Study of heat exchange on the glass furnace external surface and energy economy increasing by the way of bathtub walls regulated cooling

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Abstract. The duration of the work campaign and the level of fuel consumption average per work campaign determine the energy efficiency of bathtubs glass melting furnaces working with fossil fuel. This paper shows the investigation of a PHOENICS model of heat transfer on the outer surface of the furnace enclosure. An algorithm is proposed for the thermotechnical calculation of the furnace, which uses adequate ratios in finding the value of heat loss and allows to estimate the error in calculating fuel consumption. The duration of the working campaign depends on the rate of refractory fence erosion in place where it contacts with the melt. One of the ways to slow down erosion is local jet cooling of the refractory outside at the level of the melt mirror. Based on the developed PHOENICS model, it was examined the efficiency of the jet heat transfer for thermal state monitoring of the fence. The results can be used to increase the energy efficiency of glass melting furnaces.

1. Introduction
The container glass production is characterized by significant scale and high energy intensity all over the world. The main consumers of fuel and energy resources are regenerative glass melting furnaces with a capacity of up to 600 tons per day with a specific energy consumption of at least 4.500 MJ / (kg of molten glass), while the theoretical minimum is 2.050 MJ/kg with a cullet mass fraction of 30%. A high level of energy saving potential indicates the availability of reserves to increase the energy efficiency of glass melting furnaces [1].

The energy efficiency of bathtubs glass melting furnaces working with fossil fuel depends on a number of parameters. The most significant are the duration of the work campaign and the average for this period specific fuel consumption. The heat flux through the refractory \( Q_{env} \) has a significant share in the expendable part of the heat balance. Fuel consumption is directly proportional to \( Q_{env} \) or the corresponding heat flux density \( q_{env} \). These values depend on temperature of the inner surface of the refractory, the intensity of convective and radiative heat transfer on the outer surface of the enclosure, as well as the degree of erosive wear of the refractory, which appears due to contact of the inner surface with aggressive media – molten glass and high-temperature gases.

The most intense erosion of the refractory is observed on the inner surface of the side walls of the furnace enclosure in the region of the free surface (mirror) of the molten glass melt. It destroys the enclosing surface and can lead to reduction of the working overhaul campaign. One of the ways to
slow down the erosion process is to use forced cooling of the walls outer surface at the level of the melt mirror in order to low the temperature of the internal refractory surface. Such cooling can be arranged by local jet blowing.

On the outer surface of the fencing elements like the arch, side walls and the hearth there is free-convective and radiation heat exchange with the environment. In the zone of local blowing, forced convection is realized. A reliable calculation of \( q_{env} \) under both free and forced convection conditions has a big interest for engineering practice, energy audit, design and research.

This paper presents the results of heat transfer mathematical modeling through the refractory elements of a glass melting furnace under free convection conditions on the enclosure surface and forced convection during jet blowing of the fencing surface local zone.

2. Study of heat transfer through furnace refractory with free convection on outdoor surface

2.1. Literature data

Heat flux density \( q_{env} \) determines with the equations:

\[
q_{env} = a_{sum}(T_{out} - T_{env}) ; \quad a_{sum} = a_{conv} + a_{rad} ;
\]

\[
a_{rad} = e_{out}^4(T_{out} + 273)^4 - (T_{env} + 273)^4 \bigg/ (T_{out} - T_{env}) ,
\]

where \( T_{out} (°C) \) – temperature of the refractory outer surface, \( T_{env} (°C) \) – temperature of environment, \( a_{sum} (W/m^2/K) \) – total heat transfer coefficient of the refractory outer surface, \( a_{conv} (W/m^2/K) \) – convective heat transfer coefficient, \( a_{rad} (W/m^2/K) \) – radiative heat transfer coefficient, \( e_{out} \) – blackness of the outer surface, \( \sigma_0 = 5.67 \times 10^{-8} \text{ W/m}^2\text{K}^{-4} \) – Stefan-Boltzmann constant.

The following formulas are used for calculation of the heat transfer coefficient for free convection on a vertical wall:

\[
\text{Nu} = \begin{cases} 
0.75 \text{Ra}^{0.25} \left( \frac{Pr_{env}}{Pr_{out}} \right)^{0.25} , & \text{Ra} < 6 \times 10^{10} ; \\
0.15 \text{Ra}^{1/3} \left( \frac{Pr_{env}}{Pr_{out}} \right)^{0.25} , & \text{Ra} \geq 6 \times 10^{10} ;
\end{cases}
\]

\[
\text{Nu} = \begin{cases} 
0.726 (\text{Ra} \cdot \Phi(Pr))^{0.20} , & \text{Ra} < 10^9 ; \\
0.241 (\text{Ra} \cdot \Phi(Pr))^{0.25} , & \text{Ra} \geq 10^9 ;
\end{cases} \quad [3]
\]

\[
\text{Nu} = \begin{cases} 
0.67 (\text{Ra} \cdot \Phi(Pr))^{0.25} , & 10^4 < \text{Ra} < 10^9 ; \\
0.15 (\text{Ra} \cdot \Phi(Pr))^{1/3} , & \text{Ra} > 10^{12} ;
\end{cases} \quad [4]
\]

\[
\text{Nu} = \begin{cases} 
0.825 + 0.387 \text{Ra}^{1/6} \left[ 1 + (0.492/Pr)^{9/16} \right]^{8/27} , & 10^4 < \text{Ra} < 10^{13} . \quad [4, 5]
\end{cases}
\]

Formulas (2) – (5) are obtained under the condition of constant heat flux density on the wall, formula (5) – under the condition of constant wall temperature [4]. In formulas (2) – (5), the determining size \( L \) is the length of the heat exchange surface, with the defining temperature

\[
T_{def} = 0.5(T_{out} + T_{env}) .
\]
To calculate the average heat transfer coefficient for free convection on the arch (horizontal flat plate with the heat transfer surface facing up), correlations (7) in [3, 4], and equations (8) in [5] were recommended:

\[
\text{Nu} = \begin{cases} 
0.766 \text{Ra}^{1/5} \left(1 + \left(\frac{0.322}{\text{Pr}}\right)^{11/20}\right)^{-4/11}, & \text{Ra} < 10^5; \\
0.15 \text{Ra}^{1/3} \left(1 + \left(\frac{0.322}{\text{Pr}}\right)^{11/20}\right)^{-20/33}, & \text{Ra} \geq 10^5;
\end{cases}
\]

(7)

\[
\text{Nu} = \begin{cases} 
0.59 \text{Ra}^{1/4}, & \text{Ra} \in [10^4, 10^7]; \\
0.10 \text{Ra}^{1/3}, & \text{Ra} \in [10^7, 10^{11}].
\end{cases}
\]

(8)

To calculate the average heat transfer coefficient for free convection on the hearth (horizontal flat plate with the heat-transfer surface facing down), formula (9) was proposed in [3], and formula (10) in [6]:

\[
\text{Nu} = 0.56 \text{Ra}^{0.25};
\]

(9)

\[
\text{Nu} = 0.58 \text{Ra}^{1/5}, \quad \text{Ra} \in [10^5, 10^{11}].
\]

(10)

In relations (7) – (10), the defining temperature is calculated by formula (6), and the determining size \( L \) is calculated as the ratio of the heat-exchange surface area to its perimeter [4]. In formulas (2) – (10), \( \text{Nu}, \text{Pr}, \text{Ra} = \text{Gr} \cdot \text{Pr}, \text{Gr} \) are the Nusselt, Prandtl, Rayleigh, Grashof numbers [2], calculated at the defining temperature \( T_{\text{def}} \) and the determining size \( L \); \( \text{Pr}_{\text{out}} = \text{Pr}(T_{\text{out}}) – \text{Prandtl number at wall surface temperature}; \)

\( \text{Pr}_{\text{env}} = \text{Pr}(T_{\text{env}}) – \text{Prandtl number at ambient temperature}. \)

In [7] it was shown that the presented formulas give a noticeable scatter in the values of the heat transfer coefficient. Uncertainty in the choice of ratios \( \alpha_{\text{conv}} \) makes it difficult to perform the thermotechnical calculation of the furnace and obtain an adequate fuel consumption value. In this regard, it is interesting to study free-convective heat transfer on the refractory outer surface of a high-temperature industrial furnace by means of mathematical modeling in order to obtain relations of the form \( \text{Nu} = f(\text{Ra}). \)

2.2. A glassmaking furnace model

A model of a glass melting bathtub, oriented toward solving the task, was developed in the PHOENICS environment [8, 9]. The computational domain (figure 1) includes the half of the furnace started from the vertical plane of symmetry and the adjacent volume of air space. Brown, orange and gray colors indicate the layers of the fence made of different refractory materials. Yellow (upper layer) indicates the gas volume filled with high-temperature combustion products, green (lower layer) indicates the molten glass. The design and operational parameters of the model are based on the results of the thermotechnical inspection of the existing furnace in the glass packaging production system.

The model is designed for stationary thermal conditions of the furnace using the following assumptions. The gas volumes of the furnaces are isothermal, stationary and identical in composition to the product of the complete natural gas combustion; the volume of the molten glass is isothermal; gas and melt infinitely large heat capacity. The study of the models made it possible to obtain the following correlation, which are shown in table 1 [7].

An alternative approach of the heat losses defining through the furnace fencing elements can be obtained on the basis of the data presented in [10] as the dependences of the heat flux densities on \( T_{\text{out}} \). An analysis of the data performed in [7] made it possible to find out that obtained correlation
characterize the total heat transfer and to find the ratios for the total heat transfer coefficient $\alpha_{\text{sum}}$ as a function (12). The values of the coefficients $c_0, c_1, c_2$ for the fencing elements of the furnace are presented in table 2.

$$\alpha_{\text{sum}}(T_{\text{out}}) = c_0 + c_1(T_{\text{out}} - 20)c_2.$$  \hfill (11)

![Computational domain of the glassmaking furnace model.](image)

**Figure 1.** Computational domain of the glassmaking furnace model.

**Table 1.** Correlation for describing surface heat transfer

| Side wall | Arch | Hearth |
|-----------|------|--------|
| $\text{Nu} = 0.488\text{Ra}^{0.326}$ | $\text{Nu} = 0.465\text{Ra}^{0.294}$ | $\text{Nu} = 0.517\text{Ra}^{0.271}$ |

**Table 2.** Ratios of function (11) for $\alpha_{\text{sum}}$

| Fencing element | $c_0$ | $c_1$ | $c_2$ |
|-----------------|-------|-------|-------|
| Arch            | 7.09  | 0.68  | 0.562 |
| Side wall       | 7.20  | 0.56  | 0.592 |
| Hearth          | 7.20  | 0.485 | 0.614 |

2.3. **Calculating the real value of specific fuel consumption**

It is advisable to use the following algorithm, when calculating the fuel consumption for a glass melting furnace:

1. Perform an estimation of $Q_{\text{env}}$. The first option – with the definition of $\alpha_{\text{sum}}$ by formulas (11).

The second option – with the calculation of $\alpha_{\text{conv}}$ by formulas from table 1 and $\alpha_{\text{rad}}$ according to the formula (1).

2. Determine for each option specific fuel consumption – respectively $b_1$ and $b_2$.

3. Define the desired value $b$ as the arithmetic mean of the values $b_1$ and $b_2$. The discrepancy between $b_1$ and $b_2$ may be considered as an error in defining the specific fuel consumption:

$$b = 0.5(b_1 + b_2) \pm \Delta b : \Delta b = 0.5|b_1 - b_2|.$$
3. Study of heat transfer through fencing of the furnace with local jet blowing of the external surface

3.1. Formulation of the problem

To increase the service life of the refractory which contact with the melt, the outer surface of the walls of the glass melting furnace is forcibly cooled at the level of the molten glass melt mirror [11 – 16]. Air cooling is the most commonly used. It is known cases of airborne cooling [11].

Local jet blowing is usually realized by placing near the cooled zone of the air distribution box along the entire length of the melting zone of the furnace. Air goes direct from the duct to a vertical wall through a slot or through nozzles located at the side of the duct with a certain step.

Authors have been developed a PHOENICS model to explore the characteristics of jet heat transfer. Figure 2 shows the calculation region, including a fragment of the side wall of the furnace at the level of the melt mirror (regions 2, 6 and 7), the gas volume of the working space (region 1), molten glass (regions 4 and 5), the environment (region 3), and the air collector 8. A collector with a slotted nozzle with width S is placed symmetrically relative to the level of the melt mirror (zero level along the 0y axis directed deep into the bath) at a distance h from the wall. The air flow at the exit of the nozzle has a velocity \( V \), m/sec.

![Figure 2. The calculation area of local bathtub walls jet cooling.](image)

A wall of corundum refractory is in contact with the gas volume (region 2). The lateral fence of the melt pool is made of chromoxide (layer 6 with a thickness of \( \delta_2 = 0.200 \) m) and mullite (layer 7 with a thickness of \( \delta_3 = 0.300 \) m).

As well as in [16], it was assumed that part of the volume of glass melt adjacent to the fence (region 5) forms a fixed layer with a thickness of \( \delta_1 = 0.050 \) m, and the other volume of the melt glass (region 4) moves. The origin of the 0x axis lies at the boundary of regions 4 and 5.

A stream of air rushes onto the wall and spreads over its surface. The heat removal at the level of the melt mirror is intensifying and thereby reducing the Tin is reducing (at the border of regions 5 and 6, i.e., at \( x = \delta_1 \)). As a result, the process of erosion of the refractory slows down.

It is accepted that the melt temperature at the boundary (regions 4 and 5) is 1400 °C, the ambient temperature (region 3) is 20 °C. Outlet air temperature is 20 °C. A standard \( k-\varepsilon \) turbulence model is used.
3.2. Research results
The process of jet cooling causes the changes in a row of parameters related to the outer surface of the wall: temperature decrease, intensification of heat flux density to the environment and heat transfer coefficients at the wall surface.

In the course of the study, the main task was to determine dependences $T_{in}(y), T_{out}(y)$ for given values of the parameters $V$, $h$, $S$. Depth $y$ varied in the range up to 0.500 m. Initially from the physical picture of the process, a linear-fractional (hyperbolic) function of the form (12) was chosen to approximate these dependences

$$\varphi(y) = \frac{a_0y + a_1}{y + a_2}.$$  \hspace{1cm} (12)

The approximation was performed in the Mathcad using the least squares method [17]. The heat flux density and heat transfer coefficient were calculated by equations (13):

$$q_{env}(y) = \frac{T_{in}(y) - T_{out}(y)}{\delta_2/\lambda_2 + \delta_3/\lambda_3}; \hspace{0.5cm} \alpha(y) = \frac{q_{env}(y)}{T_{out}(y) - T_{env}}.$$  \hspace{1cm} (13)

![Figure 3. Dependence $T_{in}$ on depth at various blowing speeds: 1 – $V$ = 0; 2 – $V$ = 10; 3 – $V$ = 30; 4 – $V$ = 50 m/sec](image1)

![Figure 4. Dependence $T_{out}$ on depth at various blowing speeds: 1 – $V$ = 0; 2 – $V$ = 10; 3 – $V$ = 30; 4 – $V$ = 50 m/sec](image2)

![Figure 5. Dependence $\alpha$ on depth at various blowing speeds: 1 – $V$ = 0; 2 – $V$ = 10; 3 – $V$ = 30; 4 – $V$ = 50 m/sec](image3)

![Figure 6. Dependence $T_{in}$ on depth at various nozzle size: 1 – $S$ = 0.008; 2 – $S$ = 0.014; 3 – $S$ = 0.028 m](image4)
The results of the study for various $V$ are presented at figures 3–5. From the data in figure 3 it follows that at the level of the melt mirror (at $y = 0$) even at $V = 10$ m/sec the effect of a noticeable decline of $T_{in}$ is achieved, which grows proportional to blowing speed. The decrease of $T_{out}$ already at $V = 10$ m/sec extends to a depth of 0.5 m. At the same time, at figure 4, the reduction effect of $T_{out}$

Figure 7. Dependence $T_{out}$ on depth at various nozzle size:
1 – $S = 0.008$; 2 – $S = 0.014$; 3 – $S = 0.028$ m

Figure 8. Dependence $\alpha$ on depth at various nozzle size:
1 – $S = 0.008$; 2 – $S = 0.014$; 3 – $S = 0.028$ m

Figure 9. Dependence $T_{in}$ on depth at various gap size between the nozzle and the wall:
1 – $h = 0.01$; 2 – $h = 0.05$; 3 – $h = 0.10$ m

Figure 10. Dependence $T_{out}$ on depth at various gap size between the nozzle and the wall:
1 – $h = 0.01$; 2 – $h = 0.05$; 3 – $h = 0.10$ m

Figure 11. Dependence $\alpha$ on depth at various gap size between the nozzle and the wall:
1 – $h = 0.01$; 2 – $h = 0.05$; 3 – $h = 0.10$ m
is observed on a much longer stretch of the wall. Intensification of blowing significantly affects on the value of the heat transfer coefficient (figure 5). So, it is observed that at the level of the melt mirror in the absence of blowing \( \alpha = 4.7 \text{ W/m}^2\text{/K} \), and at \( V = 50 \text{ m/sec} \ \alpha = 41.1 \text{ W/m}^2\text{/K} \).

The impact of nozzle size at a constant value of airflow velocity \( V = 30 \text{ m/sec and} \ h = 0.050 \text{ m} \) is illustrated in figure 6 – 8. This characteristic practically does not effect on the temperature of the refractory inner surface (figure 6). With an increase of value \( S \) at a constant blowing rate, the mass flow rate of cooling air rises, which leads to a decline of \( T_{\text{out}} \) (figure 7) and growth of the heat transfer coefficient (figure 8).

The exploration results of the influence on the gap size between the nozzle and the wall at a constant value of airflow velocity \( V = 30 \text{ m/sec} \) and the nozzle size \( h = 0.014 \text{ m} \) are presented in figure 9 – 11. With decreasing \( h \) intensification of local cooling is observed, which manifests itself in a decline of \( T_{\text{in}} \) (figure 9) and a reduction of \( T_{\text{out}} \) at the level of the melt mirror (figure 10). As follows from figure 11, in comparison with the base case (curve 2), the heat transfer coefficient at the transition to \( h = 0.010 \text{ m} \) (curve 1) can increase in 1.6 – 1.7 times.

4. References

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