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MATHEMATICAL AND CFD MODELING OF THERMAL PROCESSES IN THE RADIANT TUBE BY NATURAL GAS COMBUSTION

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The working process within the radiant tube (RT) is analysed by means of CFD modelling. Influence of the assembly of combustion components input into the GT (of burner facility design) and of operation characteristics of fuel combustion has been analysed. Consideration of the 3D transport processes including momentum (aerodynamics), mass- and heat transfer (including radiative heat exchange with account of selectivity of gas medium radiation), on the one hand, and of macrokinetics of fuel mixture combustion (by 20 components), on the other hand, as well as of macrokinetics of NO_x formation has been used as the mathematical model of the combustion / conjugate heat exchange constituents within GT. A turbulence impact on combustion process is taken into account. The boundary conditions statement and the computation grid formation are of the great importance. To minimize the estimated time and owing to axial symmetry of the sectors of GT's circle, the quadrant (the sector of 90° angle), limited by outer cylindrical surface of GT, by two radial planes and the end face surfaces. It has been stated, that the flue gases recirculation to the flame root makes the most important factor influencing the power efficiency of GT and temperature pattern uniformity by GT length.
Basing upon analysis of some design schemes of the input of fuel and oxidant, the best of the last providing the excellent GT power characteristics as well as available NO_x issue, have been recommended.

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МАТЕМАТИЧЕСКОЕ И CFD–МОДЕЛИРОВАНИЕ ТЕПЛОВЫХ ПРОЦЕССОВ В РАДИАЦИОННОЙ ТРУБЕ ПРИ СЖИГАНИИ ПРИРОДНОГО ГАЗА

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С использованием процедур CFD-моделирования анализируется рабочий процесс внутри радиационной трубы (РТ) в зависимости от конструкции узла ввода компонентов горения в трубу (от конструкции горелочных устройств) и режимных характеристик процесса сжигания топлива. Используется 3-мерная (3D) схематизация процессов переноса импульса (аэродинамика), массы и теплоты (в т.ч. лучистого обмена с учетом селективности излучения газовой среды), с одной стороны, макрокинетики горения топливной смеси (по 20 компонентам), с другой, а также макрокинетики образования NO_x. Учитывается влияние турбулентности на процесс горения.
Важнейшее значение имеет постановка граничных условий и формирование расчетной сетки. С целью сокращения времени счета и благодаря осевой симметрии секторов РТ в качестве объекта выбран единичный квадрант (сектор с углами 90°), ограниченный наружной цилиндрической поверхностью РТ, двумя радиальными плоскостями и торцевыми поверхностями.
Установлено, что важнейшим фактором, определяющим энергетическую эффективность РТ и равномерность температурного поля по ее длине, является рециркуляция продуктов сгорания к корню факела.
На основе анализа нескольких конструктивных схем систем ввода топлива и окислителя рекомендованы конструкции, обеспечивающие наилучшие энергетические характеристики РТ и минимальный выход NO_x.

Nomenclature and main notations

a	– absorptivity;
B_f, kg/s	– mass fuel consumption;
Ef_F, %	– efficiency of fuel heat utilization released by combustion: *by nominal GT heat capacity* $Ef_F = 100\ Q_{use}/Q_{nom}$
Ef_H, %	– efficiency of total heat input (including heat released by combustion Q_{comb} and (air + gas) preheating Q_a+Q_f) utilization. The total heat input is equal to difference between excess total enthalpies of initial fuel-oxidant mixture and that of combustion products (CP) (both taken at standard temperature) *by nominal RT capacity* $Ef_H = 100\ Q_{use}/(Q_{nom} + Q_a + Q_f)$
F, F_s, m^2	– cross section and side surface of the channel under consideration;
$\tilde{k}$	– kinetic energy of turbulence (see below);
$\dot{m}$, kg/s	– mass flow rate;
Q_{use}, W	– useful (for ingots to be preheated) heat emitted by RT within the furnace per time unit, W;
Q_{comb}, W	– heat released by fuel combustion in the cold air-oxidant per time unit;
Q_{nom}, W	– nominal combustion heat per time unit;
r	– recirculation ratio, is defined by relation between ejected back flow (ingoing from inner tube (recirculation insert)into the direct flow within the annular channel between outer and inner tubes) to the initial fuel-air flow (or to the CP flow at the exit of RT). If $r < 0$ (sign (-)), it means that the direct flow of the gases from annular channel is ingoing into the recirculation insert;
p_{st}, Pa	– static pressure;
T, K	– temperature;
w, m/s	– local flow velocity;
w_{in}, w_{en}, m/s	– averaged velocity from standpoint of (momentum by mass flow rate for the injecting and ejected flows;
[NO_x], ppm	– volume NO_x concentration;
x, y, z	– coordinate axes;
ΔH_a	– surplus (excess) enthalpy of the oxidant (combustion air) flow rate under preheated air initial temperature T_a ;
ΔH_f	– surplus (excess) enthalpy of fuel (natural gas) flow rate under initial temperature T_f;
$\Delta I_a, \Delta I_f$, J/kg	– specific total enthalpy of the mass unit of oxidant (combustion air) and fuel (natural gas) – correspondingly;
ε	– emissivity of emitting surface;
$\tilde{\varepsilon}$	– dissipation rate of kinetic energy;

λ	– oxidant (combustion air) excess factor;
μ	– discharge coefficient of the opening;
σ_0 =	$5.67 \cdot 10^{-8}$ W/($m^2 \cdot K^4$) – constant of radiation (Stefan–Boltzmann's constant) for absolute black body;
ρ, kg/m^3	– medium density;
$\frac{\rho\tilde{\varepsilon}}{\tilde{k}}, \frac{kg}{m^3 \cdot s}$	– characteristic value for account the turbulence influence on combustion rate accordingly EBU (Eddy-Break Up) model;
Ω_{st}	– mass stoichiometric ratio;

Subscripts

a	– air;
f	– fuel;
F	– combustion system as a while;
en	– related to the jets of combustion products, recirculating from central to annular channel;
in	– related to the jets of fuel and oxidant introduced into combustion chamber located in the annular space between Emitting and Recirculation tubes;
ing	– material to be preheated (ingots);
RT	– radiant tube (on the whole and for the outer surface);
T	– at theoretical combustion temperature.

1. Thermal and environmental demands to Radiant Tubes designs and operation. The problems to be solved in accordance with the tasks.

Radiant tubes (RT) are seemed to make one of the most popular types of the heating systems in the steel and alloy branches of industry as well as by non-metals thermal treatment [1]. RT mainly is intended for indirect heating of the ingots within heat treatment furnaces by condition of provision the neutral or special gas atmosphere inside the furnace space. The following requirements should be brought to any design of the RT of various types: single – ended, of U-, P-, W-shape:

- ***high efficiency*** (the biggest "available heat" [2, 3]). The problem is solved by means of use of advanced combustion and heat recovery systems;
- ***high uniformity of temperature pattern*** along (by the length) surface; is provided by proper organization of combustion and fluid dynamics processes as well as by use of advanced control combustion operation system. For example *on/off or pulse* firing system [2-5] is that being mostly suited to the requirements;
- ***low-emission combustion system***. Minimum exit NO_x and CO fraction in the combustion products is reached by means of GT's special design and operation regime [1, 2, 4-7];
- ***acceptable values of the combustion components pressure at the tube (GT) inlets as well as the pressure drop*** by the GT length.

Glow Tubes represent the variety of radiant tubes and make the new type of heaters, intended for substitution the electric heaters for gas fuel ones in the fuel furnaces. Glow Tube (GT) represents the Radiant Tube of small cross-section size.

The Glow Tube submitted for consideration as the single – ended radiant tube, the characteristic properties making the small absolute sizes and great ratio of the length to diameter.

Mathematical modeling and numerical simulation has been performed for option the proper GT design and its geometry optimization.

The main attention has been paid to computation of main GT's characteristics determining serviceability of the GT design under consideration in respect of operation requirements and fitness to the last (limit temperatures of the GT assemblies including emitting and recirculation tubes – firstly from the thermal resistance standpoint) and usefulness of GT from the standpoint of energetical efficiency and of ability to ensure uniformity of thermal treatment characteristics of the pieces in the furnace.

GT under consideration has been designed by Gas Technology Institute (GTI, IL, USA) as two coaxial tubes, outer of them being facility, emitting to the ingots to be preaheated within the furnace, and inner one – as the recirculation tube (Figures 1 and 2). Mentioned figures will make clear the detailed design of Radiant Tubes of GT version intended for simulation.

In principle the presented design of combustion chamber arranged between the Emitting and Recirculation tubes may be related to the number of FLOX (flameless combustion) system. The cause makes in delivering the separate fuel (natural gas) and oxidant (air) jets into incandescent space filled in with combustion products for ballasting the burning mixture and to change combustion dynamics [7,8].

2. CFD Modelling by Optimization of GT design and operation

CFD modelling of the combustion processes within the furnaces and the burner devices represents a strong means of mathematical analysis of the transport processes including heat and mass transfer in conditions of chemical transformations or geterogenous (multi-phase) interaction [9, 10].

Computational Fluid Dynamics (CFD) is a branch of computational physics that involves the numerical solution of the governing partial differential equations that describe the transport of mass, momentum and energy for a fluid (gas medium). By computations the equations are discretized over small, finite-sized control volumes, which collectively make up the entire domain where the fluid flow is to be solved. The discretization process converts the partial differential equations into nonlinear, algebraic equations. One equation is formed for each conserved variable for each control volume. Mass, momentum and energy must be conserved over each control volume and, therefore, over the entire domain. The system of coupled, nonlinear algebraic equations is then solved iteratively, subject to the specified boundary conditions, to obtain the gas velocity, density, temperature and pressure at each control volume, providing a local representation of the flow physics everywhere in the modelled air space. There are various numerical methods available to solve the equation set (see the standard texts on computational fluid dynamics modelling and numerical methods for more details).

Combustion processes are of especial significance area to be predicted by means of CFD procedures. Mathematical presentation and analysis of burner designs and operation conditions need the system of conservation equations in more general form being complicated by account of chemical thermodynamics (quasi-equilibrium approach) and chemical kinetics. The combustion kinetics interacting with turbulence need account by calculations the combustion systems.

The CFD simulation is based on the Reynolds averaged Navier-Stokes (RANS) equations and the $\tilde{k}-\tilde{\varepsilon}$ turbulence model with standard values for the constants. Additional transport equations are solved for the calculation of the energy transfer and species distribution. The near wall flow is described by the standard wall function.

The mathematical models of the working process, performed in the Radiant Tubes of special purpose are composed of the following constituents of calculation: combustion macrokinetics, transport processes (fluid dynamics, mass - and heat transfer) as well as NO_x formation block considering as the postprocessor. The models have been developed and realized in three – dimensional (3D) statement by means of original or/and standard CFD (Computational Fluid Dynamics) codes.

Table1. Main models have been used by CFD modelling.

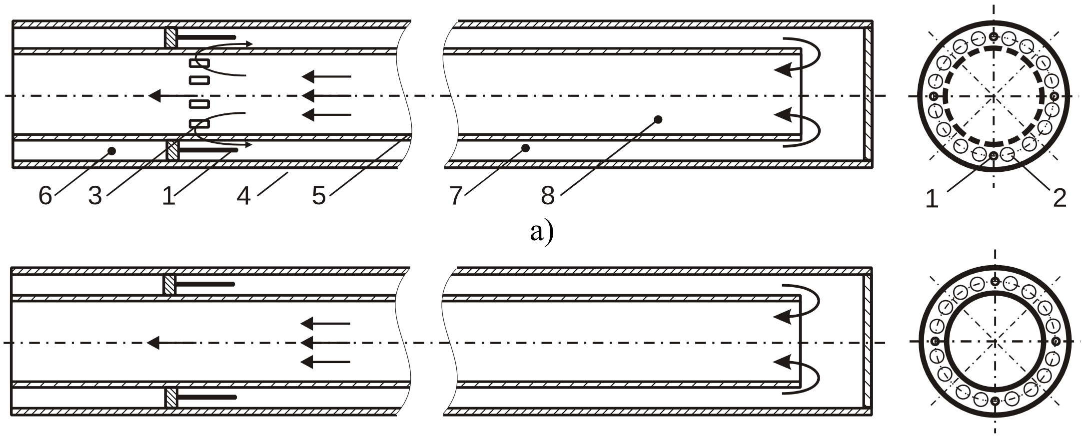
Process (model constituent)	PHOENICS code	FLUENT code
Turbulent combustion	– RNG $\tilde{k}-\tilde{\varepsilon}$ model; – Two-stage EBU model (Finite Rate Chemistry): $CH_4 + \frac{3}{2}O_2 \rightarrow CO + 2H_2O$ $CO + \frac{1}{2}O_2 \rightarrow CO_2$	Realizable $\tilde{k}-\tilde{\varepsilon}$ model for turbulent viscosity – mean square vorticity equation; – PDF equation to account interaction between chemistry and turbulence; – computation by assumption the combustion products mixture being composed of 20 components in equilibrium or near-equilibrium state.
Radiative heat transfer	Immersol code	Method of discrete ordinates

The following set of the Radiant Tube characteristics and the patterns of values have been under determination by CFD modelling: velocity vectors and the by-coordinate components, static pressure, gas phase components composition and temperatures, turbulence parameters $\tilde{k}$ and $\tilde{\varepsilon}$, turbulent viscosity, effective viscosity and ratio, vorticity magnitude and its by – coordinate components.

The boundary conditions statement and forming of the computation grid is of great importance by numerical calculation in framework of CFD processing. To reduce the time of computations and due to axial symmetry of the GT's circular sectors, the single quadrant (the sector of 90°), enclosed by outer cylinder surface of Emitting tube, by two radial planes and the but-end surfaces has been chosen as the computation element.

Usually a great number of cells and nodes are used by numerical solution the sets of simulation equations. The stringent account of the design to be simulated is of specific importance because of need to compose adequate computational grid. Some designs of GT to be computed are shown in Figures 1 and 2.

By our analysis of GT combustion system being of small sizes the computation grid include $(4\text{-}6)\cdot 10^5$ cells whereas $(1\text{-}2)\cdot 10^6$ control volume have been involved by computations of the large-scale Fume Control systems in the Steel Industry [9] (for example).



b)

Fig.1. The principal layout single-ended Radiant tube, including of the inlet (nozzles) and recirculation openings arrangement / Types of RT:
a – GT1; b – GT1* (without recirculation holes).

1– gas nozzles, number – 4;
2– air nozzle, number – 16;
3– Recirculation orifice, number – 16;
4– Emitting (outer) tube;
5– recirculation tube;
6– burner system including air preheater;
7– primary annular combustion chamber;
8– burnt (flue) gases central channel.

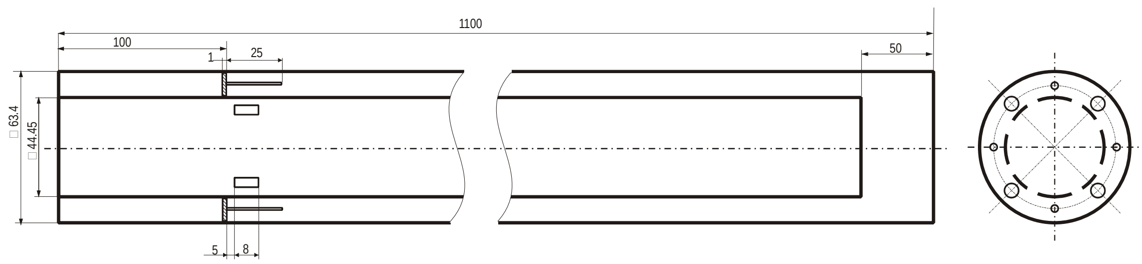


Fig.2. The principal layout of Radiant tube of GT1.0 type.

3. Heat Balance of Radiant Tube (RT). Thermal (energetic) efficiency of RT

The problems to be solved in respect to efficiency of RT operation are divided by two main tasks:

1. thermal interaction between RT (or set of RT) with the ingots arranged within the furnace;
2. internal fuel-oxidant jets combustion process within RT.

First of the tasks is considered in our presentation [11]. Both tasks are interconnected. Balanced circuit by heat transport within RT is presented in Fig. 3.

The main balance equation and components of Heat (Energy) Balance within the active section of the RT length are given below:

$$Q_{in_{RT}} = Q_{out_{RT}} \qquad (1)$$

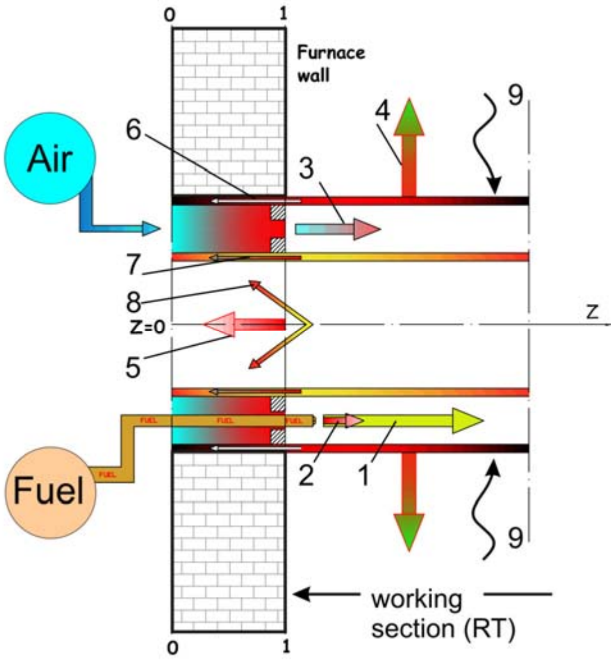
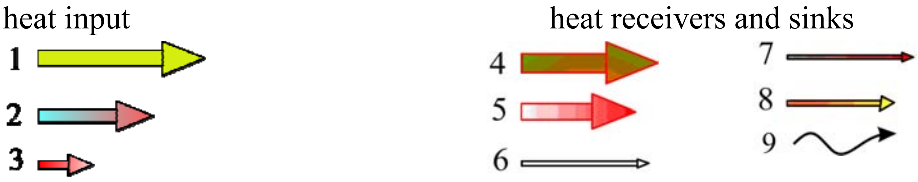


Fig. 3. Heat balance of radiant tube by stationary operation.



Control cross section 1-1

Heat input

1 – Q_f – combustion heat (low heat value by standard conditions: T=298 K): $B_f \Delta H_{f,T}$;

2 – ΔH_f – surplus (excess) enthalpy of fuel by natural gas temperature T_f: $B_f \Delta I_f$;

3 – ΔH_a – surplus (excess) enthalpy of the oxidant (combustion air) by preheated air temperature T_a;

$$Q_{in} = B_f\left(\Delta H_{f,T} + \Delta I_f + \lambda\Omega_{st}\Delta I_a\right) + \alpha_{RT} Q_{furn} \qquad (2)$$

The temperatures T_a and T_f meet to the corresponding values at the inlet the combustion chamber (after additional preheating within the section by the furnace wall length).

Heat receivers, sinks and losses

4 – Q_{emit} – total heat flux emitted by RT outer surface within the RT working section;

5 – Q_{fl} – heat content (total enthalpy) of the back flue gases flow within recirculation tube (including incompleteness combustion heat);

6 – $Q_{loss,1}$ – heat losses by heat conductivity by outer (emitting) tube length;

7 – $Q_{loss,2}$ – heat losses by heat conductivity by inner (recirculation) tube length;

8 – $Q_{loss,3}$ – heat losses by radiation from RT inner space to the tube walls outside the RT working section (to the left side of the cross-section 1-1).

9 – ($-Q_{furn}$) – heat incident on RT outer surface within the Tube working section.

$$Q_{out} = Q_{emit} + Q_{fl} + \sum_{i=1}^{3} Q_{loss,i} \qquad (3)$$

Resulting useful heat to be accounted for RT within the “ideal” furnace in the case of arrangement the single RT within the furnace:

$$Q_{use} = Q_{emit} - \varepsilon_{RT} Q_{furn} \tag{4}$$

By definition the structure of heat balance of Radiant Tube of given design (GT) including the sources of heat input and heat receivers the last are considered by division the GT assemblies for internal and external ones. The process within the GT in itself is of no importance by computation of the 1st task. For example, the recirculation process inside the GT does not influence directly on definition of heat exchange between the RT and the ingots. But indeed the recirculation process influence on temperature values and profiles inside RT at Emitting tube surface impacting the RT efficiency.

4. Interaction of mass-and heat exchange. Role of recirculation processes

By discussion the results of the RT’s working process we’ve found that power efficiency Ef_F of RT is strongly depended on mass transfer within the RT. In case of RT of designs shown, the mass exchange processes are presented both by fuel, oxidant, and combustion products mixing as well as flue gases recirculation from the inner (Recirculation) tube space into the into reaction zone within the annular space between the inner and outer (Emitting) tubes (Fig.4).

Negative recirculation (r<0) means opposite (from annular to central channel) movement of part of the reacting mixture flow and is accompanied by fuel burning incompleteness (Fig.5) and as consequence – by reduction of Ef_F (Table 2).

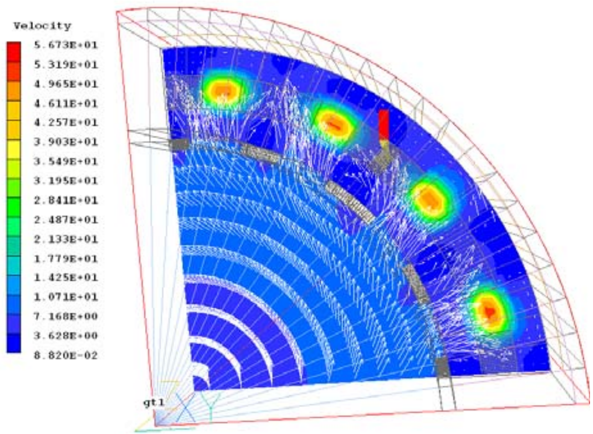


Fig.4. Recirculation of the gases through the recirculation orifices at the inner tube wall / Pattern of the velocity vectors within the plane XOY, perpendicular to the RT axis, at the cross-section z=0.105 m (distance 5 mm from the burner lid).
Mixing of gas and combustion air jets at T_a=1200 K; T_{fur}=1200 K.

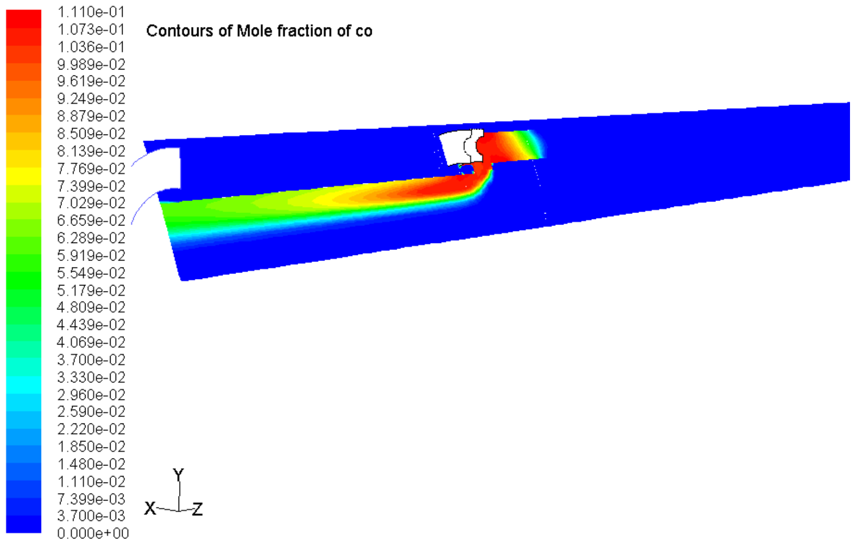


Fig. 5. Mole fraction of CO pattern within longitudinal radial section arranged in the plane of recirculation opening.

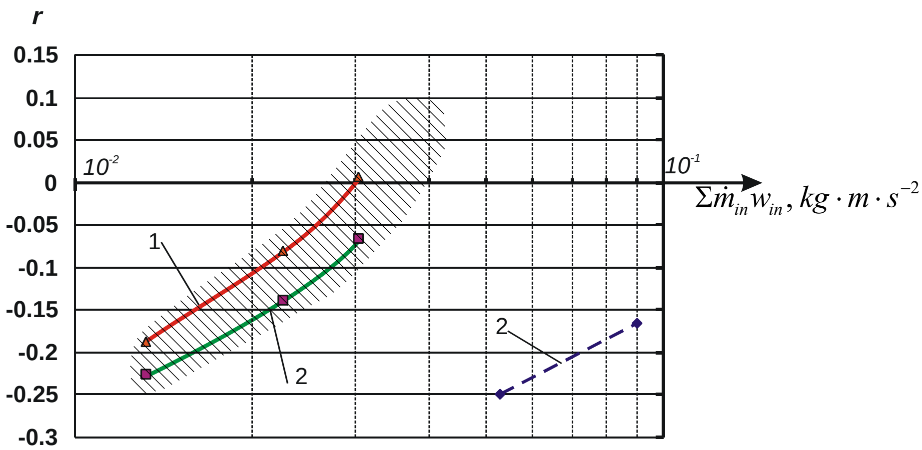


Fig. 6. Dependence of recirculation ratio r on momentum of the inlet jets (combustible +oxidant) by GT1 operation at half nominal thermal capacity ($0.5Q_{nom}$) – solid lines – and nominal thermal capacity (Q_{nom}) – dotted lines.

Furnace temperature T_{fur}, K:

1 – 773; 2 – 1173.

RT of GT1.0C (Ceramic) design. 50% of Nominal firing rate (heat capacity) $Q = 0.5Q_{nom} = 3.88$ kW. Furnace temperature T_{fur}=1500 K. Combustion air preheating temperature T_a=1200 K (negative recirculation).

4.1. Recirculation mechanism

Vacuum inside the combustion chamber, determining entrainment of the flow from the central channel into the annular channel depends on momentum $\int\limits_{F_{en}} \rho w^2 dF$ where F_{en} – the cross – section of the channel (in this case – the annular channel) where flue gases are entrained by active flows of combustible and oxidant. That's why we suppose that

$$\dot{m}_{en} w_{en} = \int_{F_{en}} \rho w^2 dF \square \int_{F_{en}} \rho w^2 dF \square \dot{m}_{in} w_{in} = T_N^{-1} \left(\dot{m}_a^2 T_a / n_a f_a \rho_{a,N} + \dot{m}_g^2 T_g / n_g f_g \rho_{g,N} \right) \quad (5)$$

Analysis of equation (5) brings to the conclusion that the most effective means of increase the inlet flows momentum is reduction of the exit cross-sections of the inlet nozzles (f_g,f_a) and the characteristic sizes of the active flows (d_a, d_g for the circular orifices) – especially.

By this reason in the final RT version titled GT 1.0 both available for metallic and ceramic radiant tubes, design we've projected the reduced inlet cross-sections of the active jets. In the GT1 version the sizes of the gas nozzles were assumed of small value as well as number of the last ($n_g = 4$). Thus both values (d_g, n_g) can't be changed and we've decided to minimize the number of the combustion air inlet orifices; in our case – to reduce 4 times the number of air orifices – from 16 (GT 1) to 4 ones (GT 1.0) by conservation both the gas and combustion air exit diameters.

Change of the RT design – from GT 1 for GT1.0 is accompanied by substantial (4 times) growth of air velocity – from 58 m/s to 232 m/s (averaged value by total orifice cross-section) and 16 times increase of required air excess pressure: more than 7.896 kPa in case of air preheating temperature $T_a = 1200$ K.

Moreover it must be taken into account that the flow is constricted by efflux through the orifice in the thin wall, and the jet fills only the part μ of formal orifice cross-section (minimum μ makes 0.61…0.63). To ensure nominal flow-rate in this case discharge velocity must be increased in μ^{-1} times and excess pressure ahead the orifice – in μ^{-2} times – up to 20 kPa.

4.2. Correlation between Heat Exchange Efficiency and Recirculation process

The characteristics of mentioned GT designs are differed one from another mainly because of various processes and of different recirculation mass-exchange ratio value by using the Glow Tubes under consideration:

- GT 1 – of "negative recirculation": $r < 0$;
- GT * – of zero recirculation: $r = 0$;
- GT 1.0 – of positive recirculation: $r > 0$.

By CFD modelling of RT the useful component of Ef_F titled Q_{use} (equation 4) is determined by portion of total enthalpy of combustion products being emitting from outer surface of RT. By this cause total volume of computation of all constituents (jets propulsion and mixing, heat exchange, combustion kinetics) is required for solution the task.

The thermal (power) efficiency of Radiant Tube as well as temperature non-uniformity by both GT tubes: inner and outer – are strongly dependent on Fluid Dynamics of the jets (Figures 6, 7) and the flows within annular combustion chamber and on direction of recirculation process within the central tube.

As to efficiency Ef_F of Radiant Tubes in dependence on recirculation process, the strong statement of following items has been determined (Tables 2-4, Figures 6,7):

- the higher is recirculation ratio r, the more would be Ef_F value;
- the negative "recirculation" ($r < 0$) by inverse flow direction of part of combustion products, causes lowering Ef_F being accompanied by fuel combustion incompleteness;

- in correspondence with the thermodynamic principles of "ideal furnace", the efficiency becomes of lower value by firing rate Q increase; thus $Ef_F\ (Q_{nom}) < Ef_F\ (0.5\ Q_{nom}\ )$ if $r > 0$.

Table 2. Comparison of Ef_F values % for metallic designs of Radiant Tubes. T_{fur} = 1173 K

T_a, K	Q_{nom}			$0.5\ Q_{nom}$		
	GT1 ($r < 0$)	GT1* ($r = 0$)	GT1.0 ($r > 1.0$)	GT1 ($r < 0$)	GT1* ($r = 0$)	GT1 ($r > 1.0$)
300	45.0	NC	57.7	46.7	NC	60.7
600	54.7	56.4	63.9	54.6	60.4	66.8
900	NC	68.7	72.0	68.4	69.2	75.2

Note: GT1, GT1*, GT1.0, GT1.0C – types of GT under consideration, NC – not computed variant of operating conditions

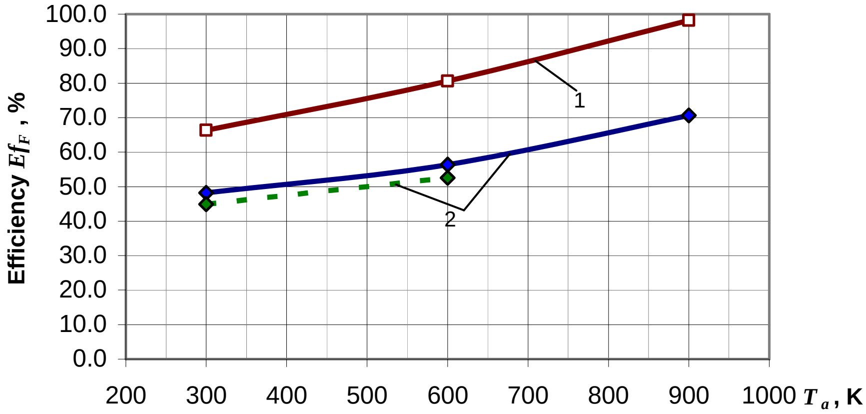


Fig. 7. Dependence of efficiency of natural gas utilization Ef_F on combustion air preheating temperature T_a for Radiant Tube GT1 version by operation at half nominal thermal capacity ($0.5Q_{nom}$) – solid lines – and nominal thermal capacity (Q_{nom}) – dotted lines.
Furnace temperature T_{fur}, K:
1 – 773; 2 – 1173.

4.3. Provision of recirculation and increase of ratio *r* value

As it was emphases above, break of flue gases recirculation as well as direct overflow of combustion components into the recirculation orifices and incomplete burning (Fig.5) cause lowering the efficiency Ef_F of natural gas utilization (Fig.7).

Fluid dynamics process within the RT inner space has strongly pronounced 3D character, both by the length of RT and by the cross-section, including radial and angular coordinates. The recirculation process direction is connected with turbulence and static pressure of flow. Decrease of pressure occurs by the jets propulsion within the enclosed space – by the channel length and between the jets (figures 8,9). The higher would be absolute value of minimum pressure p (the vacuum), the more would be recirculation ratio r.

Numerical analysis allows us to state that r value is enhancing (i.e. the r absolute value is decreased when recirculation ratio r has the negative sign) by increase the initial jets' specific momentum (ρw^2 value) by conservation the specific flow-rate value: $\rho w = const$ (Fig.6).

In the cases have been analyzed, the initial momentum value been introduced into the combustion chamber (the annular channel) is fully determined by operational parameters: gas (g) and combustion air (a) flow-rates and their initial temperatures.

By extrapolation the data presented in Fig.6. to higher values of introduced momentum we would ensure recirculation taking place and obtain the positive values of r of required magnitude.

The higher is an active flows momentum input $\sum_{(i=g,a)} \dot{m}_{i,in} w_{i,in}$, the stronger must be the recirculation process. The last is determined by the momentum of the flow of the combustion products $\dot{m}^{en} w^{en}$ has been entrained into the combustion chamber. The momentum of active flows may be increased

- due to higher velocity of flow $w_{i,in}$ of each of the input components (active jets) by conservation of the same mass flow – rate $\dot{m}_{i,in}$ (for example in our case – under variation of air flow temperature) ($i \equiv g,a$);
- due to increase of the mass flow rate of the active jets $\dot{m}_{i,in}$ ($i \equiv g, a$);
- due to increase of both mass flow rate $\dot{m}_{i,in}$ and the velocity $w_{i,in}$ ($i \equiv g, a$).

Value of $\dot{m}_{en} w_{en}$ by all the mentioned cases would be increased that means growth of $\dot{m}_{en}$ value. Influence of $\dot{m}_{en}$ growth on the recirculation ratio r value depends on the specific method of sum momentum $\sum_{(i=g,a)} \dot{m}_{i,in} w_{i,in}$ augmentation. Under condition of $\dot{m}_{i,in}$ increase by the first and the third of mentioned methods, the ratio of the flow – rates $r = \dot{m}_{en} \big/ \sum \dot{m}_{i,in}$ would be increased. But if the initial momentum is increased due to augmentation of mass flow rates of active flows, the r value may be conserved or even decreased.

For example, under isothermal conditions and / or when the chemical processes don't occurred while recirculation takes place (positive r value), the value of r is practically invariable on active jets' flow-rates for incompressible mediums of permanent initial parameters.

In the cases of absence the recirculation process by the reason of lack of the recirculation orifices and by conservation of the pressure losses by RT length, growth of combustion air preheating temperature means decrease of available RT heat capacity and reduces the direct overflow the gases from the annular channel into the central (recirculation) tube because of increase the inlet flows momentum.

5. Forming of temperature profiles within RT space. Uniformity of temperature pattern along emitting surface of RT

Very important question is an ability of the radiant tubes to ensure uniform heat reception by the ingots surface. This characteristic is interconnected with combustion process within RT.

One of the task statements consists in definition of the values and the 3D patterns of the resulting heat fluxes q_{res} received by ingots. In conditions of the settled temperature of the ingots T_{ing} and emissivity ε_{ing} the resulting flux is equal to absorbed flux minus proper (own) radiative flux:

$$q_{res,ing} = a_{ing} H_{\Sigma\, ing} - \sigma_0\, \varepsilon_{ing}\, T_{ing}^4 = \varepsilon_{ing}\left(H_{\Sigma\, ing} - \sigma_0\, T_{ing}^4\right) \text{ if } a=\varepsilon \text{ (grey body)}, \quad (6)$$

where H_Σ –heat flux incident upon ingots.

Solution of the last task represents the subject of the RT investigation by statement 1 (see chapter 3). In frame of the statement 2 conditionally is assumed that $H_{\Sigma_{ing}}$ meet to outer radiative flux Q_{emit} has been found through the values of the Emitting tube temperatures pattern the last being simultaneously used by definition Ef_F (see section 4.2).

As a result of the CFD modelling of GT of tested design and of operation regime under consideration we've succeeded in obtaining the temperature profiles by the mediums and the surfaces.

The temperature profiles by the surfaces of each of concentric tubes (Emitting and Recirculation ones) are characterized by Area-Weighted Average Temperature of the tube T_{rad} (subscript $rad \equiv emit, rec$), on the one hand, and non-uniformity ΔT_{rad}, on other hand.

We've used the simplified approach – consideration of maximum T_{rad}^{max} value by fixed T_{rad} values within the outer tube surface and the maximum deviation ΔT_{rad}^{max} of the T_{rad} value between the local maximum and local minimum values being fixed at the diagrams of the observed computed data (by the GT length accordingly various diagrams by cylinder (tube) generating lines and the cross – sections).

Both of these values – T_{emit}^{max} and ΔT_{emit}^{max} – for the Emitting Tube would be readily found by the temperature profiles watching along and across the tube temperature profiles (see Table 3).

Analysis of data shows the great value of absolute temperature non-uniformity ΔT_{emit}^{max} by surface of Emitting Tube reaching more than 200K (see Table 3). Meanwhile the Standard Deviation Temperature (SDT) makes 3…5 times lower value. The cause is that there are only small locations of the maximum temperatures at the tube surface neighboring to the zones where combustion chemical reactions take place.

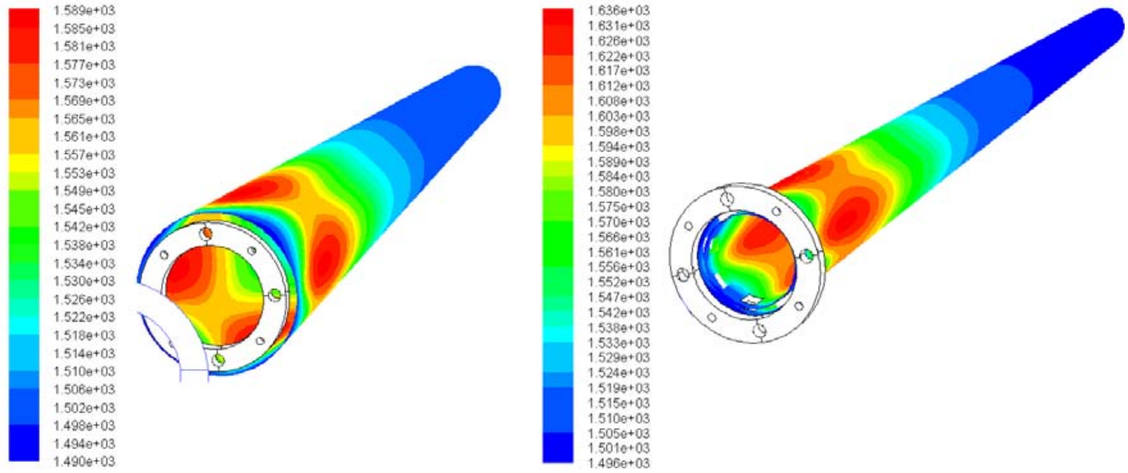
To explain this situation in Figure 8, 9 are given the temperature patterns by the length of Emitting Tube and the Recirculation Tube, the temperature range within each cross section being colored. The diagram indicates on 3D character of variation the temperatures.

Presented local character of temperature non-uniformity becomes understandable by joint consideration of Figure 8,9 with the Fig.10 related to gas-phase temperature within the annular space of radiant tube. Last figure demonstrate essential temperature non-uniformity within the reacting gases flow (up to 1000 K).

Table 3. CFD modelling of Radiant Tube. Generalized data for version GT1 and GT1*/ Influence of thermal capacity and combustion air preheating.

Furnace temperature 773 K.

Characteristics	Dimensions	50% of Nominal thermal capacity $Q = 0.5Q_{nom} = 3.88$ kW		
Initial data				
Temperature of combustion air preheating T_a	K	300	600	900
Mass flowrate of fuel (natural gas – methane)	kg/s	7.746E-05		
Mass flowrate of combustion air	kg/s	1.235E-03		
Air excess factor, λ		1.052		
Computed data				
Total Heat Transfer Rate (Useful Heat)	W	**2354**	**2592**	**2915**
Efficiency, Ef_F,	%	**60.7**	**66.8**	**75.2**
Maximum of Facet Values Temperature of the tube wall within External tube surface	K	1313	1320	1329
Minimum of Facet Values Temperature of the tube wall within External tube surface	K	1071	1093	1133
Area-Weighted Average Temperature of the tube wall within External tube surface	K	1202	1205	1209
Standard Deviation Temperature of the tube wall within External tube surface	K	47.0	46.6	46.4
Maximum Temperature of gas delivering tubes (nozzles)	K	982	994	996
Maximum Temperature of inner tube (recirculation insert)	K	1357	1366	1378
Mass-Weighted Average Temperature of flue gases at the GT's exit (outlet), T_{fl}	K	1121	1151	1191
Mass-Weighted Average Temperature of combustion air at the nozzles (orifice)	K	669	849	1030
Mass-Weighted Average Mass fraction of CO at exit (outlet)		0.0073	0.0093	0.0108
Mass-Weighted Average Mole fraction of CO at exit (outlet)		0.0075	0.0091	0.0105
Recirculation ratio, r		**0.462**	**0.577**	**0.713**
Area-Weighted Average Pressure, Air-Inlet	Pa	2257	2591	3148
Area-Weighted Average Pressure, Gas-Inlet	Pa	469	452	450
Imbalance Total Heat Transfer Rate	W	-76.0	-26.56	-47.60



a) b)

Fig. 8. Temperature pattern, K, at outer surface:
a –of Emitting tube; b – of Recirculation tube
GT of GT1.0C (Ceramic design). 50% of Nominal firing rate (heat capacity) $Q = 0.5Q_{nom} = 3.88$ kW. Furnace temperature T_{fur}=1500 K. Combustion air preheating temperature T_a=1200 K.

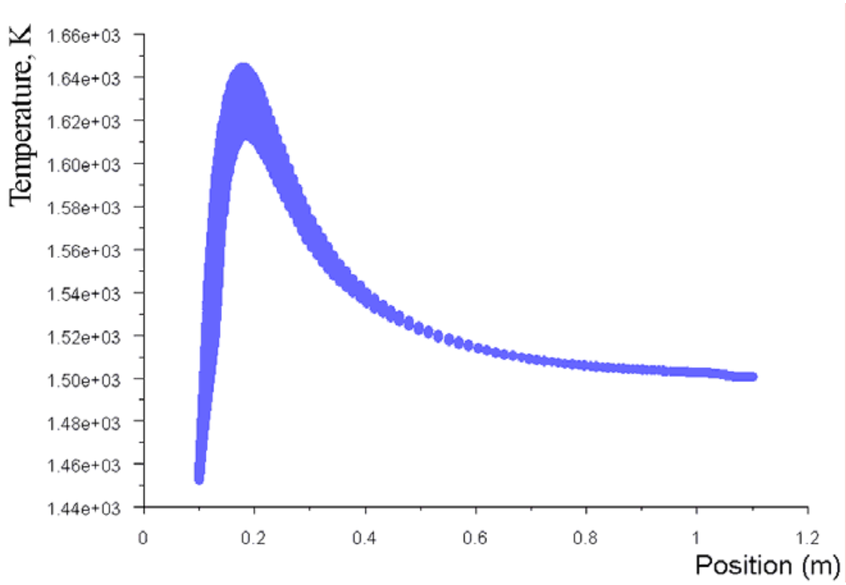


Fig. 9. Temperature profile of the Emitting tube surface along of the tube axis *z*. The values related to each cross-section are colored (the *T* range by any *z=const* (along tube axis) section).
GT of GT1.0C (Ceramic design).
Nominal firing rate (heat capacity) Q_{nom}=7.76 kW.
Furnace temperature T_{fur}=1500 K. Combustion air preheating temperature T_a=1200 K.

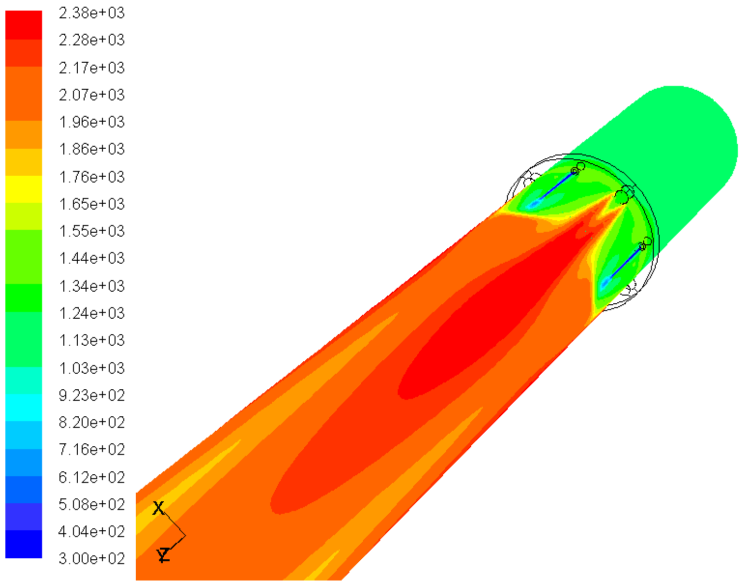


Fig. 10. Combustion products temperature, K, pattern within conditional cylinder surface, coaxial with outer and inner tubes at radius y = 0.0243 m (center line of annular channel involving axes of the gas and air nozzles).
GT of GT1.0C (Ceramic design). Nominal firing rate (heat capacity) Q_{nom}=7.76 kW.
Furnace temperature T_{fur}=1500 K. Combustion air preheating temperature T_a=1200 K.

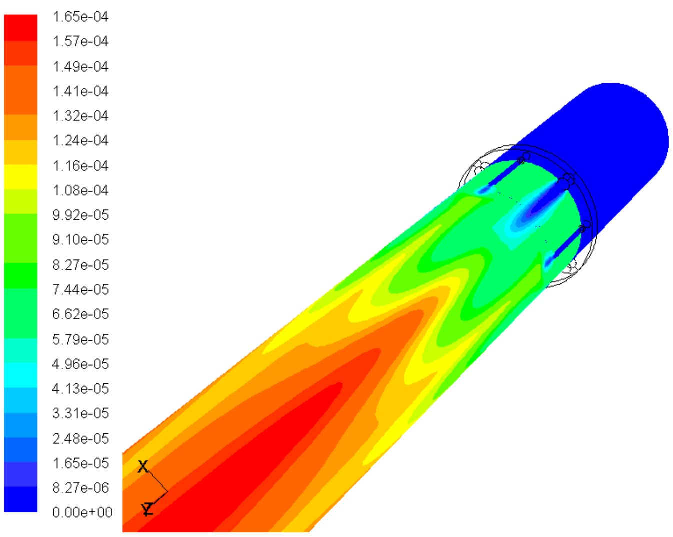


Fig.11. Mole fraction of NO species within conditional cylinder surface, coaxial with outer and inner tubes at radius y = 0.0243 m (center surface of annular channel where axes of the gas and air nozzles are located).
RT of GT1.0C (Ceramic) design. Nominal firing rate (heat capacity) Q_{nom}=7.76 kW.
Furnace temperature T_{fur}=1500 K. Combustion air preheating temperature T_a=1200 K.

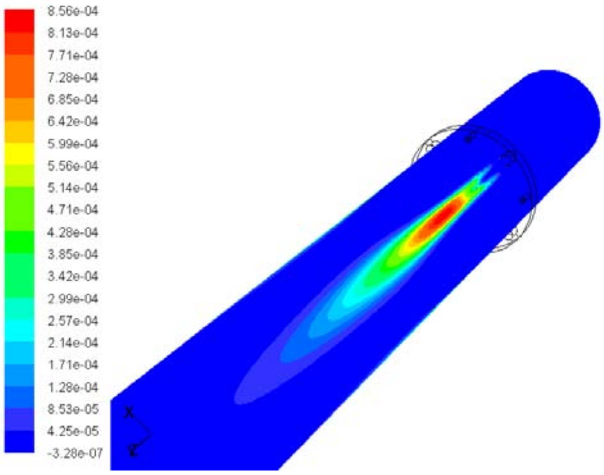


Fig. 12. Rate of Thermal NO formation, [kmole/m^3·s] within conditional cylinder surface, coaxial with outer and inner tubes at radius y = 0.0243 m (center surface of annular channel where axes of the gas and air nozzles are located).
RT of GT1.0C (Ceramic) design. Nominal firing rate (heat capacity) Q_{nom}=7.76 kW.
Furnace temperature T_{fur}=1500 K. Combustion air preheating temperature T_a=1200 K.

6. Low-emission combustion system

The developed concept of RT presents the low-emission design under condition of choice the optimum design and ensuring the proper preparation mode. In the case of negative “recirculation” (r < 0) part of initial reagents in the very beginning of burning process, overflows into the central tube. This portion of initial combustion components merges with back flue gases flow moving along the central recirculation tube to the RT exit. By this reason CO content (see Fig.5) in the checking section (RT exit) is essential. Mass-weighted average CO mass and mole fraction at the outlet is enhancing by r value is increased by absolute value in case of negative “recirculation”. [CO] concentration would be reduced if thermal output of RT is increasing by simulations $|r|$ decreasing by absolute value (in the case of r < 0).

For example $[CO]_{mass}$= 1.66% by Q_{nom}=7.76 kW and $[CO]_{mass}$= 2.81% by 0.5Q_{nom}=3.88 kW under using RT of GT1.0C version, T_{fur}=1500 K, T_a=1200 K.

NO_x formation (Fig.11) is determined mostly by thermal mechanism (Fig.12) i.e. the following factors are of main importance: maximum reached local temperatures, local composition of reacting gases ([O_2] + [N_2], Fuel), residence time within localization of maximum temperature. Analysis of data presented in Table 4 shows that first of mentioned factors (T_{max}) causes the greatest influence on [NO_x] value.

Maximum [NO_x] value (the biggest from the values obtained in our calculations) is related to the NO_x concentration by operation of the ceramic design of GT version 1.0C by following set of initial data: $Q = Q_{nom}$ = 7.76 kW, T_{fur} = 1500 K, T_a = 1200 K. Under mentioned conditions [NO_x] = 136 ppm.

Lowering the firing rate causes lowering Ef_F because of increased recirculation ratio r by absolute value (r < 0). Simultaneously [NO_x] is reduced by two reasons: because of decrease of T_{max} and of CO increase (Table 4). Usually [CO] and [NO_x] variation have an opposite tend [12].

7. Hydraulic state of RT

The pressure range of combustion components (natural gas and air-oxidant) before the burner system make an important operation characteristic of RT perfection because it determine the permissible heat output of RT.

CFD modelling allows estimating the pressure drop by RT length. Apparently it depends on maximum change of static pressure in two cross-sections: before the burner system and the exit of RT.

From the law of momentum conservation, the difference of proper values of $\int\limits_{(F)} \left(p_{st} + \rho w^2\right) dF$ in the mentioned cross-section with account of the side walls reaction (by tangential τ and normal σ_n stress components) allow us to obtain the static pressure profile (Fig.13), value of pressure drop by the RT length and necessary velocity head by air and gas pathways:

$$\frac{\partial}{\partial z}\left[\int\limits_{(F)} \left(p_{st} + \rho w^2\right) dF + \int\limits_{(F_s)} \left(\tau_z + \sigma_{n,z}\right) dF_s\right] = 0\,, \tag{7}$$

were z – coordinate and z-th component of the stresses (as the subscript).

Because of essential non-uniformity of static pressure within any cross-section at the surfaces (Fig.14) and in the flow the parameters to be defined, represent Area-Weighted Average Pressure: for Gas-Inlet and Air-Inlet correspondingly (see Tables 3, 4).

Table 4. CFD modeling of Ceramic design of Radiant Tube. Generalized data for version GT1.0C (Ceramics)/

Furnace temperature 1500 K.

Characteristics	Dimensions	Nominal thermal capacity $Q_{nom} = 7.76$ kW	50% of Nominal thermal capacity $Q = 0.5Q_{nom} = 3.88$ kW
Initial data			
Temperature of combustion air preheating T_a	K	1200	1200
Mass flowrate of fuel (natural gas – methane)	kg/s	1.549E-04	7.746E-05
Mass flowrate of combustion air	kg/s	2.471E-03	1.235E-03
Air excess factor, λ		1.052	1.052
Computed data			
Total Heat Transfer Rate (Useful Heat)	W	5004	1940
Efficiency, Ef_F,	%	64.5	50.0
Maximum of Facet Values Temperature of the tube wall within External tube surface	K	1647	1590
Maximum of Values Temperature External tube (inner surface)	K	1661	1599
Minimum of Facet Values Temperature of the tube wall within External tube surface	K	1454	1491
Area-Weighted Average Temperature of the tube wall within External tube surface	K	1530	1512
Standard Deviation Temperature of the tube wall within External tube surface	K	45.9	27.2
Maximum Temperature of gas delivering tubes (nozzles)	K	1331	1402
Maximum Temperature of inner tube (recirculation insert)	K	1719	1635
Mass-Weighted Average Temperature of flue gases at the GT's exit (outlet), T_{fl}	K	1572	1716
Mass-Weighted Average Temperature of combustion air at the nozzles (orifice)	K	1284	1361
Mass-Weighted Average Mass fraction of CO at exit (outlet)		0.0166	0.0281
Mass-Weighted Average Mole fraction of CO at exit (outlet)		0.0158	0.0261
Recirculation ratio, *r*		-0.052	-0.154
Area-Weighted Average Pressure, Air-Inlet	Pa	4056	1101
Area-Weighted Average Pressure, Gas-Inlet	Pa	2400	1259
Imbalance Total Heat Transfer Rate	W	94.4	76.64
NO, Mass-Weighted Average Mole fraction		1.361E-04	6.31E-05

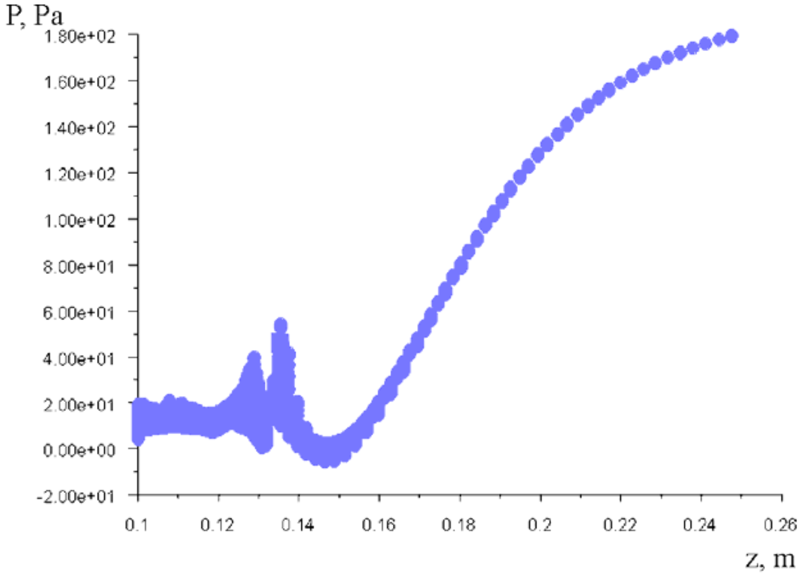


Fig. 13. Profile of static pressure at the surface of recirculation insert along of the tube axis *z*. The values related to each cross-section are colored (the range by any *z=const* section).
GT of GT1.0C (Ceramic design). Nominal firing rate (heat capacity) Q_{nom}=7.76 kW. Furnace temperature T_{fur}=1500K. Combustion air preheating temperature T_a=1200K.

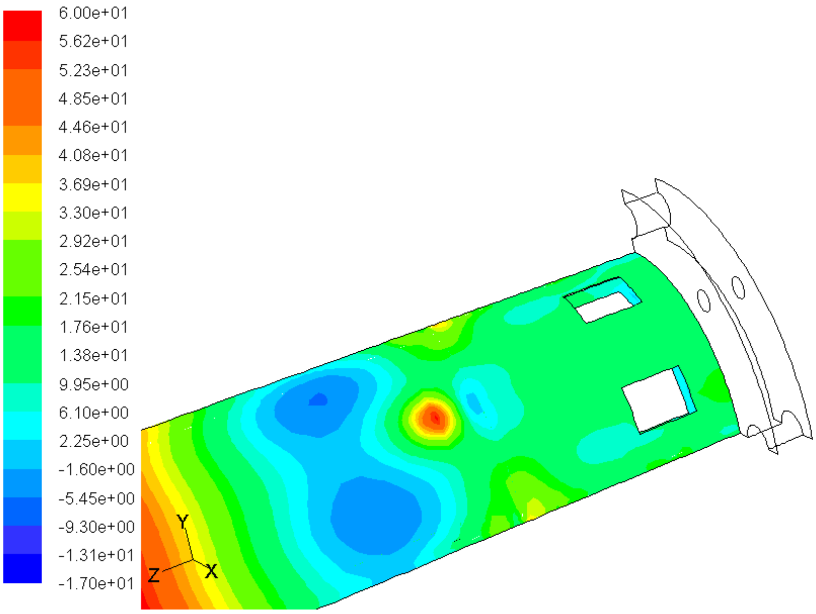


Fig. 14. Static pressure pattern at the surface of recirculation insert.
GT of GT1.0C (Ceramic design).
Nominal firing rate (heat capacity) Q_{nom}=7.76 kW. Furnace temperature T_{fur}=1500 K.
Combustion air preheating temperature T_a=1200 K.

Conclusion

1. The combustion process taking place in Radiant Tube has been studied by means of CFD modelling to optimize the RT design (in the case under consideration – of GT versions) and operation mode from the combination of power, production processes and operation conditions (thermal and hydraulic perfection),) and environmental requirements. Each of listed demands has been accounted by proper criterion:

- power perfection – by RT efficiency;
- process characteristics – by temperature uniformity within Emitting tube surface (non-uniformity, absolute and statistically averaged (STD)), available heads of fuel and oxidant;
- ecological impact – by NO_x and CO effluence at the RT exit.

2. Some designs of RT of GT versions (GT1, GT1*, GT1.0, GT1.0C) have been proposed by GTI and modified in cooperation with the Gas Institute. Main characteristics of RT designs are compared basing upon numerical simulation by means of CFD procedures from the following standpoints and criteria: power, production process and operation conditions, environmental, mentioned above in item 1.

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