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Temperature distribution in the outlet of cross-flow heat exchangers

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Abstract—The well-known equation $\dot{Q} = kA\,\Delta T_m$ for the heat transfer contains the adiabatic mixed temperature ΔT_m and does not permit the determination of the temperature distribution in the outlet cross-section of a heat exchanger. In general, this temperature distribution is not homogeneous, but considerably deviates locally from the adiabatic mixed temperature according to the tube pitch, the Reynolds number of the flow and the turbulence intensity. In order to examine these influences, temperature distributions behind tube bundles with three rows of plain tubes with transversal and longitudinal pitch-to-diameter ratios of $a = 2.58$, $b = 1.29$ and $a = b = 1.29$ were measured by calibrated resistance thermometers. Fluid velocities in the minimum cross-section ranged from 5 to 20 m s^{-1} and free stream turbulence intensities from 0.8 to 16.6%. The tubes were heated by saturated steam and cooled by air in cross flow. The measured temperature distribution in the outlet cross-section of the heat exchangers can be characterised as mixed, partially mixed or unmixed. A sufficiently small transversal and longitudinal pitch-to-diameter ratio, e.g. $a = b = 1.29$, leads to a homogeneously mixed flow and thus to a homogeneously mixed temperature distribution. A higher transversal pitch-to-diameter ratio, e.g. $a = 2.58$, leads to an unmixed flow and to an unmixed temperature distribution. A turbulence intensity $Tu = 16.6\%$ of the oncoming flow enhances the lateral mixing effect, provided that the transversal pitch-to-diameter ratio is not below $a = 2.58$. For the transversal pitch-to-diameter ratio $a = 1.29$, channel flow between the tubes prevents better lateral mixing even for a turbulence intensity $Tu = 16.6\%$. Duplicating the Reynolds number up to $Re_e = 42\,000$ for a transversal pitch-to-diameter ratio $a = 2.58$ and to $Re_e = 57\,000$ for a transversal pitch-to-diameter ratio $a = 1.29$ intensifies lateral mixing significantly for the greater transversal pitch-to-diameter ratio, $a = 2.58$.

1. INTRODUCTION

The performance of heat exchangers, and particularly of tube bundle heat exchangers, is usually determined either by means of the logarithmic mean temperature difference method, the so-called LMTD method, or the effectiveness NTU method. Both methods, described for instance in ref. [1], presume that the mean temperature difference, and therefore the temperatures of the fluid at the inlet and at the outlet, are known. Knowledge of the mean temperature difference permits calculation of the transferred heat flux, according to:

$$\dot{Q} = kA\,\Delta T_m. \tag{1}$$

For heat exchangers with two fluid streams a calculation of the mean temperature requires four different temperatures: the inlet and outlet temperature of the hot fluid and the inlet and outlet temperature of the cold fluid. In heat exchangers with two fluids in co-current flow, the logarithmic mean temperature difference ΔT_m is defined as:

$$\Delta T_m = \frac{(T_{h,out} - T_{c,out}) - (T_{h,in} - T_{c,in})}{\ln[(T_{h,out} - T_{c,out})/(T_{h,in} - T_{c,in})]}. \tag{2}$$

In counter-current flow heat exchangers the logarithmic mean temperature difference is given by:

$$\Delta T_m = \frac{(T_{h,out} - T_{c,in}) - (T_{h,in} - T_{c,out})}{\ln[(T_{h,out} - T_{c,in})/(T_{h,in} - T_{c,out})]}, \tag{3}$$

where the indices h and c denote the hot and cold fluid, respectively. If the effectiveness NTU method is preferred, the effectiveness ε is defined by:

$$\varepsilon = \frac{\dot{q}}{\dot{q}_{max}} = \frac{C_h(T_{h,in} - T_{h,out})}{C_{min}(T_{h,in} - T_{c,out})} = \frac{C_h(T_{h,in} - T_{h,out})}{C_{min}(T_{h,in} - T_{c,out})}, \tag{4}$$

where C is the heat capacity rate of the fluid flow. According to the definition of the effectiveness ε, equation (4), also outlet temperatures ought to be known.

Nevertheless, fluid temperatures in the inlet cross-section denoted by the index in and temperatures in the outlet cross-section denoted as out are mostly non-uniform. Their distribution depends on parameters, such as tube arrangement, fluid velocity and turbulence intensity. Both methods, the LMTD and the effectiveness NTU, assume in the outlet cross-section either a homogeneous temperature distribution as it

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NOMENCLATURE

A	heat transfer area [m^2]	Tu	turbulence intensity
a	transversal pitch-to-diameter ratio	ΔT_m	mean temperature difference [K]
b	longitudinal pitch-to-diameter ratio	t_E	ambient temperature [°C]
C	heat capacity rate	t_{exp}	measured temperature [°C]
c_1, c_2	coefficients in equation (9)	Δt	temperature difference [K]
c_{pL}	heat capacity of air [kJ (K^{-1} kg)$^{-1}$]	u	flow velocity [m s^{-1}]
d_a	tube outside diameter [m]	U_S	output voltage [V].
ε	effectiveness		
Δh_V	enthalpy of vaporisation [kJ kg^{-1}]		
k	heat transfer coefficient [W K^{-1} m^{-2}]	Indices	
K	number of experiments	c	cold
$\dot{M}$	mass flow [kg s^{-1}]	ch	channel
n	current number	e	smallest cross-section
Δp_{12}	static pressure difference [Pa]	h	hot
$\dot{Q}$	heat flux [W]	in	in
$\dot{q}$	heat flux density [W m^{-2}]	K	condensate
ρ	density [kg m^{-3}]	L	air
Re	Reynolds number	max	maximum
s_l	distance between tube columns [m]	min	minimum
s_q	distance between tube rows [m]	o	free stream
T	temperature	out	out.

occurs in a fully mixed flow or temperature differences as they occur in an unmixed flow. Even if a flow is homogeneously mixed in each plane perpendicular to the flow direction, its temperature can still vary in the flow direction. In an unmixed flow, temperature differences can occur perpendicular to the flow direction, but according to Taborek [2] heat fluxes due to these temperature differences can be neglected.

In technical apparatus, however, the flow is partially mixed and the outlet temperature distribution is unknown [2, 3]. To date one cannot predict to what extend a partially mixed flow is formed, which leads to uncertainties in design and prediction of the performance.

The purpose of this investigation is to measure temperature distributions at the outlet cross-section of tube bundle heat exchangers. In the experiments heat exchangers with three rows of plain tubes and different tube arrangements for various fluid velocities and turbulence intensities were studied. According to the outlet temperature distribution the flow can be characterised as mixed, partially mixed or unmixed.

2. APPARATUS

The experiments were made with an open wind tunnel, described in detail in refs. [4, 5], Fig. 1. Air flows through the inlet nozzle (a) and the inlet housing with integrated flow rectifiers and calming screens to the test section (b), where it is heated by passing the hot tube bundle (c). The air is then cooled to ambient temperature in a finned radiator (d). A fan (e) with a performance of 37 kW and a flow rate of 2 m^3 s^{-1} blows the air through the test section (b). The air passes a flow rectifier with an integrated sound absorber. The fan motor is sound-isolated by an absorbing chamber. For low air velocities an additional intake (g) was installed. The tubes are heated by saturated steam from an electric steam boiler with a capacity of 96 kW. The steam line between boiler and dome is electrically heated. The steam passes through the steam dome (h) to the vertically arranged tubes and condenses on their inside. The condensate is cooled in a counter-current cooler (i) and collected in beakers (k). Figure 2 shows the test section with a tube bundle with three rows of plain tubes, the location of temperature measurements (T_1–T_4) and the differential pressure measuring locations (p_1 and p_2). The test section has a square cross-section and a length of 258 mm. Plain aluminium tubes were installed vertically, fixed with slipped bushes and sealed with O-rings. To vary the turbulence intensity of the oncoming air stream, turbulence generating grids were installed in a distance of x_1 = 100 mm in front of the first tube row. This investigation was performed with biplanar grids described in refs. [6–9]. The turbulence intensities are given in Table 1 together with the geometrical and experimental parameters.

Figure 3 illustrates the outlet cross-section of the heat exchanger with an installed temperature probe. The temperatures were measured by a platinum resistance thermometer and a constant current anemometer. The sensing element was a single platinum wire of 0.4 mm length and of 1 μm diameter. Local

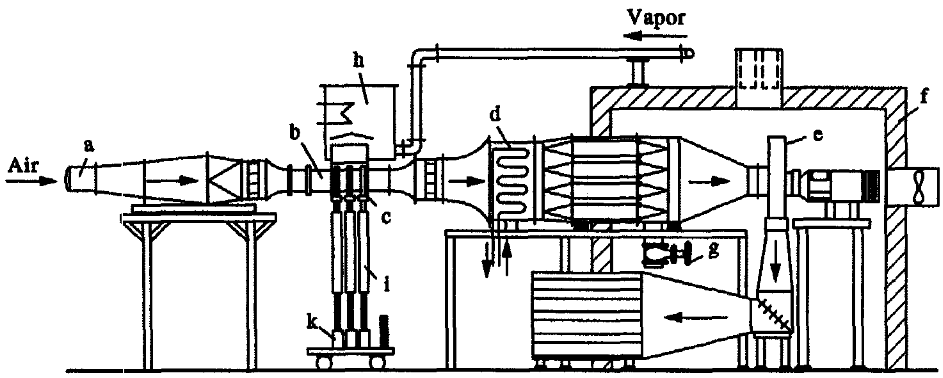

Fig. 1. Experimental set-up.

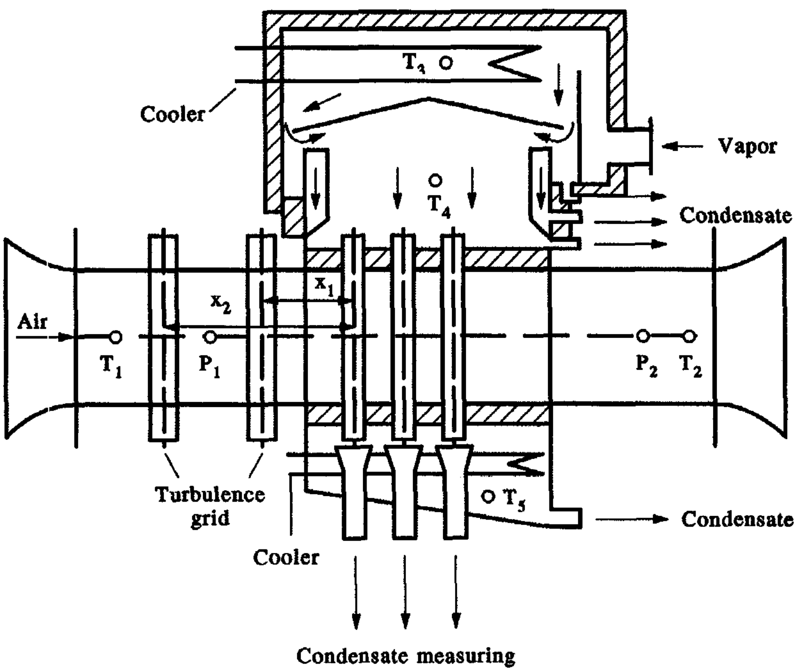

Fig. 2. Test section.

Table 1. Geometrical dimensions and experimental parameters

Transversal pitch-to-diameter ratio, a	Longitudinal pitch-to-diameter ratio, b	Flow velocity, u_0 (m s^{-1})	Turbulence intensity, Tu (%)	z-Coordinate (mm)
2.58	1.29	10	0.8	241.5, 216.5, 166.5, 116.5, 66.5, 16.5
2.58	1.29	7.5	0.8	116.5
2.58	1.29	20	0.8	116.5
2.58	1.29	7.5	6.7	116.5
2.58	1.29	7.5	16.6	116.5
1.29	1.29	10	0.8	241.5, 166.5, 116.5, 66.5
1.29	1.29	5	0.8	116.5
1.29	1.29	10	16.6	116.5

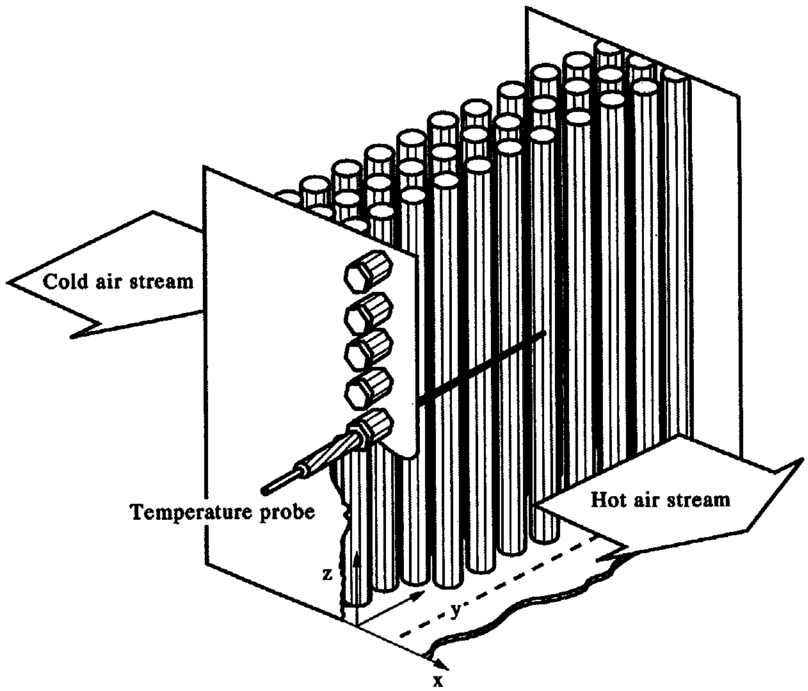

Fig. 3. Outlet cross-section with temperature probe.

temperatures could thus be measured. The origin of the coordinate system x, y, z for the platinum wire is located as indicated in Fig. 3 at the bottom and on the side wall of the heat exchanger, 25.8 mm behind the centreline of the third tube row. The distance in the y-direction could be varied between 33 mm and 224 mm in steps of 1 mm. The distance in the z-direction ranged from 16.5 to 241.5 mm in steps of 25 mm.

The experimental parameters such as transversal and longitudinal pitch-to-diameter ratio, flow velocity, turbulence intensity and the distance in the z-direction, described as the z-coordinate, are listed in Table 1.

The flow velocity in the smallest cross-section u_e depends on the free stream flow velocity u_o and on the transversal pitch-to-diameter ratio:

$$a = s_q/d_a, \tag{5}$$

where s_q is the distance between two tube columns and d_a is the outside diameter of the tubes according to:

$$u_e = u_o(a-1)/a \tag{6}$$

The longitudinal pitch-to-diameter ratio is defined by:

$$b = s_l/d_a, \tag{7}$$

where s_l is the distance between two tube rows. The designations column and row are adopted here from Ishigai and Nishikawa [10]. They defined row as a line of tubes perpendicular to the flow direction and column as a line of tubes in the flow direction.

The maximum flow velocity in the smallest cross-section was restricted to $u_e = 36\ \mathrm{m\ s^{-1}}$ for a transversal pitch-to-diameter ratio $a = 2.58$ and to $u_e = 13.5\ \mathrm{m\ s^{-1}}$ for a transversal pitch-to-diameter ratio $a = 1.29$. Due to velocity fluctuations caused by flow separation and turbulent velocity fluctuations generated by turbulence grids, the flow velocity in the smallest cross-section was kept below these values, as Table 1 indicates, in order to avoid damage of the sensing wire.

3. TEMPERATURE MEASUREMENTS

To measure local temperatures in gas flows the sensing element should have a small size to yield a high geometrical resolution. It should not disturb the gas flow and its material properties, e.g. temperature coefficient and electrical resistance, should be constant. Because temperatures at stationary points in a turbulent flow fluctuate with time, the response time of the sensing element must be short enough to detect high frequency temperature fluctuations.

The experiments were conducted with a platinum resistance thermometer because platinum meets the requirements over a wide temperature range. The electrical resistance was determined from the electrical current through the wire, the measured voltage being between 4 and 6 mV $\mathrm{K^{-1}}$. Hashemian and Peterson [11] reported a strong linear correlation between temperature and voltage for resistance thermometers. Corrsin [12] pointed out that hot-wire anemometers are very appropriate to measure the temperature in turbulent gas flows. Quick reacting and low noise level Wheatstone bridges permit measurement of high

Table 2. Experimental parameters and results

Arrangement	u_o ($m\ s^{-1}$)	Re_e	Tu (%)	Δp_{12} (Pa)	$\dot{M}_K$ ($g\ s^{-1}$)
$a = 2.58$	10	20 100	0.8	45.89 ± 0.39	0.67 ± 0.024
$a = 2.58$	7.5	15 730	0.8	25.98 ± 0.39	0.55 ± 0.0005
$a = 2.58$	20	41 930	0.8	177.3 ± 0.29	1.16 ± 0.005
$a = 2.58$	7.5	15 730	6.7	38.54 ± 0.29	0.64 ± 0.0065
$a = 2.58$	7.5	15 730	16.6	69.14 ± 0.1	0.7 ± 0.0052
$a = 1.29$	10	57 110	0.8	1260.5 ± 17.95	3.12 ± 0.08
$a = 1.29$	5	28 550	0.8	319.20 ± 4.9	1.95 ± 0.01
$a = 1.29$	10	57 110	16.6	1442.5 ± 25.01	3.45 ± 0.18

frequency temperature fluctuations [13]. Because of the low current intensity of 1–0.2 mA the temperature rise of the wire is negligible and thus the temperature of the gas flow independent of the flow velocity. The high frequency of the Wheatstone bridge, up to 2 kHz [14], permitted quick measurements of high frequency temperature fluctuations. The platinum wire employed had a diameter of 1 μm and a length of 0.4 mm and thus did not essentially disturb the gas flow. For exact and accurate measurements, calibration is essential in order to eliminate possible interference effects of the electrical equipment.

As a reference temperature for calibration, a quartz-thermometer with an accuracy of 0.01 K was used. Temperatures and output voltages could be correlated by:

$$T = c_1 + c_2 U_S, \tag{8}$$

with a mean relative error of approximately 0.05%. The calibration procedure had to be repeated at regular intervals because structural changes of the probe material during the experiments could occur, resulting in a change of the coefficients. Dust from the air could be deposited on the probe surface, as well. As a consequence slight changes in the coefficients could be noticed and the coefficients had to be adapted to these. Also the output voltage had to be adjusted when the ambient temperature changed. The temperatures indicated are mean values from 10 experiments. Because of temperature fluctuations behind the apparatus, the voltage of the anemometer also fluctuates with time. A mean value is obtained from an integration:

$$U_m = \frac{1}{\Delta t}\int_{\Delta t} U_S(x, y, z)\,\mathrm{d}t, \tag{9}$$

performed by a signal-analyser with an integration time of 1 s.

4. EVALUATION OF THE MEASUREMENTS AND PRESENTATION OF THE RESULTS

Table 2 lists the experimental parameters. Figure 4 presents the measured temperatures for a transversal pitch-to-diameter ratio $a = 2.58$, a flow velocity $u_o = 10\ m\ s^{-1}$, a turbulence intensity $Tu = 0.8\%$ and for a distance $z = 116.5$ mm. The experiments represented by crosses and circles were taken at two successive days.

When the ambient temperature changed differences up to 3 K could be registered. Each experimental point therefore had to be adjusted to the actual ambient temperature according to:

$$\Delta t(y, n) = t_{\exp}(y, n) - \Delta t_E, \tag{10}$$

where $\Delta t(y, n)$ is the temperature difference number n in the distance y and $t_{\exp}(y, n)$ is the measured temperature number n in the same distance y and:

$$\Delta t_E = (n+K)/K\, t_E(0) - t_E(K), \tag{11}$$

where $t_E(0)$ is the ambient temperature at the beginning of a series of experiments and $t_E(K)$ the ambient temperature at the end of these series. K is the number of experiments.

After correction of the experiments of Fig. 4 by means of equations (10) and (11), one obtains the results presented in Fig. 5. As Fig. 5 reveals, changes of the ambient temperature are now eliminated.

5. RESULTS

The flow pattern inside a tube bundle influences the heat transfer and hence the temperature distribution behind the bundle. The flow around a tube in the first row of the bundle is similar to that around a single cylinder [15, 16]. In tube bundles with three rows of in-line tubes, as investigated, tubes of the second and third row are placed in the wake of the preceding row. It is apparent that the shear flow separating from the preceding tube influences the flow of the following tube. It is of crucial importance for the heat transfer whether the sheet-flow rolls up in front of a tube and whether vortices are formed. Kostic and Oka [17] examined the flow around two cylinders in in-line arrangement for Reynolds numbers, Re, between 12 000 and 40 000 and longitudinal pitch-to-diameter ratios, b, between 1.6 and 9. For a large longitudinal pitch-to-diameter ratio a tube is not touched by the vortex of its preceding tube. Behind the preceding tube the shear flow rolls up and separating vortices lead the main flow into the gap between the cylinders. High velocity in combination with strong temperature gradients yield a good heat transfer from the following

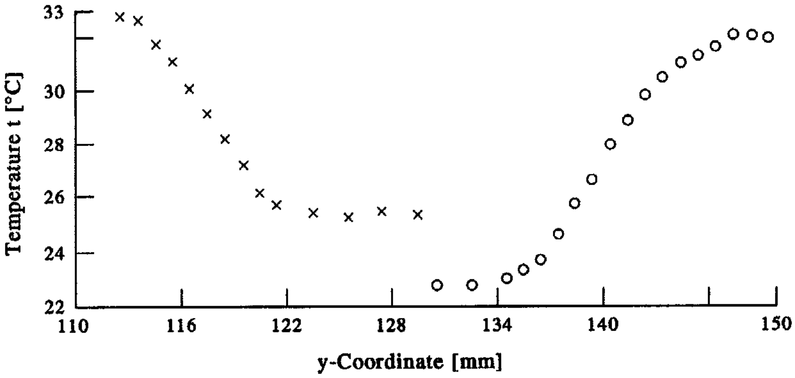

Fig. 4. Temperature distribution.

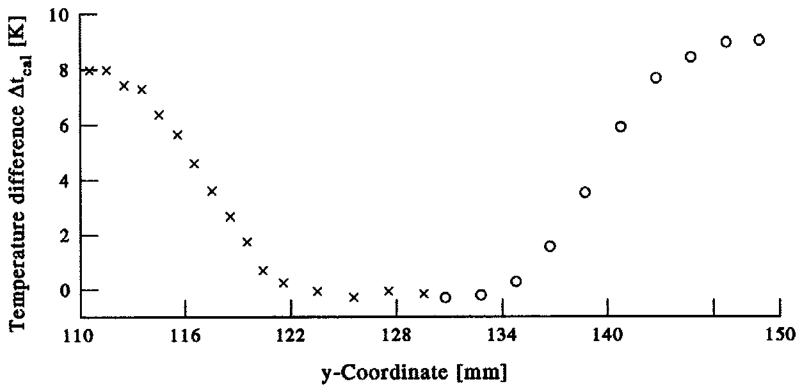

Fig. 5. Corrected temperature distribution.

tube. The flow around the following tube corresponds to that of a single cylinder at high Reynolds numbers, according to the high turbulence in the wake of the preceding tube. With decreasing longitudinal pitch-to-diameter ratio, a tube is placed into the vortex region of its preceding tube. Between both tubes a vortex region is built-up, without temperature or velocity exchange. The heat transfer on the rear side of the preceding tube and on the front part of the following tube is low [18].

Figure 6 shows temperature differences for a heat exchanger with a transversal pitch-to-diameter ratio $a = 2.58$, a flow velocity $u_o = 10$ m s^{-1}, a turbulence intensity $Tu = 0.8\%$ and for a distance $z = 216.5$ mm. Numbers 1–4 indicate the columns. The distance between the tube axes perpendicular to the flow direction is 51.6 mm, i.e. twice the distance of the tube axes in flow direction. Behind the tubes four regions with elevated temperatures are noteworthy. The temperature difference Δt amounts to approximately 9 ± 0.5 K. In addition, the maximum temperature difference is shifted to the middle of the test section, located at $y = 129$ mm, due to fluctuations of the static pressure differences between the vortex regions, an effect known as the Coanda Effect. It describes the flow attachment of a slit jet close to a wall. Values of $\Delta t < 0$ in the gap between the tubes result from a weak velocity sensitivity of the temperature probe. These values indicate an unmixed flow. Cold air flows without heat exchange through the gap between the tube columns. A similar result was obtained by Ishigai *et al.* [19] and Ishigai and Nishikawa [10] for Reynolds numbers between 1400 and 33 000 and for transversal pitch-to-diameter ratios $a < 2$. Lohrisch [20] visualised an equivalent flow with ammonium chloride.

Hot condensate found above the top plate and the higher heat transfer coefficient inside at the top of the tubes due to the thin condensate film lead to higher temperatures close to the top plate of the heat exchanger and behind the tubes, as shown in Fig. 7. This figure is valid for a transversal pitch-to-diameter ratio $a = 2.58$, a flow velocity $u_o = 10$ m s^{-1}, a turbulence intensity $Tu = 0.8\%$ and for a distance $z = 241.5$ mm, represented by crosses, and a distance $z = 216.5$ mm, represented by circles. Between the tubes the temperatures do not considerably depend on the z-coordinate. According to Taborek the flow behind this tube arrangement can be characterised as unmixed.

Different results were obtained for a heat exchanger with transversal pitch-to-diameter ratio $a = 1.29$, Fig. 8, which again is valid for a flow velocity $u_0 = 10$ m s^{-1}, a turbulence intensity $Tu = 0.8\%$ and for a distance $z = 241.5$ mm, represented by circles, and

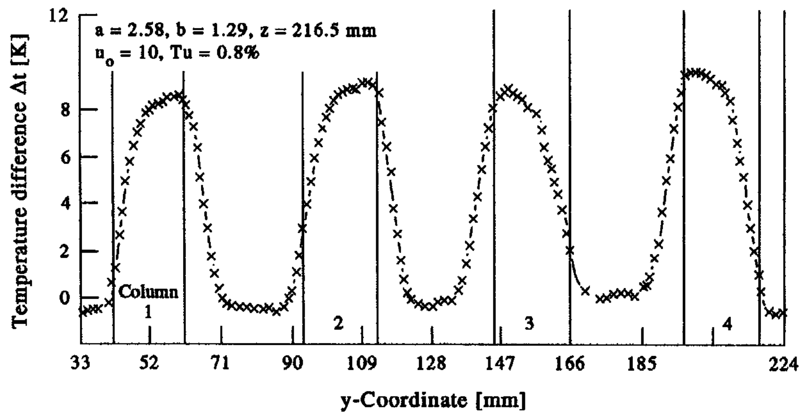

Fig. 6. Temperature differences.

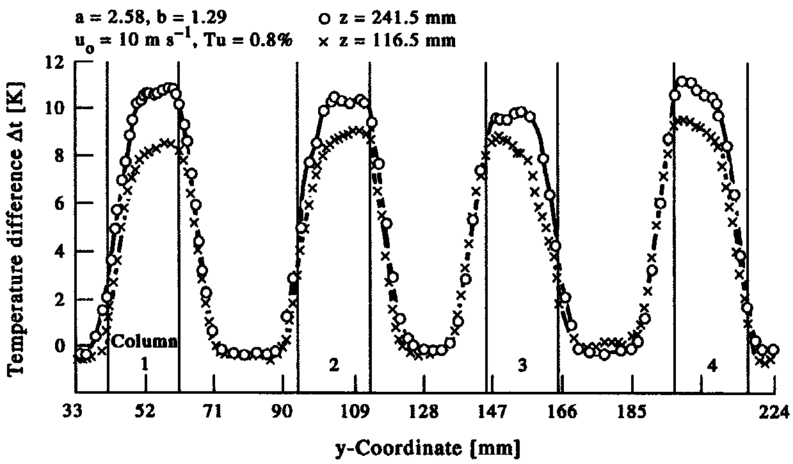

Fig. 7. Temperatures differences for $z = 241.5$ mm and $z = 216.5$ mm.

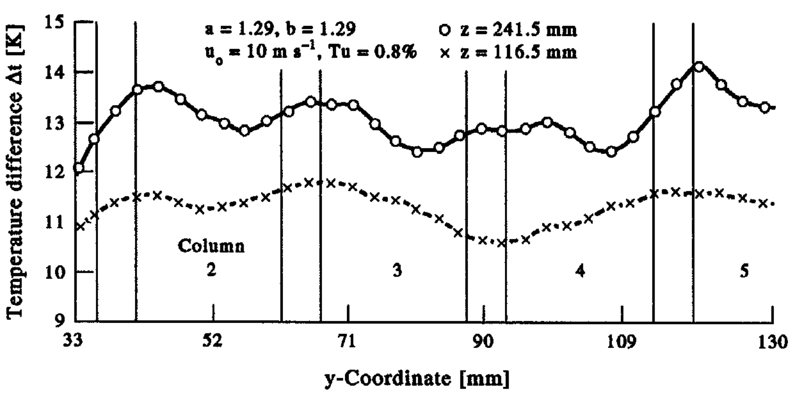

Fig. 8. Temperature differences for $z = 241.5$ mm and $z = 116.5$ mm.

for a distance $z = 116.5$ mm, represented by crosses. Neither regions with pronounced high nor low temperature differences can be observed. The temperature difference in the gap between the tube columns does not differ from the values along the hot tubes. This is caused by lateral mixing between heated and unheated layers. The difference between minimum and maximum temperature difference for all z-coordinates amounts only to 2 K. According to these results the flow in the outlet cross-section of a tube bundle heat exchanger with a transversal pitch-to-diameter ratio $a = 1.29$ can be characterised as homogeneously mixed.

Lateral mixing occurs due to the Coanda Effect.

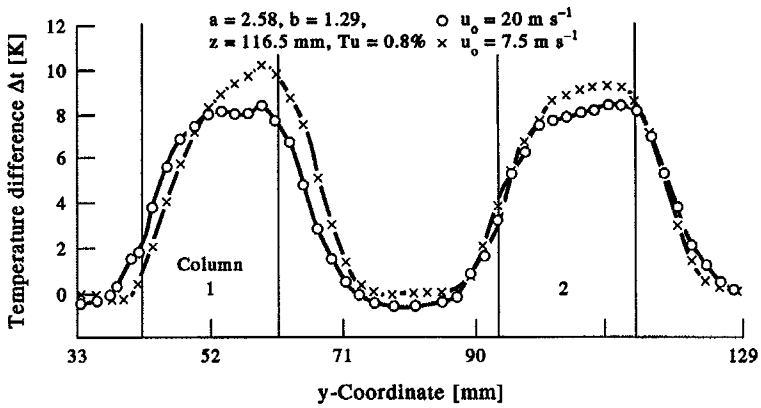

Fig. 9. Temperature differences for various fluid velocities.

Slit jets will be deflected from the main flow direction into the wake of adjacent cylinders. The deflection is caused by fluctuations of the static pressure between the vortex region of adjacent cylinders. As Bearman and Wadcock [21] reported, shear layers behind adjacent cylinders mix before vortices are formed. Ishigai and Nishikawa [10] found an increasing deflection of slit jets in a tube bundle with two rows in in-line arrangement for a transversal pitch-to-diameter ratio $a = 1.4$. For a transversal pitch-to-diameter ratio $a = 1.2$ they observed a deflection of flow by the Schlieren method. Karman vortices did not appear and the velocity distribution became distorted. Aiba *et al.* [22] studied velocities and turbulence intensities between the first and second tube row of an in-line tube bundle with transversal and longitudinal pitch-to-diameter ratio $a = b = 1.2$ in the range of Reynolds numbers $10\,000 \leqslant Re_e \leqslant 60\,000$. They found fluctuating interferences of separated shear layers leading to instabilities. The flow was deflected and increasingly mixed. At Reynolds number $Re_e = 41\,000$, the flow inside the tube bundle was totally deflected. The experiments shown in Fig. 8 confirm these effects.

Figure 9 presents temperatures differences for a heat exchanger with a transversal pitch-to-diameter ratio $a = 2.58$, a flow velocity $u_0 = 20$ m s^{-1} represented by circles and a flow velocity $u_0 = 7.5$ m s^{-1} represented by crosses. For $u_0 = 7.5$ m s^{-1} the maximum temperature difference behind the tubes amounts approximately to 10 K and for $u_0 = 20$ m s^{-1} to 8 K. This can be explained by the following considerations.

The heat flux:

$$\dot{Q} = \dot{M}_K \Delta h_V, \tag{12}$$

inside the tubes equals the heat flux absorbed into the air:

$$\dot{Q} = \dot{M}_L c_{pL} \Delta T_m, \tag{13}$$

where the air flow is given by:

$$\dot{M}_L = \rho_L A_{ch} u_o. \tag{14}$$

From equations (12)–(14) one obtains:

$$\dot{Q} = \rho_L A_{ch} u_o c_{pL} \Delta T_m = \dot{M}_K \Delta h_V, \tag{15}$$

or

$$\frac{\rho_L A_{ch} c_{pL}}{\Delta h_V} = \frac{\dot{M}_K}{\Delta T_m u_o} = \text{constant}. \tag{16}$$

Hence, we obtain:

$$\Delta T_m \approx \frac{\dot{M}_K}{u_o}, \tag{17}$$

and for the ratio of temperature differences:

$$\frac{\Delta T_{m1}}{\Delta T_{m2}} = \frac{\dot{M}_{K1} u_{o2}}{\dot{M}_{K2} u_{o1}}. \tag{18}$$

With the respective flow velocities given in Table 2, the temperature ratio is:

$$\frac{\Delta T_{m,7.5\,m\,s^{-1}}}{\Delta T_{m,20\,m\,s^{-1}}} = \frac{0.55\,g\,s^{-1} \cdot 20\,m\,s^{-1}}{1.16\,g\,s^{-1} \cdot 7.5\,m\,s^{-1}} = 1.26. \tag{19}$$

The temperature ratio read from Fig. 9 is:

$$\frac{\Delta T_{m,7.5\,m\,s^{-1}}}{\Delta T_{m,20\,m\,s^{-1}}} = \frac{10\,K}{8\,K} = 1.25, \tag{20}$$

and is in good agreement with the result from equation (19). For a heat exchanger with transversal pitch-to-diameter ratio $a = 1.29$, flow velocities $u_o = 5$ m s^{-1} and $u_o = 10$ m s^{-1} one obtains:

$$\frac{\Delta T_{m,7.5\,m\,s^{-1}}}{\Delta T_{m,20\,m\,s^{-1}}} = \frac{1.95\,g\,s^{-1} \cdot 10\,m\,s^{-1}}{3.12\,g\,s^{-1} \cdot 5\,m\,s^{-1}} = 1.25, \tag{21}$$

and from Fig. 8 one reads:

$$\frac{\Delta T_{m,7.5\,m\,s^{-1}}}{\Delta T_{m,20\,m\,s^{-1}}} = \frac{14\,K}{11.5\,K} = 1.22, \tag{22}$$

which is also in good agreement. It indicates that an increase of flow velocities has no significant influence on the temperature difference distribution and thus on the mixing effect. An increase of the turbulence intensity of the oncoming air stream leads to a significant increase of heat transfer mainly in the first

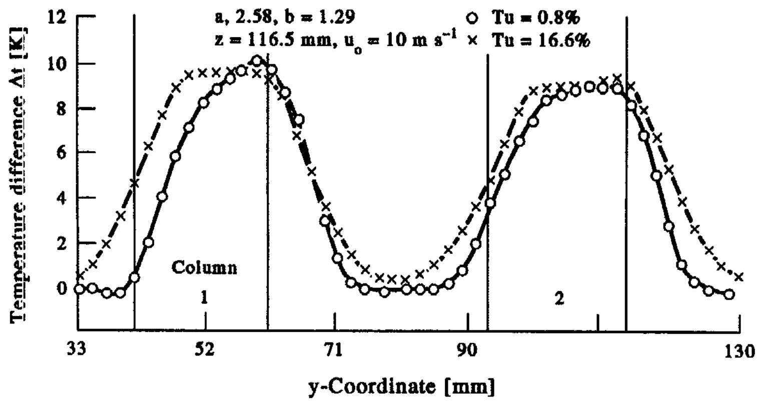

Fig. 10. Temperature differences for different turbulence intensities.

row. The pressure drop increases only slightly as shown in previous papers [4–9, 23, 24].

Figure 10 presents temperature differences behind a tube bundle with a transversal pitch-to-diameter ratio $a = 2.58$, a flow velocity $u_0 = 7.5$ m s^{-1} for a distance $z = 116.5$ mm and for turbulence intensities $Tu = 0.8\%$ and $Tu = 16.6\%$. The region with the maximum temperature difference extends over a wider range for $Tu = 16.6\%$ than for $Tu = 0.8\%$. A maximum temperature difference of 10 K that extends over a wide range y is observed for a turbulence intensity $Tu = 16.6\%$, whereas for $Tu = 0.8\%$ a maximum of 10.2 K is observed over a smaller range y. For a turbulence intensity $Tu = 16.6\%$, the temperature difference between the tube is always positive, whereas for a turbulence intensity $Tu = 0.8\%$ temperature differences are negligible. High turbulence intensities cause an intensive temperature exchange between the heated and the cold air. The lateral mixing between the tube columns is however dampened due to the small longitudinal pitch-to-diameter ratio, $b = 1.29$. This leads to channel flow and thus to an unmixed flow behind the third tube now.

6. CONCLUSIONS

Depending on the transversal and longitudinal pitch-to-diameter ratio a mixed, unmixed or partial mixed flow appears behind a tube bundle in in-line arrangement. Behind a tube bundle with three rows of plain tubes and a transversal and longitudinal pitch-to-diameter ratio $a = 2.58$, $b = 1.29$ regions between the tubes without noticeable temperature increase and regions behind the tubes with a temperature increase can be observed. The maximum temperature difference in the outlet cross-section amounts in the experiments to approximately 9 K. Due to the small longitudinal pitch-to-diameter ratio, $b = 1.29$, the cold air between the tube columns flows without heat exchange as channel flow. Lateral mixing between heated and unheated layers does not occur. An increase of the Reynolds number from $Re_c = 15\,000$ up to approximately $Re_c = 42\,000$, or an increase of the turbulence intensity of the oncoming flow from $Tu = 0.8\%$ to $Tu = 16.6\%$, did not recently affect the temperature distribution behind the bundle, because the flow was mainly influenced by the longitudinal and transversal pitch-to-diameter ratio and could be characterised as unmixed.

The flow behind a bundle with small transversal and longitudinal pitch-to-diameter ratios $a = b = 1.29$ is approximately homogeneously mixed. Lateral mixing between heated and unheated layers leads to a homogeneous temperature distribution in the outlet cross-section. A decrease of the Reynolds number of $Re_c = 28\,000$ caused no significant change in the temperature distribution and the flow pattern. An increase of the turbulence intensity of the on-coming flow enhances the lateral mixing. The outlet temperature distribution, however, is mainly determined by the strong deflection of the flow, because the small transversal and longitudinal pitch-to-diameter ratio is decisive for the temperature distribution.

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