Heat transfer in packet of parallel plates in the short-term processes

Yu A Kirsanov1*, D V Makarushkin 1, A E Yudakhin 1, A Yu Kirsanov 2

1 Institute of Energy and Advanced Technologies - structural unit FRC «Kazan Scientific Center RAS»
2 FSBEI HE «KNRTU–KAI», Kazan

E-mail: *kirsanov-yury@mail.ru

Abstract. The description of the experimental stand (laboratory regenerative air preheater), the method for measuring the unsteady flow temperature and the results of experimental heat transfer studies of a package of plates (nozzles) of alloy steel AISI-430 with a 2 mm thickness for short-time periods with different durations are presented. The experimentally obtained mean Nusselt numbers are generalized by the criterial equation.

1. Introduction
In many power plants thermal processes take place during a limited period of time. Such power plants include regenerative air preheaters (RAPH) used in steam generators, air cooling and air separation machines, gas turbine engines, metallurgical furnaces, ventilation and heating systems [1-3].

Due to a cyclic heat exchange of the nozzle with a cold and hot heat carrier the heat transfer process in each cycle period is unsteady. It is known that the unsteady nature of external conditions qualitatively and quantitatively affects on local heat transfer [3-7]. Therefore, the heat transfer coefficient of the nozzle averaged over the period, which is important for the calculation and design of the RAPH, becomes dependent, among other factors, on the duration of the periods, that is, on the frequency of regeneration [3, 4]. However, this dependence is not fully understood and in most cases is not taken into account by the developers of industrial RAPH [1, 2].

The work elaborates on the description of the test bench, experimental procedure and data processing in measurements of the heat transfer coefficient of the plate pack surface in the laboratory RAPH. We also demonstrate the effect of the regeneration cycle duration on the heat transfer coefficient averaged over the period, obtaining the criterion dependence of the Nusselt number on the duration of the period and the flow regime of the heat transfer agent for a specific packing.

2. Stand with a laboratory RAPH
The stand consists of air blowers, air ducts with flowmeters, a heater on one of the air ducts, a working area, a moving unit, an automatic control system (ACS) and an automated measuring system (AMS). In the working area, the body of which is molded from fiberglass and having a cavity of a square section of $50 \times 50$ mm, the test nozzle is located. Thermocouples are installed in the inlet and outlet of the working area to measure the non-stationary temperatures of air flows at the inlet and outlet of the RAPH. In addition, in front of the working site there are standard thermometers. The air flow was...
measured with the help of narrowing devices with low-flow diaphragms [8]. The flow switching, that is, the periodic connection of the RAPH to the ducts of the cold and hot air ducts, is carried out by means of electric drives.

AMS includes a personal computer (PC) and an I/O device from National Instruments PCI-6251, connected to the PCI bus of the PC [9]. To create the AMS software, the LabVIEW graphical programming environment was used (License agreement number: 777455-03 Serial number: M71X16236). The readings from mV to degrees were carried out according to the individual calibration characteristics of thermocouples. AMS registered the following parameters: flow temperatures at both nozzles of the working section, the surface temperature and the center of the nozzle, pressure differences on the flowmeters and the RAPH. Readings of stationary parameters (barometric pressure, temperatures and relative humidity of ambient air, temperatures and pressures before flowmeters, standard thermometers) were taken manually.

The explored packing was a package of parallel plates of alloy steel AISI-430 1 mm thick and 100 mm long. The plates are welded in pairs by contact welding, forming 13 plates 2 mm thick, set at a distance of 2 mm from each other. Thermocouples are welded between thin plates and on the surface of one of them to measure the temperatures of its central point and surface. Thermocouples from chromel-kopel wires with a diameter of 0.2 mm are welded by a contact welding machine.

3. Method for measuring the heat transfer coefficient of the nozzle of the RAPH

The heat transfer coefficient of the nozzle surface in a separate period is given by the formula

$$\alpha = \frac{Q}{F_w(T_f - T_w)}$$

where $\alpha$ is the heat transfer coefficient, W / (m² K); $Q$ is the heat load transferred through the nozzle from the hot heat carrier to the cold, W; $F_w$ is the complete surface of the nozzle, m²; $\overline{T_f} = (T_f' - T_f'')/2$ is the average temperature of heat carrier, °C; $T_f'$ and $T_f''$ are the temperatures of the heat carrier at the inlet and outlet of the working section in the period under review, °C; $\overline{T_w}$ is the average surface temperature of the nozzle, °C.

The thermal load can be represented as the thermal power perceived or transmitted by the nozzle at the current time

$$Q_w = M_w c_w \frac{dT_w}{d\tau}.$$  

Here $M_w$ is the mass of the nozzle, kg; $c_w$ is the specific heat of the packing, J / (kg K); $T_w$ is the average nozzle temperature, K.

The stand allows to measure all the quantities entering into formulas (1) - (2). In figure 1 lines 1-4 show the AMS recorded changes over the cycle of nozzle and air temperatures in two experiments, the duration of the working periods of which was 2.6 and 40 seconds. The cycle is conditionally divided into four calculation periods. In the experiment with $\tau_p = 2.6$ s, the calculated periods I and III coincide with the working periods. The duration of the first two calculation periods coincides with the duration of the last two. Between the second and third periods there is a period of time $\Delta\tau_{per}$ during which there is a switch from the cold period to the hot one. A similar time interval exists after the calculation period IV (not shown in the figure).

The most abrupt changes in time of air flows temperatures are observed in the first and third periods, that is, immediately after the flows switching. Obviously, when changing the flows, surrounding individual thermocouples, their readings due to inertia are different from the actual temperatures of the flows $T_f'$ and $T_f''$. In the second and fourth calculation periods, the rate of change
of the flow temperatures is small or equal to zero, and therefore $T_f$ and $T_f'$ are accepted to the indications of the corresponding thermocouples.

3.1. Calculation of the actual values of rapidly varying flow temperatures

The works [10-13] are devoted to the method of measuring non-stationary flow temperatures.

The equation of thermal conductivity for a thermocouple junction in relative variables has the form [11]:

$$dθ/dt + θ/t_*= θ_f/t_*,$$

where $θ = (T - T_*)/T_*$; $t = τ/τ_p$; $T$ is temperature of the junction, $K$; $T_*$ is temperature of the heat carrier at the end of the transition period, $K$; $T_*$ is temperature scale, $K$; $T_{max}$ and $T_{min}$ are the maximum and minimum temperatures, respectively, of the hot and cold flows, $K$; $τ$ is time from the beginning of the period, $s$; $τ_p$ is duration of the transition period, $s$; $τ_p = ρcV/(αF)$ is characteristic thermocouple time, $s$; $ρ$ is density, kg/m$^3$; $c$ is the heat capacity, J/(kg K); $F$ and $V$ are the surface area, m$^2$, and volume, m$^3$, of the junction.

The general solution of the ordinary differential equation (3) [14] is

$$θ(t) = θ_0 \exp(-t/t_*) + \frac{1}{t_*} \int_0^t θ_f(η) \exp\left(\frac{η-t}{t_*}\right) dη.\quad(4)$$

Here $θ_f(t)$ is the true dependence of the flow temperature on time, in which we use the exponential function

$$θ_f(t) = \exp(a_0 + a_1t).\quad(5)$$

The required coefficients $a_0$ and $a_1$ are found from the comparison of solution (4) with the regression equation that approximates the thermocouple readings $θ_p(t)$

$$θ_p(t) = \sum_{i=0}^k b_i t^i,\quad(6)$$

where $b_i$ is the regression coefficients.
3.2. Determination of the coefficients of the exponent (5)

Integration of equation (4) after substitution of the function (5) gives an expression for the calculated thermocouple temperature:

\[ \theta(t) = \theta_0 \exp(-t/t_*) + \exp(a_0) [\exp(a_1 t) - \exp(-t/t_*)]/(1 + a_1 t_*) . \]  

(7)

To find the unknowns \(a_0\) and \(a_1\), two conditions are set:

\[ \frac{d\theta(t)}{dt} = \frac{d\theta_p(t)}{dt} , \]  

(8)

\[ \int_0^1 \theta(t) dt = \int_0^1 \theta_p(t) dt , \]  

(9)

from which we obtain the equations

\[ a_0 = \ln(\theta_0 + b_1 t_*) , \]  

(10)

\[ \theta_0 t_* [1 - \exp(-1/t_*)] + \exp(a_0) [\exp(a_1) -1]/a_1 - t_* [1 - \exp(-1/t_*)] = \overline{\theta} . \]  

(11)

The solution of these equations allows to find the unknown coefficients \(a_0\) and \(a_1\). Here \(\overline{\theta}\) is the average temperature of the flow during the period

3.3. Results of correction of thermocouple readings and their analysis

The coefficient of heat transfer of the thermocouple junction, necessary for calculating the characteristic time of the thermocouple \(\tau_*\), was calculated from the formulas [15]:

\[ \alpha = Nu / \lambda_f , \]

where \(Nu = Nu_{i,min} + \sqrt{Nu_{lam}^2 + Nu_{sur}^2} ; Nu_{lam} = 0.664 \ Re_i^{0.5} Pr_i^{1/3} ; Pr = \nu_f / \alpha_f \) ; \( Re_f = w_f l / \nu_f \)

\[ Nu_{i,min} = \begin{cases} 0.3 \text{ for the cylinder,} \\ 2 \text{ for the sphere;} \end{cases} \]

\[ Nu_{lam} = 0.337 \ Re_i^{0.8} Pr_i^{1/3} / (1 + 2.443 \ Re_i^{0.1} (Pr_i^{2/3} - 1)) ; \]

\[ l = \begin{cases} d_i \pi/2 \text{ for the cylinder,} \\ d_{ic} \text{ for the sphere;} \end{cases} \]

When determining the characteristic time of thermocouples \(\tau_*\), the thermophysical properties of their junctions were calculated on the assumption of the equality of volume fractions of metals [16-18]: \(\rho = 8800 \ \text{kg/m}^3; \ c = 424 \ \text{J/(kg K)}\).

The evaluation of the effect of radiation and heat removal on thermocouple wires on the readings of thermocouples measuring air flow temperatures under experimental conditions showed that they are small (about 1% and 0.5 K, respectively) and were not taken into account in the calculations.

Calculated temperatures of heat carriers and nozzles after adjusting the thermocouple readings at the inlet and outlet of the working section are determined by the expression

\[ \theta_f = \theta_\infty \pm \exp(a_0 + a_1 t) + \sum_{l=0}^k c_l t^l \]

and are shown in figure 2. Here \(\theta_\infty\) is the temperature limit at the end of I or III of the calculation period; the sign "+" before the exponent corresponds to the cold period, the sign "-", to the hot period; the polynomial on the right-hand side determines the calculated temperature in the II or IV calculation period.
3.4. Measurement of the plate heat transfer coefficient

The calculated current values of the temperature of the heat carriers and nozzle, shown in figure 2, enable to calculate the heat transfer coefficient of the plate package by the formulas (1) and (2). The results of calculating the heat transfer coefficient for both experiments are given in figure 5 as a function of the Nusselt number $\text{Nu} = \frac{\alpha d_e}{\lambda_f}$ from time. Here $d_e$ is the equivalent diameter of the channels between the plates.

The character of the dependence of the heat transfer coefficient on time, shown in figure 3, qualitatively agrees with the classical concepts [5] explaining such a behavior of the heat transfer coefficient by increasing the thickness of the boundary layer in time tending to the steady-state value. A similar character of the change in the heat transfer of a plate when the temperature of the coolant changes abruptly is confirmed by theoretical studies [6].

Figure 5 also shows the values of the Nusselt number recommended for stationary heat exchange conditions S.S. Kutateladze ($A_T = 1.85$ [19]) and A.A. Zhukauskas ($A_T = 1.38$ [20]) for calculating the average over the length of the channel values of the heat transfer coefficient for a laminar regime$^1$.

---

$^1$ The correction recommended in the literature for non-isothermal flow is omitted in view of the weak dependence of the Prandtl number on air on temperature.
where \( d \approx 2\delta \) is the equivalent diameter of the channel, m; \( \delta \) is distance between plates, m; \( C_f \approx 1.904(d/l)^{0.173} \) is correction for the length of the channel [3].

From figure 3 it follows that calculations using equation (12) give mean values of average heat transfer coefficients in the experiment with a long period and by 35...38% in the short range, which are understated by 14...15%.

3.5. Measurement of the average over the period of heat transfer

The average value of the heat transfer coefficient for the working period was calculated using formulas (1) and (2), in which the values of \( \overline{T_w} \) and \( \overline{T_f} \) were calculated from the regression equations used for approximation of direct measurements of the corresponding temperatures.

The results are shown in figure 4 as a dependency \( \text{Nu}_{cp} = f(\text{Re}) \). One can notice the stratification of points and regression lines by duration of periods: shorter periods correspond to more intensive heat transfer of plates. Moreover, the points and regression lines are located much higher than the lines described by the criterial equation (12).

The points shown in figure 4 are generalized by the regression equation

\[
\frac{\text{Nu}_{cp}}{\text{Nu}_{st}} = 1 + 1.2 \, \text{Ho}^{-0.2} \left( \frac{\text{Re}}{1000} \right)^{0.58},
\]  

shown in figure 5 by the dotted line. The correlation coefficient of the line and points was 80%. The limits of applicability of equation (13): \( 139 < \text{Ho} \equiv \rho \overline{w} \overline{T} \tau_p / l < 11750; \ 1.04 < \text{Re} \cdot 10^{-3} < 5 \). The determining temperature was the average temperature of the heat carrier over the period.

The presented results confirm the conclusions made in the recent studies [3, 4] that the duration of periods significantly affects the heat transfer coefficients of the plate-shaped packing of the regenerative air heater.
4. Summary

1. A laboratory stand with automated control and measurement systems of air flow parameters in a regenerative air preheater has been developed.
2. With reference to the experimental conditions, a method was developed for measuring the non-stationary temperatures of air flows, taking into account the inertia of the thermocouples.
3. Dependences of the current values of Nusselt numbers in relatively short periods are obtained, confirming the results of known theoretical and experimental studies.
4. The period-averaged Nusselt numbers for short periods of time have been measured and generalized by the criterial equation, convenient for subsequent use in engineering practice.

Acknowledgments
The work was carried out under the Treaty on Creative Cooperation between the FSBEI HE «KNRTU–KAI» and FRC KazRC RAS № 7 of 28.01.2015.

References
[1] Hausen H 1976 Wärmeübertragung im Gegenstrom, Gleichstrom und Kreuzstrom (Springer) p 432
[2] Dobryakov T S, Migaj V K et al: Air Heaters of Boiler Plants (Leningrad.Energy) p 184 [in Russian]
[3] Kirsanov Yu A 2007 Cyclic Thermal Processes and the Theory of Thermal Conductivity in Regenerative Air Heaters (Moscow: Fizmatlit) p 240
[4] Kirsanov Yu A 2003 Izv.VUZov. Aviacionnaya Tehnika I pp 31-34
[5] Koshkin V K, Kalinin E K et al 1973 Nonstationary Heat Transfer (Moscow: Mashinostroenie) p 328 [in Russian]
[6] Vilenskiy V D 1974 Teplofizika Vysokih Temperatur 12(5) 1091–1104
[7] Padet J 2005 J. Braz. Soc. Mech. Sci. & Eng. 27 74-95
[8] Kremlevsky P P 1975 Flow Meters and Quantity Counters (Leningrad: Mashinostroenie) p 776 [in Russian]
[9] Kirsanov Y A, Yudakhin A E, Kirsanov A Y 2017 High Temperature 55(1) pp 114-119
[10] Yaryshev N A 1990 Theoretical Basis of Non-Stationary Measurement of Temperature (Leningrad: Energoatomizdat) p 256 [In Russian]
[11] Ahtmann M, von Wolfersdorf J, Meyer G 2015 ASME. J. Turbomach 137(12)
[12] Bernhard F 2004 Technische Temperaturmessung (Springer-Verlag, Berlin) p 1460
[13] Garnier B, Lanzetta F 2015 In situ realization/characterization of temperature and heat flux sensors. Advanced Spring School «Thermal Measurements & Inverse techniques», Domaine de Françon, Biarrtiz
[14] Kamke E 1971 Differentialgleichungen Lösungsmethoden und Lösungen (Springer Fachmedien Wiesbaden) p 246
[15] Reference book on heat exchangers. In 2 vol. 1987 ed V K Petukhov, V K Shikov (Moscow: Energoatomizdat) p 560
[16] Babichev A P, Babushkina N A, Bratkovskiy A M et al. 1991 Physical Quantities. Reference Book (Moscow: Energoatomizdat) p 1232
[17] Tables of Physical Quantities. Reference book 1976 (Moscow: Atomizdat) p 1008.
[18] Nikonov N V 2015 Thermocouples. Types, Characteristics, Designs, Production (Moscow: OOO “MTK-Metotehnika”) p 62
[19] Kutateladze S S 1990 Heat Transfer and Hydrodynamic Resistance (Moscow: Energoatomizdat) p 367
[20] Zukauskas A A 1982 Convective Transfer in Heat Exchangers (Moscow: Nauka) p. 472