THE IMPACT OF THE HEIGHT OF ROTATING-HEARTH FURNACE WORKSPACE ON FUEL CONSUMPTION

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Abstract
Steel and iron making industries have been driven by the requirements of maintaining or improving product quality and minimization of production costs. Within these requirements the overall energy management and energy efficiency play important role. Since the rotating-hearth furnace in the seamless pipe rolling mill has the nominal heating capacity of 27 t/h, and the real needs are of an average 13 t/h, the aim of the conducted study was to investigate whether it is possible by decreasing the height of rotating-hearth furnace workspace to reduce fuel consumption. In the article's introduction, a short description of the furnace with the measured values of heating regime, which served as the basis for calculations, is given. Namely, for the purposes of calculation is necessary along the furnace to know the temperatures of flue gases and the surfaces of linings and heated charge. The intensity of the charge heating depends on the radiation heat transfer from the flue gases and the linings onto the charge and the convection heat transfer from the flue gases onto the charge. It was established with the calculations to what extent the decrease in the furnace space height affects the coefficients of heat transfer by radiation and convection, and on the other hand, the heat loss through the furnace linings on the environment. In the end of the article, the suggested solutions are explained.

Keywords: rotary hearth furnace, steel charge, heat transfer, furnace efficiency

1 Introduction

During the last decades, global energy and market conditions have motivated energy users to improve energy utilization - this means reducing the fuel consumption - in order to remain competitive. For this reason, a research in the thermal processing industry has been directed toward one or more of the following: improving process productivity, improving thermal efficiency, increasing process temperature, improving process temperature uniformity, improving process quality and reducing the pollutant production [1-8].

The goal of a reheating furnace is to heat the steel charge to the minimum temperature consistent with achieving the correct temperature and metallurgical properties at the finishing stands of the mill. Uniformity of the temperature within the load, minimisation of local temperature gradients, avoidance of surface defects such as skid marks, overheating marks and oxidation scale represent the characteristics of the ideal product of the reheating operation [9-14].

The rotary-hearth furnace of the nominal heating capacity (throughput capacity) of 27 t/h is located before the rolling stand of type “pilger” in the line for seamless pipe production. Because of the lower production capacity of the seamless tube mill in the relation to the furnace heating capacity, the furnace works with the decreased furnace output of 13-15 t/h. For this reason, the preheated zone is not fired.

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Since it has been shown [15] that the height of the combustion chamber has a major impact on the intensity of heat transfer, the aim of the conducted study was to investigate whether it is possible by decreasing the height of rotating-hearth furnace workspace to reduce fuel consumption, i.e. to what extent the decrease in the furnace space height affects the coefficients of heat transfer by radiation and convection, and on the other hand, the heat loss through the furnace linings on the environment.

2 Short description of the rotary hearth furnace

The rotary hearth furnace under consideration is designed to heat the charge of round cross-section (diameter of 294÷428 mm), which is used for the production of seamless pipes. The furnace has the nominal heating capacity (throughput capacity) of 27 t/h. The charge is heated up to the final heating temperature of 1260÷1280°C. The furnace basic dimensions are: Medium diameter 14000 mm, External refractory lining diameter 19220 mm, Internal refractory lining diameter 8764 mm, Hearth surface 180 m^2, Hearth width 3988 mm, Height of furnace workspace or Combustion chamber height 1940 mm, and Wall refractory thickness 506 mm.
The burners are installed in the zones: Preheating zone (13 burners of 640 kW each), Heating zone (13 burners of 640 kW each) and Soaking zone (8 burners of 640 kW each), in the mode to provide the movement of the flue gases in the direction opposite to the movement of the charge (**Fig. 1**).

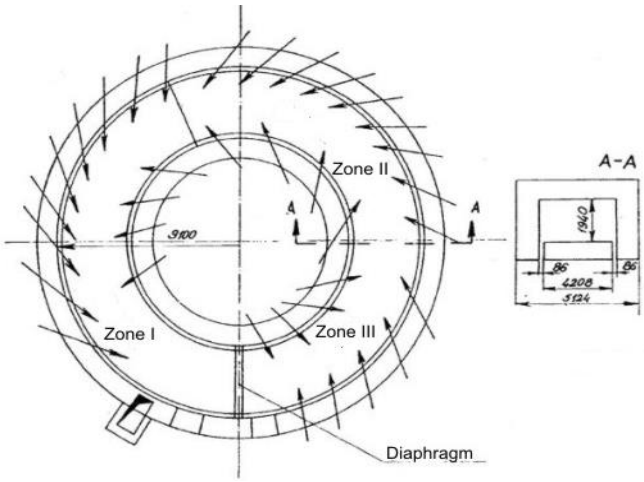

Fig. 1 Zone and burner arrangement in the rotary hearth furnace

The natural gas has the lower heating value of about 34 MJ/m^3 depending on the chemical composition of the supplied gas. In this paper, for the calculation requirements, the lower heating value of natural gas of 34181 kJ/m^3 was used.

3 Measured values of the heating regime

The parameters of the heating regime, which was being monitored, are measured for the steel charge of round cross-section of diameter *ø* 410 mm, length 0.880 m, mass 910 kg, and quality X52. The furnace was charged by 15 billets per hour, stacked in two rows with step of $k = d + b$

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= 820 mm, where *d* is the charge diameter, and *b* is the intercharge gap (thereby $b = d$). The time of charge heating was 6 h and 10 min., so that the throughput capacity was 13.65 t/h. In the **Fig. 2** the measured surface temperatures of walls and charge along the furnace are shown.

For the purpose of accurate calculation, the furnace is divided into seven segments: the first zone in three segments, and the second and third zone into two segments. According to the measured temperature of furnace wall and the charge, the temperature of the flue gases is determined according to data from tables of Heiligenstaedt [16] for ψ =0.2, where ψ is the view factor, i.e. relationship between the surfaces of charge and wall. The results are shown in **Table 1**.

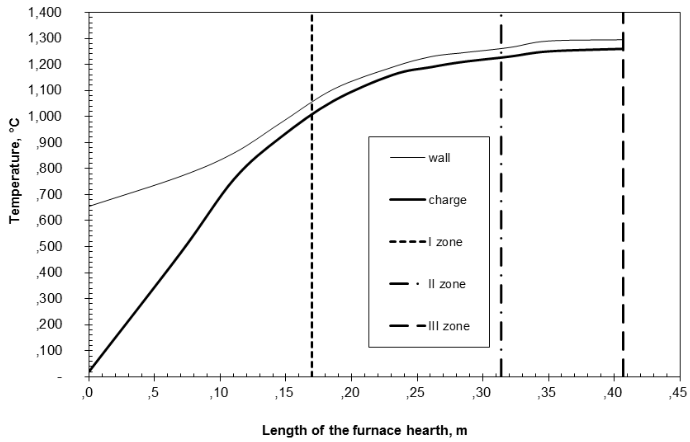

Fig. 2 Temperatures of walls and charge along the furnace

Table 1 Temperatures of the walls, charge and flue gases of the segments

Temperature, ºC		Segments						
		I	II	III	IV	V	VI	VII
Wall								
- starting-point	$\vartheta_{w,s}$	720	820	920	1200	1280	1270	1270
- end	$\vartheta_{w,e}$	820	920	1200	1280	1270	1270	1270
- middle	$\vartheta_{w,m}$	770	870	1060	1240	1275	1270	1270
Charge								
- starting-point	$\vartheta_{m,s}$	20	320	560	990	1160	1230	1250
- end	$\vartheta_{m,e}$	320	560	990	1160	1230	1250	1260
- middle	$\vartheta_{m,m}$	170	440	775	1075	1195	1240	1255
Flue gases								
- starting-point	$\vartheta_{g,s}$	830	920	1020	1320	1400	1380	1380
- end	$\vartheta_{g,e}$	920	1020	1320	1400	1380	1380	1380
- middle	$\vartheta_{g,m}$	875	970	1170	1360	1390	1380	1380

Consumptions of the natural gas and combustion air per zones are given in **Table 2**.

The calculation results of the composition and quantity of flue gases by burning natural gas in excess air of n =1.1 are shown in **Table 3**.

Table 2 Consumption of the natural gas and combustion air per zones

Consumption of the media per zones	Natural Gas (m^3)	Combustion Air (m^3)		Preheated Air Temperature (˚C)
		Set (m^3)	Achieved (m^3)	
I zone	Unfired	0	0	0
II zone	512	4514	4566	238
III zone	298	2668	2672	238

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Table 3 The composition and quantity of flue gases in m^3/m^3 of natural gas

V_{air}	V_{O_2}	V_{CO_2}	V_{H_2O}	V_{N_2}	V_g
10.57	0.101	1.028	1.984	7.794	11.0

4 Calculation of the coefficients of heat transfer

Complete calculations of heat transfer coefficients were derived by the method and the expressions given in refs. [17, 18], according to which many furnaces are designed and can be considered as sufficiently reliable for expert calculations.

Based on the temperatures of charge and flue gases, shown in **Table 1**, the heat transfer coefficient were calculated using the following expressions:

- Radiation heat transfer coefficient, α_r,

$$\alpha_r = C_{red.,g\text{-}m} \cdot \frac{\left(\frac{\vartheta_g + 273}{100}\right)^4 - \left(\frac{\vartheta_m + 273}{100}\right)^4}{\vartheta_g - \vartheta_m}, \quad W/(m^2K) \tag{1}$$

where $C_{red,g\text{-}m}$ is called the reduced radiation coefficient or the coefficient of radiation of gas to metal in the system flue gases-lining-intercharge gap. Since it is also found [19] that the distance between the charges has a major impact on the radiative heat transfer, the influence of the gap radiation on the charge was calculated by the view factors between the gap and the charge, $\varphi_{gap\text{-}m}$, taking into account the reflection of radiation from the adiabatic hearth, in a way $C_{red,g\text{-}m} = C_{g\text{-}w\text{-}gap} \cdot \varphi_{gap\text{-}m}$. The value $\varphi_{gap\text{-}m} = 0.87$ was read from the diagram for $k/d = 0.82/0.41 = 2$, and the value for $C_{g\text{-}w\text{-}gap}$, i.e. the radiation coefficient in the system flue gases-lining-intercharge gap, was obtained by expression

$$C_{g\text{-}w\text{-}gap} = C_0 \cdot \varepsilon_g \cdot \varepsilon_{gap} \cdot \frac{\varphi_{w\text{-}gap} \cdot \left(1 - \varepsilon_g\right) + 1}{\varphi_{w\text{-}gap} \cdot \left(1 - \varepsilon_g\right)\left[\varepsilon_{gap} + \varepsilon_g \cdot \left(1 - \varepsilon_{gap}\right)\right] + \varepsilon_g}, \; W/(m^2K^4) \tag{2}$$

where $C_0 = 5.77$ $W/(m^2K^4)$ is radiation coefficient of absolutely black body, ε_g is emissivity of flue gases, $\varphi_{w\text{-}gap} = l/(A\text{-}B)$ is the portion of effective flux falling from the lining to the intercharge gap (l is the charge length, A is the surface of furnace workspace per 1 m furnace length, and B is the hearth width), and ε_{gap} is emissivity of intercharge gap calculated by relation

$$\varepsilon_{gap} = \left[1 + \frac{k}{\pi \cdot d + k} \cdot \left(\frac{1}{\varepsilon_m} - 1\right)\right]^{-1} \tag{3}$$

where ε_m is emissivity of the charge.

- Convective heat transfer coefficient, α_k,

$$\alpha_k = 0.023 \cdot \left(\frac{\lambda}{D_{ekv}}\right) \cdot Re^{0.8} \cdot Pr^{0.4} \cdot k_t \cdot k_l \tag{4}$$

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The relation is valid for turbulent flow ($Re > 5\cdot10^3$) in ducts. D_{ekv} is the equivalent diameter, which is equal to four cross-sectional areas of the duct divided by the total (wetted) perimeter, Re is Reynolds number, Pr is Prandtl number, λ is thermal conductivity of the flue gases, k_t is correction factor that takes into account non-isothermal flow of the flue gases, k_l is correction factor that takes into account non-stabilized flow in the beginning of the duct.

a) Results of calculation of the radiation heat transfer coefficients

Results of calculation of the radiation heat transfer coefficients for the furnace space height of h =1.94m, h =1.50m and h =1.20m are shown in **Table 4**. The results of calculation show that a reduction in the furnace space height causes a decrease in the value of the radiation coefficient of heat transfer, because the equivalent mean beam length $l \approx 3.6 \cdot V_g/A_s$, where V_g is the total volume of the gas body and A_g is the total surface area, together with the partial pressures of H_2O and CO_2 in the flue gases have, beside the flue gas temperature, decisive impact on the value of the emissivity of flue gases. This decrease is not as significant due to the large hearth width. In the last rows of **Table 4** the percentage reductions in the radiation heat transfer coefficient in relation to the existing height of 1.94 m are shown.

Table 4 Radiation heat transfer coefficients

	Middle of the segment						
	I	II	III	IV	V	VI	VII
$\vartheta_{g,s}$, °C	875	970	1170	1360	1390	1380	1380
$\vartheta_{m,s}$, °C	170	440	775	1075	1195	1240	1255
h =1.94 m							
α_r , W/(m^{2}K)	58.9	97.2	183.2	290.5	294.4	343.6	348.4
h =1.50 m							
α_r , W/(m^{2}K)	58.3	95.2	173.8	281.1	321.3	333.0	337.6
$\Delta\ \alpha_r$, %	-1.0	-2.0	-5.1	-3.2	-8.4	-3.1	-3.1
h =1.20 m							
α_r , W/(m^{2}K)	57.4	93.1	176.8	275.2	321.3	323.6	328.1
$\Delta\ \alpha_r$, %	-2.5	-4.2	-3.5	-5.3	-8.4	-5.8	-5.8

b) Results of calculation of the convective heat transfer coefficients

Results of calculation of the convective heat transfer coefficients for the furnace space height of h =1.94m and h =1.20m are shown in **Table 5**.

Table 5 Convective heat transfer coefficients

	ZONE I			ZONE II			ZONE III		
	Start	End	Middle	Start	End	Middle	Start	End	Middle
ϑ_g , °C	830	1340	1085	1340	1400	1370	1400	1450	1425
ϑ_m, °C	20	750	385	750	1100	925	1100	1180	1140
h = 1.940 m									
α_k, W/(m^{2}K)	1.734	1.732	1.573	1.732	1.628	1.408	0.991	0	0.568
h = 1.20 m									
α_k, W/(m^{2}K)	2.732	2.729	2.705	2.729	2.566	2.222	1.562	0	0.908
$\Delta\alpha_k$, %	+36	+36	+41.8	+36.5	+23.2	+36.6	+36.5	0	+37.4

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It is important to note that the heat transfer by convection was calculated for the case of gas flow in the channel with flat walls, but there is actually flow over the round billets for which an empirical expression in literature was not found. Therefore, it can be expected that the actual heat transfer coefficient by convection is slightly higher, but in any case is the order of magnitude shown in **Table 5**, and have no a significant impact on the total heat transfer in high temperature zones II and III, which is typical for this type of reheating furnace. In the last line of **Table 5**, the increase of the convective heat transfer coefficient is expressed in %.

c) Results of the calculation of heat loss through the walls of the furnace on the environment

Results of the calculation of heat loss through the walls of the furnace on the environment for the furnace space height of h=1.94m, h=1.50m and h=1.20m are shown in **Table 6**.

The meaning of the symbols in **Table 6** is as follows: ϑ_w is the temperature of inner surface of zone walls, q is the heat flow density, $\Phi_{w,\mathrm{zone}}$ is the total heat flow through the walls of each zone, $\Phi_{w,\Sigma}$ is the total heat flow through the furnaces walls.

Table 6 Results of the calculation of heat loss through the walls

	ZONE I		ZONE II		ZONE III	
ϑ_w , °C	1043	1043	1233	1233	1233	1233
q , W/m^2	547	555	731	745	731	745
	h = 1.94 m					
$\Phi_{w,\mathrm{zone}}$, W	$4.322 \cdot 10^4$		$4.111 \cdot 10^4$		$2.648 \cdot 10^4$	
$\Phi_{w,\Sigma}$, W	$1.108 \cdot 10^5$					
	h = 1.50 m					
$\Phi_{w,\mathrm{zone}}$, W	$3.341 \cdot 10^4$		$3.179 \cdot 10^4$		$2.047 \cdot 10^4$	
$\Phi_{w,\Sigma}$, W	$8.568 \cdot 10^4$					
	h = 1.20 m					
$\Phi_{w,\mathrm{zone}}$, W	$2.673 \cdot 10^4$		$2.543 \cdot 10^4$		$1.638 \cdot 10^4$	
$\Phi_{w,\Sigma}$, W	$6.854 \cdot 10^4$					

5 Discussion of results

By reducing the furnace space height from 1.94 onto 1.50 m, taking into account the calculated values of the radiation heat transfer coefficient, the heat transfer by radiation is reduced by an average of 3.7%. On the other hand, the increase of the convection heat transfer is slightly so that it can be neglected. It could be approximated that the reduction in radiation heat transfer prolongs the heating time with the same percentage, i.e. from 6h 10min on 6h 23min.

Taking into account the natural gas consumption of 810 m^3/h, the prolongation of the heating time of 13 minutes would cause the increase of natural gas consumption for 176m^3 per the heating time, and the decrease of the throughput capacity for 0.5 t/h.

Taking into account the lower heating value of natural gas of 34.181 MJ/m^3 and the values of heat flows through the walls in the **Table 6**, by reducing the furnace space height from 1.94 m onto 1.50 m, the heat flow through the walls would be reduced for 25 120 W. This means that the decrease in heat loss through the walls is 557.664 MJ, which is equivalent to the decrease in consumption of natural gas of 17 m^3 per the heating time.

It can be concluded that the reduction in the furnace space height does not contribute to fuel economy, but on the contrary, to the fuel consumption.

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A significant reduction in fuel consumption by reducing the furnace roof would be achieved by the application of the roof heating with flat-flame burners. In this way of heating the heat from flue gases is not transferred directly to the load but indirectly from the furnace roof. Such way of heating can reduce the height of furnace workspace up to 0.8 m achieving the fuel savings of 40% and increasing the quality of heating [20-22].

6 Conclusions

It is not recommended to lower the furnace space height of II and III zone because the fuel savings or increasing the quality of heating cannot be achieved. In addition, it is necessary to put down the burners compared to the existing height, to keep the required existing distance from the furnace roof, which would result in additional costs. Otherwise, this type of furnaces with this heating mode, using the conventional burners, usually have the height of working space of 1.5 m to 2.0 m. Reducing the height lower than 1.5 m would cause uneven load heating with possible local overheating of the load or even the furnace roof;

It may be possible to lower the furnace roof in the first zone onto a level of 1.2 m, because the reduction of heat transfer by radiation is largely compensated by the increase of heat transfer by convection, and the heat loss through the furnace walls is also reduced. Since this zone is not fired, it is not necessary to descend burners, and the occurrence of non-uniform heating is not possible. The transition from II to I zone must be carried out gradually inclining the roof to avoid a significant increase in resistance to flow of flue gases.

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