REGENERATIVE HEATING OF AIR DURING THE COMBUSTION OF A GAS IN RADIATION TUBES WITH A DISPERSED HEAT CARRIER

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This article examines and empirically realizes the regenerative heating of air in the combustion of a gaseous fuel in radiant tube-heaters with cascade fluidization and counter-current recirculation of an intermediate dispersed heat carrier. Such an arrangement is found to be considerably more efficient than recuperative heating.

The combustion of a gaseous fuel in radiant tube-heaters has come into wide use for the heating and chemicothermal treatment of metal, as well as in radiative drying and the firing of ceramics [1-3]. Utilization of the heat of flue gases and recuperative heating of air increase the thermal efficiency of such tubes to 70-75%. However, the welded or cast heat-resistant recuperators used for this purpose significantly increase the metal content of the tubes and make their fabrication more difficult.

Burning the gaseous fuel in radiant tubes with a dispersed heat carrier [4] eliminates the need for recuperators and provides for regenerative heating of air by the same carrier. In this case, it is used not only to intensify heat transfer in the combustion zone, but also to regenerate the heat of the outgoing gases. In the process, the temperature of these gases is reduced to 150-200°C and combustion efficiency is raised to 90-95% [5]. Such an economical and relatively simple design of gas heater is quite promising and is deserving of special attention. It will be the subject of the present article.

By analogy with the designs of well-known regenerative heat exchangers and air heaters which use a dispersed heat carrier [6-8], counter-current regenerators were installed at the top and bottom ends of the vertical radiation tube. These chambers cool the flue gases and heat the air as the particles of heat carrier spill over them (Fig. 1). To obtain maximum efficiency from the counter-current flow, the chambers were designed so as to provide for cascade fluidization of the heat carrier. Fluidization takes place on perforated grids with a specified bed height. Overflow pipes are located between the grids [9, 10]. The novelty and simplicity of the design lies in the fact that all of the functions which are normally performed by separate components are performed by one component; gas combustion takes place in the middle, radiating part of the radiation tube. As shown in Fig. 1, this part of the tube is located in the working space of the heating furnace, while the regenerative chambers with the grids are located in the top and bottom lining of the furnace. This prevents heat losses in these chambers.

Along with internal circulation of the particles in the combustion zone [4], general external recirculation of the carrier takes place. In the process, the particles spill down the exterior of the tube over its entire length and pneumatically recirculate in the opposite direction – as indicated by the arrows in Fig. 1. Here, they absorb the heat of the counter-flowing stream of flue gases in the top chamber, pass through the combustion zone, and release their heat to the flow of cold air in the bottom chamber. Figure 1 also presents a thermocyclogram of the heat carrier. In the figure, temperatures t_{1-5} denote the external particle recirculation loop, while temperatures T_{I-III} denote the internal particle circulation loop in the combustion zone. In accordance with the thermocyclogram, the flue gases are cooled from the temperature t_{cmb} to the value t_{ot}, while the air is heated from the initial temperature t_0 to the value t_a.

There are two possible variants for the passage of the particles of heat carrier through the combustion zone – with and without mixing of the external and internal loops in this zone. Figure 1 shows only the first variant. In the second variant, the particles travel through a special by-pass tube in the middle part of the design and go directly from the top to the bottom

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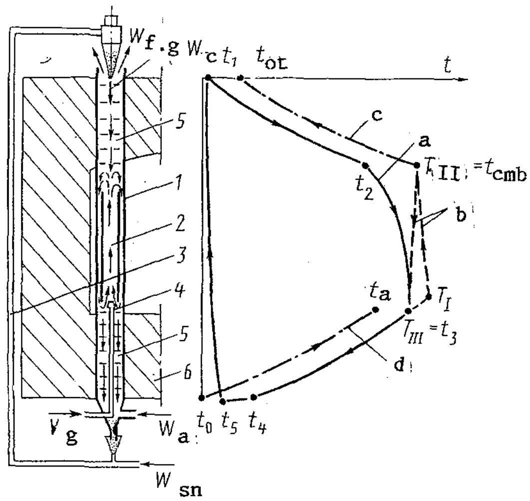

Fig. 1. Sketch and theoretical thermocyclogram of a gas radiation tube with a dispersed heat carrier and regenerative air heating: 1) radiation tube; 2) spherical tube-insert in the combustion zone; 3) air line for recirculation of the dispersed heat carrier; 4) gas inlet pipe; 5) counter-flow regenerators with grids for cascade fluidization of the heat carrier; 6) lining of heating furnace; V_g, feed and volumetric flow rate of gas; $W_{c,f.g.,a}$ and W_a^{pr}, feed, discharge, and water equivalents of the dispersed heat carrier, flue gases, and combustion air and the air for pneumatic recirculation of the heat carrier; a) external system-wide recirculation loop for the heat carrier; b) internal circulation loop for the carrier in the combustion zone; c) cooling of flue gases; d) heating of air.

chamber without mixing with the particles circulating in the combustion zone. Although this variant is more complex, it is more efficient and makes it possible to rapidly pass a large amount of even relatively cold heat carrier through the combustion zone. This scheme has its advantages, as will be discussed below.

Regardless of the variant used, the thermal efficiency (EFY) of the radiation tube is determined by the following components of the heat balance:

$$\mathrm{EFY} = \frac{Q_{rad}}{Q_{cmb}} = 1 - \frac{Q_{h4} - Q_{h1}}{Q_{cmb}} - \frac{Q_{ot}}{Q_{cmb}}, \qquad (1)$$

where Q_{cmb} and Q_{rad} are the heat released by the fuel and the useful thermal radiation from the tube in the middle (radiating) part; Q_{h1} and Q_{h4} are the amounts of heat delivered and consumed during the motion of the heat carrier according to the thermocyclogram in Fig. 1 when the temperature of the carrier is t_1 and t_4, respectively; Q_{ot} is the heat of the outgoing flue gases.

Here, we should note that electrical power consumption is not taken into account when the thermal efficiency of the tube is determined from Eq. (1), i.e. no allowance is made for the heat and fuel used to supply the combustion air or – in our case – the heat and fuel used to supply the air for pneumatic recirculation of the heat carrier. We assumed that, for the given high efficiency of the tube, the total consumption of air for these purposes is no greater than its consumption in other radiation tubes of the same capacity (and for which the energy costs of the air supply are also ignored in the determination of gas combustion efficiency [1-3]).

Assuming as usual [6-8] that the heat capacities of the gases, air, and heat carrier are constant, using a well-known index of the efficiency of counter-current heat exchange with cascade (multistep) fluidization [10], and allowing for the approximate equality of the temperatures

$$t_3 \simeq T_{\mathrm{I-III}} = t_{\mathrm{rop}} \tag{2}$$

by making several simple substitutions we reduce the initial equation (1) to the following theoretical form [5]:

$$\mathrm{EFY}_{\mathrm{I}} = 1 - \frac{t_{\mathrm{cmb}}}{t_{\mathrm{th}}} + \frac{t_{\mathrm{cmb}}}{t_{\mathrm{th}}} \frac{1}{R} \left[2\eta - 1 + \frac{(1-\eta)^2}{1+R_{\mathrm{sn}}} \exp\left(-\frac{\alpha_0 F_{\mathrm{sn}}}{1+R_{\mathrm{sn}}}\right)\right]$$
$$\left(\eta = \frac{R - R^{n+1}}{1 - R^{n+1}};\ R = \frac{W_{\mathrm{a}}}{W_{\mathrm{c}}};\ R_{\mathrm{sn}} = \frac{W_{\mathrm{a}}^{\mathrm{pr}}}{W_{\mathrm{c}}}\right), \tag{3}$$

where $W_{a,c}$ are the discharge heat capacities (water equivalents) of the air and heat carrier; W_a^{pr} is the same for the consumption of air for pneumatic recirculation of the carrier; $t_{th,cmb}$ are the theoretical and actual combustion temperatures of the gas in the tube; n is equal in each chamber to the number of stages of fluidization of the heat carrier; α_0 and F_{sn} are the heat-transfer coefficient and surface area of the air line which transfers heat to the surrounding medium with the temperature t_0. The subscript 1 denotes thermal efficiency in the first regeneration variant, with mixing of the heat carrier in the combustion zone.

It is interesting to examine extreme applications of the given theoretical formula. In particular, with ideal recirculation of the heat carrier (without heat losses in the air line), Eq. (3) is simplified to:

$$\mathrm{EFY}_{\mathrm{I}}^{\max} = 1 - \frac{t_{\mathrm{cmb}}}{t_{\mathrm{th}}} + \frac{t_{\mathrm{cmb}}}{t_{\mathrm{th}}} \frac{\eta^2}{R}. \tag{3'}$$

Conversely, in the case of pneumatic recirculation of the carrier with maximum heat losses, we have

$$\mathrm{EFY}_{\mathrm{I}}^{\min} = 1 - \frac{t_{\mathrm{cmb}}}{t_{\mathrm{th}}} + \frac{t_{\mathrm{cmb}}}{t_{\mathrm{th}}} \frac{2\eta - 1}{R}, \tag{3''}$$

where the corresponding recirculation conditions are given in the form $\alpha_0 F_{sn} = 0$ or $\alpha_0 F_{sn} = \infty$.

Returning to the second variant – in which the heat carrier, without mixing in the combustion zone, travels directly from the top chamber to the bottom chamber with the temperature

$$t_3 = t_2, \tag{4}$$

we find by analogy with (3)-(3″) that [5]:

$$\mathrm{EFY}_{\mathrm{II}} = 1 - \frac{t_{\mathrm{cmb}}}{t_{\mathrm{th}}} + \frac{t_{\mathrm{cmb}}}{t_{\mathrm{th}}} \frac{\eta^2}{R} \left[1 - \frac{(1-\eta)^2}{1+R_{\mathrm{sn}}} \exp\left(-\frac{\alpha_0 F_{\mathrm{sn}}}{1+R_{\mathrm{sn}}}\right)\right], \tag{5}$$

$$\mathrm{EFY}_{\mathrm{II}}^{\max} = 1 - \frac{t_{\mathrm{cmb}}}{t_{\mathrm{cmb}}} + \frac{t_{\mathrm{cmb}}}{t_{\mathrm{cmb}}} \frac{\eta}{(2-\eta) R}, \tag{5'}$$

$$\mathrm{EFY}_{\mathrm{II}}^{\min} = 1 - \frac{t_{\mathrm{cmb}}}{t_{\mathrm{th}}} + \frac{t_{\mathrm{cmb}}}{t_{\mathrm{th}}} \frac{\eta^2}{R}. \tag{5''}$$

Without recirculation of the particles in the case $W_c = 0$, both formulas degenerate into the familiar definition for the thermal efficiency of radiation tubes without air preheating, when $t_{ot} = t_{cmb}$:

$$\mathrm{EFY}_{\mathrm{I}}^{*} = 1 - \frac{t_{\mathrm{cmb}}}{t_{\mathrm{th}}}. \tag{6}$$

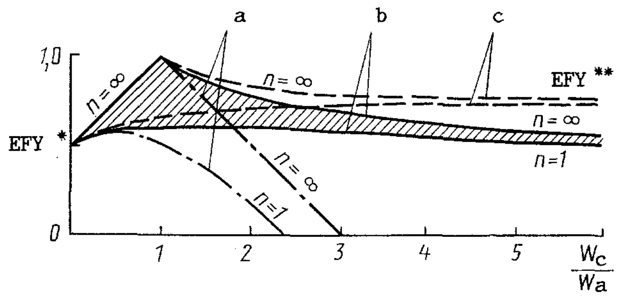

Fig. 2. Theoretical region and limits of change in the efficiency (EFY) of the radiation tube with normal combustion of the gas: a) according to (3″); b) according to (3′) and (5″); c) according to (5′).

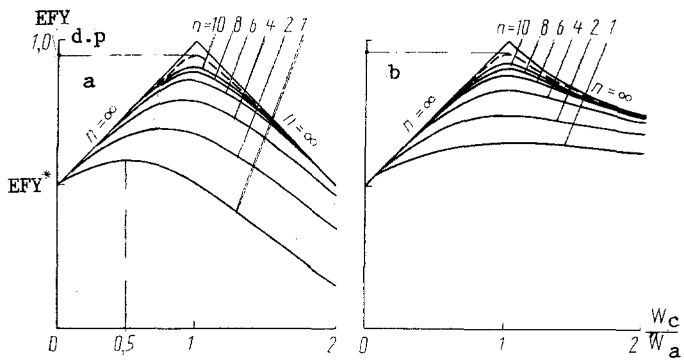

Fig. 3. Theoretical efficiency (EFY) of a tube in the region of the optimum ratio of water equivalents $W_c/W_a = 1$ ($t_{cmb} = 1000°C$, $t_{th} = 2000°C$): a) minimum efficiency according to (3″); b) maximum efficiency according to (3′).

It follows from a comparison of (3) and (5) that a higher efficiency is achieved in the second variant than in the first variant. This is evident from the fact that Eq. (3′) is comparable to (5″) only with ideal pneumatic recirculation:

$$\mathrm{EFY}|_{\mathrm{I}}^{\max} = \mathrm{EFY}_{\mathrm{II}}^{\min}. \tag{7}$$

Figure 2 shows the theoretical regions and limits of change of the efficiency of the given radiation tubes in all of the above-described cases and circulation variants. For greater clarity, the region of agreement (in accordance with Eq. (7)) is hatched. In the given case, it is the most realistic field of values of efficiency when the latter is examined in relation to the number of fluidization stages n and the ratio of the water equivalents W_c/W_a. Apart from the initial point (EFY*) efficiency at $W_c/W_a = 0$, the theoretical maximum efficiency = 1 at n = ∞, and the ratio $W_c/W_a = \infty$, it is interesting to examine the two characteristic efficiency limits with an infinite increase in carrier flow rate to $W_c/W_a = \infty$:

$$\lim(\mathrm{EFY}_{\mathrm{I}}^{\max}) = \lim(\mathrm{EFY}_{\mathrm{II}}^{\min}) = \mathrm{EFY}^{*} = 1 - \frac{t_{cmb}}{t_{th}}, \tag{8}$$

$$\lim(\mathrm{EFY}_{\mathrm{II}}^{\max}) = \mathrm{EFY}^{**} = 1 - 0{,}5\frac{t_{cmb}}{t_{th}}, \tag{8'}$$

With sufficiently intensive particle circulation, the second expression makes it possible to achieve a relatively high value of efficiency = 0.7-0.75 even at n = 1. In this case, the first variant, with a shift, clearly is inferior to the second variant. It even

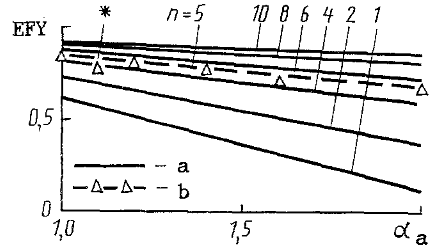

Fig. 4. Dependence of the efficiency (EFY) of the tube on the coefficient of excess air α_a with an increase in the number of fluidization stages n: a) calculation; b) experiment.

results in negative values of efficiency during pneumatic particle recirculation. Here, in accordance with (3″), the heat losses formally exceed the heat released in the combustion zone (Fig. 2a).

Figure 3 shows the region of values of efficiency near $W_c/W_a = 1$. This data allows us to make a judgement as to a reasonable restriction on the number of fluidization stages – the solution is determined by the convergence of the maximum and minimum of efficiency in each variant. In particular, the difference for n = 4 is already no greater than 5%. In this case, efficiency in the first circulation variant (in accordance with (3′)-(3″)), is 77-80%.

The upper limit of the number n and the value of efficiency itself is determined by the dew point $t_{d.p.}$ of the flue gases. When they cool to below this point, water vapor in the first fluidization stage condenses into the upper regenerating chamber. Taking $t_{ot} = T_{d.p.}$ in this case and thus determining the cascade reflux ratio in this chamber in the form [10]

$$\eta_{d.p} = \frac{t_{cmb} - t_{d.p}}{t_{cmb} - t_1} R \simeq \left(1 - \frac{t_{d.p}}{t_{cmb}}\right) R \qquad (t_{cmb} \gg t_1), \tag{9}$$

we find the sought value of n by established methods

$$\eta_{d.p} = \frac{R - R^{n+1}}{1 - R^{n+1}} = \left(1 - \frac{t_{d.p}}{t_{cmb}}\right) R. \tag{9'}$$

In particular, the simple method of successive substitution shows that with a known temperature $t_{d.p.} = 50\text{-}60°C$ and the combustion temperatures $t_{cmb} = 1000°C$ in Figs. 2 and 3, the limiting number of fluidization stages is roughly $n_{d.p.} = 20$. At this number, the thermal efficiency of the tube reaches 95-96% in all of the variants (see Fig. 3).

The theoretical formulas and estimates obtained above are valid with normal combustion of the gas in the given radiation tubes when the excess air coefficient $\alpha = 1$. When $\alpha > 1$, heat release from the gas in the combustion zone is expressed in the form

$$Q_{cmb} = \frac{W_{f.g}}{\alpha} t_{th} , \tag{10}$$

and theoretical formulas (3) and (5) show a linearly decreasing dependence on α. Of interest is the weakening of this dependence with an increase in the number of stages n, as shown in Fig. 4. Along with the theoretical data, the experiments conducted at n = 5 showed that an increase in excess air to $\alpha = 2$ lowers the efficiency of the regenerator from 85% to only 70%. Meanwhile, a similar excess of air in recuperative tubes in [1-3] reduced their efficiency by 40% or more.

The above empirical data was obtained on laboratory specimens of the investigated tubes [5]. The heat carrier in the specimens was circulated in accordance with the first variant, with the displacement of the particles in the combustion zone as depicted in Fig. 1. Here, a simple span-type radiation tube with a diameter of 102 × 5 mm and a length (height) of up to 3 m was placed on a separate laboratory stand with thermal insulation simulating the lining of a heating furnace. The test was set up so that the temperature in the combustion zone was at least 1000°C. Natural gas was burned in the tube at a rate of about 0.5 m^3/h. The rate of flow of combustion air was varied within the range $\alpha = 1\text{-}2$. The dispersed heat carrier inside and outside the tube was quartz (bank) sand of the 0.4-0.63-mm fraction. The ratio of the water equivalents of the two portions of sand was kept close to $W_c/W_a = 1$. In a separate case, this quantity was especially raised to roughly $W_c/W_a = 1.5$. The latter value exceeded the theoretical optimum (see Fig. 3) and actually led to a marked decrease in efficiency (denoted in Fig.

4 by the asterisk). Nevertheless, in each case the temperature of the flue gases decreased to 150-200°C, while the air was heated to 700-740°C. Here, flue-gas temperature was measured with a special "convective" thermocouple shielded by a fine steel mesh to reduce the radiation component of the measurement error.

The efficiency of the tube was evaluated empirically with the use of Eq. (1), where the heat content of the components was determined by the flow rates of the gas, air, and heat carrier and their temperatures at the outlet of the tube. No allowance was made for the reverse flow of heat from the particles into the top of the tube. Thus, the estimate was based on a minimum value corresponding to complete loss of the residual heat of the particles in the air line. It can be seen from Fig. 4 that minimum value of efficiency for normal combustion was 85%, which shows the economy of the above-examined regeneration scheme and the promise of using such tube-heaters not only in industrial furnaces, but in many other types of energy-intensive applications in which a thermal efficiency of 90-95% is a realistic goal.

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