Determining heat losses in a reheat furnace: a case study

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Abstract:

In a previous study of an actual reheat furnace with a capacity of 45 ton/h, it was found that a staggering 20% of the input energy was unaccounted for in the heat balance. It was hypothesized that this missing energy was lost to the environment as heat losses. In this work, the main losses are identified and quantified. First, the heat transfer through the wall of the furnace was determined. For this, an extensive measurement campaign was performed. Based on the measured wall temperatures and emissivity values, the heat transfer from the walls for the operating conditions at the time of the measurements was estimated. The heat rejected through the walls amounts to approximately one fifth of the total heat loss. Secondly, when the furnace door is opened, a relatively large flow rate of hot gas leaves the furnace, and a net heat loss occurs due to the radiative heat exchange between the furnace interior and the environment. As the aforementioned heat losses are very difficult to measure, a simplified theoretical model was made based on physical principles. The corresponding results indicate that the opening of the furnace accounts for a large part of the remaining heat loss.

Keywords:

Reheat Furnace, Heat Losses, Experimental, Case Study

1. Introduction

One of the commonly used operations used in steel production is hot rolling. In this process hot slabs are rolled in a mill to reduce the thickness. In order to increase the temperature of the slabs to the required temperature, they are first reheated in a reheat furnace. As the discharge temperature of the slabs is typically over 1000 °C, reheat furnaces have a rather large energy use.

The reheat furnace under consideration is a walking beam type furnace. The cold slabs are charged into the oven on one side of the furnace. They are transported towards the length of the furnace using walking beams, which is a system of stationary and moving skids. Gas fired burners in the furnace are controlled to provide the proper temperature levels in each part of the furnace.

In order to get high quality steel, it’s important that the average temperature of the slabs satisfies the requirements of the hot mill. The temperature has to be controlled in a relatively narrow range of a couple of tens of degrees Celsius. Both a temperature that is too low as a temperature that is too high are detrimental to the process. Apart from the requirements on the average temperature, it is also important that the temperature distribution in the slab is uniform. Too large temperature gradients in the slab at the time of discharge also result in reduced quality.

A slab which does not satisfy the requirements of the milling process at discharge has to be rejected. A large discharge rate has a profound negative impact on both the throughput of the furnace, as well as on the energy efficiency. It comes as no surprise that there is therefore a relatively large amount of scientific literature on the modeling and controlling of such furnaces.

Most models make use to some extent of the zone method, developed by Hottel and Sarofim [1]. The furnace is discretized into different volume zones and surfaces. Each of the volume zones and surfaces are assumed to be at a uniform temperature. The radiative exchange can then be expressed between all surfaces and gas zones.

Most of these models are focused on the prediction of the state of the slabs when they are discharge, as well as the optimal control of the burners to achieve the required state with the least amount of energy.

However, suboptimal control of the furnace temperature is not the only reason for reduced energy efficiency in a furnace. There are also losses associated with the imperfect insulation of the furnace walls, the charging and discharging process and leakage flows through gaps and cracks.

In a specific case study of interest, around 20% of the heat balance was unaccounted for. It has hypothesized that these losses could be due to the previously mentioned effects. However, there are only very few studies which investigate these effects.

Han et al. have done a lot of work on the simulation of walking beam furnaces. In one paper [2], they investigate the efficiency of a furnace using computational fluid dynamics. They model the surface interior using a 2D geometry. The conductive losses through the wall are analysed by imposing the outer temperature of the wall to 70° C and specifying the conductivity of the wall material. The radiative balance is then solved to determine the inner wall temperatures and the temperature profiles of the different slabs at different positions. A loss to the water cooled walking beam skid system of 12% of the input energy to the local zone is assumed.

As a result they find a conductive loss through the wall of 4% of the total heat input. The rest of the energy input is used for the slab heating (47.8%), lost to the skid system (12.4%) or found in the exhaust gases extracted from the furnace (35.8%). It should be remarked that convective or radiative losses due to charging and discharging were not considered. Furthermore, the wall loss is largely dependent on the assumed outer wall temperature and the skid loss follows directly from the chosen value for the loss coefficient.

Another study on the thermal efficiency of furnaces is by Filipponi et al. [3]. These authors consider a bogie hearth furnace, used for heating ingots for a hot forging process. This type of furnace differs from a walking beam furnace in that a movable hearth is extracted from the furnace, which is loaded outside the furnace before being brought back in. As the loading process is going on, the furnace door stays open. As a result, the opening times of the bogie hearth furnace are a lot longer than for a walking beam furnace. For the bogie hearth this is from about 10 minutes up to one hour, whereas the discharging process of a walking beam furnace lasts around one minute. Nevertheless, as a new slab is charged every few minutes, the walking beam furnace also spends a significant amount in the open state.

The authors performed a 3D transient simulation of the furnace and its immediate environment. The furnace under consideration is 4.7 m high, 6.9 m wide and 18.55 m deep. The temperature in the furnace is initially 1500 K, but as the burners are switched off during the loading and unloading process, this temperature decreases in time. The gases leaving the furnace decrease in temperature by 400 °C over the first 160 seconds of the opening of the door.

The radiative heat flux is at most 4 MW, whereas the peak convective heat flux is 8 MW. On average in 600s time, the radiative flux is 0.5 MW, the convective flux is 8.7 MW and the conduction losses are less than 0.1 MW. Even though the situation is quite different from a walking beam furnace, these results strongly indicate that the convective losses are important if the furnace door spends a significant amount of time in the open state on average.

It was therefore decided to do an analysis of these losses for the specific case of the walking beam furnace. The goal of this study is to have a first order of magnitude estimate. The conductive losses through the wall will therefore be measured in situ, to avoid needing to make assumptions on the outer wall temperature. The convective and radiative losses will be estimated using basic first principles models. This is sufficient to get an order of magnitude estimate and identify the main parameters, especially since the exact geometrical details are not available.

2. Measurement of the wall temperatures

As the furnace of the case study in question was due for maintenance at the time of the measurements, it was expected that the thermal insulation was degraded. As such, using the thermal conductivity values from the manufacturer would likely underestimate the losses.

An alternative approach to estimating the losses is therefore pursued. As a first step, the local temperature of the furnace wall is measured at several locations. This is done both by infrared thermometry and contact thermometry.

In order to determine the temperature from infrared thermometry, the surface emissivity needs to be known. In principle the emissivity could also be determined from the knowledge of the contact temperature and the measured IR temperature, if the temperature of the reflective environment is known. This proved not to be feasible.

The data showed many instances where the measured black body temperature was higher than the temperature determined by contact thermometry. This is only possible if there is reflection from surfaces at a higher temperature than the surface being measured, or due to uncertainty on the temperature measured by the contact thermometer. This could be the case due to the very dusty state of the furnace walls. As the environment was at a lower temperature than the surface being measured, the temperature obtained by the contact measurement can be interpreted as a lower bound on the real temperature.

As the goal is to make an order of magnitude estimate, it was decided to approximate the surface temperature by taking the average of the temperature determined from the IR sensor and the contact temperature. The emissivity of the surface is assumed to be 1 for the emissivity correction of the IR sensor. This is a reasonable value, given that the soot which covers the furnace walls is nearly a black body radiator. This assumption leads to an underprediction of the temperature, as in reality the sensor also measures some radiation that is reflected of the surface being measured, which comes from surfaces that are likely at a lower temperature.

In total 92 local temperature measurements are made distributed over the entire furnace area. The entire surface of the furnace is subdivided in several zones. The average temperature for each zones of the furnace is determined by averaging over the local temperature measurements that were made for each zone. The bottom of the furnace was not accessible due to the walking beam mechanism. The temperature on this surface was therefore estimated to be the same as that of the sides, for the purpose of determining the losses in a later step.

The result is given in Table 1.

Table 1. Estimated surface temperatures on the furnace

Zone	*Average temperature [°C]*	*Maximum temperature [°C]*
Top heated zone	174	255
Sides heated zone	134	198
Bottom heated zone	134	-
Top unheated zone	86	-
Sides unheated zone	68	-
Bottom unheated zone	68	-

No maximum values are given for the unheated zone, as for the unheated zone only two measurements per zone were done, due to the expectation of the losses being smaller for these surfaces.

The results show that the wall temperature on the heated zone of the furnace is significantly larger than assumed in the paper of Han et al., where it was 70° C. This indicates that by the time maintenance is performed, the insulation has indeed degraded significantly. Some hotspots were also noticed, with wall temperatures of up to 255° C being measured.

3. Estimation of the conductive losses

The conductive losses through the furnace walls result in a heat transfer from the furnace walls to the environment. This heat loss occurs both through convection and radiation. Evaluating these losses require some geometrical parameters, which are given in Table 2.

Table 2. Geometrical parameters of the furnace

Zone	*length [m]*	*width [m]*	*height [m]*
Heated zone	17.6	11.4	4
Unheated zone	46.4	11.4	4

The temperature of the environment T_∞ is assumed to be equal to 30°. Both the air in the surroundings of the furnace as the surfaces for the radiant heat exchange are assumed to be at this temperature.

3.1. Estimation of convective losses

The convective losses are estimated by using correlations for the convective heat transfer coefficient. The top surfaces are evaluated using convection from a top facing heated plate. The correlation of Raithby and Hollands is used, as described in the textbook of Lienhard [4].

Convection from the bottom surface of the furnace is considered as a convection from downwards facing hot plate, evaluated with the correlation of Lloyd and Moran [5]. The side walls of the furnace are modelled as vertical plates, the convection coefficient is determined using the correlation of Churchill and Chu [6].

The resulting heat transfer coefficients *h* are given in Fig. 1.

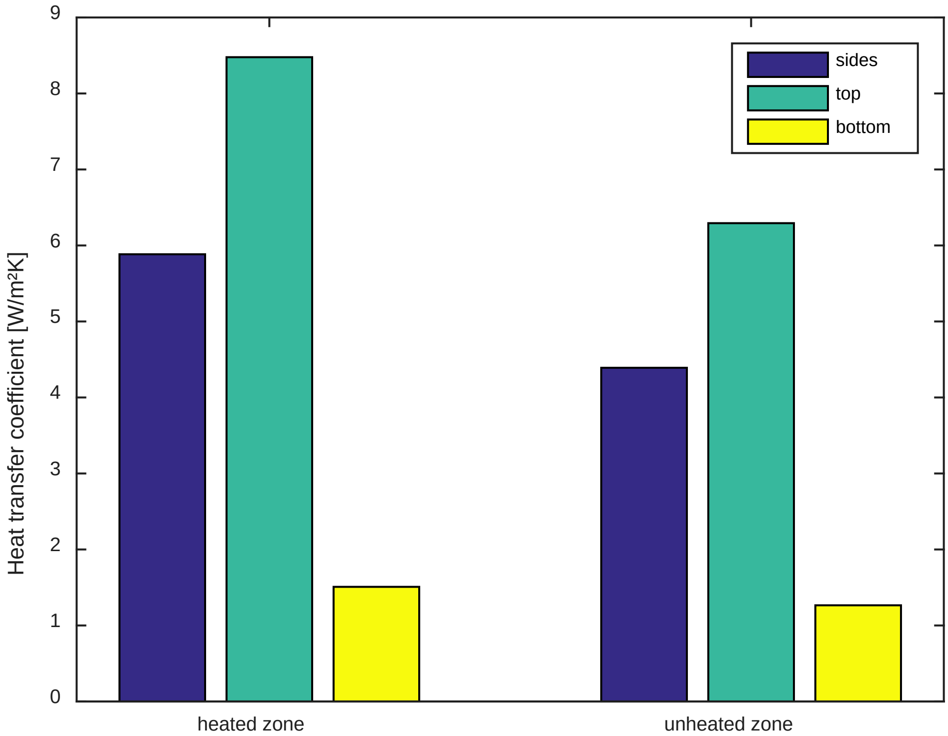

Fig. 1. Values for the heat transfer coefficient on the different surfaces.

The heat transfer rate for a surface j follows from the definition of the heat transfer coefficient *h* and is given by (1).

$$\dot{Q}_{j,conv} = A_j h (T_j - T_\infty), \qquad (1)$$

3.2. Estimation of radiative losses

The radiative losses are calculated using the Stephan-Boltzmann law. The surface emissivity is set to one, the view factor to the environment is also one, as the furnace geometry is convex. The radiative exchange law then takes the particularly simple form of (2).

$$\dot{Q}_{j,rad} = A_j \sigma (T_j^4 - T_\infty^4), \qquad (2)$$

3.3. Results

By summing the convective and radiative contribution of each surface, the total heat loss per surface is obtained. This is shown in Fig. 2.

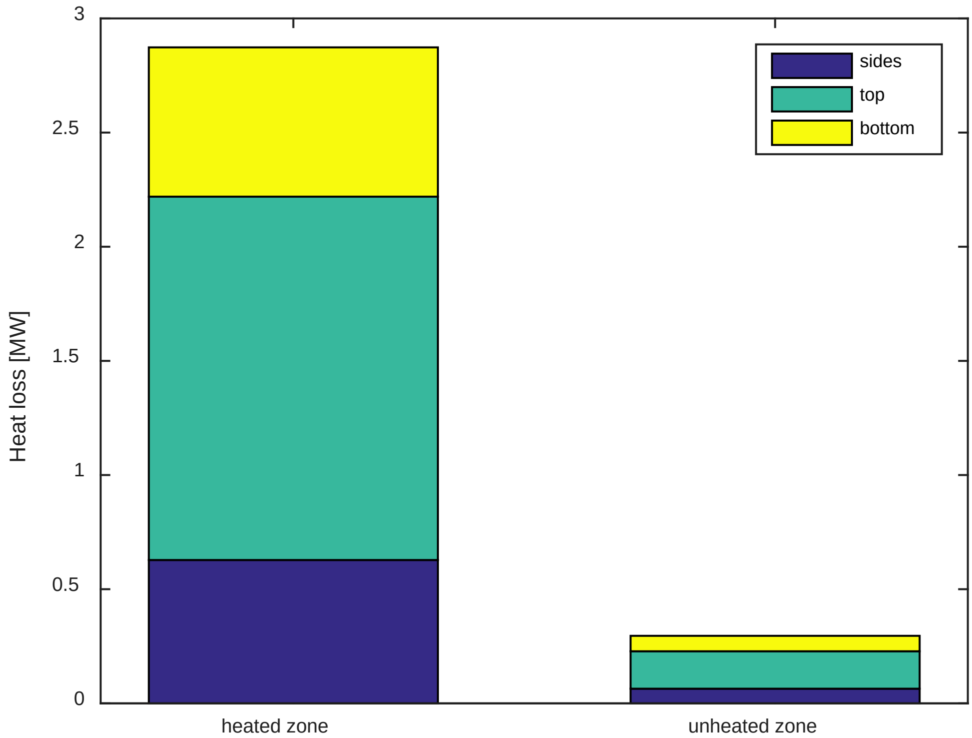

Fig. 2. Heat loss through the walls for different surfaces.

It is apparent that the main contribution to the heat loss is from the top surfaces. There are two main reasons for this. First, the surface area on top is larger than the surface area of the sides. For example for the heated zone, the total surface area on the top surface is 11.4x46.4 = 529 m^2, whereas the total surface area on the sides is 2x4x46.4 = 371 m^2.

The second reason is that the temperature of the top surface is higher than that of the other surfaces. This results in an increase in both the radiative heat transfer and the convective heat transfer per unit surface area.

Both effects are comparable in magnitude and result in the largest contribution to the conductive losses through the wall stemming from the ceiling of the furnace. It should be noted however than in reality there is a lot of instrumentation and piping located above the furnace, which obstructs the flow. It is very difficult to estimate the impact of these obstructions on the heat transfer rate.

The overall heat loss due to conductive heat losses through the furnace walls is estimated to be around 3 MW. Given that the total surface area of the furnace is 2000 m², this corresponds to an average loss of 1.5 kW/m².

The average power supplied to the furnace during the experiments was 70 MW. The conductive loss is around 4.3% of the heat input to the furnace, which is similar to the value obtained by Han et al. [2].

4. Estimation of the losses due to the opening of the door

4.1. Introduction

In order to estimate the heat loss associated with opening the furnace doors on the discharge side, a simple first principles model is made. The furnace is modelled as an enclosure which is kept at a constant pressure by controlling the air flow rate to the burners and the rate of flue gases extracted from the furnace. A schematic 2D representation is given in Fig. 3.

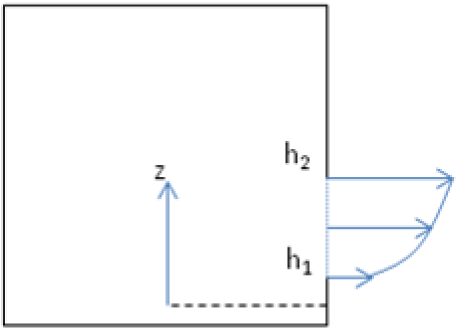

Fig. 3. Schematic 2D representation of furnace with open door

A coordinate system is introduced, where the z=0 position corresponds to the location of the pressure sensor. At this location, the pressure difference with the atmosphere is measured. The control algorithm tries to maintain this pressure difference at a constant value. For the assessment of the losses, the control will be assumed to be perfect. This corresponds to a constant pressure difference with the atmosphere being maintained at all times. This pressure difference is indicated with Δp and is equal to 12 Pa.

The distance between the pressure and the bottom of the furnace opening is indicated with h_1. The distance to the top of the opening is determined by the bottom of the furnace door and is indicated with h_2. The profile for h_2 as a function of time is given in Fig. 4. A new slab is charged every three minutes, the door takes ten seconds to open and close and remains open for 40 seconds in between. The bottom of the door opening is located approximately 0.7m below the pressure sensor.

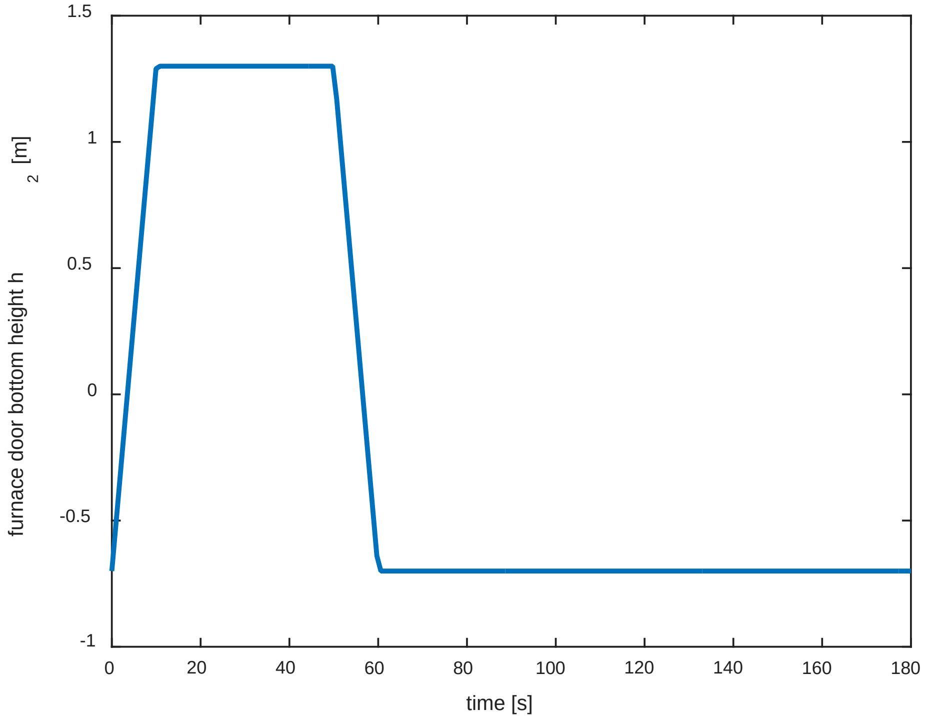

Fig. 4. Profile for the height of the bottom edge of the furnace door

4.2. Convective loss

Sufficiently far from the furnace opening, the velocity is assumed to be negligibly small. This allows writing the static pressure profile in the furnace using the hydrostatic equation. All pressure are expressed in gauge pressure, relative to the atmospheric pressure at the same height of the sensor. The result is given in (3).

$$p_{furnace}(z) = \Delta p - g\rho_i z\,, \qquad (3)$$

The pressure distribution is one-dimensional, purely determined by the height with respect to the pressure sensor. The density ρ_i which appears in this equation is the density of the gas inside of the furnace.

The pressure outside of the furnace door opening is also determined. The simplifying assumption is made that there is negligible entrainment of outside air at the boundaries where the hot gas plume leaves the furnace. This implies that the air of the surroundings is approximately stagnant at this location. As a consequence, the pressure profile can again be determined using the hydrostatic equation, which results in (4).

$$p_{outside}(z) = 0 - g\rho_o z\,, \qquad (4)$$

This equation is very similar to (3), with the exception that the static pressure difference with respect to the atmosphere at height z=0 is by definition zero, and the density ρ_o that appears is the density of the surrounding gas outside of the furnace.

The next assumption that is made is that the streamlines are parallel to each other and perpendicular to the door opening. In reality there will be some streamline curvature as the escaping gases will rise up due to buoyancy effects. However, with the assumption that this effect can be neglected, the pressure profile at the location of the door opening can be determined. If there is no streamline curvature, the pressure for a point in the door opening is equal to the pressure of the surroundings.

This fact follows from the conservation of momentum in directions perpendicular to the streamlines.

The densities are calculated using the ideal gas law for air and an atmospheric pressure p_{atm} of 1 bar. The temperature of the gas in the furnace is approximately equal to 1200 ° C. The resulting densities are given in (5) and (6)

$$\rho_i = \frac{p_{atm}}{RT_i} = 0.24 \ \frac{\mathrm{kg}}{\mathrm{m}^3} \quad (5)$$

$$\rho_i = \frac{p_{atm}}{RT_i} = 1.2 \ \frac{\mathrm{kg}}{\mathrm{m}^3} \quad (6)$$

Due to the temperature difference, the density of the air in the furnace is a lot lower than the air outside of the furnace.

Using these densities, (3) and (4) can be represented in graphical form. Fig. 5 shows the relation between the pressures and the height with respect to the sensor. For the sake of clarity, the pressures are shown with respect to the atmospheric pressure at a height of 4m above the sensor. The dotted line indicates the estimated location of the bottom of the door opening, at z=h_1=-0.7 m.

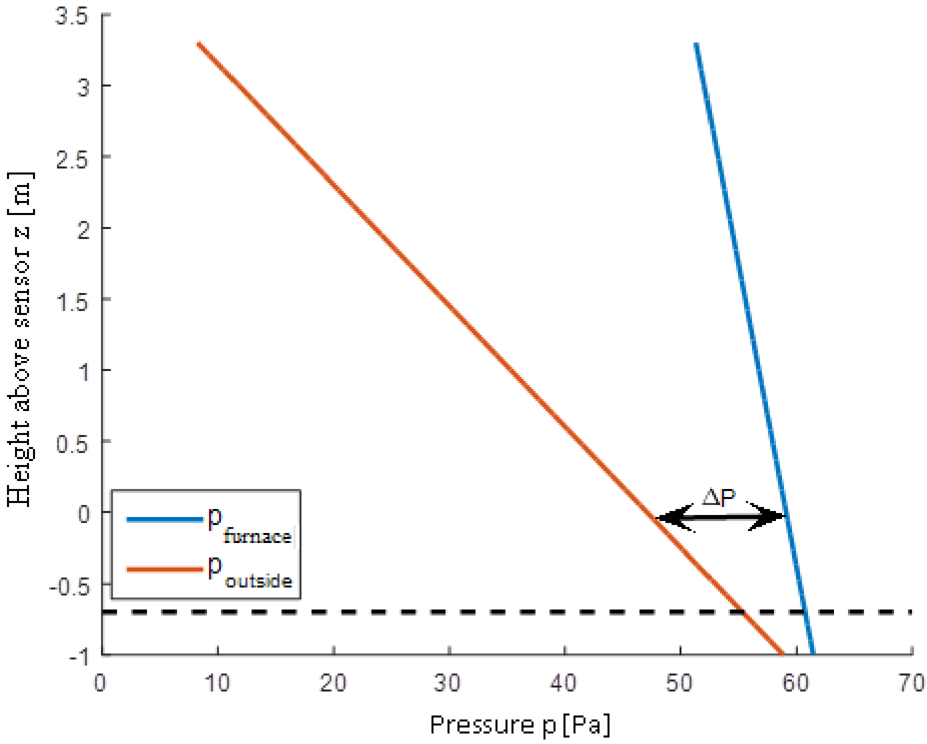

Fig. 5. Pressure profiles relative to atmospheric pressure at z=4m. Dotted line indicates bottom of door opening.

Due to the difference in density, the pressure decreases much slower with increasing height inside of the furnace than outside of the furnace. This results in a pressure difference between both locations, which is called the stack effect.

Writing out the Bernoulli equation on a streamline from the stagnant region of the furnace to a point located in the door opening at the same height, the velocity magnitude at the location of the door opening can be determined. The Bernoulli equation for this streamline is given by (7).

$$\frac{1}{\rho_i} p_{opening}(z) + \frac{1}{2} v^2{}_{opening}(z) - \frac{1}{\rho_i} p_{furnace}(z) = 0 \qquad (7)$$

Replacing the pressure at the opening by the pressure of the surroundings at the same height and solving for the velocity at the location of the opening results in the velocity profile at the opening. Equation (8) provides the velocity magnitude at the door opening as a function of the height above the sensor.

$$v_{opening} = \sqrt{2\frac{\Delta p + gz(\rho_o - \rho_i)}{\rho_i}} \qquad (8)$$

The volumetric flow rate leaving the furnace can now be calculated by integrating the velocity profile over the opening surface. Because in reality there are also flow contraction and friction effects, the effective surface area available to the flow needs to be reduced. This is taken into account by means of a discharge coefficient C_d. The door opening is approximated as a sharp edged orifice, for which the discharge coefficient is approximately 0.6. This results in the expression for the volumetric flow rate given by (9), where *W* is the width of the opening.

$$\dot{V} = C_d W \int_{h_1}^{h_2(t)} v_{opening}(z) dz \qquad (9)$$

In this equation the transient behaviour such as the intertia of the gas flow has been neglected. It is assumed that at each instance in time, the flow rate is equal to the steady state equilibrium flow rate for the instantaneous opening of the door. This is an approximation of reality, but the correct transient behaviour cannot be analysed without taking into account the inability of the control to maintain the maintain the imposed pressure difference at the location of the sensor.

The heat loss associated with this volumetric flowrate is determined by the heat transfer rate necessary to heat up new air from ambient temperature to the temperature in the furnace, i.e. from 20° C to 1200 °C.

With Cp the average specific heat capacity at constant pressure, the instantaneous heat transfer rate required to keep the furnace at constant temperature is given by (10).

$$\dot{Q}(t) = \dot{V} \rho_i c_p (T_i - T_{amb}) \qquad (10)$$

Fig. 6 shows the instantaneous convective heat loss as a function of time.

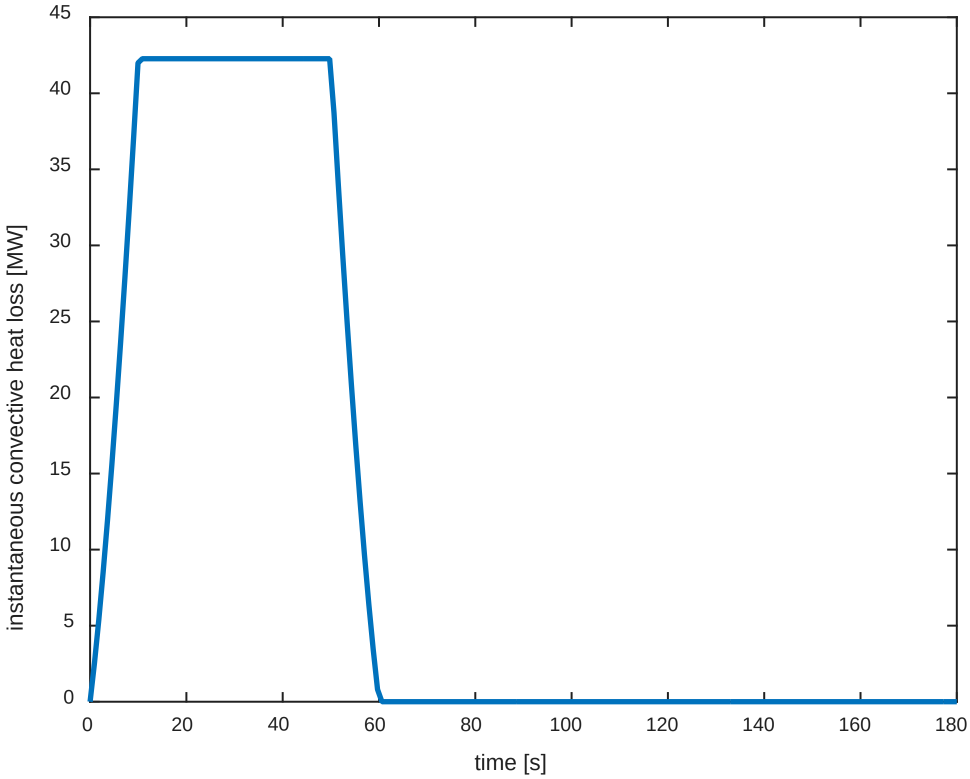

Fig. 6. Instantaneous convective heat loss as a function of time

It is immediately apparent that the instantaneous convective heat loss is nearly a factor of 4 higher than calculated by Filipponi et al. [3], even though the total opening area is almost the same and the opening height is even larger in the case of Filipponi et al.. This is mainly caused by the assumption in this paper that the air temperature and pressure in the furnace is maintained at the nominal value when the furnace is open. There is outflow over the entire surface of the door opening in this case. In the case of Filipponi et al. there was no additional air being supplied to the furnace, which means that inflow occurs over the bottom half of the door opening. As a result, the average pressure difference between the air in the furnace and that outside of the furnace is greatly reduced.

The time averaged convective heat loss is 11.5 MW, which is significantly larger than the conductive loss. As Fig. 6 also reveals that the instantaneous heat transfer rate is a strong function of the time and therefore of the door opening, it is clear that optimization of the opening profile of the door can offer some significant cost savings. It is advised to open the door only as high as necessary and as briefly as possible.

4.3 Radiative loss

From the point of view of the environment, the furnace with its opening forms a cavity. As the opening area of the furnace is much smaller than the surface of the furnace interior, the effective emissivity of the fictitious surface formed by the door opening is approximately one. This is valid regardless of the emissivity of the inner surfaces or the absorptivity of the gas in the furnace.

A good estimate of the radiative heat transfer to the environment is then obtained by assuming the door opening is a fictitious black body radiator at the interior temperature of the furnace which exchanges heat with the surroundings. This is expressed by (11) and shown graphically in Fig. 7.

$$\dot{Q}_{rad}(t) = W(h_2(t) - h_1)\sigma(T_i^{\ 4} - T_\infty^{\ 4}) \qquad (11)$$

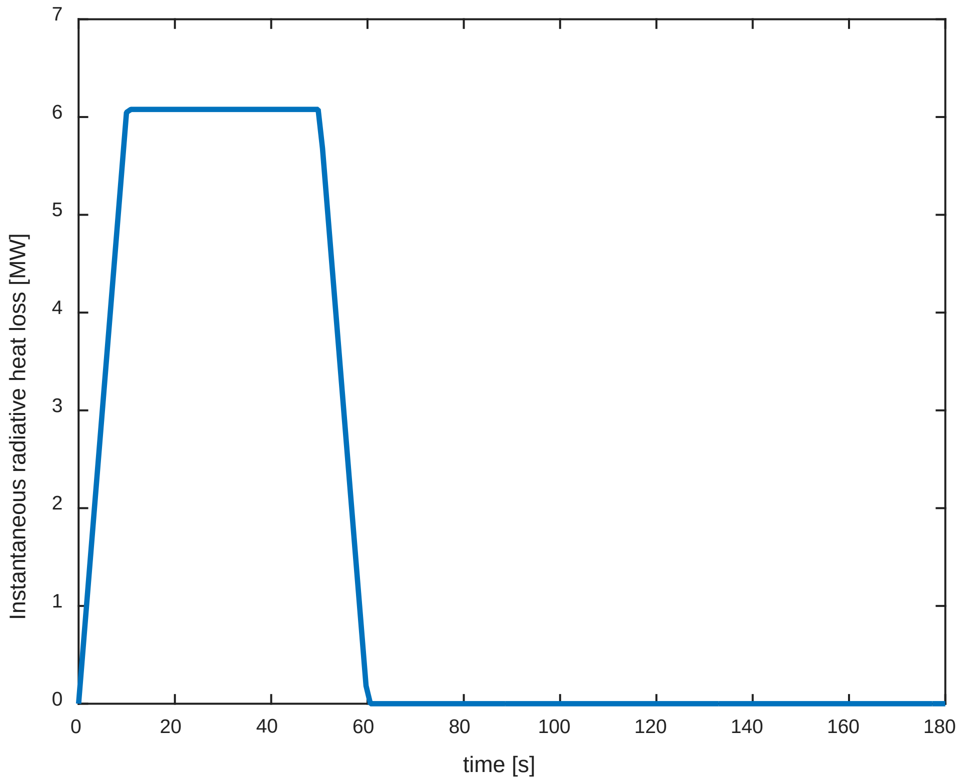

Fig. 7. Instantaneous radiative heat loss as a function of time

The heat transfer rate is proportional to the surface area of the door opening and therefore also varies piecewise linearly, according to the opening profile of the door. The peak heat flux is significantly lower than the peak convective heat flux. This is in agreement with the findings of Filpponi et al. [3].

The time averaged radiative loss is 1.7 MW. Even though it is smaller than the conductive losses through the well, the radiative loss through the door opening cannot be neglected.

4.3 Total loss through the furnace door opening

The total loss through the furnace door opening is found by the sum of the radiative and convective part. This results in a total average heat transfer rate of 13.2 MW, which mostly consists of the convective contribution. This corresponds to 19% of the 70 MW heat input to the furnace during the measurements.

Including the 4.3% conductive loss through the furnace walls results in losses which are 23% of the input energy to the furnace.

4. Conclusions

An order of magnitude estimate was made of the thermal losses occurring in a specific case of a walking beam furnace. A measurement campaign revealed that the surface temperature on the outside of the furnace was much larger than expected. This showed that by the time maintenance was performed on the thermal insulation, the performance had degraded significantly. The heat loss through the insulation was estimated to be 1.5 kW/m². This accounts for 4.3% of the input energy to the furnace.

Based on simple first principles modelling, an order of magnitude estimate was made of the convective and radiative losses associated with opening the furnace doors on the discharge side.

The total average heat loss corresponding to this phenomenon was found to be equal to 13.2 MW, which accounts for 19% of the input energy.

Together with the conductive losses through the furnace walls, the total losses are estimated to be 23% of the input energy to the furnace. This is close to the 20% losses which were estimated based on the energy balance, which is a good result, given the very basic assumptions which were made in the study.

The heat loss associated with the opening of the furnace doors has been found to be a large contributor to the overall losses of a walking beam furnace. These losses should be accounted for when investigating the efficiency of a walking beam furnace.

The convective losses associated with opening the furnace door are the largest loss, followed by conductive losses through the furnace walls. The radiative loss when the door is open is the smallest contributor to the overall loss.

Nomenclature

A	area, m²
C_d	discharge coefficient, -
c_p	specific heat capacity at constant pressure, J/(kg K)
h	heat transfer coefficient, W/(m^2 K)
	height, m
j	indexing variable, -
g	gravitational acceleration constant, m²/s
p	pressure, Pa
$\dot{Q}$	mass flow rate, kg/s
T	temperature, °C
t	time, s
$\dot{V}$	volumetric flow rate, m³/s
W	width, m
z	height, m

Greek symbols

ρ	density, m³/kg
σ	Stefan-Boltzmann constant, W/(m²K^4)
Δp	Difference between static furnace pressure and ambient pressure at sensor height

Subscripts and superscripts

amb	ambient, outside the factory hall
conv	convective
rad	radiative
i	inside the furnace
o	outside the furnace
∞	of the environment, far from the surface
1	relating to the bottom edge of the door opening
2	relating to the top edge of the door opening

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