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PRESSURIZED RECUPERATOR FOR HEAT RECOVERY IN INDUSTRIAL HIGH TEMPERATURE PROCESSES

CIŚNIENIOWY REKUPERATOR DO ODZYSKU CIEPŁA Z PRZEMYSŁOWYCH PROCESÓW WYSOKOTEMPERATUROWYCH

Recuperators and regenerators are important devices for heat recovery systems in technological lines of industrial processes and should have high air preheating temperature, low flow resistance and a long service life. The use of heat recovery systems is particularly important in high-temperature industrial processes (especially in metallurgy) where large amounts of thermal energy are lost to the environment. The article presents the process design for a high efficiency recuperator intended to work at high operating parameters: air pressure up to 1.2 MPa and temperature of heating up to 900°C. The results of thermal and gas-dynamic calculations were based on an algorithm developed for determination of the recuperation process parameters. The proposed technical solution of the recuperator and determined recuperation parameters ensure its operation under maximum temperature conditions.

Keywords: recuperator, heat transfer, flow, high temperature processes

Rekuperatory i regeneratory są ważnymi urządzeniami systemów odzysku ciepła w ciągach technologicznych procesów przemysłowych i powinny charakteryzować się wysoką temperaturą podgrzewania powietrza, niewielkimi oporami przepływu, a także długim czasem eksploatacji. Stosowanie układów do odzysku ciepła ma szczególne znaczenie w wysokotemperaturowych procesach przemysłowych (zwłaszcza w hutnictwie), gdzie tracone są do otoczenia duże ilości energii cieplej. W artykule zaprezentowano projekt procesowy wysokosprawnego rekuperatora przeznaczonego do działania przy wysokich parametrach roboczych: ciśnienia powietrza do 1.2 MPa i temperatury podgrzania do 900°C. Wyniki obliczeń cieplnych i gazodynamicznych uzyskano w oparciu o opracowany algorytm do wyznaczania parametrów procesowych rekuperacji. Zaproponowane rozwiązanie techniczne rekuperatora i wyznaczone parametry rekuperacji umożliwiają jego działanie w maksymalnych warunkach termicznych.

1. Introduction

In recent years, a systematic growth of energy consumption has been observed, which results in increasing use of fossil fuels. In industrial processes, particularly in high-temperature metallurgical ones, large amounts of energy are lost. Advanced designs of industrial furnaces contribute to improvement of production quality and reduction in energy consumption. To achieve this, more efficient recuperators and regenerators, intended to preheat combustion air or low calorific gases, are utilised. They mostly form the central part, built within the flue, or they are installed in exhaust gas extractors immediately at the burners. Currently used designs of tubular recuperators ensure that the combustion air is heated to about 400°C despite a far higher potential resulting from the temperature of exhaust gases that leave the furnace chamber (800–1300°C). This potential can also be applied for production of electrical energy in steam systems with the Rankine cycle [1] or for a solution with a gas turbine (see Fig. 1) [2, 3]. Leaving the furnace chamber, the exhaust gases lose heat in the recuperator where compressed air, delivered via the turbogenerator compressor, is heated. The heated air reaches the gas turbine and, after expansion, flows through the heat recovery recuperator. The turbine contains a combustion chamber where, if proper amounts of natural gas are added, the gas inlet temperature can be raised and its operating parameters can be stabilised.

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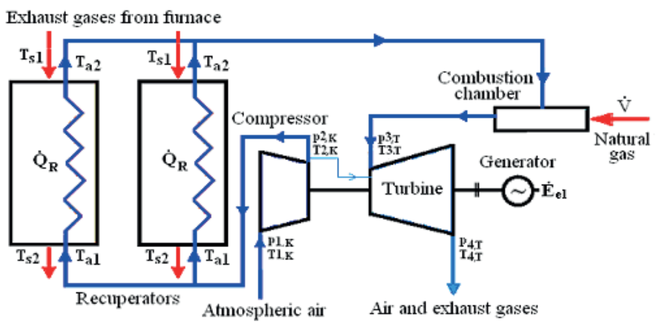

Fig. 1. A schematic diagram of the energy recovery system

Modern designs of gas turbines ensure a high thermodynamic efficiency; their application for waste heat recuperation in high-temperature industrial furnaces allows for production of large amounts of electrical energy [3]. Efficiency of exhaust gas energy conversion into electrical energy in gas turbine systems depends on the temperature and pressure of the gaseous medium that is delivered to the turbine. These parameters are determined by design and material characteristics of the heat recuperator and the pipelines. Therefore, a recuperator that operates within the energy recovery system must be characterised by high air preheating temperature, low flow resistance and a long service life. These parameters depend on the applied recuperator design solution and a proper selection of its construction materials.

In the paper, results of calculations for a tubular recuperator (Fig. 2), designed for the energy recovery system of a metallurgical furnace (Fig. 1) where compressed air (12 bar) is heated to reach above 700°C [2], are presented. The analysed energy recovery system consists of two furnaces with recuperators which cooperate with a turbine unit of specific operating parameters (Fig. 1). The recuperator (360 m^2), presented in Fig. 2, is a multicomponent system consisting of six consecutive, geometrically identical modules built within the flue where the air flow from the compressor is separated into two streams.

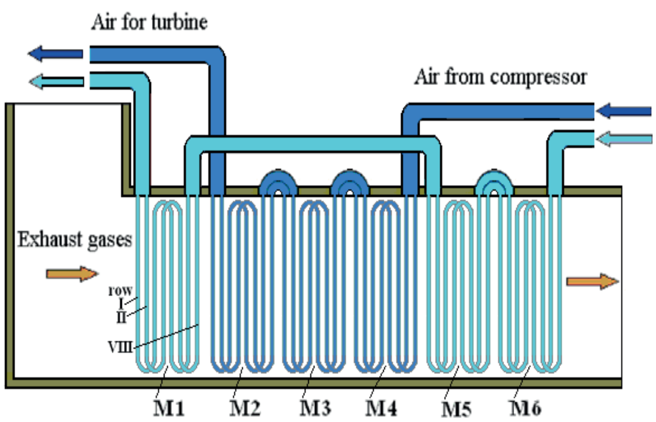

Fig. 2. A schematic diagram of the six-module recuperation system

Considering the first flow (M6+M5+M1), air from the compressor enters the M6 module and then, via the M5, reaches the M1 module. As for the other flow (M4+M3+M2), air from the compressor enters the M4 module and then, via the M3, reaches the M2 module. Numbers of the modules grow according to the direction of exhaust gas flow from the furnace to the stack. Each module contains 28 identical tubes that are bent to form a shape of serially linked UU letters that are arranged in an adjustable system. In the analysed recuperation system of a six-module recuperator with a linking tube system, cross-flow of exhaust gases and air as well as, but only in a small section, countercurrent flow are observed. The exhaust gases flow across eight rows of tubes in each module.

2. Characteristics of the system of high-temperature furnace and gas turbine unit

The power of generated electrical energy, $\dot{E}_{El}$, that depends on the operating parameters of applied turbine unit, can be calculated using the following equation:

$$\dot{E}_{el} = \left[\dot{m}_T c_{p,T} T_{3,T}\left(1-\frac{T_{4,T}}{T_{3,T}}\right)\eta_{i,T}\eta_{m,T} - \dot{m}_K c_{p,K} T_{1,K}\left(\frac{T_{2,K}}{T_{1,K}}-1\right)\frac{1}{\eta_{i,K}\eta_{m,K}}\right]\eta_G \quad (1)$$

where: $\dot{m}_T$ - gas mass flow in the turbine, kg s^{-1}; $l_{i,T}$ - specific work of the turbine, kJ kg^{-1}; η_{mT} - mechanical efficiency of the turbine; $\dot{m}_K$ - mass flow of gases in the compressor, kg s^{-1}; $l_{i,K}$ - specific work of the compressor, kJ kg^{-1}; η_{mK} - mechanical efficiency of the compressor; η_G - efficiency of the generator; $c_{p,T}$ - mean specific heat capacity of the air in the turbine, kJ (kg K)$^{-1}$; $T_{3,T}$ - temperature of the air before the turbine, K; $T_{4,T}$ - temperature of the air behind the turbine, K; η_{iT} - internal efficiency of the turbine; $c_{p,K}$ - mean specific heat capacity of the air in the compressor, kJ (kg K)$^{-1}$; $T_{1,K}$ - temperature of the air before the compressor, K ; $T_{2,K}$ - temperature of the air behind the compressor, K; η_{iK} - internal efficiency of the compressor.

Electrical power, generated by the energy recovery system, depends on the parameters of the gas flow that enters the turbine. When natural gas is not added to the turbine combustion chamber, it is replaced by compressed air. The rate of heat flow, $\dot{Q}_R$, transferred without loss from the exhaust gases to the preheated air, is determined by the following balance equation:

$$\dot{Q}_R = \dot{C}_s\left(T_{s1} - T_{s2}\right) = \dot{C}_a\left(T_{a2} - T_{a1}\right), \quad (2)$$

where: $\dot{C}_s$ - heat capacity rate of the exhaust gases, W K^{-1}; T_{s1}, T_{s2} - temperatures of the exhaust gases at the inlet and outlet of the recuperator, K; $\dot{C}_a$ - heat capacity rate of the air, W K^{-1}; T_{a1}, T_{a2} - temperatures of the air at the inlet and outlet of the recuperator, K. The heat flow rate, $\dot{Q}_R$, for the A surface where the heat transfer coefficient (k) is constant, can also be calculated using the following equation:

$$\dot{Q}_R = k\,A\left(T_s - T_a\right)_m , \quad (3)$$

where $(T_s\text{-}T_a)_m$ is a mean difference between the exhaust gas and air temperatures on the A surface.

The addition of natural gas results in the increase of temperature of gases that are delivered to the $T_{3,T}$ turbine and in enhancement of generated electrical power. An analysis of the operation of energy recovery system with the natural gas addition is presented in the paper [3] where exhaust gases (833°C) that leave the furnace heat the air (14.5 kg/s, 12 bar) in the recuperator from the initial temperature of 362°C to 700°C. In this system, $Q_{Rek} = E_T$ and the electrical energy, E_{el}, is generated from about 18 % of the heat delivered to the recuperator (Fig. 3). Addition of 400 m^3n/h natural gas results in the increased temperature of inlet gases in the $T_{3,T}$ turbine, and the produced electrical energy accounts for about 48% of the enthalpy of these gases, E_T.

Efficiency of the natural gas energy use for production of additional electrical energy, $\eta_{GZ\text{-}Eel}$, can be expressed as follows [3]:

$$\eta_{GZ-Eel} = \frac{\dot{E}_{el,GZ} - \dot{E}_{el,0}}{\dot{V}_{GZ}\, W_{d,GZ}}, \tag{4}$$

where: $\dot{V}_{GZ}$ - the rate of natural gas flow delivered to the combustion chamber of gas turbine, $m^3_n\, s^{-1}$; $W_{d,CZ}$ - the calorific value of natural gas, J m^{-3}n; $\dot{E}_{el,GZ}$ - electrical power generated with addition of $\dot{V}_{GZ}$ of natural gas, W; $\dot{E}_{el,0}$ - electrical power generated without addition of natural gas, W.

For the analysed flow of added natural gas, $\dot{V}_{GZ}$, efficiency of the natural gas energy use is higher than 42%, which is an attractive option for gas turbines.

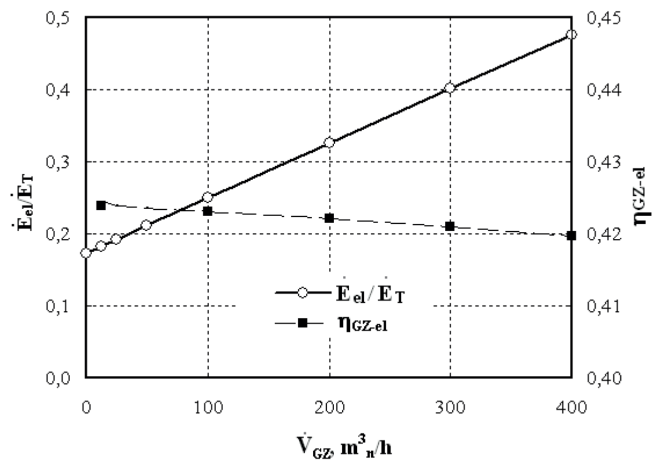

Fig. 3. The electrical energy flow rate and $\eta_{GZ\text{-}Eel}$ versus the natural gas flow

3. Methodology of recuperator calculations and a discussion of the results

Considering economical benefits, high enthalpy of gases entering the turbine should be achieved without any addition of the natural gas, $E_T = Q_R$. This requires such organisation of operating conditions of the recuperator system to obtain a highest possible temperature of the preheated air, but to maintain a permissible temperature of the tube walls. Due to the economical aspects, various materials should be used for the manufacture of individual sets of tubes (modules) of the recuperator. Therefore, the recuperator calculations in order to determine the maximum temperature of tube walls for the specific modules of the recuperation system are necessary.

Final temperatures of the exhaust gases, T_{s2}, and the air, T_{a2}, in the entire recuperator or its part (separate computational items) can be calculated if non-dimensional temperature parameters, Φ and P, are known. They are determined as follows:

$$\Phi = \frac{T_{s1} - T_{s2}}{T_{s1} - T_{a1}}, \tag{5}$$

$$P = \frac{T_{a2} - T_{a1}}{T_{s1} - T_{a1}}. \tag{6}$$

The temperature parameters, Φ and P, for the heat cross-flow, can be determined according to the solution proposed by W. Nusselt [4, 5]. Considering the literature relationships [6-8] used for determination of heat transfer efficiency, temperature functions (Φ and P) for various conditions of media flow were developed [9-14]. The functions can be used for calculations of exhaust gas and air temperatures in tubular recuperators: for the whole A surface or its separate part.

The temperature parameters are functions of Φf(a,b) and P=f(a,b) of the non-dimensional thermal parameters:

$$a = \frac{k\,A}{\dot{C}_s}, \tag{7}$$

$$b = \frac{k\,A}{\dot{C}_a}. \tag{8}$$

Considering the recuperator balance equation, relations of these parameters are as follows: $R = a/b = C_a/C_s = \Phi/P$; hence, $P = \Phi/R = \Phi\, b/a$ and $\Phi = P\,R = P\, a/b$.

The analysed recuperator system (Fig. 2) consists of six linked tube sets (modules), being separate computational items of different thermal parameters: a and b. To determine these parameters, the heat transfer coefficient, k, for the A surface of each module must be determined. This can be done by calculating the convective heat transfer coefficients: α_s for the exhaust gases, α_a for the air, as well as thermal conductivity, λ_m, of the wall that separates these media. Typically, tube surfaces in industrial furnace recuperators are always contaminated to some degree. When the A surface is contaminated by a deposit layer, its thickness (g_o) and thermal conductivity (λ_o) must be determined. For the tube wall, the heat transfer coefficient, k, can be calculated with the use of the following equation:

$$\frac{1}{k} = \frac{1}{\alpha_a \frac{A_a}{A}} + \frac{s_m}{\lambda_m \frac{A_m}{A}} + \frac{g_o}{\lambda_o \frac{A_o}{A}} + \frac{1}{\alpha_s \frac{A_s}{A}}, \tag{9}$$

where A is the computational surface area of the recuperator, A_m and A_o are mean surface areas of the metal wall and the wall with a deposit layer, while A_a and A_s are internal and external surface areas of the tube walls.

To calculate the coefficient (α_a) of convective heat transfer from the tube wall to the preheated air, the Dittus-Boelter equation was used. It allows for determining the Nusselt number, $Nu_a=\alpha_a d_w/\lambda_a$, for the air flowing in the tube, applying the corrective Michiejew equation [9]. To calculate the convective heat transfer coefficient for the exhaust gases, α_s, the flow of heat delivered to the recuperator surface, A, through: convection (q_{sk}) as well as exhaust gas radiation (q_{sr}) and radiation of the fuel walls (q_{wr}), must be determined [12]. When the convective heat transfer coefficients for the exhausted gases (α_s) and the air (α_a) are determined, temperature of the wall that separates both media can be calculated. Assuming that the metal wall that separates the exhaust gases and the air is not contaminated by deposits as well as when its insignificant thermal resistance is neglected, its temperature, $T_{śc}$, can be calculated:

$$T_{śc} = \frac{T_s \alpha_s + T_a \alpha_a}{\alpha_s + \alpha_a}. \tag{10}$$

Calculations for the energy recovery system (Fig. 1) were performed with the use of software where computational algorithms for recuperators (presented in papers [10-16]) are applied. It was assumed that the air mass flows in all tubes of a specific module were balanced. More complicated procedures occur during energy recovery at other metallurgical processes [17,18]. Calculations of the temperature distributions for exhaust gases and air in the module tubes were performed according to the methodology presented in the paper [14]. Thermal resistance of the deposit layer was determined based on the equations presented in the paper [13]. Air pressure loss in the recuperation system was calculated for non-isothermal flow in the tubes. Numbers of local resistances were determined using the literature data as well as results of gasodynamic modelling of physical objects and mathematical CFD methods [2, 11, 15, 19]. The subjects of simulation were variants of the energy recovery system operation without addition of natural gas, with the air flow of 14.5 kg/s, 12 bar, for three temperatures of exhaust gases leaving the furnace: T_{s1}= 850°C, 900°C and 950°C. The calculation results, presented in Figs. 4 and 5, suggest that with the exhaust gas temperature of T_{s1}= 850°C, the air will be heated to T_{a2}=733°C, which allows for generation of about 1.16 MW of electrical power, while for T_{s1}= 900°C, it will be T_{a2}=767°C and 1.39 MW as well as for T_{s1}= 950°C: T_{a2}=801°C and 1.62 MW.

The temperature of preheated air, T_{a2}, is markedly affected by thermal resistance of the deposit layer on the recuperator tube surfaces as it causes the temperature reduction of about 50 K, which results in the decrease in generated electrical power of approx. 0.4 MW.

Results of the calculations of exhaust gas and air temperatures in the recuperation system are presented in Figs. 6, 8, 10. In the analysed three calculation cases, the outlet air temperature of the M1 module is higher than that of the M2 module – for the exhaust gas temperature of T_{s1}= 850°C, they are: T_{a2M1}= 708°C and T_{a2M2}= 722°C; for T_{s1}= 900°C: T_{a2M1}= 738°C and T_{a2M2}=754°C; for T_{s1}=950°C: T_{a2M1}=767°C and T_{a2M2}=785°C. The wall temperature distributions along the tube length in all modules are presented in Figs. 7, 9, 11.

The calculated maximum tube wall temperatures in individual modules are listed in Table 1. It was demonstrated that within the temperature range of T_{s1}= 850°C to T_{s1}= 950°C for the exhaust gases entering the recuperation system, the values of tube wall maximum temperature in individual modules are as follows: Module 1 - T_{Rek}= 767°C ÷ 861°C, Module 2 - T_{Rek}= 758°C ÷ 838°C, Module 3 - T_{Rek}= 707°C ÷ 780°C, Module 4 - T_{Rek}= 622°C ÷ 688°C, Module 5 - T_{Rek}= 637°C ÷ 680°C, Module 6 - T_{Rek}= 570°C ÷ 609°C.

TABLE 1

Maximum temperatures of the tube walls, T_R

Module	Ts1 = 950oC		Ts1 = 900oC		Ts1 = 850oC	
	Row 1	Row 2	Row 1	Row 2	Row 1	Row 2
1	861 oC	844 oC	822 oC	808 oC	779 oC	767 oC
2	838 oC	827 oC	804 oC	795 oC	766 oC	758 oC
3	780 oC	763 oC	752 oC	737 oC	720 oC	707 oC
4	688 oC	660 oC	667 oC	643 oC	643 oC	622 oC
5	680 oC	668 oC	665 oC	655 oC	646 oC	637 oC
6	609 oC	590 oC	599 oC	582 oC	585 oC	570 oC

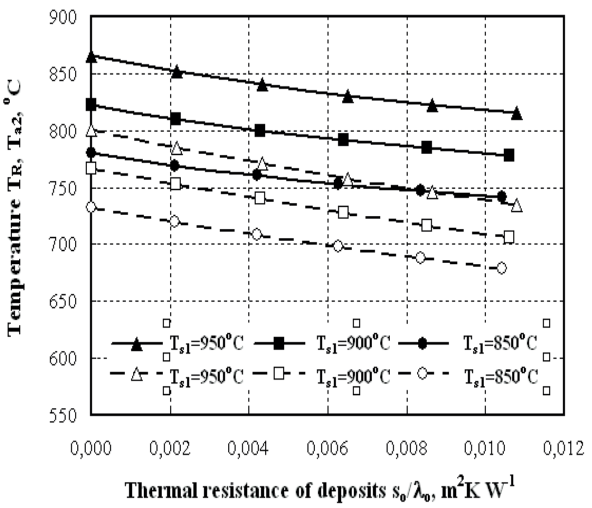

Fig. 4. Temperatures of the tube wall, TR (solid line), and the air, Ta2 (dotted line), versus thermal resistance of the deposit layer on the tubes

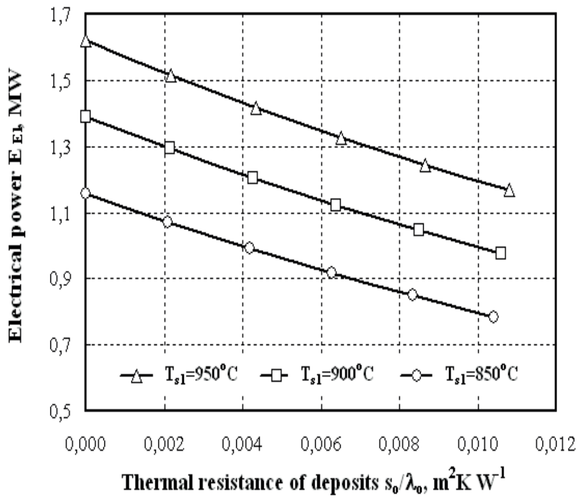

Fig. 5. Electrical power versus thermal resistance of the deposit layer on the tubes

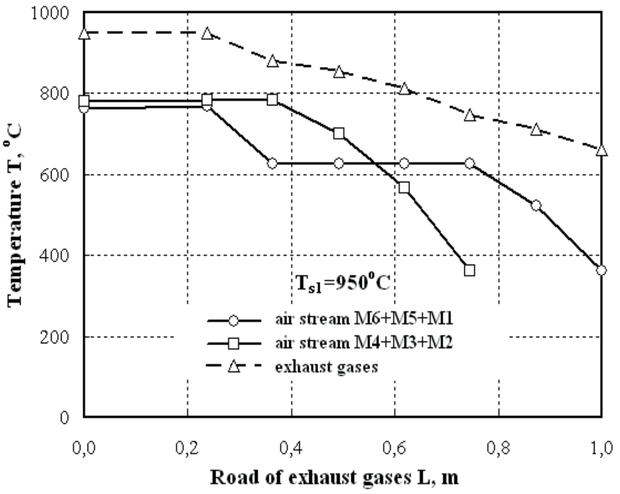

Fig. 6. Temperatures of exhaust gases and air in the recuperation system for T_{s1}=950°C

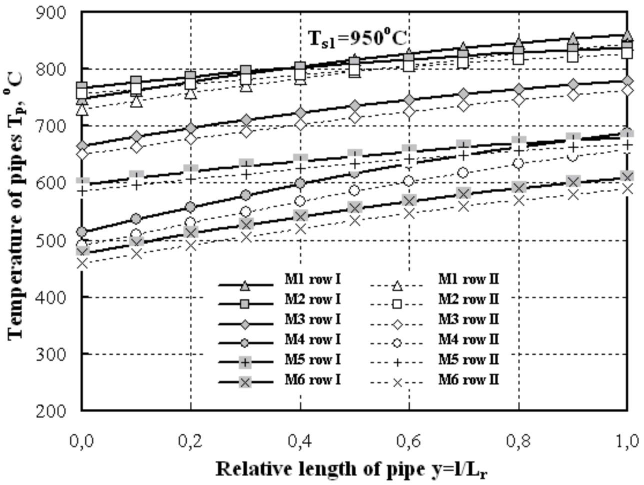

Fig. 7. The tube wall temperatures in rows 1 and 2 of the modules for the exhaust gas temperature of Ts1=950oC

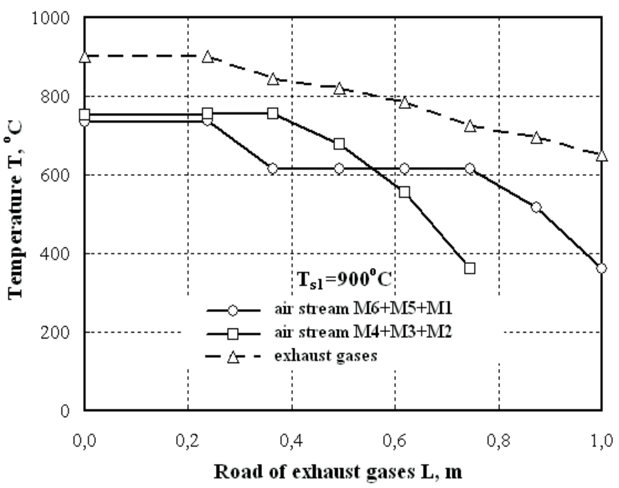

Fig. 8. Temperatures of exhaust gases and air in the recuperation system for Ts1=900oC

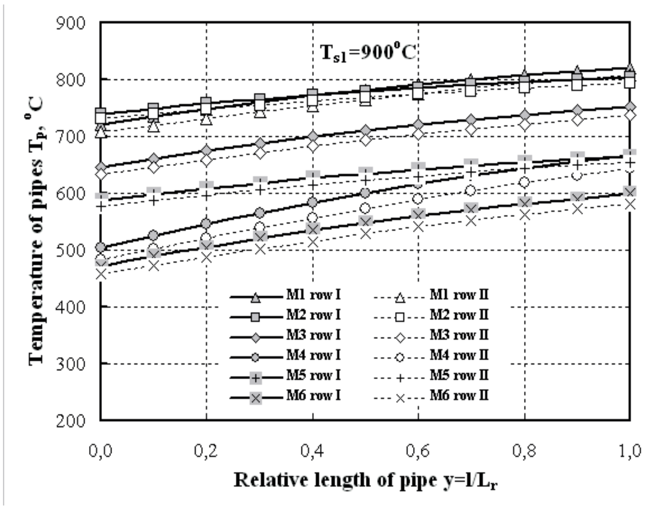

Fig. 9. The tube wall temperatures in rows 1 and 2 of the modules for the exhaust gas temperature of Ts1=900oC

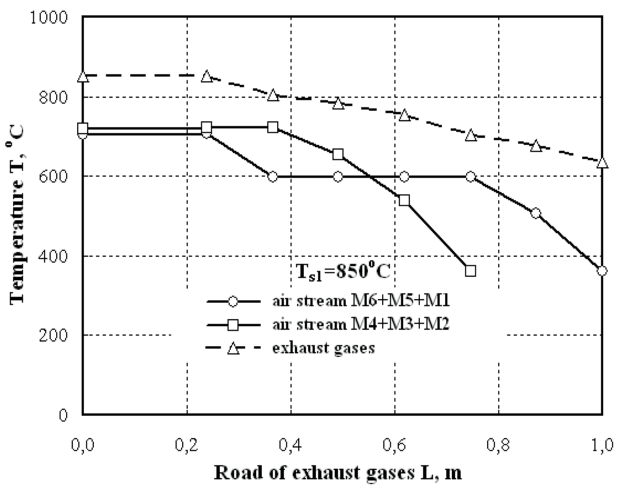

Fig. 10. Temperatures of exhaust gases and air in the recuperation system for Ts1=850oC

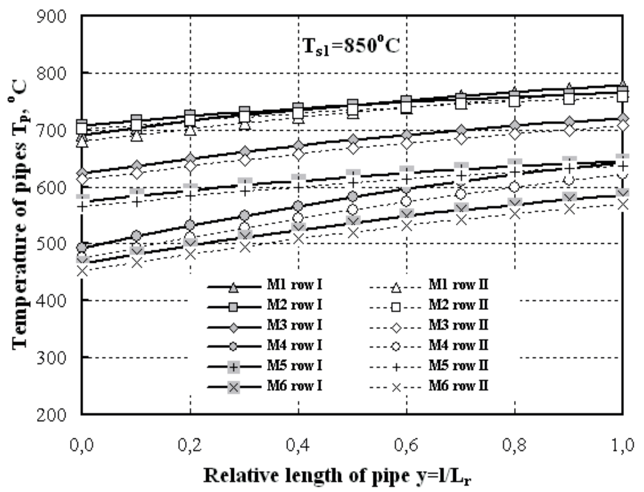

Fig. 11. The tube wall temperatures in rows 1 and 2 of the modules for the exhaust gas temperature of Ts1=850oC

4. Conclusions

The analysis of calculations for the energy recovery system shows that the generated electrical power, $\dot{E}_{el}$, significantly depends on the temperature of compressed air, preheated in the recuperator, which is limited by the maximum permissible temperature of the tube walls, T_R. When the tubes are not contaminated by any deposit, electrical power generated at T_R=750°C is about $\dot{E}_{el}$ =1.0 MW and increases to approx. $\dot{E}_{el}$ =1.6 MW at T_R=850°C. To obtain good energy effects with the energy recovery system, optimal selection of materials for recuperator tubes that are adjusted to operate under extreme thermal conditions is required.

Increased thermal resistance of deposits within the Figs. 4 and 5 range results in about 50 K reduction of temperature of compressed air that enters the turbine and in the decrease in generated electrical power of approx. 0.4 MW. To obtain the assumed effects of energy recovery system application, development of an effective cleaning system for the tube surfaces is necessary.

For the exhaust gases that leave the furnace, within the temperature range of 850°C to 950°C, the tube wall temperature in the entire recuperation system can reach the values of 570°C to 860°C. If during determination of strength parameters of the materials in individual modules, such a high variety of wall temperature is considered, the costs of recuperation system will be markedly reduced.

The method of module linking in the recuperation system (Fig. 2) ensures that the final temperature and pressure loss of the M6+M5+M1 and M4+M3+M2 air flows are well levelled. Considering air pressure at the compressor outlet, the loss is about 1.35% of its value. In the analysed computational cases, the temperature of air heated in the recuperation system without tube deposit is T_{a2}= 715°C – 776°C. Air temperature at the M1 module outlet is only 14 K to 18 K higher than that in the M2 module, which is beneficial for solutions of compensation of pipeline temperature elongation.

To obtain possible best effects of electrical energy production, mathematical calculation simulations are necessary for the selection of turbine unit with operating parameters that are optimally adjusted to cooperate with the furnace and the recuperation system.

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