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CHARACTERISTICS OF HEAT FLOW IN RECUPERATIVE HEAT EXCHANGERS

A simpified model of heat flow in cross–flow tube recuperative heat exchangers (recuperators) was presented in this paper. One of the purposes of this investigation was to analyze changes in the values of some parameters of heat transfer in recuperators during combustion air preheating. The logarithmic mean temperature (Δt_m) and overall heat transfer coefficient (U), are two basic parameters of heat flow, while the total heated area surface (A) is assumed to be constant. The results, presented as graphs and in the form of mathematical expressions, were obtained by analytical methods and using experimental data. The conditions of gaseous fuel combustions were defined by the heat value of gaseous fuel Q_d = 9263.894 J·m^{-3}, excess air ratio λ = 1.10, content of oxygen in combustion air $\nu(O_2)$ = 26%$_{vol.}$, the preheating temperature of combustion air (cold fluid outlet temperature) t_{co} = 100–500°C, the inlet temperature of combustion products (hot fluid inlet temperature) t_{hi} = 600–1100°C.

Fuel enters significantly into manufacturing costs, and in some industries represents one of the largest expenses. Inprovements in the overall energy efficiency of an industrial plant, reflected in increased values of the output/input ratio, are primarily achieved by reducing not only the energy input values, but also the energy losses at the output side of a process step [1–4]. W ith nearly all industrial processes consuming primary energy, a considerable part of heat is lost as waste gas. The major energy loss from industrial furnaces is *via* the heat of waste gas, the temperature of which generaly exceeds the interval of 600–1400°C. The recuperative utilization of waste gas heat for the process, from which the flue gas heat is obtained, is one of the most effective methods of energy recovery. Due to the reduction of the fuel quantity, less waste gas reaches the atmosphere, whereby the efficiency is increased. By installing a recuperator in the waste gas flue to preheat the combustion air, however, the flame temperature is increased, the heat transfer efficiency improved and the overall fuel consumption reduced [5–7].

The total losses of heat in waste flue gases can be minimized by providing the proper amount of air for combustion. The amount of waste flue gases can be minimized and the heating rate of the unit can be increased by the oxygen enrichment of the combustion air. With the same fuel input, enriched air for combustion increases the flame temperature of a given fuel, thereby inproving the heat–transfer rate and increasing production: Alternatively, the fuel input may be decreased when enriched air is used to maintain the same production rate as obtained with fuel using ordinary air. Increased production rates almost always reduce the heat losses per unit of product in any high temperature furnace [3,5,8–10].

The combustion behaviour is significantly affected by the temperature of the reactants, the amount of gas recirculation and the temperature of the flue gases. High temperature air combustion technology has had a significant impact on the design and development of advanced industrial furnaces for energy savings. One of the major contributions to fuel economy in metallurgical plants has been the recovery of waste heat by different types of heat exchangers [1,2,8].

THEORETICAL FORMULATION

There are many reasons for interest in the thermal process in recuperators. The literature already contains a number of thermal analyses of complex processes of heat transfer phenomena during air preheating. In order to facilitate modelling it is necessary to define the main parameters of the process and to quantify the variables that determine the thermal conditions during heat exchanger operation [3–8]. Two basic parameters used to calculate the performance of a given heat exchanger are the overall heat transfer coefficient (U) and the logarithmic mean temperature value (Δt_m). The overall heat transfer coefficient depends on the geometric configuration of the wall separating two fluids and the convective heat transfer coefficient on each side. The overall coefficient for a certain geometry can be defined as the quantity which yields the total rate of local heat transfer between the cold and hot fluids [1,3,4]. For a cross–flow exchanger, in any incremental portion where A is the heat exchanger total surface area, in which an increment of transfer surface area (dA) is exposed between the two fluids, the incremental heat transfered is

$$dq = U\,dA\,\Delta t \qquad (1)$$

where Δt is the temperature difference between the two fluids in the increment dA, and U is the overal heat trans-

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fer coefficient (based on the area being used) between the two fluids. After integration of Eq. (2), the total heat transfer in the entire heat exchanger is given in the form

$$q = (U\,\Delta t)_m\,A, \tag{2}$$

For several cases it may be assumed that an appropriate average value for U can be found that applies throughout the exchanger, so parameter Δt_m can be defined as the mean temperature difference. Eq. (2) can be written as

$$q = U\,\Delta t_m\,A \tag{3}$$

Equations (2) and (3) imply the existence of a mean temperature difference (Δt_m), defined as

$$\Delta t_m = \int_0^A (\Delta t)\,dA \tag{4}$$

It has been determined that the desired temperature difference can be expresed as the logarithmic mean temperature difference, i.e. the mean temperature differences at the two ends of the heat exchanger. The four fluid temperatures, which are presumed known, are: hot fluid inlet (t_{hi}), and hot fluid outlet temperature (t_{ho}), cold fluid inlet (t_{ci}) and cold fluid outlet temperature (t_{co}). In terms of these four fluid terminal temperatures, the logarithmic mean temperature difference (Δt_m), for the counterflow case, is:

$$\Delta t_m = \frac{(t_{hi} - t_{co}) - (t_{h0} - t_{ci})}{\ln \dfrac{(t_{hi} - t_{co})}{(t_{ho} - t_{ci})}} \tag{5}$$

Finally, the overall heat transfer coefficient can be calculated from Eq. 3, as

$$U = \frac{q}{\Delta t_m A} \tag{6}$$

The heat transfer area can be minimized by maximizing the overall heat transfer coefficient and the logarithmic mean temperature difference for a given heat transfer.

RESULTS AND DISCUSSION

The results presented in this paper were obtained by combining an experimental technique with an analytical method of combustion calculation of a chosen type of gaseous fuel. They are related to the work of a special type of continuous furnaces for steel billet heating prior to hot rolling. The furnace is fired with gaseous fuel and the air for combustion is preheated and enriched (temperature above 20°C; the air contains more than 21% O_2).

The working conditions of fuel combustion were determined by:

– coefficient of air excess (upper value in the interval 1.05–1.10), $\lambda = 1.10$

– air enrichment rate (content of oxygen in combustion air), $\nu(O_2)_{co} = 26.0\%_{vol}$

– inlet temperature of the hot fluid (combustion products), $t_{hi} = 600–1100°C$

– the outlet temperature of the hot fluid, t_{ho}, was calculated

– inlet temperature of the cold fluid (cold combustion air), $t_{ci} = 20°C$

– outlet temperature of the cold fluid (preheated air), $t_{co} = 100–500°C$.

The values of λ, $\nu(O_2)$ and t_{hi} were chosen according to the required thermal conditions and temperature profile in the furnaces for lowalloy steel billet heating.

The chemical composition and heating value of the fuel are given in Table 1.

Table 1. Chemical composition of gaseous fuel

Analysis, vol %							
H_2	CO	CH_4	C_2H_4	CO_2	O_2	H_2O	N_2
27.47	15.44	10.19	1.15	3.99	2.48	1.65	37.63

Lower heating value of the fuel: 9263.894 $J \cdot m^{-3}$

The results of the mathematical calculation of heat transfer in the recuperator are available in the form of graphs, as well as in table form. All the presented graphs can be classified in the following way:

– changes of the amount of heat transfered to the combustion air (combustion air enthalphy, heat content of air), q_{ho}, depend on the air preheating temperature, the coefficient of excess air, the air enrichment rate, Fig. 1;

– the overall heat transfer coefficient, Figs. 2 and 3;

– the mean logarithmic temperature difference, Fig. 4 and 5;

The derived expressions, for the calculation of U and the Δt_m mean values, are available in Table 2.

The amount of heat transfered to the combustion air (heat content of air) can be defined as a funcion of

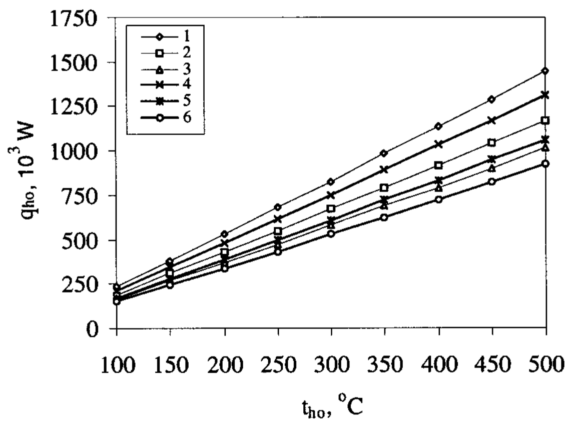

Figure 1. Variation in the enthalpy of preheated air with air temperature, for different values of the hot fluid inlet temperature: 1 – 600°C; 2 – 700°C; 3 – 800°C; 4 – 900°C; 5 – 1000°C; 6 – 1100°C

the air preheating temperature and other conditions of fuel combustion. The dependance is shown in Fig. 1 for two values of the excess air ratio and for three values of the oxygen content in the combustion air. The elements of the recuperative heat exchanger, used for preheated air were plane tubes (the separating wall had a cylindrical geometry without fin arrays to enhance heat exchange).

The values of the air preheating temperature range from 100 to 500°C. Curves 1, 2 and 3 show the conditions of combustion for λ = 1.10, while curves 4, 5 and 6 describe the variation in the values of the heat content in air for λ = 1.0. Curves 1 and 4 are related for $\nu(O_2)$ = 21%, curves 2 and 5 to $\nu(O_2)$ = 26%, while curves 3 and 6 to $\nu(O_2)$ = 30%. On the basis of the shape of the curves in Fig. 1 it may be concluded that with increasing air temperature the values of the heat content in air also increase in the total investigated range of λ and $\nu(O_2)$ values.

During fuel combustion at lower values of λ, less air supplied for combustion is needed, therefore the amount of transferred heat is also reduced. By increasing content of oxygen in the combustion air, the amount of air per unit of fuel is decreased as a result of which the amount of transferred heat also decreases.

In the analysis, according to the obtained results, the mean logarithmic temperature difference (Δt_m) can be considered depend on the conditions of the heat exchange process, i.e. on the inlet and outlet temperature of the combustion products. Figure 2 describes Δt_m as a function of the inlet temperature of the combustion products, while the air temperature varies in the interval 100–500°C. The dependence of this parameter on the air preheating temperature, for the analysed range of inlet temperatures of the combustion products is shown in Figure 3.

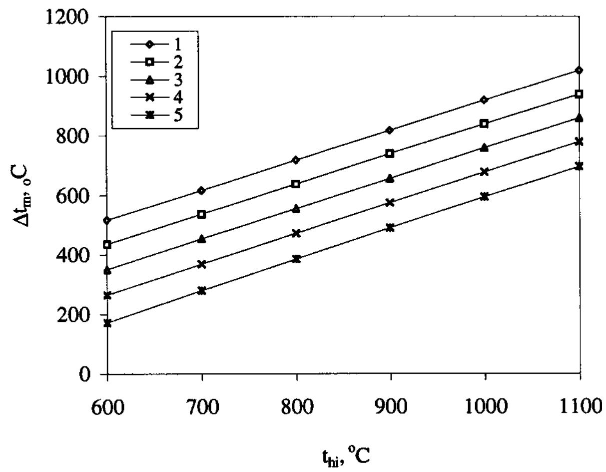

Figure 2. Changes of the logarithmic mean temperature difference as a function of the inlet temperature of the hot fluid, for different values of the air preheating temperature: 1 – 100°C; 2 – 200°C; 3 – 300°C; 4 – 400°C; 5 – 500°C

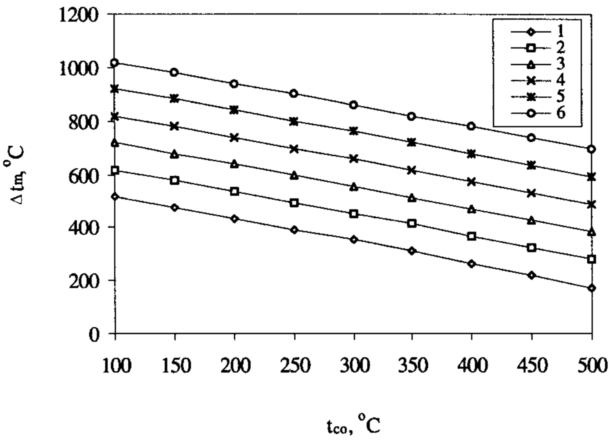

Figure 3. Changes in the logarithmic mean temperature difference as a function of the air preheating temperature, for diferent values of the inlet temperature of the hot fluid: 1 – 600°C; 2 – 700°C; 3 – 800°C; 4 – 900°C; 5 – 1000°C; 6 – 1100°C

According to the given data it may be concluded that with increasing air temperature, the mean logarithmic temperature difference decreases, for the total range of inlet temperatures of the combustion products, while by increasing the inlet temperature of the combustion products, the mean logarithmic temperature difference is increased for the whole interval of air temperature values.

The overall heat transfer coefficient (U) was calculated as the sum of various resistances to heat transfer that might be encountered. Its basic form is:

$$1/U = 1/h_h + Rf_h + \delta/k + Rf_c + 1/h_c \qquad (4)$$

where:

h_h and h_c – are the individual (local) heat transfer coefficients (measure of transferring heat from the hot fluid to the wall and from the wall to the cold fluid), $W \cdot m^{-2} \cdot {}^{o}C^{-1}$

Rf – the fouling resistance, measured as the thickness of the fouling layer divided by the thermal conductivity of the fouling material, $W^{-1} \cdot m^2 \cdot {}^{o}C$

δ/k – the resistance of the heat transfer surface, measured as the wall thickness divided by the thermal conductivity of the wall material, $W^{-1} \cdot m^2 \cdot {}^{o}C$

k – thermal conductivity, $W \cdot m^{-1} \cdot {}^{o}C^{-1}$

The parameters which can affect the local heat transfer coefficients, the resistance of heat transfer and the overall heat transfer coefficient are: the coefficient of air excess, the air enrichment rate and the air preheating temperature. Thus, the changes in the overall heat transfer coefficient, described as a function of the air preheating temperature and the inlet temperature of the combustion products are presented in Figs. 4 and 5.

The rate ot air preheating temperature increase, as well as the increase of the amount of transferred heat in the recuperator are directly proportional to an increase in the overall heat transfer coefficient values. It was ob-

*Table 2. Derived expressions for determining the mean values of U and Δt_m**

Parameter (y)	Expression derived on the basis of the model	Independent variable (x)	Expression valid for the mean value of
$\Delta t_m = f(t_{co})$	$y = -1\cdot10^{-4}x^2 - 0.76\,5\,x + 843.43$ $R^2 = 1$	$t_{co} = 100 - 500^oC$	$t_{hi(m)}$
$\Delta t_m = f(t_{hi})$	$y = -2\cdot10^{-5}x^2 + 1.0504\,x - 275.71$ $R^2 = 1$	$t_{hi} = 600–1100^oC$	$t_{co(m)}$
$U = f(t_{co})$	$y = 2\cdot10^{-3}x^2 - 0.0264x + 4.8173$ $R^2 = 0.9934$	$t_{co} = 100–500^oC$	$t_{hi(m)}$
$U = f(t_{hi})$	$y = -2\cdot10^{-7}x^3 + 5\cdot10^{-3}x^2 - 0.547\,x + 209.1$ $R^2 = 0.9978$	$t_{hi} = 600–1100^oC$	$t_{co(m)}$

*All the expressions in Table 2 can be used for the conditions of fuel combustion described by $v(O_2) = 26\%$, and $\lambda = 1\,.1\,0$.

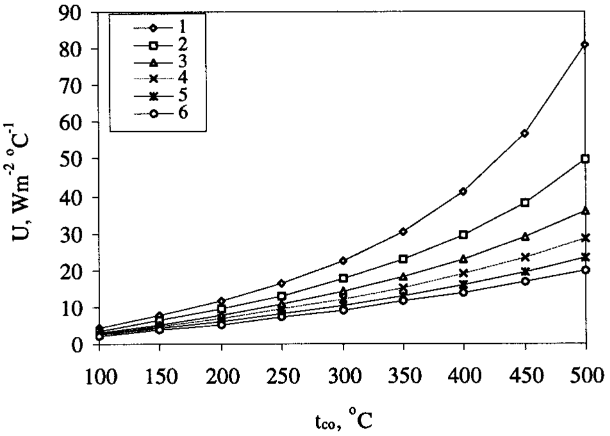

Figure 4. Overall heat transfer coefficient as a function of the air preheating temperature for different inlet temperatures of the hot fluid: 1 – 600°C; 2 – 700°C; 3 – 800°C; 4 – 900°C; 5 – 1000°C; 6 – 1100°C

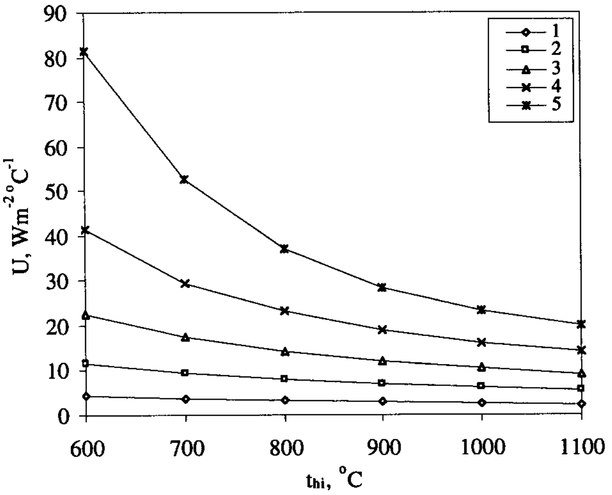

Figure 5. Overall heat transfer coefficient as a function of the inlet temperature of the hot fluid, for different values of the air preheating temperature: 1 – 100°C; 2 – 200°C; 3 – 300°C; 4 – 400°C; 5 – 500°C

served that the overall heat transfer coefficient substantially decreased by increasing the inlet temperature of the combustion products for the total range of air preheating temperature values. That can be explained according to Eq. (6) and the statement that parametre U is indirectly proportional to the mean logarithmic temperature difference, when the heat exchanger total surface area in the recuperator is constant (A = const).

CONCLUSIONS

A the more detailed picture of the thermal work of a recuperator can be obtained by the determining the dependance of the main process parameters, i.e. by establishing a functional dependance between the overall heat transfer coefficient, the mean logarithmic temperature difference, the air preheating temperature, the inlet and outlet temperature of the combustion products. It is also necessary to analyze the conditions of fuel combustion in the furnace because the produced combustion products are used as the bearer of heat during recuperator operation.

There are several ways for reaching a greater potential for heat transfer during the operation of a counter–cross flow recuperator. One of them is to maximize the overall heat transfer coefficient. The goal is to minimize the surface area requirement for the given heat transfer rate. If the surface area for the exchange of heat is constant, maximizing of the overall transfer coefficient can be reached by minimizing the value of the logarithmic mean temperature difference. The way to maximize the overall heat transfer coefficient is by maximizing the individual (local) heat transfer coefficients, and by minimizing the local heat transfer resistances. A high temperature of preheated air used for combustion is recommended for fuels with low heating values. With the preheating of combustion air, the working temperature and fuel economy are improved, the heat transfer efficiency increased and the overall fuel consumption can be significantely reduced.

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IZVOD

KARAKTERISTIKE TOPLOTNOG TOKA U REKUPERATIVNIM IZMJENJAČIMA TOPLOTE

(Naučni rad)

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Rekuperacija toplote, tj. predgrijavanje vazduha potrebnog za sagorijevanje goriva, smatra se jednim od najefikasnijih metoda iskorišćenja toplote izlaznih dimnih gasova (produkata sagorijevanja). Povišenjem temperaturnog nivoa vazduha prije procesa sagorijevanja goriva, omogućen je porast temperature sagorijevanja i temperature radnog prostora peći. Osim toga, ubrzava se proces sagorijevanja i poboljšavaju se uslovi toplotnog rada peći. Kod kontinuiranih peći za zagrijavanje čeličnih poluproizvoda prije plastične prerade postupkom toplog valjanja, vazduh se, najčešće, predgrijava u intervalu temperatura 100–600°C.
Obogaćivanjem vazduha potrebnog za sagorijevanje goriva, tj. povećanjem sadržaja kiseonika iznad 21 %vol, raste temperatua plamena i ekonomije goriva, a intenzivirani su procesi toplotne razmjene u rekuperatoru i u radnom prostoru peći.
Zbog značaja rekuperativnih izmjenjivača toplote, kao posebnih uređaja za predgrijavanje vazduha koji se koristi za sagorijevanje goriva u metalurškim pećima, uopšte, u ovom radu je naveden dio rezultata izučavanja osnovnih veličina toplotnog rada procesa rekuperacije toplote i parametara koji utiču na promjenu njihovih vrijednosti.
U radu je data analiza procesa prenosa toplote u rekuperativnom cijevnom izmjenjivaču toplote koji se koristi za predgrijavanje vazduha (hladni fluid), potrebnog za sagorijevanje goriva u kontinuiranoj peći loženoj gasovitim gorivom. Kao fluid koji prenosi toplotu (topli fluid), koriste se izlazni produkti sagorijevanja goriva (iz peći za koju se koristi predgrijavanje vazduha). Rad rekuperatora karakteriše unakrsno–protusmjerna šema strujanja fluida. Parametri procesa toplotnog toka u rekuperatoru su srednja logaritamska temperaturna razlika (Δt_m) i ukupni koeficijent prenosa toplote (U), dok je ukupna površina za razmjenu toplote u rekuperatoru (A) konstantna.
Rezultati, prikazani u radu u grafičkoj i tabelarnoj formi, dobijeni su eksperimentalnim i računskim metodama. Uslovi sagorijevanja goriva u peći definisani su sledećim parametrima: toplotnom moći goriva, Q_d = 9263,894 J·m^{-3}, stepenom obogaćenja vazduha (sadržajem kiseonika u vazduhu potrebnom za sagorijevanje), $\nu(O_2)$ = 26%, koeficijentom viška vazduha, λ = 1,10, temperaturom predgrijavanja vazduha, t_{co} = 100–500°C i temperaturom produkata sagorijevanja ispred rekuperatora, t_{hi} = 600–1100°C.

Ključne reči: Razmenjivači toplote • Rekuperacija • Vazduh • Predgrevanje •

Key words: Heat exchangers • Recuperative tube • Recuperators • Air • Preheating •