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Development of Calculation Technique by Designing the Radiative Recuperators Advanced in Frame of EUREKA Program

The EUREKA project «REPLACE NG» is focused on development and application of radiative (radiative-convective) recuperator of advanced design for heat recovery of flue gases by combustion the fuels including those of low calorific value (LCV gases) in the industrial furnaces of various types. It was supposed that the recuperators are suitable for preheating the flows of combustion air or LCV gases. Recuperators of both purposes could be used by installation at the furnaces fired with LCV gases. Mentioned equipment — recuperator of RRD type — was developed by Gas Institute of NASU (Ukraine). It was supposed accordingly preliminary program and schedule to involve Co. TÜKI (Hungary) as the project partner for testing the recuperator arranged at the bell type furnace of Cold Rolling Mill Plant of ISD DUNAFERR Group under the furnace firing with different process gases (alternative fuels) originated by several metallurgical productions. The present paper demonstrates the proposed calculation technique suitable for comparison the thermal state of the new type heat exchangers and the radiative recuperators of traditional wide spread design basing upon computation of heat exchange processes within parallel-current and counter-current flows. The system of simplified differential one-dimensional heat balance and heat exchange equations was composed and used as the computation background. The computation model provides an opportunity to analyse an influence of heat transfer coefficients, mass and volume flow rates and heat fluxes by process of an air preheating due discovering and application the generalizing auxiliary dimensionless parameters. The advantages of new approach to the recuperators designing even by neglecting of an enhancement of heat transfer intensity from both sides of heat exchange surface(s) or from both media flows are connected with increase the heat exchange surface in case of recuperator of RRD type in comparison with ordinary one-stage radiative recuperators. Application of secondary (intermediate) adiabatic emitters arranged in channels for flue gases and air (or LCV gas) flows provides the additional opportunities to increase an air (or LCV gas) preheat. *Bibl. 11, Fig. 8, Table 2.*

Key words: bell type furnace, combustion air, combustion products (flue gases), counter-current flow, heat exchange coefficient, heat transfer coefficient, low calorific value fuel (LCV gas), parallel-current flow, radiative recuperator, temperature of heat transfer medium.

1. Presentation of the international project

The «REPLACE NG» project has been submitted to EUREKA program by international consortium under the co-ordination of the Gas Institute, NASU in the year 2011. The purpose of application made development of special firing and heat recovery appliances suitable for use both in case of natural gas (NG) and the alternative fuels (AF) use and promoting the substitution of NG for AF. The project has been accepted by the EUREKA Forum in Brussels in June of 2011. The Research and Development Company TÜKI has been recognized as the participating structure from the Hungarian side by the Project performing.

One of the decisive tasks of the project is connected with the recuperators advancement resulting in development of updated designs of convective [1] and radiative [2] recuperators. Division of this heat exchange equipment by two mentioned types is conditional enough in principle because of both constituents of heat transfer takes place under the plant operation. But the higher is working temperature of the furnace operation the higher would be radiation input.

1.1. Concept of development the novel types of heat recovery facilities. High-efficient two-passing radiative recuperators of RRD type (description). The distinguished feature of the recuperators of both types accordingly Gas Institute's proposal makes application of adiabatic (secondary, intermediate) emitters — the inserts arranged in radiatively transparent media (for tube convective type recuperators) or within flow of emitting combustion products moving along flue gases channels (for basket type radiatiative recuperators). Application of the inserts within the channels in media of both types serves to provide enhancement of heat flux by radiatiation from the heat exchange surface between two media and to increase area of reception surface to transfer the resulting heat flux absorbed by flow to be preheated by convection.

Radiative heat transfer provides the main component in the heat flux to reception surfaces from the side of 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 wide spread in the furnaces of various purpose where combustion products temperature makes 1300–1700 K at the inlet to recuperator or at furnace exit [3, 4]. By this reason the great demands are made of thermal stability of metallic heat exchange surfaces in recuperators, the counter — flow scheme being used in case of 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 through the annular slot [2]. The casings are made of steel sheet of 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. 1, a) and double-sided heating (Fig. 1, b).

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 wasteful. Heat exchange sheet partition between «hot» and «cold» flows is insufficient by its area.

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 [5]. These facilities increase the heat flux by radiation to inner shell and thus are enhancing the heat efficiency of recuperator.

The original design of radiative two-passage recuperator of RRD type (see Fig. 2) has been developed in Gas Institute of National Academy of Sciences of Ukraine [6]. The relevant design is characterized by rational mutual arrangement the heating surfaces as well as by secondary emitters installation within the flue channels. 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 transferred to air flow as the secondary heat-transfer medium.

High efficiency of RRD type recuperator under consideration is provided due two-stage air preheat.

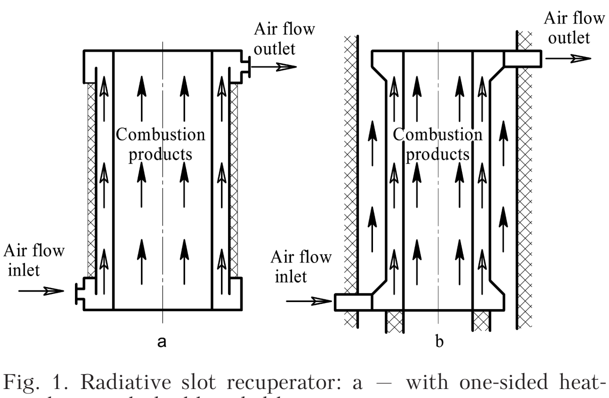


Fig. 1. Radiative slot recuperator: a — with one-sided heating; b — with double-sided heating.

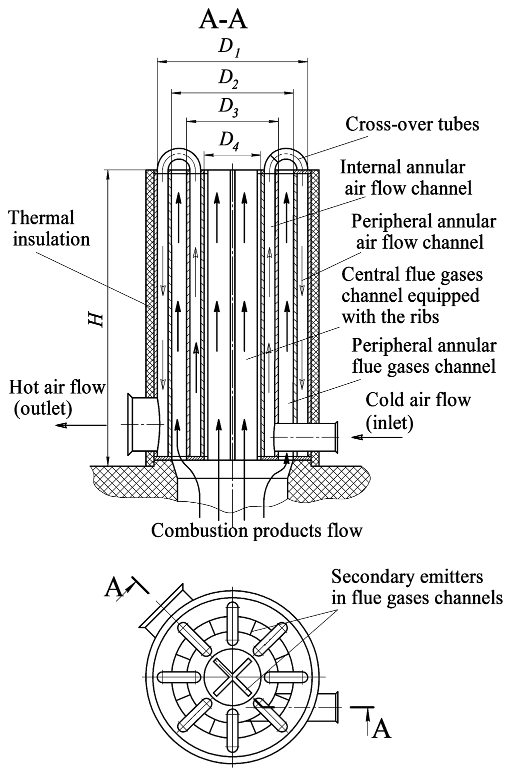


Fig. 2. Radiative recuperator of RRD type design developed by Gas Institute of NASU [2]. Combustion products flow is shown by thick lines, air flow — by plane (thin) lines.

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 flow: 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 is formed by 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.

Due enhancement the heat exchange process the final temperature of media to be preheated is increasing while the heat exchange surface temperature is decreasing and thus the service life of the facility on the whole should be prolonged.

1.2. Stages and matter of experimental tests. The advantages of new equipment by using the process gases, biogases and the gasification products of industrial wastes, solid and liquid biofuels for firing purposes need of special confirmation. The recuperator design ensures an enhancement of heat transfer providing lower level of temperature of heat exchange surfaces and thus reducing the corrosion effect by action of gas flows. It makes possible the efficient utilization the heat content of LCV flue gases in wide range of technologies. An increase of efficiency of application of blast furnace gas (BFG) and coke oven gas (COG) as well as of the mixture of BFG and COG for high temperature processes and technologies is of great importance.

Because of similar reasons the application of an new heat recovery appliances is also advantageous by gasification technologies. The production and application of gaseous products from carbon based or other solid organic fuels by partial oxidation of the mentioned materials is one of main targets of the power engineering for the next decades. These technologies are providing the engineering basis of the CO_2 free «clean-carbon» programs. The heat exchanger to be developed in framework of the present project provides more safe and economic operation of the facility.

The tests of heat exchanger operating with LCV gases of corrosive flue and, if is it required, the steps for further development have to be made. In the framework of the EUREKA project Co. TÜKI will undertake performing of this part of the task.

The Supporting Contract (EUREKA HU 12–1–2012–003) has appeared due approval by the International Development Agency concluded with the Co. MAG. Starting of the program began in February 2013.

Co TÜKI is intending to realize its investigation program through two directions. In the first phase the recuperator to be tested will be installed at bell-type furnace operating with mixture of blast furnace gas (BFG) and coke oven gas (COG). The tests will be carried out under real working conditions. By means of these studies will be get appropriate data regarding the operation features of the facilities.

In the second phase of these trials the recuperator will be installed at the special experimental furnace of 200 kW capacity. The furnace will be fired with the blast furnace gas and coke oven gas as well as with the combination of two fuels of various composition. A database on the temperature and pressure losses and further measuring variables will be recorded jointly with the Ukrainian partner on the temperature and pressure losses and further measuring variables. For LCV gases the determination of the optimal design has to taken into account the heat transfer factor, the minimal heat exchange surface temperature, the smallest pressure losses and minimal danger of deposition.

The great complex of modelling researches, numerical analysis and the experimental tests carried out with the tube convective recuperator sections arranged at the large – scale firing rig along with the proper validation data have confirmed convincely the justice and adequacy of initial statements used by development the novel approach to designing the high temperature recuperators [7].

It must be stated that till present all tests with application of the recuperators of new designs have been carried out exclusively by firing the furnace (furnace chamber) with natural gas.

1.3. The trial furnace with RRD type recuperator. For the purpose of the trials with LCV gases the RRD type radiative-convective recuperator [patent] developed by the Gas Institute will be used. The chosen type can be explained by the related market demands. The high temperature is characterizing the majority of the utilized technologies and they require involving the possibilities of the radiation heat transfer [8]. Naturally, the convection is also an important part of the combined heat transfer and the increase of its intensity is of great importance especially within temperature range inherent for heat treatment furnaces.

In the first phase of the investigations the recuperator will be installed at the bell-type furnace of the Cold Rolling Mill Ltd. of the Co. ISD DUNAFERR [9].

This furnace set is consisted of thirty sockets and by twenty four of these ones the TÜKI applied 2–2 pieces of bag basket type recuperators at each furnace for combustion air preheating. The furnaces are fuelled with blast furnace gas and coke-oven gas in ratio of 4 : 1, providing BFG and COG mixture of average heating value of 5.6–6 MJ/m^3.

The furnaces are of batch-type operation. Temperature of the flue gas in the second part of the up heating phase resp. during the soaking period is in the range of 800–900 °C. The Volume flow rate of an air to be preheated in recuperator is about 600 m^3/h.

The gas mixture is burned by means of 6 Pyronics HS7 type burners installed on the mantle of the heating bells [7]. The flue gases flowing by side of the internal protective bell transfer a certain part of its heat content and leave the heating bell through its upper section.

The Fig. 3 shows the heating bell equipped with the recuperators of RRD type. Their location is identical to the present arrangement of exsting heat recovery plants. The recuperators of perpendicular axis are located on the outer mantle of the bell. The flue gases leaving the furnace gets to the recuperator through the baffle element of 90° and the compensator. An air operated ejector located along the centre ensures the appropriate effect of sucking. After its blending with the flue gases, the mixture leaves the facility at the bottom and flows to the flue gases channel.

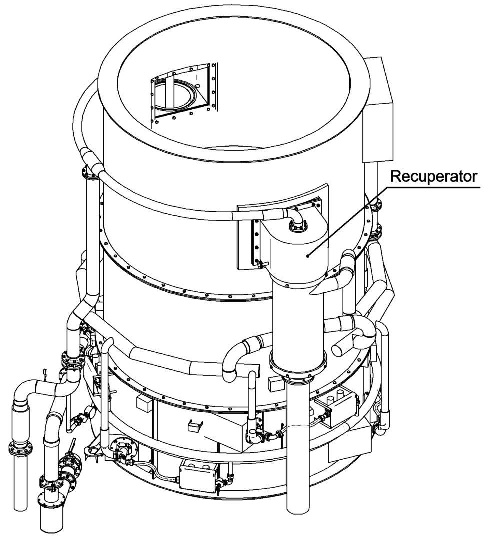


Fig. 3. Coil annealing bell-type furnace with installed recuperators.

By operation of new design of developed recuperator of RRD type (Fig. 2 and Fig. 4) the flue gases are flowing in two parallel pathways: within central internal channel and through intermediate peripheral annular space. The direction of moving flows along the axis from the recuperator top to the bottom. The air is flowing through recuperator in two lines connected in turn. In the first one a parallel flow heat transfer is realized by heating of the air channel from both sides. After turning the air is flowing in contrary direction. Within the outer air channel a counter flow heat transfer takes place. The outer mantel is furnished with heat insulation. It has been foreseen to keep the existing flue gases and air connections of the recuperator by designing the RRD type recuperator for arrangement at the bell type furnace for testing to be carried out.

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 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.

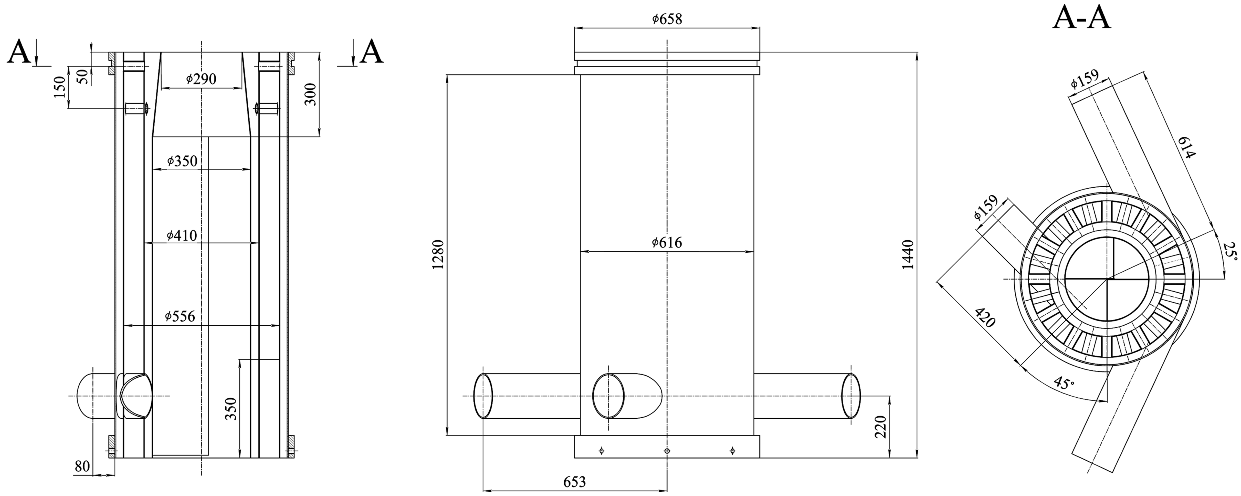


Fig. 4. Design layout of new type of experimental recuperator for the bell-type furnace.

2. Development of technique with differential equations for comparative calculations

The RRD type recuperator represents the novel developed modification of the wide spread heat utilizing equipment. Comparing with the «traditional» chimney recuperators, they have about the double length of pathways for the air and roughly three times bigger heat transfer surfaces. In order to define the surplus of the waste utilization, calculation model was developed in frame of EUREKA project basing on simple differential approach to heat exchange evaluation between flows separated by cylindrical shells.

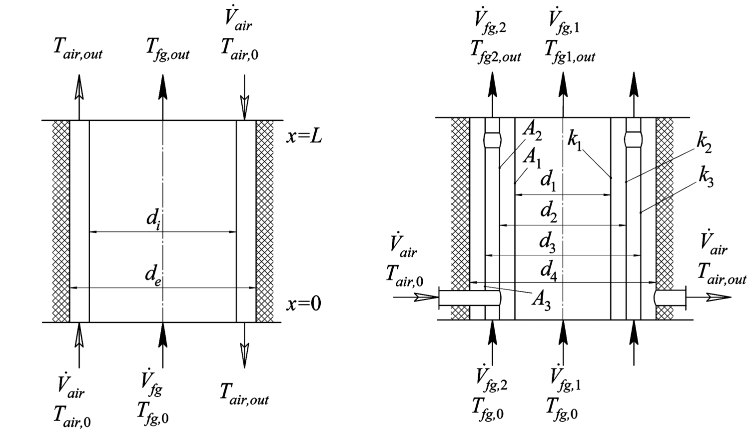
The model concerns of both RRD and «tube in tube» recuperators designs. The last one (BR) was chosen as basic equipment for comparison the characteristics under consideration. Fig. 5 shows the calculation sketches and introduces the nomenclature of main parameters. Initial values have been introduced below for generalization the obtained calculations results as follows.

The scale of the air flow preheating can be expressed by the ratio

$$\gamma_T = (T_{air,out} - T_{air,0}) / (T_{fg,0} - T_{air,0}) = (T_{air,out}/T_{fg,0}) \times [(1 - T_{air,0}/T_{air,out})/(1 - T_{air,0}/T_{fg,0}). \quad (1)$$

The quotient of enthalpies (physical heat contents) characterising the degree of heat utilization:

$$\gamma_q = (c_{air}\, \dot{V}_{air}\, T_{air\ out}) / (c_{fg}\, \dot{V}_{fg}\, T_{fg,0}). \quad (2)$$



Parallel (left) and counterflow (right) models of BR recuperator

Model of RRD recuperator

Fig. 5. Calculation layout for the models under consideration: A_1, A_2, A_3 — heat exchange surfaces between primary (combustion products and secondary (air flow) heat transfer media; k_1, k_2, k_3 — heat exchange coefficients for respective heat exchange surfaces.

The channels design accordingly models don't include neither primary flow directing nor swirl inducing elements.

2.1. Basic correlations. Bernoully's equation for flue gases flows and for air flow as well indicates conservation of the mechanical energy value. The matter is that the flow movement process within the central and annular flue gases channels takes place simultaneously with heat transfer process. As a result the energy conservation differential equation corresponding to the 1st thermodynamics law has the following form

$$\begin{cases} dq = du + p\,dv + d\,(w^2/2) \\ dq = di - v\,dp + d\,(w^2/2) \\ dq = c_p\,dT - dp/\rho + d\,(w^2/2) \end{cases} \quad . \qquad (3)$$

The last equation in the system (3) represents differential form of the equation for «total energy *e*» [10] (in case under consideration — for non-reacting flow).

It means that supplied heat is mainly consumed for flow preheating and acceleration (change the velocity (dynamic) head or kinetic head). As a result the pressure losses are greatly increased in comparison with isothermal (low-temperature) flow. Simultaneously a portion of internal energy dq_{diss} is dissipated because of substantial growth of hydraulic resistance by flow movement. In above given equations the following nomenclature is used: q — supplied heat amount; internal energy; i — enthalpy; w — flow velocity; T, p, ρ, v — temperature, static pressure, gas (air) density and specific volume at given p and T conditions, c_p — specific heat content.

Moreover, account of the pressure losses with friction (only) is insufficient because of more strong impact of temperature on other accompanying constituents, first of all on local pressure losses: those division the flue gases flow before its entering into the flue channels, by inlet to and outlet of the pathways.

$$\dot{Q} = c_{pm}\,\Delta T - \Delta p / \rho_m + 0.5\,\Delta w^2 + \sum_{(i)} \varsigma_i\, w^2 \qquad (4)$$

or the transferred heat flux Q_a between 1 and 2 cross sections along the pathway or the considered flow and channel,

$$\dot{Q} = c_{pm}\,(T_2 - T_1) + (p_1 - p_2)/\rho_m + \\ + 0.5\,(w_2^2 - w_1^2) + \sum_{(i)} \varsigma_{fr\,i}\, w^2 + \sum_{(j)} \varsigma_{loc\,j}\, w^2. \qquad (5)$$

In equations (4) and (5) the subscript «m» means averaged value within temperature range while «i» and «j» are referred to friction and local resistances by pathway between mentioned arbitrary chosen cross-sections 1 and 2. The averaged values of heat capacity (specific heats at constant pressure) of the heat exchanging media are used.

2.1.1. Parallel flow. The relation of the temperatures change within the parallel flow could be found basing upon the heat balance equation from initial to current cross-section of the channel:

$$T_{air} = T_{air,0} + \Psi\,(T_{fg,0} - T_{fg}), \qquad (6)$$

where factor Ψ has been the «water value quotient» modified by the efficiency of the heat exchanger:

$$\Psi = \eta\,(c_{fg}\,\dot{V}_{fg}) / (c_{air}\,\dot{V}_{air}). \qquad (7)$$

Total heat flux transferred to the air volume flow rate $\dot{V}air$ as function of local temperatures of heat transfer media can be expressed by the differential equation:

$$d\dot{Q}_{air} = c_{air}\,\dot{V}_{air}\,dT_{air} = k_i\,d_i\,\pi\,(T_{fg} - T_{air})\,dx. \qquad (8)$$

Substituting the temperature of the flue gas from equation (6) by means of integration procedure, we obtain the solution in form of exponential equation for current location (cross-section x along the tube pathway):

$$T_{air} = [\Psi\,T_{fg,0} + T_{air,0} - (T_{fg,0} - T_{air,0}) \times \\ \times \exp\left(-\frac{k\,d_1\,\pi\,x}{c_{air}\,\dot{V}_{air}} \cdot \frac{1+\Psi}{\Psi}\right)] / (1+\Psi). \qquad (9)$$

By using the simplified indicator

$$\beta = (k\,F_i) / (c_{air}\,\dot{V}_{air}) \qquad (10)$$

the temperature of preheated air at the recuperator exit of length «L» could be obtained in following form

$$T_{air,out} = [\Psi\,T_{fg,0} + T_{air,0} - \Psi(T_{fg,0} - \\ - T_{air,0}) \cdot \exp\left(-\beta\frac{1+\Psi}{\Psi}\right)] / (1+\Psi). \qquad (11)$$

The air preheating ratio at this stage of heat exchange makes

$$\gamma_{parallel} = \Psi\,[1 - \exp\left(-\beta\frac{1+\Psi}{\Psi}\right)] / (\Psi+1). \qquad (12)$$

2.1.2. Counter flow. Comparing with equation (6) the temperature relation regarding heat exchange media in case under consideration will be changed. Temperature pattern will be presented for the pathway site from current till exit cross-section:

$$T_{air} = T_{air,0} + \Psi\,(T_{fg} - T_{fg,\,out}). \qquad (13)$$

Similarly, the temperature of air at the tube exit is somewhat different from the same by parallel flow

$$T_{air,out} = T_{air,0}\,(1-\Psi) + T_{fg,0}\,\Psi\,[1- \\ -\exp\left(-\beta\frac{1-\Psi}{\Psi}\right)] / [1 - \Psi \cdot \exp\left(-\beta\frac{1-\Psi}{\Psi}\right)] \qquad (14)$$

where the definition of the parameter «Ψ» is unchanged. The following expression demonstrates the recuperator's opportunity for the second stage of air preheating

$$\gamma_{counter} = \Psi\,\{1 - \exp\,[-\beta\,(1-\Psi)/\Psi]\} / \\ / (1 - \Psi\,\exp\,[-\beta\,(1-\Psi)/\Psi]. \qquad (15)$$

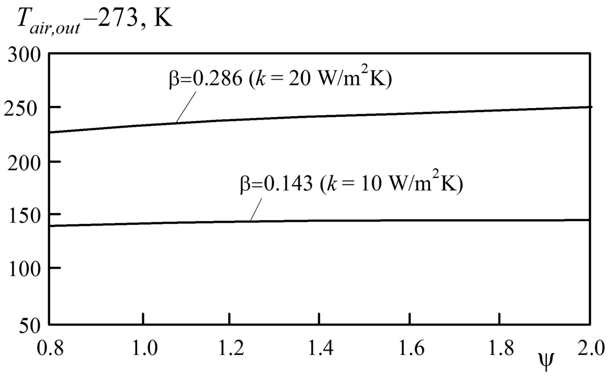


Fig. 6. The effect of parameter «Ψ» on the outlet temperature of preheated air.

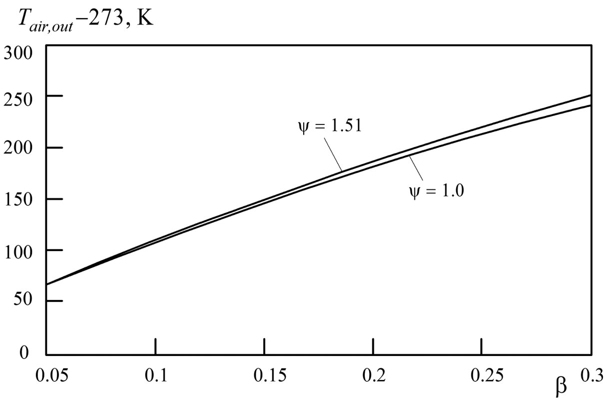


Fig. 7. The effect of parameter «β» on the outlet temperature of preheated air.

The equations 12 and 15 show that efficiency of the heat exchanger has been defined by parameters «Ψ» and «β». In order to get their real influence, we've carried out some evaluative calculations. They are based upon characteristics given in Fig. 6 while operation parameters of the bell type furnace should be taken into account as well.

It has been stated that the calculated air temperature is practically independent on flow direction. The difference at the investigated combinations of parameters didn't exceed the value of 4 °C.

Fig. 7 introduces the effect of parameter «β» being proportional to the heat exchange coefficient «k». Independently from the value «Ψ» the function under consideration is nearly linear — harmonising with equations (6) and (13). The effect of parameter «Ψ» could be seen in the Fig. 7. Its influence is minimal. At higher «β» values (higher heat exchange coefficient) difference of $T_{air,out}$ values becomes slightly increasing, but even doubling of the flue gases volume flow rate results only in about 15 °C growing of exit temperature.

Within the RRD recuperator the surfaces A_1 and A_2 are serving for heat exchange process being realized in parallel flow. Through A_3 surface the second stage of heat exchange performing and of air preheating takes place in counter flow. Therefore the heat balance equations of the flowing media are as follows (see Fig. 5, directions of heat balance consideration — along the axis of the system, upwards from the recuperator's enter cross-section):

— for the «primary» flue gases within the internal tube

$$-c_{fg}\,\dot{V}_{fg,1}\,dT_{fg,1} = k_1\,d_1\,\pi \times$$
$$\times\,(T_{fg,1} - T_{air,1})\,dx\cdot 1/\eta; \qquad (16)$$

— for the inner annular pathway — first line of air (parallel flow)

$$c_{air}\,\dot{V}_{air}\,dT_{air,1} = -\,c_{fg}\,\dot{V}_{fg,1}\,dT_{fg,1} +$$
$$+\,k_2\,d_2\,\pi\,(T_{fg,2} - T_{air,1})\,dx. \qquad (17)$$

The heat balance of the outer flue gases flow the «second» line could be presented in differential form

$$-c_{fg}\,\dot{V}_{fg,2}\,dT_{fg,2} = -\,c_{air}\,\dot{V}_{air}\,dT_{air,2} +$$
$$+\,k_2\,d_2\,\pi\,(T_{fg,2} - T_{air,1})\,dx, \qquad (18)$$

while the same for outer pathway — second line of air (counter flow) — will be expressed by equation

$$-c_{air}\,\dot{V}_{air}\,dT_{air,2} + k_3\,d_3\,\pi \times$$
$$\times\,(T_{fg,2} - T_{air,2})\,dx. \qquad (19)$$

The system of equations can't be solved by direct formulas. It requires an application of numerical technique. In order to simplify the calculation, it was considered, that comparison requires the exit temperatures only. Applying the defining values — mean logarithmic temperature differences — the system of the differential equations (16)–(19) could be substituted by the final equations (20)–(24). The numbers in the lower indexes are related to the pathway (lines, trajectories) of the flue gases regarding to the heat transfer surfaces.

The system has to be solved by iterations technique. As the first step it is expedient to estimate temperature $T_{air,m}$. As its initial value we've used the exit air temperature of BR recuperator (see equation (11) for exit air preheating temperature by co-flow (parallel-current flow) heat exchange case) multiplied by the ratio of heat transfer surfaces (see equation 24).

$$T_{fg,1,out} = T_{fg,0} - \beta_1(\Psi_1 / \eta_1)^{-1}[T_{fg,0} - T_{air,0} - \left(T_{fg\,1,out} - T_{air\,m}\right)] / \ln \frac{T_{fg,0} - T_{air,0}}{T_{fg\,1,out} - T_{air,m}}; \quad (20)$$

$$T_{air,m} = T_{air,0} + \Psi_1(T_{fg,0} - T_{fg\,1,out}) + \beta_2[T_{fg,0} - T_{air,0} - (T_{fg\,2,out} - T_{air,m})] / \ln \frac{T_{fg,0} - T_{air,0}}{T_{fg\,2,out} - T_{air,m}}; \quad (21)$$

$$T_{air,out} = T_{air,m} + \Psi_2(T_{fg,0} - T_{fg\,2,out}) - \eta \cdot \beta_2[T_{fg,0} - T_{air,0} - (T_{fg\,2,out} - T_{air,m})] / \ln \frac{T_{fg,0} - T_{air,0}}{T_{fg\,2,out} - T_{air,m}}; \quad (22)$$

$$T_{air,out} - T_{air,m} = \beta_3[T_{fg,0} - T_{air,out} - (T_{fg\,2,out} - T_{air,m})] / \ln \frac{T_{fg,0} - T_{air,out}}{T_{fg\,2,out} - T_{air,m}}; \quad (23)$$

$$T_{air,m}^{(init)} = \{[\psi \cdot T_{fg,0} + T_{air,0} - \psi\,(T_{fg,0} - T_{air,0}) \cdot \exp\,(-\beta \frac{1+\psi}{\psi})] / (1+\psi)\} \cdot \frac{d_1 + d_2}{d_i}. \quad (24)$$

The last mentioned value meets to ratio of total heat exchange surface for recuperator of RRD type to that of single surface («i») for recuperator of BR type.

It must be taken into account that ratio of surfaces in equation (24) as a first approximation meets to ratio of total heat fluxes transferred in recuperators RRD and BR. The last exceeds the real ratio of intermediate temperature $T_{air,m}$ in RRD recuperator to $T_{air,out}$ in $BR_{parallel}$.

The computation equations do not account the possible effect of alteration the transfer processes by each side of heat exchange surface influencing on convective and radiative constituents like the increased turbulence, difference of coefficients k_1 and k_2, and the modified temperature differences between the media. They compensate partly one another, hence is it acceptable as initial condition.

The calculation with the equations (20), (21) and (23) results the value of $T_{fg,1,out}$, $T_{fg,2,out}$ and $T_{air,out}$. The preliminary date of $T_{air,m}$ by equation (24) can be checked and corrected by equation (22).

The input parameters for the comparative calculation are summarized in the Table 1.

The same values of heat exchange coefficients k_i for each of heat exchange surfaces have been supposed. The calculations have been carried out with two values of k_i = k: 10 and 20 W/(m²·K) (see Table 2).

The data of the Table 2 show, that at k = 10 W/(m²·K) the exit temperature of air preheating by application of RRD type recuperator has been more than doubled in comparison with ordinary type plant. By case of k = 20 W/(m²·K) the exit air temperature will be 90 % higher for RRD recuperator the respective temperature for ordinary recuperators of BR type.

The heat exchange coefficient «k» depends on the heat transfer coefficients from both sides according to the well known equation [10]:

$$k^{-1} = \alpha_{fg}^{-1} + \alpha_{air}^{-1}. \quad (25)$$

Table 1. The input data of the models

Characteristic	$\dot{V}_{air}$	$\dot{V}_{fg}$	$\dot{V}_{fg,1}$	$\dot{V}_{fg,2}$	d_i	d_e	d_1	d_2	d_3	d_4	L	$T_{air,0}$	$T_{fg,0}$	η
Dimension	m³/s	m³/s	m³/s	m³/s	m	m	m	m	m	m	m	K	K	–
Value	0,11	0,17	0,1	0,07	0,43	0,48	0,35	0,41	0,55	0,61	1,5	293	1273	0,9

Table 2. Comparison of characteristics of standard of recuperators BR with the developed one of RRD type

Type of recuperator	k	$T_{air,out}$	$T_{air,m}$	$T_{fg1,out}$	$T_{fg2,out}$	γ_t	γ_q
	W/(m²·K)	K (°C)	K (°C)	K (°C)	K (°C)	–	–
$BR_{parallel}$	10	417 (144)	–	1191 (918)	–	0,126	0,081
$BR_{counter}$	10	418 (145)	–	1190 (917)	–	0,128	0,082
RRD	10	589 (316)	483 (210)	1172 (899)	982 (709)	0,302	0,170
$BR_{parallel}$	20	516 (243)	–	1125 (852)	–	0,227	0,137
$BR_{counter}$	20	520 (247)	–	1123 (850)	–	0,231	0,139
RRD	20	740 (467)	603 (330)	1100 (827)	864 (591)	0,456	0,263

Generally, the velocity of flue gases is limited and it determines the intensity of the whole heat exchange process for the furnaces of moderate level of working temperatures. The RRD type recuperator is supplied with additional internal emitters (inserts) in order to improve the heat transfer coefficient at the flue gases side. In spite of limiting impact of heat transfer at the air side of heat exchage surfaces ($\alpha_{air} < \alpha_{fg}$) the effect of hot side is growing with insreasing the flue gases temperature due increase the temperature head $T_{fg} - T_{air}$. More adequate and reliable data will be defined in the process of trials at the bell type furnace.

The effect of «k» factor by BR recuperator application is somewhat more intensive. Its double value results in about 80 % higher gain of an air preheating exit temperatures (see Table 2) while in case of RRD type heat exchanger the same change of heat ex change coefficients (k = 20 against 10 W/(m^2·K) causes only 50 % gain of $T_{air,out} - T_{air,0}$.

This difference for RRD is bigger than the same for BR by absolute value.

Determination of the actual «k» factors requires an accurate calculation of convective and radiative heat transfer coefficients from both sides. Obviously, these values will be different for each heat exchange surfaces.

The range of value «Ψ» makes about 0.9 by natural gas combustion. While in case of LCV gases using it could be of significantly varied value. For example by blast furnace gas application it makes more than Ψ = 2.0. However, as could be seen from the Fig. 6, the exit air temperature is slightly influenced by Ψ value.

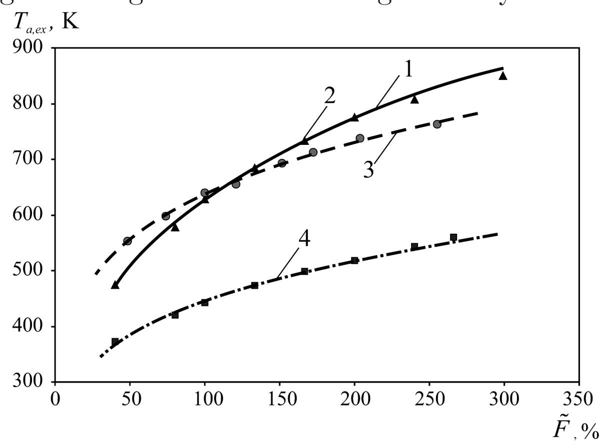


Fig. 8. Dependence of combustion air preheating temperature $T_{a,ex}$, K, on relative value of recuperative heat exchange surface $\tilde{F}$, %: 1 — curve by data of paper [8]; 2 — the dots accordingly our computations for radiative recuperator of RRD type by increase of heat exchange surface due to rise the recuperators height H; 3 — accordingly our computations for radiative recuperator of RRD type (Fig. 2) 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.

3. Numerical analysis of thermal characteristics for RRD type recuperator by alteration the geometry of heat exchanger

In Fig. 8 the impact of extension the heat the heat exchange surface (the relative area $\tilde{F}$) upon an temperature of air preheating $T_{a,ex} = T_{air,out}$ within the recuperator of bell-type furnace is shown in curve 1 [8]. 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) [11]. 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 % [8].

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 [8].

The results obtained in paper [8] are compared with our data. The $T_{a,ex}$ temperatures accordingly our computations are marked as the dots and joined by respective curves (Fig. 8) in dependence on $\tilde{F}$ value influence. The correlations between $\tilde{F}$ value and exit air temperature $T_{a,ex}$ for radiative recuperators of new original design of (RRD type — radiative recuperators of double-passage) [6] (Fig.2).

The following conditions of recuperator's efficiency comparison by variation 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 D_i and H — respectively the recuperators height and diameters of shells; F_0 — area of basic (of one-sided) design heat exchange surface.

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, D_1 — diameters of heat exchange surfaces (see Fig. 2).

In the Fig. 8, 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 (see Fig. 1, a). The recuperator's design is distinguished with parallel (current) flow of combustion products and air flow to be preheated [2]. As could be seen from the Fig. 8 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 ($\widetilde{F}$ = 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 = 0.87 m. Inner diameter of central flue gases channel D_4 = 0.8 m. Width δ of air slot channels is equal to 0.025 m. The recuperators height makes H = 3 m.

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 flow rates 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.

Conclusion

1. Creation of highly efficient recuperators of advanced design has been envisaged in frame of performing the joint European project accordingly EUREKA program. The final purpose of mentioned development makes provision the conditions of high temperature operation of the industrial furnaces by replacement the hydrocarbons for alternative fuels. These heat exchangers are intended for preheating both the combustion air and/or low-calorific gas fuels.

2. The radiative recuperator of basket type has been chosen for further improvement as universal and of low hydraulic resistance design for the furnaces of periodic and continuous operation. For enhancement the power efficiency of recuperator and to increase the transferred resulting heat fluxes the following approaches have been sdudied: increase of heat transfer surface and rise of heat transfer coefficient due affect of intermediate (secondary) emitters arranged within the heat transfer channels for mediums, firstly — for flue gases.

3. The paper presents a simplified calculation technique for prediction the heat transfer process in the basket-type recuperator of new design. This approach has been proposed for analyzing of coaxial-flow system of heat transfer media movement. The results of the calculation have proved, that almost twice increase of air preheat temperature can be achieved by enhancing of the heat exchange surface in new recuperator design. Enhancement of coefficient «k» by application of internal secondary emitters was not considered. The effect of combined account of both factors of increase of an air preheat under application of RRD type recuperator: enhancement of coefficient «k» and of growth of heat exchange surface — influence was predicted by numerical calculations and would be defined through the trials of recuperators installed at industrial furnaces.

4. Recuperator of RRD type provides an increase of combustion air preheat temperature $T_{a,ex}$ in comparison with slot type radiative recuperator. Both balance calculation techniques: differential and averaged engineering approaches show the similar impact of increase the heat exchange surface on increment of $T_{a,ex}$ value in comparison with traditional heat exchangers. For RRD recuperators under consideration the value of mentioned air preheating temperature increase can exceed 200 °C.

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Разработка методики расчета с целью создания высокоэффективного радиационного рекуператора в рамках выполнения программы «EUREKA»

Проект «REPLACE NG» («Замещение природного газа») направлен на разработку и использование радиационного (радиацинно-конвективного) рекуператора новейшей конструкции для утилизации теплоты уходящих газов при сжигании топлив, включая низкокалорийные (LCV) газы, в промышленных печах различного назначения. В соответствии с планом работ предполагалось, что разрабатываемые рекуператоры должны быть пригодны для нагрева воздуха горения и самих низко калорийных газов. Рекуператоры двойного назначения могут использоваться при установке на печах, отапливаемых низкокалорийным газом. Упомянутое оборудование — рекуператор типа РРД — разработано Институтом газа НАНУ (Украина). В соответствии с предварительной программой и графиком работ было намечено привлечение компании «ТЮКИ» (Венгрия) в качестве партнера по проекту на разных этапах, в частности, при испытаниях рекуператора, монтируемого с этой целью на колпаковой печи в цехе холодной прокатки комбината «ИСД ДУНАФЕРР». В ходе испытаний предусмотрено опробовать отопление печи различными технологическими газами металлургических производств (альтернативным топливом). В настоящей работе представлен вывод расчетных зависимостей, обеспечивающих прямое сравнение теплового состояния теплообменника нового типа и радиационных рекуператоров традиционной широко распространенной конструкции. Методика базируется на расчете процессов теплообмена в условиях прямо- и противоточного взаимного движения теплоносителей в каналах. Упрощенные дифференциальные уравнения теплового баланса и теплообмена для разных участков рекуператора были использованы в качестве базовой системы при получении расчетных зависимостей. Благодаря отысканию и применению обобщающих вспомогательных безразмерных параметров расчетная модель обеспечивает возможность анализа влияния коэффициентов теплообмена, массового и объемного потоков теплоносителей и тепловых потоков на результирующее температурное распределение в аппарате в процессе подогрева воздуха. Преимущества нового подхода к проектированию рекуператоров даже в случае пренебрежения ростом интенсивности теплообмена с обеих сторон теплообменной поверхности объясняются существенным увеличением области теплообмена (в случае рекуператора типа РРД) по сравнению с обычными одноходовыми радиационными рекуператорами. Использование вторичных (промежуточных) адиабатных излучателей в каналах для дымовых газов и воздуха (или потоков низкокалорийного газа) обеспечивает дополнительные возможности повышения подогрева воздуха (или низкокалорийного газа). *Библ. 11, рис. 8, табл. 2.*

Ключевые слова: воздух горения, колпаковая печь, коэффициент теплоотдачи, коэффициент теплопередачи, низкокалорийное топливо, противоток, прямоток, продукты сгорания, радиационный рекуператор, температура теплоносителя, тепловой поток.

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Розробка методики розрахунку з метою створення високоефективного радіаційного рекуператора в рамках виконання програми «EUREKA»

Проект «REPLACE NG» («Заміщення природного газу») направлений на розробку та використання радіаційного (радіаційно-конвективного) рекуператора новітньої конструкції для утилізації теплоти викидних газів при спалюванні палив, у тому числі низькокалорійних (LCV) газів, у промислових печах різного призначення. У відповідності до плану робіт передбачалося, що розроблювані рекуператори мають бути придатними до нагріву повітря горіння та самих низькокалорійних газів. Рекуператори подвійного призначення можуть використовуватися при установці на печах, опалюваних низькокалорійним газом. Таке обладнання — рекуператор типу РРД — розроблено Інститутом газу НАНУ (Україна). У відповідності до наміченої програми та графіка робіт було намічено залучення компанії «ТЮКІ» (Угорщина) як партнера по здійсненню проекта на різних етапах, а саме: при випробуваннях рекуператора, змонтованого з цією метою на ковпаковій печі у цеху холодної прокатки комбінату «ІСД ДУНАФЕРР». Під час досліджень передбачено випробувати опалювання печі різними технологічними газами металургійного виробництва (альтернативним паливом). У цій роботі представлено виведення розрахункових залежностей, які забезпечуватимуть пряме порівняння теплового стану теплообмінника нового типу та радіаційних рекуператорів традиційної широко розповсюдженої конструкції. Методика базується на розрахунку процесів теплообміну в умовах прямо- та протиточного взаємного руху теплоносіїв у каналах. Спрощенні диференційні рівняння теплового балансу та теплообміну для різних зон рекуператора були використані як базова система при отриманні розрахункових залежностей. Завдяки відшуканню та застосуванню узагальнюючих допоміжних безрозмірних параметрів розрахункова модель забезпечує можливість аналізу впливу коефіцієнтів теплообміну, масового та об’ємного потоків теплоносіїв та теплових потоків на результуючий температурний розподіл в апараті у процесі підігріву повітря. Переваги нового підходу до проектування рекуператорів, навіть не беручи до уваги зростання інтенсивності теплообміну з обох сторін теплообмінної поверхні, пояснюються суттєвим збільшенням області теплообміну (у разі рекуператора типа РРД) у порівнянні зі звичайними одноходовими радіаційними рекуператорами. Використання вторинних (проміжних) адіабатних випромінювачів у каналах для димових газів та повітря (чи потоків низькокалорійного газу) забезпечує додаткові можливості підвищення підігріву повітря (чи низькокалорійного газу). *Бібл. 11, рис. 8, табл. 2.*

Ключові слова: повітря горіння, ковпакова піч, коефіцієнт тепловіддачі, коефіцієнт теплопередачі, низькокалорійне паливо, протиток, прямоток, продукти згоряння, радіаційний рекуператор, температура теплоносія, тепловий потік.

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Газогорелочные устройства для сжигания биогаза в котлах

На основе проведенных ранее экспериментов впервые выведены критерии подобия для инженерного расчета горелочных устройств для сжигания биогаза, а также его сжигания совместно с природным газом. Установлено, что основное отличие биогаза от природного газа вызвано наличием в его составе более 30 % углекислого газа и его влиянием на плотность, теплотворность смеси газов и нормальную скорость распространения пламени. Показано, что без изменения конструкции и режимов подачи топлива сжигание биогаза в горелочных устройствах, разработанных для сжигания природного газа, практически не возможно. Рассмотрено несколько примеров горелочных устройств для работы на биогазе, рассчитанных для сжигания природного газа. Приведенные данные были положены в основу переоборудования горелочных устройств для совместного и раздельного сжигания биогаза и природного газа. *Библ. 7, рис. 2, табл. 2.*

Ключевые слова: биогаз, горелочные устройства, котлы, критерии подобия.

В последнее время использование биогаза в качестве топлива для промышленных котлов приобретает все большую актуальность. Это вызвано как минимум тремя причинами: 1) высокой стоимостью природного газа; 2) необходимостью очистки городских и промышленных сточных вод, в результате которой как остаточный продукт образуется биогаз; 3) усилением внимания к выбросу парниковых газов, в первую очередь CO_2, CH_4.

Котельный парк Украины насчитывает десятки тысяч котлов коммунального хозяйства, несколько тысяч котлов промышленности и электростанций, большинство которых оснащено горелочными устройствами для сжигания природного газа. Использование биогаза в промышленных котлах в Украине крайне ограничено. При этом биогаз обычно подают в горелочные устройства, которые разработаны для природного газа. Реже применяются горелочные устройства, разработанные для сжигания биогаза, зарубежных фирм.

Рассмотрим возможность использования для сжигания биогаза существующих горелочных устройств, разработанных для сжигания природного газа.

В табл.1 приведены данные Института газа НАН Украины о составе биогаза различного