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Direct flame impingement heating for rapid thermal materials processing

G.K. Malikov [a], D.L. Lobanov [a], K.Y. Malikov [a], V.G. Lisienko [a], R. Viskanta [b,*], A.G. Fedorov [b]

[a] *Ural State Technical University, Ekaterinburg, Russian Federation*

[b] *Heat Transfer Laboratory, School of Mechanical Engineering, Purdue University, 1288 Mechanical Engineering Building, West Lafayette, IN 47907-1288, USA*

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Abstract

Combined experimental and theoretical investigations have been carried out to study heat/mass transfer and combustion in the direct flame impingement (DFI) furnace for rapid heating of metals in materials processing. A large-size industrial DFI furnace, equipped with a multiflame combustion system, has been instrumented for performing detailed fluid dynamics and heat transfer measurements. The mean and local pressure, fuel mass fractions, temperatures and convective/radiative heat fluxes have been measured and are reported for high jet velocities (up to 230 m/s) and firing rates. In the case of natural gas–air firing, the convective heat fluxes as high as 500 kW/m^2 were recorded with relatively 'cold' refractory wall temperatures (<1400 K). The combustion gas temperature varied between 1500 and 1800 K. A simplified two-dimensional theoretical model was developed to analyze gas flow, flame jet combustion and heat/mass transfer in the DFI furnace. The model developed has been validated against the experimental data and was used to obtain a fundamental understanding of the physical processes taking place in the furnace. In addition, the model has been used as a tool to optimize design and operation of the DFI furnace.

1. Introduction

Natural gas fired rapid heating furnaces, taking advantage of enhanced convective heat transfer from the combustion products to the load, allows one to increase productivity, to improve heating uniformity, to reduce fuel consumption, pollutants emissions, scale formation and load metal oxidation [1–5]. During the past several decades, industrialized countries have developed rapid heating processes using different approaches [1–5,7–9]. However, these furnaces are not widely used by metallurgical and materials processing industry due to lack of reliable data and methodologies for optimal furnace design and operation.

British Gas Corporation has pioneered development of rapid heating furnaces. In such furnaces, the hot gases (combustion products) were generated via combustion taking place inside the special tunnel pre-burners prior to introducing the combustion products into the working space of the furnace [3,7,10]. However, use of pre-burners makes the furnace design and operation more complex and, thus, less reliable; it also results in an increase of the flue gas temperature inside the tunnels that leads to an increase in NO_x formation. In addition, high temperature gradients, which develop inside the tunnel pre-burners, reduce significantly the lifetime of the furnace.

The Direct Flame Impingement (DFI) technology [4,5,11] offers an attractive alternative for many applications in materials processing since it increases significantly the combustion efficiency as well as the total heat transfer rates from the combustion products to the load; it also leads to a decrease in the refractory temperature

* Corresponding author. Tel.: +1-765-494-5632; fax: +1-765-494-0539.

E-mail address: viskanta@ecn.purdue.edu (R. Viskanta).

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Nomenclature

d	internal diameter of the nozzle
h	enthalpy
H	distance from the nozzle tips to the load
k	turbulent kinetic energy
m	species mass concentration
P	normalized dynamic pressure $= P/P_0$
Q	calorific value
r	radial coordinate
S	source term in transport equation or surface area
T	temperature
U, V	axial and radial jet velocities
x	coordinate along jet axis
ε	rate of turbulence energy dissipation
ϕ	generalized scalar variable
ρ	density

by as much as 150–200°C as compared to conventional radiant furnaces [11–13]. In addition to the above mentioned benefits, the rapid heating furnaces with DFI technology are more compact and require 1.5–2 times less refractory materials. Startup of the DFI furnace takes 10–20 min instead of days for the conventional furnace [4,5]. Owing to a decrease in the flue gas and refractory temperatures, the NO_x emission is 2–3 times lower in the DFI furnaces [11–13].

In the DFI furnace, the heating of the workpiece is accomplished via numerous high-speed burning jets of the pre-mixed fuel-air mixture introduced into the furnace chamber through the nozzles in the refractory walls. As such, no combustion chambers, tunnels or flame holders are needed for the process. The DFI process takes advantage of the high velocity of the fuel–air jets (up to the Mach number equal to (1) to insure high convective heat fluxes from the hot combustion products to the load in addition to radiant heating of the load from the hot refractory walls. Owing to complexity of physicochemical processes, successful application of the multiple flame jet system requires more elaborate furnace design and optimization of operating parameters such as (1) nozzle geometry and dimensions, (2) nozzle arrangement (i.e., the distance between the individual nozzles and between the groups of nozzles in the longitudinal and transverse directions), (3) specific total number of nozzles (i.e., per unit area of the load surface), (4) distance between the nozzles and the metal load to be heated, (5) fuel supply distribution between the furnace zones and the nozzle groups, and (6) firing rates and velocity of the gas leaving the nozzles [11,12].

A number of experimental and theoretical studies of the flame impingement has been performed over the years, and comprehensive reviews are available [13–15]. It should be emphasized that most of the previous works related to this topic have been concerned with an idealized situation of the low velocity pre-mixed single jet burning at ambient conditions [6,13,14]. Only recently, Malikov et al. [12] presented the experimental data on heat/mass transfer for an array of high speed flame jets arranged at the top and the bottom of the load in the furnace within a rectangular combustion space. In contrast to commonly accepted view that the flame separation (instability) is more likely to occur with an increase in the jet velocity [6], Malikov et al. [12] demonstrated for the first time that the combustion in the high jet velocity system (up to the speed of sound) may be sufficiently stable if the furnace environment is confined and maintained at the high temperature.

Although the experimental data reported [12] have been obtained under conditions similar to those in the actual operating furnaces [4,5], the geometry of the test furnace was rectangular (not cylindrical) and much simplified as compared to actual industrial DFI furnaces. Therefore, the purpose of this paper is to present a study of combustion and heat transfer in the actual large-scale industrial DFI furnace and to demonstrate very high efficiency of DFI furnaces for rapid heating of round metal bars and tubes. To this end, a detailed experimental study has been undertaken to obtain data for a DFI furnace. The measurements of the dynamic pressures, concentrations of combustion products, temperatures and the heat fluxes (both convective and radiative) from the hot combustion products to the impinging surface of the load (workpiece) are reported in this paper. A simplified two-dimensional fluid dynamics, heat/mass transfer and combustion model developed and described in our earlier paper [12] has been used to simulate the processes. The model predictions of the flow, convective and radiative heat transfer have been validated by comparing the calculations against the experimental data.

2. Experiments

The test section was designed as an integral part of an industrial gas-fired DFI furnace that has been used for heating of continuously supplied tube before the rolling mill [5,16]. The furnace chamber was cylindrical in shape and was lined with the refractory bricks. The internal diameter of the test furnace was 0.3 m and it was 1.2 m long (see Fig. 1). The flue gases were exhausted from the furnace combustion space through two chimneys arranged at both ends of the test chamber. A water-cooled, cylindrical calorimeter (0.108 m in diameter) was installed along the furnace axis to simulate a cold tube to

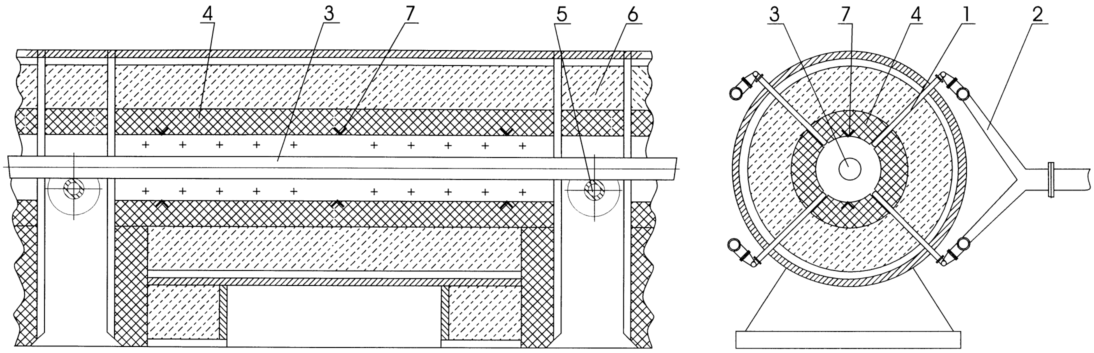

Fig. 1. Schematic diagram of the experimental furnace: 1 – nozzles, 2 – gas-mixture manifold, 3 – water cooled calorimeter, 4 – rerfractory, 5 – rolls, 6 – flue chimney box, 7 – thermocouple locations.

be heated and to measure the heat fluxes from the hot combustion products to the workpiece (i.e., tube). The normalized distance between the nozzle tips and the calorimeter (H/d) was 4, with d being the nozzle diameter. Our previous experiments [12] indicate that this distance is sufficient to achieve high mean temperatures of the burning jet before the hot gases (combustion products) impinge on the surface of the workpiece. Two variants of the nozzle arrangement with different number of nozzles were employed: (i) 4-row arrangement with 24 nozzles of 4 mm diameter per row (i.e., totally 96 nozzles) and spacing 50 mm $\times$ 85 mm in axial and circumferential directions, respectively; and (ii) 4-row arrangement with 12 nozzles of 6 mm diameter per row (i.e., totally 48 nozzles) and spacing 100 mm $\times$ 85 mm in axial and circumferential directions, respectively. Note that in the latter case, the normalized distance from the nozzle tips to the calorimeter was 16.

A standard 50 kPa industrial air blower was used to achieve high mixture velocities at the nozzle tip (exit) up to 400 m/s. Pre-heated or ambient air-natural gas mixture was supplied through the manifold and distributed uniformly between individual nozzles. The firing rates studied were in the range of 250–600 kW to insure rapid heating of the tubes. The test section was specially designed to ignite the fuel–air mixture in the vicinity, but downstream of the nozzle tips, so that the combustion would continuously take place up to the surface of the calorimeter. The strong recirculation zones created by the gas stream leaving the nozzles provide intense and stable combustion over a wide range of jet velocities. The firing rates in excess of 20 m^3 per hour of natural gas were used to guarantee stable ignition and combustion of the reactants.

Owing to very high velocities and temperatures of the flame jets near the load surface, large convective heat fluxes have been measured by the calorimeter. Also, the uniform refractory and load temperatures and complete fuel combustion within the furnace have been observed as a result of the intense internal recirculation and mixing of the combustion products within a confined space of the test chamber. In addition, high velocities of burning jets have been instrumental in preventing an occurrence of the flashback.

A number of tests has been performed to study thermal performance of the DFI furnace over the following range of operating parameters.

- Equivalence ratio of the gas–air mixture was varied from 1.0 to 1.05.
- Heat input per unit calorimeter surface area was varied from 0.5 to 1.27 MW/m^2.
- Velocity at the nozzles tip (exit) was varied from 65 to 165 m/s for the ambient air and from 50 to 120 m/s for the pre-heated (up to 750 K) air.
- Temperature of the combustion gases was varied from 1500 to 1900 K.
- Temperature of the furnace refractory was varied from 1200 to 1400 K. (Note that during the tests the surface temperature of the calorimeter remained in the range between 420 and 460 K.)

The dynamic pressure along the axis of the burning jet was measured at several locations using specially designed thin quartz probes with a diameter of 0.5 mm. The probes were calibrated in cold free jet flow with known velocity distribution to provide the accurate data for the high velocity jet flows. The local velocities were then determined from the measured dynamic pressure profiles.

The local flame temperatures of the high velocity jets were measured using the thermocouple probes made from 0.1 mm diameter (platinum–platinum–13% rhodium) wire, which was covered with a 0.01 mm thick silicon film to prevent the surface temperature rise due to catalytic combustion [6,19,20].

The local gas temperatures in the low velocity regions of the furnace in the regions (e.g., outside of the free jet core in the wall jet area) were measured using a suction pyrometer. Gas species (CO, CO_2, CH_4, H_2 and O_2) concentrations were measured by using gas chromatography. All species measurements were normalized by using the calorific value of methane Q_g in order to estimate the weighted average fuel concentration ($m_g = \sum_i m_i Q_i / Q_g$, where Q_i is the species calorific value and m_i is the mass fraction of the species CO, H_2 and CH_4). The overall heat flux to the calorimeter was calculated from the amount of heat absorbed by the cooling water. The refractory temperature was measured with (platinum–platinum–13% rhodium) wire (0.5 mm in diameter) installed flush with the surface of the refractories, as shown at the locations in Fig. 1. The temperature of exhaust gases was also recorded with the same kind of thermocouples. Further details on the measurement systems can be found elsewhere [12].

The experimental uncertainties have been estimated to be the following: the gas temperature measurements are accurate to ±50 K and the refractory temperatures to ±10 K; the uncertainty of the dynamic pressure is ±5%; since the velocity is calculated from the dynamic pressure, its relative uncertainty is even lower than ±5%; the uncertainty in the average heat flux measurements is estimated to be between ±2 and 3% [12].

To check the experimental setup and instrument calibration, first the preliminary experiments with hot and cold air blowing (without combustion) were carried out. The temperature and flow field measurements were obtained and a good agreement between the results and those published in the literature for a single jet and array of jets [21–23] has been obtained.

3. Theoretical analysis

The physical processes taking place in the DFI rapid heating furnace are complex and, in general, are defined by a set of mass, momentum, species and energy conservation equations in three-dimensional formulation. In addition, the combustion kinetics has to be accounted for in the problem formulation. Clearly, solution of the model equations for the entire furnace domain including more than 96 burning jets impinging on the metal load is a rather costly and time consuming problem, even for the most powerful computers. Our experience with the furnace design shows that sometimes it may be necessary to make hundreds of calculations to find the optimal nozzle arrangement and the set of suitable operating parameters [12,24]. Therefore, for practical applications it is necessary to formulate a simplified, yet physically sound, theoretical model of the furnace that (1) gives an acceptable accuracy for engineering purposes, and (2) produces solutions on a personal computer in reasonable time. The intent of the model is not only to obtain understanding of the processes but also to use it as an engineering analysis and design tool. Therefore, a simplified two-dimensional axisymmetric model for a single flame jet is employed in this study. The detailed description of the model, the governing equations and the method of solutions are reported in our earlier paper [12] and, therefore, are not repeated here. Instead, only the summary of main features of the physical/mathematical model is presented.

The model includes a mass conservation (continuity) equation and set of two-dimensional conservation equations for the linear momentum and scalar variables such as the enthalpy, turbulent kinetic energy and its rate of dissipation and concentrations of the species [18]. The Arrhenius equation accounting for turbulent eddy-break-up and local temperature fluctuations [12,18] was used as a combustion kinetics model to express the reaction rate. The radiative heat transfer is treated using a zonal exchange method on the coarse grid [17] which is then coupled with the model conservation equations and are solved on the fine mesh. The model was further simplified by assuming that (i) cross-flow effects are negligible, (ii) interaction between neighboring flame jets is negligible, and (iii) temperature and velocity fields are similar for all flame jets (indeed, the measurements have shown the maximum deviation between the different jets to be 5%).

Briefly, the model entails numerical simulation of combustion, radiation and the turbulent transport of mass, momentum, chemical species and energy within the furnace. The general form of the mass-weighted averaged conservation equations for steady, two-dimensional axisymmetric flow are expressed as [12]:

Conservation of mass:

$$\frac{1}{r}\frac{\partial}{\partial r}(r\rho V) + \frac{\partial}{\partial x}(\rho U) = 0. \tag{1}$$

Conservation of the general scalar:

$$\begin{aligned}\frac{1}{r}\frac{\partial}{\partial r}(r\rho V\phi) + \frac{\partial}{\partial x}(\rho U\phi)\\ = \frac{1}{r}\frac{\partial}{\partial r}\left(r\Gamma_\phi\frac{\partial\phi}{\partial r}\right) + \frac{\partial}{\partial x}\left(\Gamma_\phi\frac{\partial\phi}{\partial x}\right) + S_\phi,\end{aligned} \tag{2}$$

where ϕ stands for velocity components (U and V), k the turbulent kinetic energy, ε the dissipation rate for kinetic energy of turbulence, m_i the species mass concentration and h is enthalpy. In this equation, S_ϕ is the source term (due to combustion, for example), and Γ_ϕ is the temperature dependent diffusion coefficient Γ_ϕ for given dependent variable ϕ, respectively. A standard k–ε turbulence model [25,26] is employed, and the definitions are not repeated here for the sake of brevity. The boundary conditions are identical to those used in our previous study [12] and, therefore, are omitted from the discussion as well.

An orthogonal grid of 33 × 25 in x- and r-directions, respectively, was used in all calculations. The grid was non-uniform near the solid surfaces so that the closest nodes were placed at the distance about $0.05d$ from the solid surface. The two momentum equations were written in the pressure–velocity form and were solved using an established procedure [17,27]. The zonal radiation exchange method [17] was used to predict radiative heat transfer. The main features of this method include the following: (1) The enclosure walls are divided into finite width parallel planes to speed up the calculation of the radiation exchange factors for the zones by taking into account the selective spectral emission of the gases. (2) Simple expressions are used to subdivide radiation heat fluxes between the nodes in any zone, which proved to be valid for a wide range of opacities.

4. Results and discussion

In the results presented, the distances are normalized by the nozzle diameter (d), and the flow velocities are scaled using the baseline fuel–air mixture volumetric flow rate (m^3/s at 273 K)$/(0.785d^2N)$, where N is the number of nozzles. Note that the numerical scaling factor 0.785 in the previous expression comes from projecting the actual geometry of the furnace onto the geometry of a single-jet-based computational domain [12]. The dynamic pressures and fuel mass concentrations were normalized by their respective values at the exit of the nozzle.

Fig. 2 shows a decrease in the normalized dynamic pressure along the jet axis as a function of the normalized distance (x/d) from the nozzle exit. For the purpose of model validation, the isothermal measurements [21] and model predictions are compared for the hot (denoted by '×') and cold (denoted by '+') air jets without combustion. An excellent agreement between the results of numerical simulation and the measurements has been obtained. Note that neglect of the temperature fluctuation term in the model equation leads to predictions of lower temperatures by about 200–250 K as compared to the experimental data. Fig. 2 also indicates that the dynamic pressure for the burning jets decays with the distance at considerably smaller rate than that for the isothermal jets. For instance, at $x/d = 16$, the burning and the isothermal jets have dynamic pressures at the centerline of 0.28 and 0.15, respectively. Hence, it can be expected that the impinging flame jets produce much higher convective heat fluxes than the isothermal impinging jets of combustion products in the conventional radiant heating furnaces with tunnel pre-burners. The theoretical predictions indicated by solid lines in Fig. 2 agree with the measurements within 7%. Although not presented here, the numerical calculations showed that the local mass flow rates of the circulating hot combustion products between the jets is about 6–7 times higher than the mass flow rate supplied through each nozzle. This explains the excellent mixing of the combustion gases and, consequently, the complete combustion of fuel within the furnace.

Fig. 3 presents local temperature distributions along the axis of the flame jets for different velocities of the air–fuel mixture at the nozzle exit. The burning jet temperature increases rather fast downstream from the nozzle tip and it reaches relatively high values in the impingement zone ($x/d > 17$). The measurements and calculations show that most of the furnace volume is occupied by the hot gases at a fairly uniform temperature, and the cold (below 1000 K) jet cores occupy less

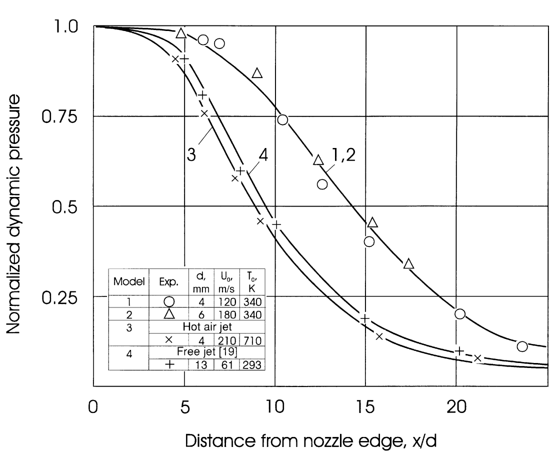

Fig. 2. Variation of normalized dynamic pressure along the flame jet axis.

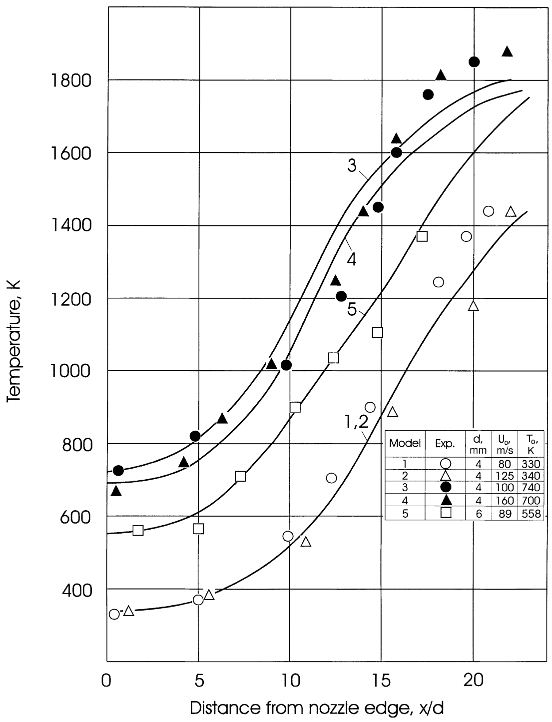

Fig. 3. Variation of temperature along the flame jet axis.

than 1% of the furnace internal volume. It is clear from Fig. 3 that the length of the cold jet core is equal to about 17–18 nozzle diameters for an ambient air and about 10 nozzle diameters for a pre-heated air. Although the core temperature of the flame just before its impacting the calorimeter surface (at the stagnation point) can be rather low (for instance, less than 1100 K for small separation distances between the nozzle exit and the calorimeter), yet the combustion process accelerates rapidly in the wall jet region.

Finally, it should be pointed out that the temperature along the flame jet axis increases from 340 to 1300–1400 K for the ambient air and from 700 to 1600–1800 K for the pre-heated air within a short distance of about $h/d = 20$, thereby ensuring high convective heat transfer rates from the combustion products to the load. Furthermore, as expected, pre-heating of the air decreases considerably the distance between the nozzles and the load which is guaranteeing the completion of the combustion process. This allows one to design very compact and highly efficiency natural gas-fired furnaces [11,12].

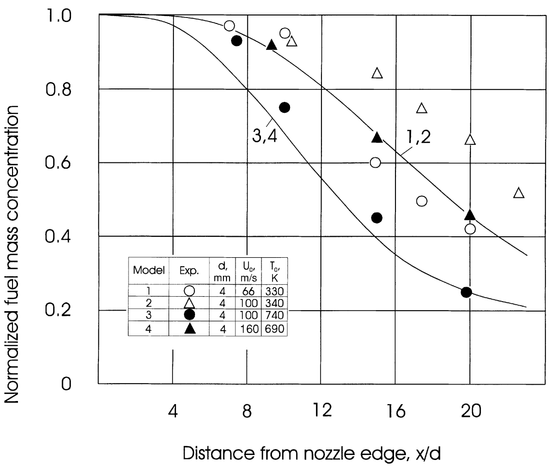

Fig. 4. Variation of scaled fuel concentration along the flame jet axis.

The furnace atmosphere was found to be non-luminous. No visible flame eddies were observed, and no combustibles were found in the gas samples extracted from the zones beyond the thin jet flames. The cold core of the flame jets appeared to be slightly blue and extended into the impingement regions. No soot deposits were detected on the cold surfaces of the calorimeter.

Fig. 4 compares the experimental data and the calculations of the fuel concentrations along the flame jet axis. As expected, owing to combustion, the fuel concentrations decrease with the distance from the nozzle. However, not all fuel is burned as the jet reaches the stagnation point. Therefore, the combustion process continues with enhancement in the near-wall jet deflection region over the surface of the workpiece (calorimeter in our case). In spite of the fact that the water-cooled calorimeter is cold, the chemical reactions are not quenched. This is because the combustion is stabilized and essentially completed through the interaction between the hot combustion products from the adjacent flame jets.

Fig. 5(b) (top panel) depicts behavior of the average combustion product and refractory temperatures with an increase in the velocity of the air jets leaving the nozzles. Clearly, very high degree of the uniformity in both temperatures is observed in the DFI furnace. The refractory temperatures [averaged as $T^4 = (\sum T_i^4)/N$, where N is the number of thermocouples installed in refractory walls] are 200–300 K lower than those encountered in the conventional furnaces at comparable specific firing rates and heat fluxes [2–5]. This allows one to use relatively inexpensive refractory materials in the design of the DFI furnaces. At the maximum firing rates, the local combustion gas temperatures do not exceed 1900 K, whereas in the high velocity tunnel burners installed in the conventional heating furnaces the local temperatures may exceed 2000–2300 K when using pre-heated air [12]. In addition, excellent mixing conditions and relatively low combustion temperatures provide the low levels of nitrogen oxides. The NO_x concentration in the final combustion products was only 35–50 ppm at 2% excess oxygen.

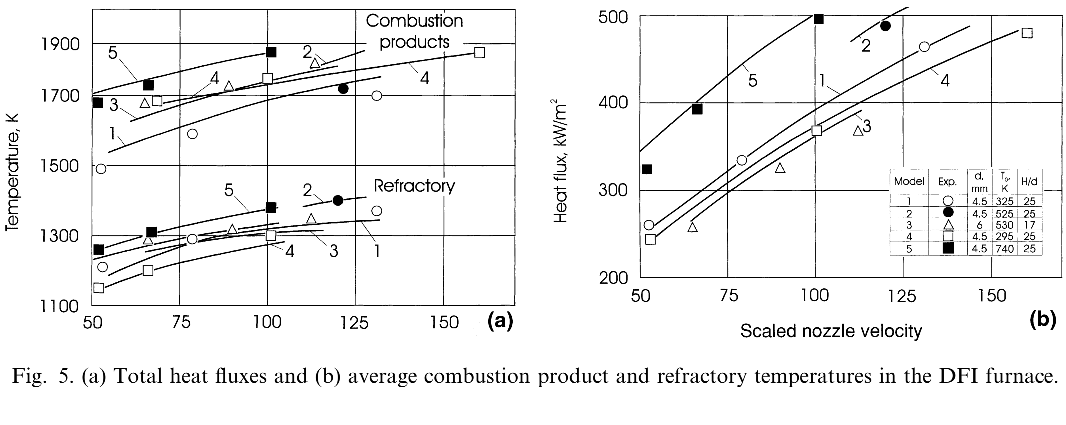

Fig. 5. (a) Total heat fluxes and (b) average combustion product and refractory temperatures in the DFI furnace.

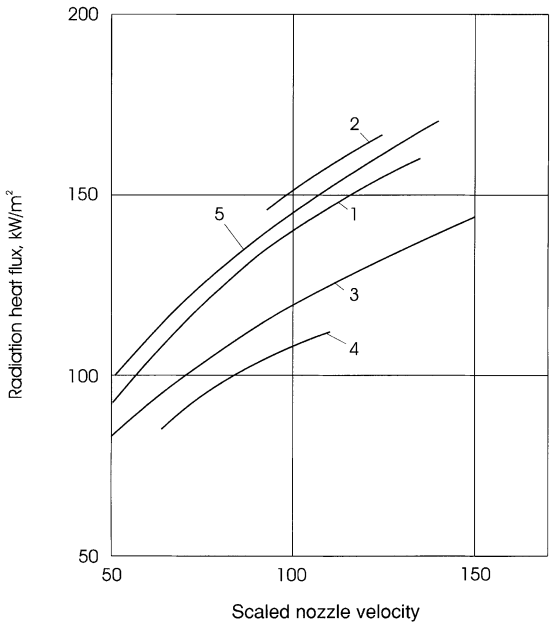

Fig. 6. Model predictions of the radiative heat fluxes at the calorimeter surface (refer to Fig. 5 for legend).

Figs. 5(a) (bottom panel) and 6 show the measured and calculated total and radiative heat fluxes, respectively, at the calorimeter surface for different jet velocities. Note that only the theoretical results are given in the case of radiative heat fluxes. Clearly, the radiative fluxes account for only about 1/3 of the total heat flux, while the remaining 2/3 are contributed via extremely efficient convective heat transfer from the hot jets impinging on the surface of the metal load. Indeed, the convective heat fluxes are not only about two times larger than the radiative fluxes, but they are also 5–10 times larger than the convective heat fluxes found in conventional furnaces [2]. This fact provides the reason for extremely fast start-up of the DFI furnace as compared to the conventional heating technologies.

5. Concluding remarks

A fundamental investigation of the DFI heating process has been undertaken with an aim to understand and quantify the mechanisms of heat transfer from burning jets impinging on the load surface, thereby providing a sound foundation for design and optimization of novel industrial furnaces for rapid thermal materials processing. Based on the results obtained, the following conclusions can be drawn:

1. If properly designed, the DFI heating system can utilize very high jet velocities (up to Mach number of unity) to achieve an extremely efficient, rapid and uniform heating of the load via combined convective–radiative heat transfer from the hot chemically reacting gases in the jet without causing the flame instability. In contrast to conventional heating furnaces, as much as 60–70% of the total heat transferred from the hot combustion products to the load is by convection rather than by thermal radiation from the refractories.
2. The high velocities and temperatures in the impingement zone enhance the convective heat transfer from the hot combustion gases to the load. The total heat fluxes up to 500 kW/m^2 can be readily achieved at the surface of the load, while maintaining relatively low refractory temperatures (below 1400 K). The measured maximum NO_x concentrations of 35–50 ppm were significantly lower than those found in the conventional furnaces equipped with the tunnel burners (150–200 ppm).
3. A relatively small distance of less than 100 mm between the nozzle tips and the workpiece has been shown to be sufficient for combustion gases to attain the temperatures of the level 1400–1800 K in the vicinity of the load surface. This gives an opportunity to design highly compact furnaces for rapid heating of metals.
4. A simplified two-dimensional theoretical model for turbulent flow, combustion, and heat transfer developed has been found to be fairly accurate and very efficient (as far as the computer resources are concerned) in predicting thermal performance of the DFI furnace as compared with the experimental data. As such, the model can be used as an effective tool in optimizing design and operating conditions of the DFI furnace.
5. The DFI system has low thermal inertia and requires only minimal initial heat up before operation. Consequently, this results in greater flexibility of production scheduling and processing efficiency. The proposed DFI rapid heating technology has potential (as demonstrated [4,5,11]) for heating metal bars, rods, slabs or strips, including reheating of slabs and billets and continuous heating of bars and strips in ferrous or non-ferrous metals manufacturing.
6. Key issues associated with practical implementation of DFI process affecting homogeneity of the target surface and metal temperatures, impact of nozzle configuration (e.g., number, spacing, size) on attaining temperature uniformity for heating metal bars of different shapes and sizes and overall heating performance need to be investigated.

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References

[1] A.G. Menshikov, Thermal treatment of railroad rails with high-velocity jet heating, Stal 6 (1994) 59–61 (in Russian).

[2] V.N. Assaturov, P.G. Krasnokutsky, P.A. Berkovskaya, Rapid Jet Heating of Metal, Technika, Kiev, 1984 (in Russian).

[3] J. Masters, Productive developments in heating and heat treatment, Metall. Metal Forming 41 (1) (1974) 13–22.

[4] G.K. Malikov, F.R. Sclar, G.K. Kabakov, DFI-furnace operating in tube reducing tube mill, Stal 7 (1983) 80–82 (in Russian).

[5] V.G. Lisienko, G.K. Malikov, Yu.K. Malikov, I.Yu. Medvedev, D.L. Lobanov, Efficiency of jet flame impingement heating in industrial furnaces, Stal 6 (1996) 45–48 (in Russian).

[6] A. Milson, N.A. Chigier, Studies of methane and methane flames impinging on cold plates, Combust. Flame 21 (3) (1973) 295–305.

[7] E. Dailey, Application of gas-fired rapid heating, Metall. Furnaces Suppl. 61 (12) (1994) FUS1, FUS4, FUS6.

[8] A.L. Bergauz, E.I. Rosenfeld, Enhancement of Efficiency of Fuel Combustion in Thermal and Heating FuNedra, Nedra, Leningrad, 1984 (in Russian).

[9] J.M. Beer, N.A. Chigier, Impinging jet flames, Combust. Flame 12 (1968) 575–586.

[10] V.L. Gusovsky, A.E. Lifshits, Burners for Heating Furnaces, Metallurgia, Moscow, 1978.

[11] G.K. Malikov, D.L. Lobanov, F.R. Skljar, Continuous furnace for rapid heating, Russian Patent No. 726400.

[12] G.K. Malikov, D.L. Lobanov, Yu.K. Malikov, V.G. Lisienko, R. Viskanta, A.G. Fedorov, Experimental and numerical study of heat transfer in a flame jet impingement system, J. Inst. Energy 72 (1999) 2–9.

[13] R. Viskanta, Heat transfer to impinging isothermal and flame jets, Exp. Thermal Fluid Sci. 6 (1993) 111–134.

[14] C.E. Baukal, B. Gebhart, Review of flame impingement heat transfer studies. Part 1: experimental conditions. Part 2: measurements in complex geometry ducts, Combust. Sci. Technol. 104 (4–6) (1995) 339–385.

[15] R. Viskanta, Convective and radiative flame impingement heat transfer, Int. J. Transport Phenomena 1 (1998) 22–28.

[16] G.K. Malikov, F.R. Scljar, V.A. Korshunov, Section furnace for rapid heating of elongated billets, Russian Patent No. 852947.

[17] V.G. Lisienko, G.K. Malikov, Y.K. Malikov, Zone-node method for calculation of radiant gas flows, Numer. Heat Transfer 22 (1) (1992) 1–24.

[18] E.E. Khalil, D.B. Spalding, J.H. Whitelaw, The calculation of local flow properties in two-dimensional furnaces, Int. J. Heat Mass Transfer 18 (6) (1975) 775–791.

[19] D. Bradley, K.J. Matteus, Measurements of high gas temperatures with fine wire thermocouples, J. Mech. Eng. Sci. 10 (4) (1968) 299–305.

[20] B.S. Soroka, Heat Transfer Enhancement in Fuel Furnaces, Technika, Kiev, 1992 (in Russian).

[21] C.D. Donaldson, R.S. Snedeker, A study of free jet impingement. Part 1, J. Fluid Mech. 45 (1971) 281–319.

[22] D.M. Kercher, W. Tabakoff, Heat transfer by a square array of round jets impinging perpendicular to a flat surface including the effect of spent air, Trans. ASME J. Eng. Power 92 (1970) 73–82.

[23] R. Gardon, J. Cobonpue, Heat transfer between a flat plate and jets of air impinging on it, in: Proceedings of the International Heat Transfer Conference. Part 2, ASME, New York, 1961, pp. 454–460.

[24] Y.K. Malikov, V.G. Lisienko, G.K. Malikov, N.A. Chusovitin, Computer design, industrial application and operation of a jet recuperator, Stal 9 (1986) 98–100 (in Russian).

[25] B.E. Launder, D.B. Spalding, A numerical computation of turbulent flows, Comput. Methods Appl. Mech. Eng. 3 (2) (1974) 269–289.

[26] H. Ramamurthy, S. Ramadhyani, R. Viskanta, A two-dimensional axisymmetric model for combusting, reacting and radiating flows in radiant tubes, J. Inst. Energy 67 (1994) 90–100.

[27] P.F. Galpin, J.P. Van Doormal, G.B. Raithby, Solution of the incompressible mass and momentum equations with application of a coupled equation line solver, Int. J. Numer. Meth. Fluids 5 (1985) 615–625.