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ADVANCED HEAT RECOVERY EQUIPMENT FOR HIGH TEMPERATURE COMBUSTION PLANTS

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Abstract

The fundamentals of heat recovery processes and plants (high-temperature recuperators) have been advanced basing upon complex of theoretical, computation (CFD modeling) and experimental researches. Some new designs of high temperature heat recovery facilities for industrial furnaces are under consideration: tube convective recuperators of RE type equipped with the internal inserts – secondary emitters inside the tubes – for metallurgical furnaces, and the high-efficient two-passage radiative recuperators of RRD type – for the furnaces of machinery engineering branch. Benefits of new heat exchangers comparing to the existing analogues have been estimated: provision of additional fuel saving by means of increase the combustion air preheating temperature and rise of recuperator's thermal stability due to lowering the tube walls temperature.

Key words: *combustion air preheating, heat recovery, high temperature recuperator, heat transfer enhancement, industrial furnace.*

1. INTRODUCTION

It is expedient to combine the heat generating equipment with heat recovery systems to increase the value of power efficiency of high-temperature combustion plants (furnaces, steam generators) in accordance with presentation the Carnot cycle (Kondepudi & Prigogine 1999). Heat recovery equipment (recuperators, regenerators (Gizinger, Wicker & Ballinger 2010; Honger, Gehl & Damagala 2010) provides enhancement of the upper potential of the heat transfer medium (theoretical (adiabatic) combustion temperature), on the one hand, simultaneously with decreasing the lower (basic – exhaust gases) temperature, on the other hand. Correspondingly the upper potential is presented by available heat related to hot source. Lower potential related to cold source is presented by heat reception.

Despite of exceeding the efficiency of regenerative burner system up to 30% in comparison with the recuperative system (Honger, Gehl & Damagala 2010; Teufert & Damagala 2010), the prevalence of recuperative recovery systems, both composed of the centralized recuperator or equipped with the recuperative burners (Gizinger, Wicker & Ballinger 2010) for the reheating furnaces is much more extent in the industry than the regenerative facilities.

The last time trend is to increase the temperature of combustion air (oxidant) in steel reheating furnaces till 600°C (873 K) despite of the most widespread recuperative systems are providing an air preheat till 450°C (Gizinger, Wicker & Ballinger 2010; Teufert & Damagala 2010). Operation situation becomes more complicated in case of utilization of sulfur-containing fuels, particularly the process gases (coke-oven gas). Service life of the ordinary recuperators made of high quality heat- and corrosion- resistant steel tubes usually doesn't exceed 1-2 years in the case of applications the sulphur – containing fuels (for example, coke-oven gas) and high-temperature air preheating.

2. DEVELOPMENT OF NEW CONCEPT AND DESIGN OF HIGH TEMPERATURE TUBE RECUPERATOR

Purpose of development was to elaborate the new concept, designs and mathematical models of high-temperature tube recuperators of prolonged 2 – 5 times service life in conditions of attack of corrosive media.

The following statements and ideas have been found by our researches as a basis to prolong the recuperator's service life:

1. Breaking time by rupture stresses is enhanced 12 – 20 times for the tubes and welded joints in the range of 530 – 650°C in case of only 70 – 100 °C decrease of the tube wall temperature.

2. The limiting step of heat exchange within the system "combustion products flow – recuperator tubes' walls – combustion air flow" falls on internal (within the tubes) constituent.

Increase of resulting heat transfer coefficient within the system "combustion products – tube walls – air flow" while reduction of tube wall temperature due to enhancement of internal (inside the tubes) coefficient of convective heat exchange to the tube walls – gives an opportunity for increase the service life of high temperature recuperator as well as allows us to lower the specific amount (mass) of metal unit of recuperators output.

The inserts enhancing the flow turbulence and correspondingly – the coefficient of convective heat exchange – are used last decades (North American Mfg Co 1982; Tanbour & Rahmani 2008). However significant increase of hydraulic (aerodynamic) resistance of the channel and great flow pressure losses by the length of heat – exchange tubes are restricting the possibilities of wide implementation the proposed technique.

Reduction of tube walls temperature T_w in the recuperators of proposed design (*RE*) by combustion with an air of increased preheating temperature T_a is provided due to implication of radiative component into the limiting (internal inside the tube) component of heat transfer by means of arrangement the secondary emitters – the inserts in form of radial ribs within the tube channels.

2.1. CFD modelling results

The theoretical analysis been carried out, was founded upon mathematical models of heat transfer processes within the tube site without and with the internal inserts. The calculations have been performed grounded upon

- simple averaged engineering approach with account of balance equations for the system of parallel tubes (Cone 1980; Soroka & Sandor 2002);
- zone method of complicated heat exchange within the system of surface and volume elements;
- CFD modelling.

The CFD comparative calculations were performed for the straight tube of 1m length (results are discussed below) and for the U-similar tube loop with the inserts in form of radial ribs.

In Table 1 the computation results of preheating the air stream of $\dot{m}_a$ flow rate moving in the tube arranged in cross flow of combustion products are presented. Computations have been carried out by boundary conditions of the 3rd kind (cross flow temperature and coefficient of heat transfer are given – see data above the Table).

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Main purpose of computations makes comparison of modernized recuperative section *MD* with the inserts of various type inside the 1m length tube with the ordinary (of basic design *BD*) 1m tube without the inserts. Tubes sizes (outer diameter/thickness) make 90/4.5 mm**.**

The resulting characteristics for comparison of *MD* and *BD* designs are temperatures of air preheat $T_{a,ex}$ and maximum temperatures of tube wall $T_{w,,max}$ of insert $T_{in,max}$.

The following parameters have been varied under consideration the inserts impact on thermal characteristics of heat exchange tubes: initial temperature of air flow to be preheated ($T_{a,en}$ = 300, 600, 900 K), tube emissivity (emissivity factor ε_t = 0.9); insert emissivity (emissivity factor ε_{in} = 0.9), air flow rate ($\dot{m}_a$ = 0.0802; 0.2 kg/s), number N of radial ribs serving as the insert (N = 0, 2, 4, 8). Main data for various N values and initial air flow temperatures $T_{a,en}$ are collected in Table 1.

TABLE 1. Main thermal and hydraulic characteristics of recuperative tube of various channel design / Results of CFD modelling (T_f =1223 K, $\alpha_{f\text{-}w}$ = 50 W/(m^{2}K))

$\dot{m}_a$, кg/s	0.0802				0.2	
Channel design						
Parameter	*BD*	MD with the inserts			*BD*	MD with the inserts
		Diameter partition	Star-4	Star-8		Star-4
$T_{a,en}$, K	300					
$T_{a,ex}$, K	380	420	451	482	361	390
$T_{w,max}$, K	1165	1141	1109	1065	1100	1018
$\overline{T}_w$, K	1140	1096	1059	1019	1054	954
$T_{in,max}$, K	–	1067	999	863	–	745
$\overline{T}_{in}$, K	–	972	872	689	–	570
Δp_a, Pa	88	144	203	320	435	873
$\Delta T_{w,a} = T_{w,max} - T_a$	785	721	658	583	739	628
$e = (T_{a,ex} - T_{a,en}) / \Delta p_a$	0.92	0.84	0.75	0.57	0.14	0.10
$e/e_0 = \left(e_D / e_{BD}\right)_{\dot{m}_a, T_{a,en}}$	1.00	0.91	0.81	0.62	1.00	0.73
$T_{a,en}$, K	900					
$T_{a,ex}$, K	939	955	968	983	924	938
$T_{w,max}$, K	1193	1184	1173	1159	1165	1136

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$\overline{T}_w$, K	1182	1166	1153	1138	1149	1110
$T_{in,max}$, K	–	1155	1136	1107	–	1070
$\overline{T}_{in}$, K	–	1126	1102	1061	–	1027
Δp_a, Pa	134	220	314	536	694	1587
$\Delta T_{w,a} = T_{w,max} - T_a$	254	229	205	176	241	198
$e = (T_{a,ex} - T_{a,en}) / \Delta p_a$	0.29	0.25	0.22	0.15	0.03	0.02
$\vartheta = \left(e_D / e_{BD}\right)_{\dot{m}_a, T_{a,en}}$	1.00	0.88	0.75	0.54	1.00	0.70

Note: $\vartheta_D \in \left\{\vartheta_{BD}, \vartheta_{MD}\right\}$

The cases of the most effective influence of the insert arrangement within the tube are related to variant of minimal $T_{a,in}$ – in respect of increase of air flow temperature at the tube exit and to base of bigger $\dot{m}_a$ – in respect of reduction tube wall temperature. Increase of the insert emissivity ε_{in} causes a little bit improvement the recuperative tube characteristics (increase of $T_{a,ex}$ and reduction of tube wall temperature $T_{w,,max}$ of order 20…25 K) while the max insert temperature is increased more than 200 K. Growth of air flow rate causes some additional improvement the thermal stability of the recuperator due to arrangement the inserts within the tubes.

Basing upon consideration of the presented data it has been settled the adequacy of assumed idea: the more is number of the ribs N the higher is an air flow temperature increase, the lower is the tube wall temperature T_w and the characteristic difference $\Delta T_{w,a}$. Pressure losses Δp_a is changed oppositely to the trend of variation $\Delta T_{w,a}$ and main heat exchanger criterion E (see $e \sim E$ in Table 1).

The data obtained due to calculations provide an opportunity to determine main characteristic of heat exchanger – combined heat and hydraulic criterion $E \sim e = \Delta t / \Delta p$ (Zhukauskas 1982). The e value indicates the power cost of flow preheating; the higher is E, the more acceptable is heat transfer facility. As a rule, rate of increase Δt is lower than rate of pressure losses Δp growth by enhancement the heat transfer process but some exceptions are occurred.

The numerical relative values of $\vartheta = e_{MD} / e_{BD}$ are given in the Table 1 under condition of ϑ_{BD} is assumed as 1.0.

2.2. Experimental set-up and testing

The way chosen to enhance heat exchange process within the tube by preheated air flow moving consists in arrangement the insert acting as a secondary (intermediate) emitter inside the tube. Advantages of heat transfer enhancement in high temperature plants and facilities due to installation of secondary emitters till present were studied and used only for the cases of arrangement the inserts within combustion products as emitting/ absorbing (opaque) medium (Demchenko, Dolinsky, & Sigal 2007; Günter & Sun 2006; Soroka, Val & Shpilski 1991). In conditions and designs under present consideration the inserts are arranged within the transparent medium – in the air channels.

2.2.1. The computerized firing rig has been developed and installed in the Gas Institute, Kiev. The methodology has been embodied in simultaneous tests of two recuperative sections comprising each of

three parallel U – formed loops: of *MD* and of *BD* types arranged in the same conditions: by temperature pattern and of flowing over by hot combustion products within the furnace chamber (see Fig.1). Numerical information on advantages of the new (MD) recuperator design in dependence on process parameters has been obtained providing continuous monitoring of thermal and hydraulic state of the loops by means of using analogy – to digital [A/D] converters, by operation of data transmission and on – line data processing systems.

Two most perspective designs of modernized recuperative sections have been used by the tests performing:

- of type *MD*1 equipped with the inserts of cross – shaped cross – section and arranged only within straight sites of the U – similar loops;
- of *MD*2 type distinguished of the inserts arrangement both at the straight and curvilinear sites of 3rd loop along with the inserts located at the straight sites of each of three loops.

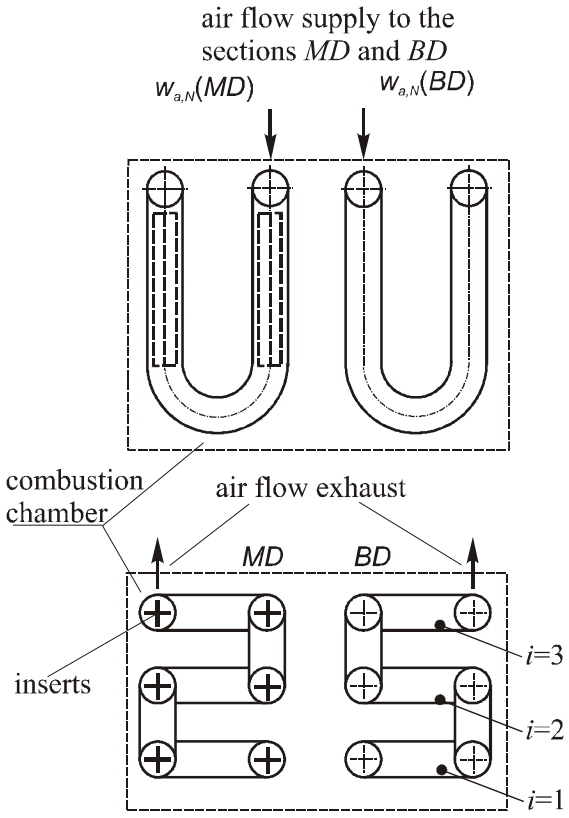


Figure 1. Structural diagram of the test sections (both of *BD* and *MD* types design) of high-temperature recuperator.

The tests include determination both hydraulic (aerodynamic) and heating constituents of combined characteristic of the recuperative sections to be studied and compared. The thermal tests have been

carried out by different furnace (flue gases) temperatures T_f in conditions of varied mass (volume) air flow rate $\dot{m}_a$ / $\dot{V}_{a,N}$.

The following main thermal parameters were studied in frame of this subsection:

- temperature of air preheating at the exit of recuperative section $T_{a,ex} = T_{a,max}$;
- temperature of the tube wall at the recuperative section T_w, including averaged value $T_{w,av}$;
- temperature of the tube insert within recuperative section T_{in};
- characteristic temperature difference $T_{w,av} - T_{a,ex}$.

Below are demonstrated dependences of outlet air preheating temperature $T_{a,ex}$ at the exit of recuperative section and temperature difference $T_{w,av} - T_{a,ex}$ for averaged wall temperature and exit air temperature as the functions of external tube temperature (within the furnace chamber) T_f – respectively for the sections to be compared: *MD*1 and *BD*.

Really experimental data demonstrate advantages of modernized recuperative sections. By relatively low temperatures within the furnace chamber ($T_f \sim 750$ K) air preheating temperature $T_{a,ex}$ (*MD*) is differed only a little-bit of $T_{a,ex}$ value for BD section (Fig. 2) while at indicated $T_f \sim 750$ K the difference $(T_{w,av} - T_{a,ex})_{BD} > (T_{w,av} - T_{a,ex})_{MD}$. Distinction between mentioned differences for *BD* and *MD* designs becomes greater when T_f is growing.

If outer temperature T_f is rising, the coefficient of outer heat transfer α_{out} from the system "furnace chamber / combustion products" to the tube walls outside become higher for both (*BD* and *MD*) sections. But for the basic design of recuperative section (*BD*) where the limiting effect of internal heat transfer tells more strongly upon resulting heat exchange than in case of *MD* section, coefficient of heat exchange $k = \alpha_{out}\,\alpha_{in}\,(\alpha_{out} + \alpha_{in})^{-1}$ in basic design being increased less than for the *MD* section. Opportunity to increase k due to α_{out} rising is higher for *MD* section than for *BD* one. By this reason air preheating temperature $T_{a,ex}$ in function of T_f as *well as* difference $T_{w,av} - T_{a,ex}$ are rising more rapidly in *MD* than in *BD* section (see Fig. 2):

$$\partial[T_{a,ex}\,(MD) - \partial T_{a,ex}\,(BD)]/\partial T_f > 0 \qquad (1)$$

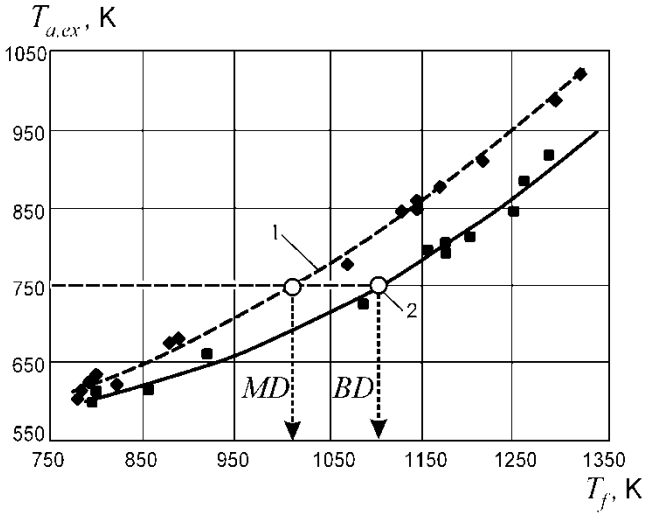


Figure 2. Dependence of air preheating temperature $T_{a,ex}$ on furnace (flue gases) temperature T_f:
1, ◆ – modernized section *MD*1 equipped with the cross-shaped inserts;
2, ■ – section of basic design (*BD*) without the inserts. Air flow by each of the compared sections $\dot{m}_a = 0.062$ kg/s = *const*.

It could be seen by observing the experimental data that air preheat to 750 K temperature in the section equipped with the inserts (*MD*) is attained by heating medium (flue gases, combustion products) temperature of 1010 K while the same air flow $m_a^{\&}$ = 0.062 kg/s preheat in the *BD* recuperative section requires temperature of T_f = 1110 K. Reduction of necessary flue gases temperature ahead of recuperator of any type serves as an ordinary means of prevention the recuperator body overheating (in case under consideration – of overheating the tube walls of recuperator).

As to characteristic temperature differences $T_{w,av} - T_{a,ex}$, the same values of mentioned difference in the recuperative sections *BD* and *MD* are related to the values T_f which are differed much stronger than by case of comparison the sections ensuring the same $T_{a,ex}$, Accordingly Fig. 3 the characteristic difference 100K = idem is provided by T_f (*MD*) – T_f (*BD*) ≈ 265 K while by difference 130K = *idem* mentioned difference T_f (*MD*) – T_f (*BD*) reaches 415K.

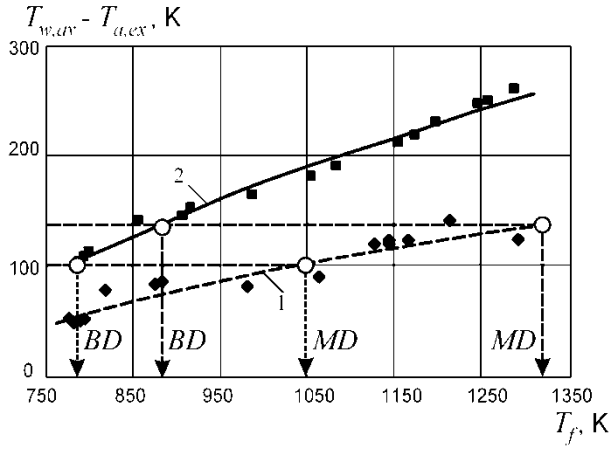


Figure 3. Dependence of temperature difference between averaged recuperative tube temperature $T_{w,av}$ and air preheating temperature $T_{a,ex}$ on furnace (flue gases) temperature T_f: 1, ◆ – modernized section *MD*1 of the cross-shaped inserts; 2, ■ – section of basic design (*BD*) without the inserts. Air flow by each of the compared sections $m_a^{\&}$ = 0.062 kg/s = *const*.

2.2.2. By processing the above mentioned data the following characteristic values have been computed to generalize thermal and heat transfer characteristics of the recuperator designs to be compared (*MD*2 and *BD*):

- amount of heat Q_a has been transferred to an air mass flow $m_a^{\&}$ (volume flow rate at normal conditions $V_{a,N}^{\&}$) by preheating within the recuperative section:

$$Q_a = c_{pm} m_a^{\&} (T_{a,ex} - T_{a,en}) = c'_{pm} V_{a,N}^{\&} \rho_{a,N} (T_{a,ex} - T_{a,en}) \qquad (2)$$

where c_{pm}, c'_{pm} – specific averaged heat capacity of air: mass and volume values – respectively; $\rho_{a,N}$ – normal air density;

- Coefficient of heat transfer k between primary (combustion products, flue gases) and secondary (air flow) heat transfer media averaged by working tube surface F_{rs} of the recuperative section (MD, BD), i.e. of all three loops together would be determine by balance equation:

$$k = Q_a / \Delta t \cdot F_{rs}\,, \qquad (3)$$

where $\Delta t = \Delta t_{f-a}$ – logarithmic mean temperature head between primary and secondary heat transfer media.

By conditions of mutual arrangement of the tested sections within the firing rig and of both heat transfer media movement the calculation equation for definition Δt would be acceptable in the next form (Hausen 1976):

$$\Delta t = (T_{a,max} - T_{a,en}) / ln\,[(T_f - T_{a,en}) / (T_f - T_{a,max})], \qquad (4)$$

T_f, $T_{a,en}$, $T_{a,max}$ – temperatures respectively: flue gases, air flow at the inlet and outlet of recuperative section under consideration;

The constituent of heat transfer coefficient – the inner (inside the tube) heat exchange coefficient α_{w-a} between the tube wall and secondary heat transfer agent – air flow has been studied experimentally within each of the i-th loop (i=1, 2, 3).

Let's consider the results of comparison the heat efficiency of the recuperative sections to be installed within the furnace (combustion chamber) simultaneously (juxtaposed in parallel).

In Fig.4 are shown respectively the thermal characteristics of the sections *BD* and *MD*2 at the same heat supply conditions, while in Fig.4 the characteristics of *MD*2 are considered under two different outer conditions. In Fig.4 are presented comparative data on heat absorption $Q_a = Q_{max}$ by air flow correspondingly within *BD* and *MD*2 in dependence on air flow through recuperative section $\dot{V}_{a,N}$. By the same outer heat source temperature (combustion products of $T_f \approx 1215 \pm 20$K) Q_{max} for *MD*2 is increased approximately 15% in comparison with *BD* while coefficient of heat transfer k is enhanced by 25 – 35%.

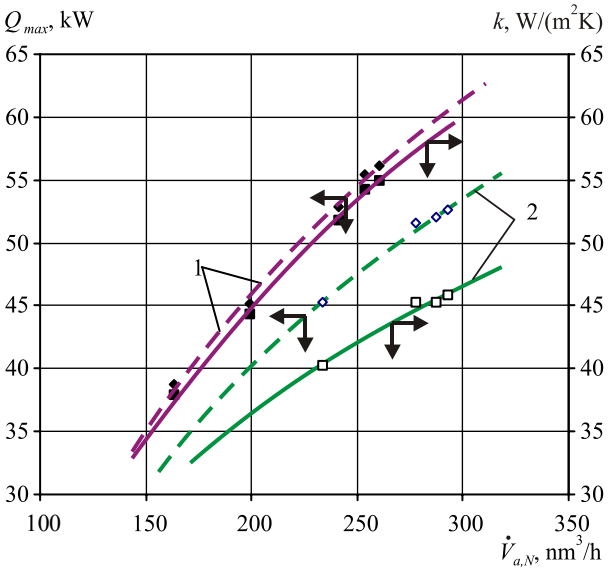


Figure 4. Comparison of the main thermal characteristics of the recuperative section of *MD*2 design (1) with those of *BD* section (2): dotted lines (◆,◇) – total heat fluxes transferred to air $Q_a = Q_{max}$, solid lines (■, □) – coefficients of heat transfer k between primary (combustion products) and secondary (air flow) media. Furnace flue gases (outer) conditions: T_f= 1215±20 K= *idem*.

By increase flow velocity within *MD*2 section by 2.2 times the relevant heat transfer coefficient *k* would be increased twice, up to 60W/(m^2·K).

Comparison of mutual disposal of the curves $Q_{max} = f_1(I\&_{a,N})$ and $k = f_2(I\&_{a,N})$ for MD2 section design with those of *BD* section ensures the conclusion on the nature of the modernized recuperator's superiority: in *BD* case the opportunities of outer heat supply to the tube surface are restrained by inner (inside the tube) possibilities of heat exchange from the tube wall to air flow. In the same time in case of modernized design an internal heat exchange is enhanced due to the insert arrangement within the tube. As a consequence the possibilities of heat reception are expanded and the limitations of internal heat exchange (coefficient α_{w-a}) impact on *k* values would be removed.

3. DEVELOPMENT OF HIGH-EFFICIENT TWO-PASSING RADIATIVE RECUPERATORS OF RRD TYPE

Radiative heat transfer represents the main component in the heat flux to reception surfaces from the combustion products. The channel for combustion products flow in the radiative recuperators has considerable size by the whole length (height) of heat exchange (reception) surface. Radiative recuperators have a vide spread in the furnaces of various purpose where combustion products temperature makes 1300 – 1700 K at the inlet to recuperator or at furnace exit (Oyelami 2012, Pioro et. al 2006). By this reason the great demands are made of thermal stability of metallic heat exchange surfaces in recuperators, the counter – flow scheme being used under inlet gases temperature of 1270 – 1470 K while by flue gases temperature over 1470 K – the co-flow scheme.

Radiative slot recuperators represent as a rule two coaxial casings where the combustion products are flowing within the inner housing while an air flow is moving by the annular slot (Tebenkow 1975). The casings are made of steel sheet 5 – 10 mm thickness, inner diameter of the central casing makes 0.5 – 3.5 m – in dependence on recuperator output and the air slot width makes 8 – 60 mm. There is an opportunity to manufacture the recuperators by layout of one-sided (Fig. 5a) and double- sided heating (Fig. 5b).

In both cases the relatively insufficient area of heating (heat exchange) surface makes shortcoming of the recuperators design because air flow is heated only within the single annular slot channel while working space of transportation the flue gases flow is used unpractical. Heat exchange surface between "hot" and "cold" flows is insufficient.

It has been proposed to use the secondary emitters in form of radial partitions to enhance heat exchange within flue channel of the slot recuperators (Kardas, Larson & Nesbitt 1975). These facilities increase the heat flux by radiation to inner shell and thus are enhancing the heat efficiency of recuperator.

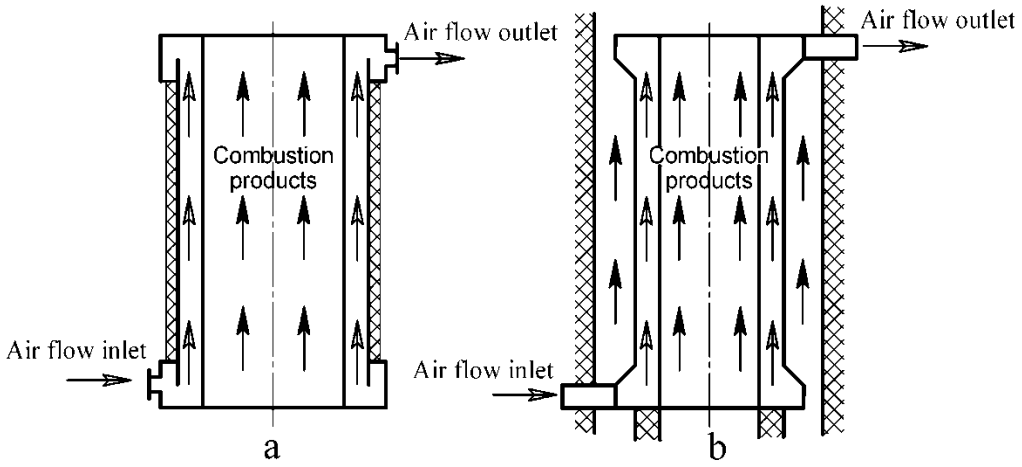


Figure 5. Radiative slot recuperator: a – with one-sided heating; b – with double-sided heating.

The original design of radiative two-passage recuperator of *RRD* type has been developed in Gas Institute of National Academy of Sciences of Ukraine (see Fig. 6) (Soroka et. al 2012). The relevant design is characterized by rational mutual arrangement the heating surfaces as we by secondary emitter's installation within the flue channel. By means of improvement the recuperator design we were succeeded in combustion air preheating temperature $T_{a,ex}$ enhancement and in increase of heat flux Q_a transferred to air flow as the secondary heat-transfer medium.

High efficiency of recuperator under consideration of *RRD* type is provided due two-stage air preheat. The 1st heating stage is performed within inner annular channel by two coaxial shells with double-sided heating from combustion products. The last flow is directed from the furnace exhaust upwards to the recuperator inlet and than is divided by two combustion products flows: central cylindrical and peripheral annular one. The 2nd heating stage represents the peripheral annular channel filled in with moving downwards air flow. The channel is surrounded from inner side with the steel shell been flowed over with the peripheral flue gases flow (of combustion products). The thermal insulation makes coating upon the external side of the outer casing of peripheral air flow. Both air channels of co-flow (parallel (-current)), and counter current flows are connected by the system of radial tubes above the recuperator top.

To increase an efficiency of heat exchange between primary (combustion products) and secondary (an air flow) heat transfer media within the central flue gases channel of the radial ribs been crossed between themselves, is arranged. The auxiliary emitters made of plane radial ribs are installed within the combustion products annular channel. Mentioned indices are ensuring increase of total heat flux to the walls of recuperator channels.

In Fig. 7 the impact of extension the heat the heat exchange surface (the relative area F^{0}/F) upon an temperature of air preheating $T_{a,ex}$ within the recuperator of bell-type furnace is shown in curve 1 (Wendt & Kühn 2011). In the last work the recuperator design doesn't presented. The modern bell-type annealing furnaces are equipped with high-efficient individual recuperators for each of burners or with centralized recuperator. The process under consideration is related to the number of low – and of middle-temperature thermal treatment process (840 – 880 K) (Mishin, Sarychev & Lebedev 1999). Meanwhile taking into account the variety of the annealing processes for steel ingots (bright and dark, spheroidizing, prehydrogen firing etc) as well as of other firing process the range of working temperatures up to 1400 K. Maximum are preheating temperature for recuperators of bell-type furnaces

makes 730 K, an average air preheat temperature by annealing process makes 630 K or a little-bit lower. In these conditions power loss with flue gases behind recuperator constitutes up to 28 % (Wendt & Kühn 2011).

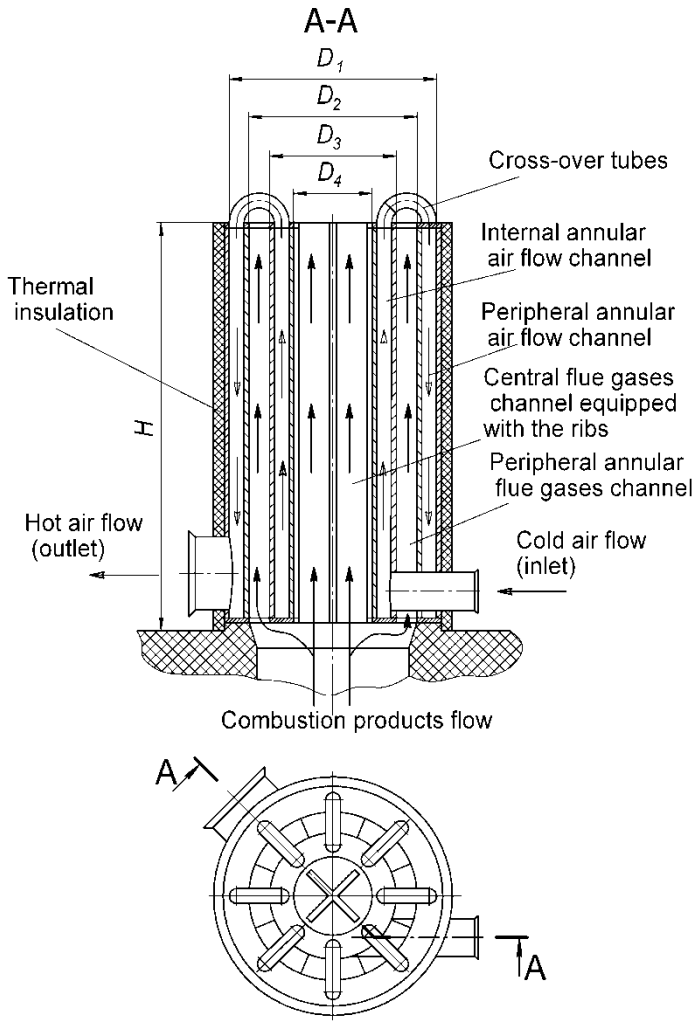


Figure 6. Radiative recuperator of *RRD* type design developed by Gas Institute, NASU. Combustion products flow is shown by thick lines, air flow – by pale (thin) lines.

Further increase air of preheating temperature till 900 K provides additional reduction of natural gas flow rate by 11 % in the technology under consideration however demands a significant enhancement of recuperators heat exchange surface (Wendt & Kühn 2011).

The results obtained in paper (Wendt & Kühn 2011) are compared with our data. The $T_{a,ex}$ temperatures accordingly our computations are marked as the dots and joined by respective curves (Fig.7) in dependence on $P_{\%}$ value influence. The correlations between $P_{\%}$ value and exit air temperature $T_{a,ex}$ for radiative recuperators of new original design of (*RRD* type – radiative recuperators of double-passage) (Soroka, Sandor & Vorobyov 2012) (Fig.6).

The following conditions of recuperator's efficiency comparison by enhancement of heat exchange surface $\tilde{F}$ in new design were tested: increase of recuperator's height or / and of diameters of shells. The criterium of heat exchange surface change $\tilde{F} = \pi(D_2 + D_3 + D_4)H / F_0$, where F_0 – area of basic (of one-sided) design heat exchange surface, H and D_i – respectively the recuperators height and diameters of shells. Ratio of diameters was supported as invariable (constant) value by variation the heat exchange surface: $D_4/D_3 = const$, $D_4 /D_2 = const$. – diameters of heat exchange surfaces (Fig.6).

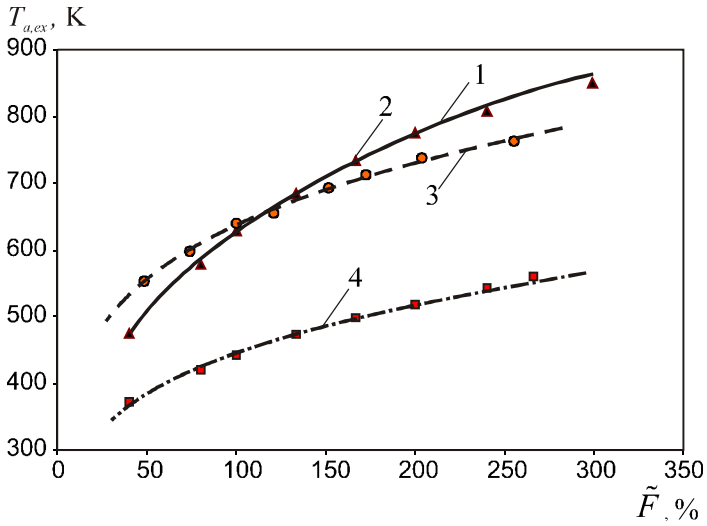


Figure 7. Dependence of combustion air preheating temperature $T_{a,ex}$, K, on relative value of recuperative heat exchange surface $\tilde{F}$, %: 1 – curve by data of paper (Wendt & Kühn 2011); 2, ▲ – the dots accordingly our computations for radiative recuperator of *RRD* type by increase of heat exchange surface due to rise the recuperators height H; 3, ● – accordingly our computations for radiative recuperator of *RRD* type (Fig. 6) by effect upon heat exchange surface by means of change the diameters of heat exchange surfaces; 4, ■ – accordingly our computations for radiative recuperators with one-sided heating by effect on the surface due to change the recuperator height H.

In the Fig. 7, curve 4 the computation dots are marked as well for radiative recuperator of traditional design likely the slot design heat exchanger with one sided heating (Fig. 5 a). The recuperator design is distinguished with parallel (current) flow of combustion products and air flow to be preheated (Tebenkow 1975). As could be seen from the Fig.7 the high efficient RRD type recuperators proposed by our team provides an opportunity for great increase an air preheating temperature (up to two times – in °C) in comparison with traditional design of radiative recuperators with one-sided heating.

The following initial data have been chosen for calculation the basic version of radiative recuperators of RRD type by evaluation the dimensions influence ($\tilde{F}$=100 %): inner diameter of outer shell D_1=1.3 m, the diameters of annular channels: for air flow delivering D_2=1.24 m, for flue gases removal D_3 = 0.87m. Inner diameter of central flue gases channel D_4 = 0.8 m. Width δ of air slot channels is equal to 0.025m. The recuperators height makes H=3m.

Combustion products temperature at the recuperators inlet makes $T_{f,en}$ = 1173 K; for air flow inlet $T_{a,en}$ = 298 K. Mass air and combustion products flows make respectively: $\dot{m}_a$ = 0.786 kg/s; $\dot{m}_{fl}$ = 0.825 kg/s.

For basic design of radiative recuperator with one-sided heating the similar sizes with *RRD* type recuperators have been taken: diameter of outer shell D_1 = 1,3 m; recuperator height H = 3 m. Width of air flow makes δ = 0.025 m. Mass air and flue gases flows as well as initial temperatures of the flows performing heat exchange process were taken of the same values as assumed above for calculation the *RRD* type recuperators.

CONCLUSIONS

1. One of the main directions of fuel saving in high temperature power and process equipment: boilers and furnaces – make heat recovery of waste (flue) gases behind the furnace exit. The recuperators present high temperature heat recovery facilities for directing portion of waste heat back, in principal process and furnace. Heat recovery process is fulfilled by means of preliminary preheat of oxidant (an air flow– first of all) and fuel (in case of low calorific (LCV) gases) – sometimes – in recuperative heat exchangers. Creation of advanced recuperators and enhancement of heat recovery degree are provided by intensification of heat exchange process and due rising the exchange surface between flue gases and medium to be preheated (combustion air, LCV – gases).

2. The recuperators are divided for convective and radiative (radiative – convective) types. Both types of mentioned heat exchangers were advanced in frame of the studying under consideration. The development results have been analyzed and presented in the paper.

For the case of tube convective recuperators two designs have been studied: of *MD*1 type – when the inserts were arranged within the straight sites (sections) of the tubes – and of *MD*2 type – when the inserts (secondary emitters) were arranged inside the tubes (fully or partially) by the whole length of straight and curvilinear sections of the tubes.

3. New concept of efficiency enhancement of tube recuperators has been proposed and tested both by computations and experimental tests. As a result the final rising the temperature of combustion air preheating as well as lowering of tube walls temperature along with reduction of characteristic temperature difference $\Delta T_{w,a}$ between air preheating and of tube walls has been proved in frame of researches. The novelty and know-how of the concept makes using of secondary (intermediate) emitters inside the tubes filled in with transparent medium (combustion air).

4. It has been stated that internal heat transfer enhancement by means of the inserts arrangement inside the tubes as well as the advantages of modernized design (*MD*) with the secondary emitters – as a whole – are increasing by flue gases temperature rise at the inlet to the recuperator. Basing upon the computation results and experimental, data a growth of air preheating temperature $T_{a,ex}$ has been proved along with reduction of tube walls temperature T_w and decrease of the appointed temperature difference $\Delta T_{w,a} = T_w - T_a$.

5. It has been found that total heat flux absorbed by *MD*2 recuperative section is enhanced by 15% in comparison with *BD* section while coefficient of heat transfer k in the system "combustion products of temperature 1215 ± 20 K – tube wall – air flow" is increased by 25…35%. The heat exchange coefficient between the tube wall and air flow is increased in modernized design of recuperator even stronger than that of k, sometimes up to 1.8 times, thus fully removing obstacles for intensification the heat transmission.

6. New design of high-efficient radiative recuperators of *RRD* type has been proposed. The developed design is distinguished with enhanced heat exchange surface and is intended for reheating and heat treatment furnaces of machinery engineering branch of industry. Application the recuperators of *RRD* type insures substantial increase of air preheating temperature along with great (up to two times)

increase of heat reception by air flow from flue gases in comparison with the recuperators of traditional design.

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