TUBULAR GAS-FIRED RADIANT HEATERS WITH CIRCULATION OF A FLUIDIZED DISPERSE HEAT CARRIER

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The combustion of gaseous fuel in radiant tube-heaters has become a common practice for the heating and chemicothermal treatment of metal. Compared to conventional flame combustion, the new method of burning a gaseous fuel in such tubes – with circulation and fluidization of the dispersed intermediate heat carrier – ensures much more rapid, uniform, and efficient heating. A study is made of the phenomenon of gravitational fluidization of heat carriers, this effect resulting in an additional intensification of heat exchange and heat transfer among the particles in such tubes.

Gas-fired radiant tube-heaters have come into wide use both here and abroad in industrial furnaces and equipment designed for heating and chemicothermal treatment (CTT) of metal [1-3]. Such tubes are heated by burning a gaseous fuel inside them. During combustion of the fuel, heat transfer from the gas jet to the wall of the tube does not generally exceed 100 $W/(m^2 \cdot K)$ and is characterized by significant local overheating of the structure. Various types of radiation tubes have been developed to improve the uniformity and intensity of the heating process. The most effective of these designs has proven to be the tube with one closed end. The fluid gases are recirculated in these tubes [3]. The gas flame in the tube ejects a portion of the flue gases and extends along the tube, making heating more uniform. This effect is achieved as a result of a deficit of air and incomplete combustion of the gas along the entire length of the flame. This situation results in incomplete combustion of the gaseous fuel. Other types of radiation tubes (span type, u-shaped, etc.) employing the same principle have been used in practice. Normally, the complete combustion of a gas in a relatively short flame leads to local overheating and burn-through of the structure in the combustion zone. Thus, this situation also poses a problem. Experience has shown that breaking up the flow and burning several jets of gas in various tube designs also fails to provide the uniformity of heating which is made possible by the presence of openings in the pipe that delivers the combustion air [3].

The above problems and deficiencies in flame combustion have forced designers to look for new and more efficient methods of burning gas in radiation tubes. One approach has been to use radiant tube-heaters with a dispersed heat carrier [4-10]. Normal combustion of gas in such tubes is accompanied by fluidization and intensive circulation of a dispersed intermediate heat carrier. Quartz sand, corundum, magnetite, and other well-known refractories have been used as the heat carrier. In contrast to normal fluidization and the associated problems of expansion and stability of the fluidized bed in the height direction [11], the circulation of the particles of the heat carrier in such tubes is of a directional and organized nature over the entire prescribed length (height) of the structure. As a result, when estimated from the ratio of the minimum and maximum temperatures of the wall, the uniformity of the heating of a tube with a length of 1 m or more is estimated to be 97-98%. Here, the savings in gaseous fuel due to completeness of the combustion process and a reduction in the number of tubes required in heating furnaces is approximately 40-50%. This method was introduced at several large industrial plants in this country during the 1980s and is protected by a number of patents [12-15, etc.].

In a manner similar to the case of blind radiation tubes with recirculating flue gases, the particles of the heat carrier in the tubes described above also circulate between the spherical internal tube-insert and the external radiating tube – as

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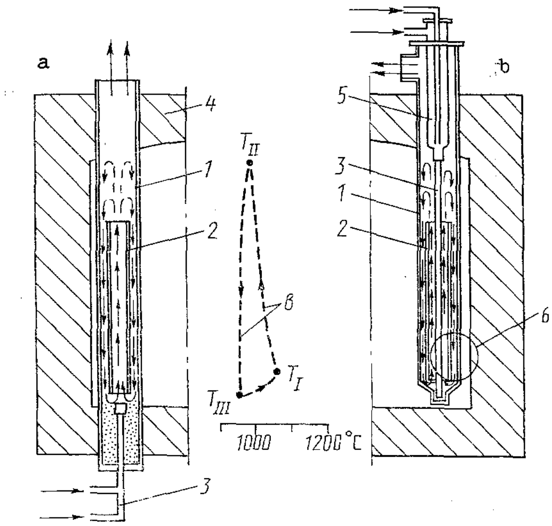

Fig. 1. Diagram illustrating the design and principle of operation of gas radiation tubes with a dispersed heat carrier: a) span-type tube; b) blind tube; c) thermocyclogram of particles; 1) radiation tube; 2) spherical tube-insert; 3) gas line with burner; 4) heating furnace; 5) recuperator; 6) fragment (see Fig. 2).

indicated by the inside arrows in Fig. 1. Here, the ejecting gas stream directs the particles upward over the tube-insert. The particles separate from the gas flow at the outlet of the insert and enter the annular channel between the tubes, i.e. they move back downward toward the ejecting gas stream along the wall of the radiating tube. During the circulation of the particles, heat is rapidly transferred directly from the combustion zone in the insert to the wall of the radiating tube without any sign of flame combustion of the gas – one sees only an incandescent and luminescent mass of circulating, dispersed heat carrier. The mass of this substance is an order of magnitude greater than the mass of the gas flow and thus ensures uniform heating of the structure. Shown to the right of the structure in Fig. 1a is one of the experimental thermocyclograms of the heat carrier in such tubes. Here, despite the higher temperature in the combustion zone, the temperature of the flow descending along the tube wall changes very slightly: its gradient in the height direction is no greater than 30°C/m at combustion temperatures on the order of 1000°C or higher.

In connection with this, one of the problems of using such radiation tubes – a problem which is also characteristic of other tube designs with a spherical internal tube-insert – is overheating of this insert relative to the external radiating tube. This problem is made worse when the reverse circulation of heat carrier along this tube occurs under conditions whereby the particles settle in a dense layer. Such a layer thermally insulates the combustion zone and causes the temperature in this zone to rise to a point exceeding the thermal resistance of the insert. The solution to this problem is complete fluidization of the dispersed heat carrier in the given tubes, i.e. not only the tube in which the flow ascends, but also the channel for the descent of the particles. Both flows should be converted to the fluidized state, it being hoped here that, under these conditions, heat transfer and heat exchange involving the particles will reach a maximum. The feasibility of such fluidization was noted in [11], while similar flows were broken up and rarified in [16-18] and were formed in [8] with an optimum theoretical ratio of tube diameters: $(D_1/D_2)^{opt} = 0.8$. In this case, the velocity and porosity of the descending disperse flow in the annular channel between the tubes corresponds to the porosity and velocity associated with counterfiltration of the gas phase – as in normal fluidization. The wall effect and particle mixing in such a flow extend across the entire channel [18], corresponding to the approximate condition [4]: $(\Delta/d_t)^{opt} = 10$, where $D_{1,2}$ are the diameters of the internal and external tubes of the structure; Δ is the channel width; d_t is the diameter of the particles of the heat carrier.

Results of such fluidization are manifest not only in an intensification of heat exchange between the particles and the wall of the radiating external tube, but also in intensive heat transfer between the ascending and descending particle flows through the wall of the internal tube-insert. Thus, direct longitudinal heat transfer by the particles from the combustion zone is an indirect but significant addition to the radiant heat transfer which takes place. Having used the particle circulation scheme shown in Fig. 1, we can determine this radial component through the ratio of the characteristic temperatures on the thermocyclogram [10]:

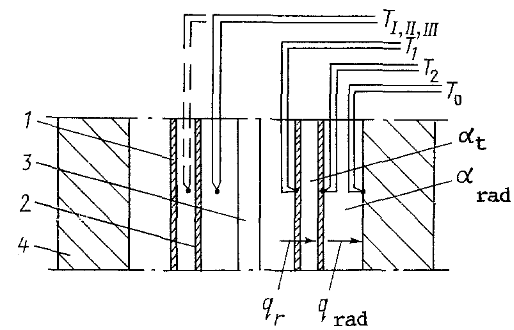

Fig. 2. Temperature measurement scheme for a blind radiation tube on a laboratory stand: q_r, q_{rad} – heat fluxes; α_t, α_{rad} – heat-transfer coefficients; $T_{I\text{-}III,1\text{-}2}$, temperatures from Eq. (2)-(4): 1) radiation tube; 2) spherical tube-insert; 3) gas line; 4) lining of stand.

$$r = \frac{Q_r}{Q_{rad}} = 1 - \frac{T_{II} - T_{III}}{T_I - T_{III}} \tag{1}$$

Here, Q_{rad} is the thermal radiation of the external tube; Q_r is heat transfer through the wall of the internal tube-insert; $T_{I,II,III}$ are the temperatures of the heat carrier in the combustion zone and at the upper and lower circulation points.

These calculations and conclusions were corroborated experimentally in tests of a blind radiation tube with a dispersed heat carrier. The tests were conducted on a laboratory stand (Fig. 2). We compared three characteristic regimes of particle circulation in the annular channel of the structure: dense settling layer; freely falling gas suspension; the gravitational particle fluidization regime described above. Due to the confined conditions in the channel – and to avoid disturbing the hydrodynamics of the flow on the transducer side – we estimated the rate of heat exchange between the particles and the wall (surface) indirectly by an original method [10] based on Eq. (1). Here, the rate is determined by the temperature gradient and reaches its maximum in the combustion region:

$$q_r = \frac{\alpha_t}{2}(T_1 - T_2), \tag{2}$$

$$q = q_{rad}\,(T_2 - T_0), \tag{3}$$

$$q_r = r q_{rad}, \tag{4}$$

where q_r and q_{rad} are the corresponding heat fluxes; α_{rad} is the coefficient of heat transfer by radiation outside the tube; α_t is the coefficient of heat transfer between the particles and the wall in the annular channel between the internal and external tubes; $T_{1,2}$ are the wall temperatures in these tubes; T_0 is the temperature of the surrounding medium (in the furnace). The coefficient 2 in Eq. (2) accounts for the dual thermal resistance of the dispersed flow between the walls of the annular channel. Meanwhile, we can ignore the curvature of the channel for diameter ratios $(D_1/D_2) > 0.5$.

It follows from (1-4) that if the temperature measurements are consistent with the temperature T_I as shown in Fig. 2, then a comparative estimate of heat exchange between the particles and the wall can be made for any particle circulation regime on the basis of the relative change in these temperatures [10]

$$\frac{\alpha_t}{\alpha_{rad}} = 2\frac{T_2 - T_0}{T_1 - T_2}\left(1 - \frac{T_{II} - T_{III}}{T_I - T_{III}}\right), \tag{5}$$

where the coefficient of heat transfer by radiation α_{rad} outside the radiation tube is found from the familiar relation [1]:

$$\alpha_{rad} = 5{,}67\,\varepsilon\,\frac{\left(\frac{T_2}{100}\right)^4 \left(\frac{T_0}{100}\right)^4}{T_2 - T_0} \tag{5'}$$

TABLE 1. Characteristics of Flows of a Dispersed Heat Carrier in an Annular Channel

Particle circulation regime	Design diameter ratio	Channel constraint index
Dense settling layer	0,6	20
Freely falling suspension	0,6*	20*
Gravitational fluidization	0,8	10
Wedging of particles	0,9	5

*With incomplete filling of the channel.

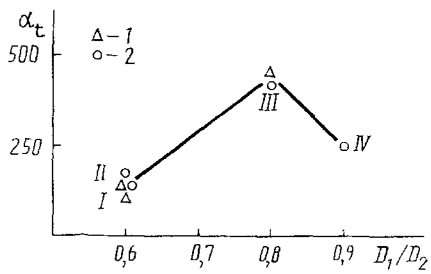

Fig. 3. Coefficient of heat transfer between the particles and the wall of the annular channel α_t, W/(m^2·K), in relation to the diameter ratio D_1/D_2 in different particle circulation regimes: I) dense settling layer; II) freely falling suspension; III) gravitational fluidization; IV) circulation with wedging of particles in the channel; 1) particles of 0.4-0.63 mm; 2) particles of 0.63-1.0 mm.

Here, the emissivity of a radiation tube of steel Kh23N18 was taken as $\varepsilon = 0.7$ (in accordance with handbook data [1]). This quantity had no qualitative effect on the results of the comparison.

The above-indicated regimes of heat-carrier circulation in the channel were established on the basis of the theoretical diameter ratio [10] and an index expressing the degree of constraint of the particles in the channel Δ/d_t. The latter quantity unambiguously characterizes the disperse flows we are discussing [18]. Since the terminology for such flows has not yet been definitively established, we have listed the regime definitions we are using in Table 1 along with corresponding values of the constraint index. Preliminary tests on a transparent (cold) model, in combination with the method of tagged particles, allowed us to visually observe these circulation regimes until the particles became wedged in the channel at the known critical value $(\Delta/d_t)_{cr} = 5$ [11].

The diameter ratios indicated in Table 1 were obtained through the appropriate selection of an internal tube-insert of several standard diameters: 63 × 3.5 mm, 76 × 4 mm, 83 × 4.5 mm. The diameter of the external radiating tube, 102 × 5 mm, was left unchanged. The indicated constraint indices were realized by appropriately choosing the heat-carrier fraction. As the latter, we used quartz (bank) sand 0.4-0.63 and 0.63-1.0 mm in diameter, in accordance with GOST 6139–78. We burned natural gas as the fuel in the tube, its combustion regime being characterized by an amount of excess air that was 10-20% above the norm. The temperature in the combustion zone was 1000-1100°C, which ensured that the tube would have a radiant heating capacity on the order of 20-25 kW/m^2 with a radiative heat-transfer coefficient of roughly 50 W/(m^2·K). Temperatures in the structure were measured by 0.5-mm-diam. chromel–alumel thermocouples embedded in the tube wall. The thermocouples were located at one level, as shown in Fig. 2.

Figure 3 shows results of analysis of these measurements using Eqs. (5)-(5′). These results clearly show the effect of gravitational fluidization, with the rate of heat exchange between the particles and the wall in the annular channel reaching 400

-450 W/(m2·K). This effect also resulted in a decrease in overheating of the structure in the combustion zone by a factor of 1.5-2 compared to the previous case, while the difference in temperatures between the spherical internal tube and radiative external tube was only 40-50°C.

The trial use of radiation tubes with a dispersed heat carrier has shown that the high degree of uniformity and intensity of their heating make it possible to realize a 40-50% increase in the heating capacity of CTT furnaces which employ mainly tubes in which gas is burned by flame combustion. Both the number of tubes used and the amount of gas consumed can also be decreased, given the same productivity and heating capacity of the unit. Finally, the service life of the tubes is increased as a result of the above-indicated excess of air and reserve capacity.

LITERATURE CITED

1. A. M. Semernin and A. E. Erinov, Gas Radiation Tubes [in Russian], Kiev (1968).
2. A. E. Erinov and A. M. Semerin, Industrial Furnaces with Radiation Tubes [in Russian], Moscow (1977).
3. E. V. Kreinin and Yu. P. Kafyrin, Combustion of Gas in Radiation Tubes [in Russian], Leningrad (1986).
4. N. V. Antonishin, E. V. Borisov, and V. A. Kalitko, in: Transactions of the Third All-Union Conference on the Design, Construction, and Use of Radiation Tubes in Industry [in Russian], Kiev (1977), pp. 141-144.
5. N. V. Antonishin, E. V. Borisov, V. V. Lushchikov, and V. A. Kalitko, Inzh.-Fiz. Zh., **38**, No. 2, 217-221 (1980).
6. E. V. Borisov, "Investigation of radiation tubes with a dispersed heat carrier," Author's Abstract of Engineering Sciences Candidate Dissertation, Minsk (1981).
7. N. V. Antonishin, E. V. Borisov, V. A. Kalitko, and V. A. Nemkovich, in: Transactions of the IV All-Union Conference on the Design, Construction, and Use of Radiation Tubes in Industry [in Russian], Kiev (1982), pp. 85-88.
8. V. A. Kalitko, Heat and Mass Transfer in Production Processes and Equipment [in Russian], Minsk (1985), pp. 26-33.
9. V. A. Kalitko and M. A. Geller, Vestsi Akad. Navuk B. SSR, Ser. Fiz. Énerg. Navuk, No. 3, 62-68 (1986).
10. V. A. Kalitko, "Intensification of heat transfer and stabilization of hydrodynamics in gas radiation tubes with circulation of a dispersed heat carrier," Author's Abstract of Engineering Sciences Candidate Dissertation, Minsk (1989).
11. S. S. Zabrodskii, Heat Transfer and Hydrodynamics in a Fluidized Bed [in Russian], Moscow–Leningrad (1963).
12. Radiant Burner: Inventor's Certificate No. 507748 SSSR: F 23 D 13/12.
13. Radiant Burner: Inventor's Certificate No. 850986 SSSR: F 23 D 13/12.
14. Method of Starting a Combustion Chamber: Inventor's Certificate No. 885705 SSSR F 23 C 11/02.
15. Method of Starting a Combustion Chamber: Inventor's Certificate No. 1375904 SSSR: F 23 C 11/02.
16. M. R. Judd and P. N. Rowe, Advances in the Theory and Practice of Fluidization [Russian translation], Moscow (1980), pp. 97-111.
17. I. I. Mukhin, in: Hydrodynamics and Heat and Mass Transfer in Granular Media [in Russian], Ivanovo (1985), pp. 52-54.
18. Z. R. Gorbis, Heat Transfer and Hydrodynamics of Dispersed Continuous Flows [in Russian], Moscow (1970).