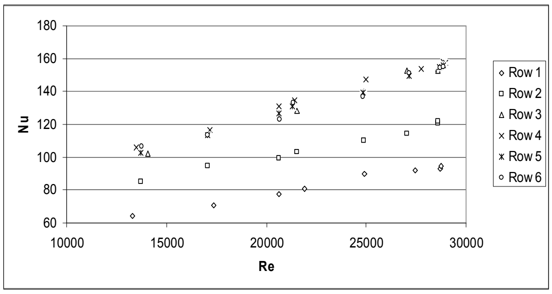

Figure 2 – Experimental data for each row

In figure 3 and 4 there are presented the comparison of the variation of Nu with Re through the experimental data and the most used correlation: Isacenko, Kays, Miheev, Zukauskas, and Grimson for the 1st row and 2nd row.

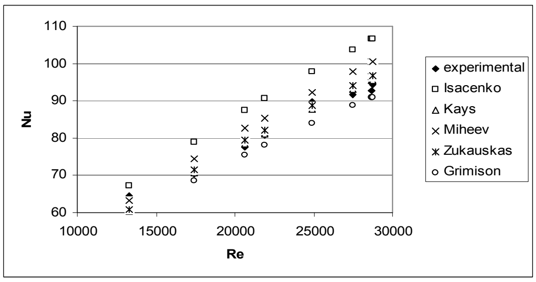

Figure 3 – Comparison through experimental data and theoretical correlation for the 1st row

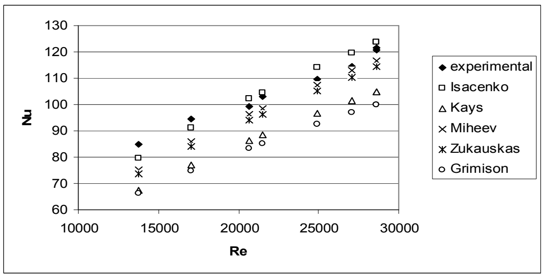

Figure 4 – Comparison through experimental data and theoretical correlation for the 2nd row.

Starting with the 3rd row there is observed that the experimental data are practically superposed. Therefore there was born the idea of adopting a model, which is assuming the unification of the heat transfer from the 3rd row above, similar with Isachenko's and Miheev's model. The results are synthesized in the figure 5.

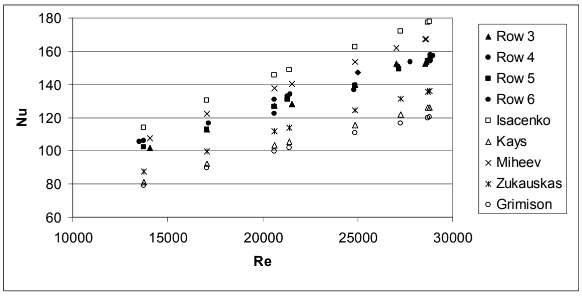

Figure 5 – Comparison through experimental data and theoretical correlation for the 3, 4, 5, and 6 rows

For the model created there is obtained the correlation between *Nu* and *Re* and determined the values of the constants *C* and *m*. Therefore, according to figure 6, on the linear regression of the experimental data for the rows 3 to 6 there was obtained same value 0.56 for the coefficients *C* and *m*. This value might be considered with precaution due a few numbers of experimental points, but however it is approached to the values existing in literature for this range of Reynolds.

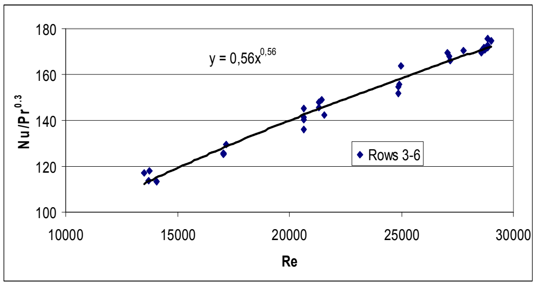

Figure 6 – Determination of the values of the constants *C* and *m*.

For the rows 1 and 2, the correction coefficients obtained have the values: ε_1 = 0.61 and respectively ε_2 = 0.79, according to the correlation obtained and presented in figure 6.

5. CONCLUSIONS

In the experiment performed there is obtained the variation between Nu and Re for Re from 10000 to 30000, our range of study. For this study there was observed that the minimum error for our regression is obtained for a coefficient of Prandtl equals to $n = 0.3$.

Comparing the experimental data for Nusselt numbers with the data obtained form figure 6, results a range of relative errors of (– 4 % ÷ 6 %) for the whole experiment achieved. The standard relative errors, related to the theoretical correlations, for this type of experiment are presented in table 2 for the rows 1 to 6.

For the rows 1 to 2 the correcting factors for ε_N have a similar value with the theoretical data presented in table 1.
The Nusselt numbers for the row 3 to 6 obtained experimentally, are not influenced anymore on the corrective factors variables ε_N of each row, therefore there was adjusted a common relationship for these rows.

Table 2 – Theoretical relative errors

Author	Relative errors	
	min.	max.
Isacenko	-6%	19%
Kays	-26%	3%
Miheev	-11%	12%
Zukauskas	-20%	4%
Grimison	-30%	-2%

As a conclusion, choosing the best correlation for a flowing throughout a bundle of tubes, should be made carefully, due to the errors that could be obtained, errors that should be avoid as much as possible. And upwards the errors should be minimums due to the estimation of the global transfer heat coefficient for the whole device.

Moreover, for example, for an air heater, using different correlations for convective heat coefficient is influenced the heat surface. In this case, the burned gases are passed through the tubes, and the air is passed transversally over outside the tubes.
Using the correlations described above, there are obtained the following relative errors for the heat surface (see table 3).

Table 3 – Relative errors for heat surface for different correlations (Reference Miheev's correlation)

Author	Global heat surface relative errors
Isacenko	5.60%
Kays	1.32%
Zukauskas	0.42%
Grimison	2.30%

Therefore for obtaining the optimum solution the correlation should be chosen very carefully. This case is obtained when the air convective coefficient is greater than the gas convective coefficient. If the air convective coefficient is smaller than the gas convective coefficient than the relative errors are smaller.

All these results are the first step in the heat exchangers research. This study may be continued with the research of others geometries, others arrangements, others fluids. Moreover a very interesting research is the simulation in FLUENT and CFX code, for all these new research directions.

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