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HEAT TRANSFER MODELLING OF THE COMBUSTION CHAMBER OF A SIDEWELL FURNACE

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ABSTRACT

World aluminum production has been increasing incessantly due to its much desired properties, including its recyclability without any property loss. Aluminum production from the ore requires 15 to 20 times more energy compared to that from recycled metal. The recycled aluminum is melted in a variety of furnaces. The sidewell furnaces are commonly used for recycled beverage cans.

A general mathematical model has been developed for sidewell furnaces. The heat transfer model of the combustion chamber is presented in this paper. A parametric study has been carried out using this model. The results of the study are discussed. The heat transfer to metal surface and consequently the furnace performance could be improved via various changes in different parameters. The most important and practical one is the preheating of the combustion air.

INTRODUCTION

Aluminum has many desirable properties: a lightweight metal, high resistance to corrosion, excellent conductor of heat and electricity, high workability (ductility), and easy recyclability. World aluminum production has been increasing drastically due to these properties. However, the primary aluminum production, i.e., starting from the raw materials, is highly energy intensive. Canada is a key player as one of the top aluminum producing and exporting countries in the world, mainly due to its rich hydro-electric power generation capacity.

In general, recycling has many benefits not only for the energy conservation, but also for the environmental protection. The energy used for the recycled aluminum is about 5 to 8% of the energy needed for the primary metal production. Aluminum is an energy-wise and environment-friendly alternative to many other metals due to its capability for unlimited recycling and the corresponding low energy consumption.

Recycled aluminum is re-melted in different types of furnaces. The recycled beverage cans are usually melted in sidewell furnaces. A schematic view of a sidewell furnace is given in Figure 1. Unlike conventional furnaces which consist of only one chamber, the sidewell furnaces have a main enclosure and a side well; and the metal circulates between the two sections via arches. The solid metal is fed to the side well continuously and melted. The combustion chamber which provides heat for the entire operation is located on the top of the main enclosure. The heat for the melting of the solid aluminum is transferred through the circulating metal.

These furnaces are not usually very efficient in terms of energy use due to a number of constraints. The gas in the combustion chamber exits at high temperatures since the gas temperature has to be above the melting point of the metal. Also, the heat transfer for melting is indirect and realized via the metal circulation between the two chambers. Thus the energy efficiency depends on the effectiveness of the metal circulation.

A mathematical model has been developed and used to improve the furnace performance by testing various design options and operating conditions. An important part of the general model is the combustion chamber model which calculates the heat transfer from the gas to the metal surface by convection and radiation. The general model, which was built based on a modular structure, includes also a flow sub-model for the liquid metal and heat transfer sub-models for the refractories and the liquid metal. Due to the modular structure of the model, each sub-model can be used individually or together (see Figure 2).

The combustion chamber model is based on an overall energy balance for the gas with the radiative heat transfer solved using a one-gas-zone model. This sub-model is relatively simple, but represents all the important phenomena occurring in the combustion chamber. The results of the parametric study on the impact of various operational and design parameters of the combustion chamber on furnace performance are presented.

Many physical processes occur in a combustion chamber. Energy released by the combustion of fuel heats up the product gas to high temperatures. Then heat is transferred from these hot combustion products to cold sink surfaces by conduction, convection and more importantly by radiation. The flow field plays an important role in determining the combustion rate and, consequently, the temperature and heat flux distributions. At the same time, combustion and heat transfer processes affect the flow field. In order to analyse such systems, all these mechanisms have to be considered simultaneously.

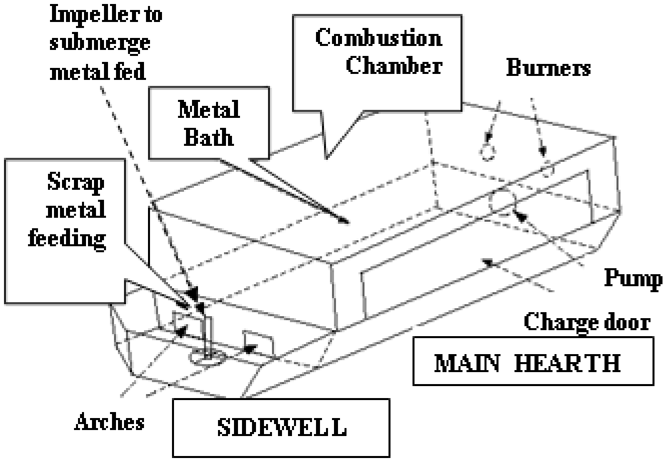

Figure 1 A schematic view of a sidewell furnace

The above phenomena and their interactions are governed by the design of the system and the operating conditions. A rigorous combustion chamber analysis requires detailed modelling of the momentum, heat and mass transfer processes with special attention to turbulent combustion and radiative heat transfer[1-3]. These models give detailed information, but they usually require longer computation times. Many engineering problems call for much simpler models which could yield answers in a relatively short time. The approach to be used depends on the purpose of the modelling work.

In simple models, the complex reacting flow is simplified by assuming a well-mixed or a plug-flow system. The plug flow assumption can be used only for very long furnaces. The well-mixed furnace model, however, can be applied to many industrial furnaces after certain modifications which allow for more realistic simulations.

In the well-mixed furnace model, it is assumed that the combustion gas in the enclosure is completely mixed, therefore no gradient exists in the gas and gas properties are uniform throughout the system. In furnaces with vigorous mixing (due to high momentum air or fuel jets) and dimensions not too different from each other (similar length, height and width), the conditions may approach those of a well-mixed chamber. In this case, the exit gas temperature is the same as the radiating gas temperature. However, the well-mixed model underpredicts the heat transfer since the actual radiating gas temperature is normally much higher than the exit gas temperature. This problem can be improved by differentiating the two temperatures and assigning a higher value to the radiating gas temperature. The model can be further extended to systems containing different types of walls by making allowance for heat transfer to more than one heat sink surface[4-5]. The well-mixed furnace models with such modifications which lead to more realistic predictions are usually referred to as the "one-gas-zone models".

In sidewell furnaces, there are two different types of surfaces: the metal surface and the refractory surface. The refractories re-radiate most of the heat they absorb and a relatively small percentage is lost to the surroundings. In this project, a one-gas-zone model was developed and adapted to sidewell furnaces. The model is presented in the next section.

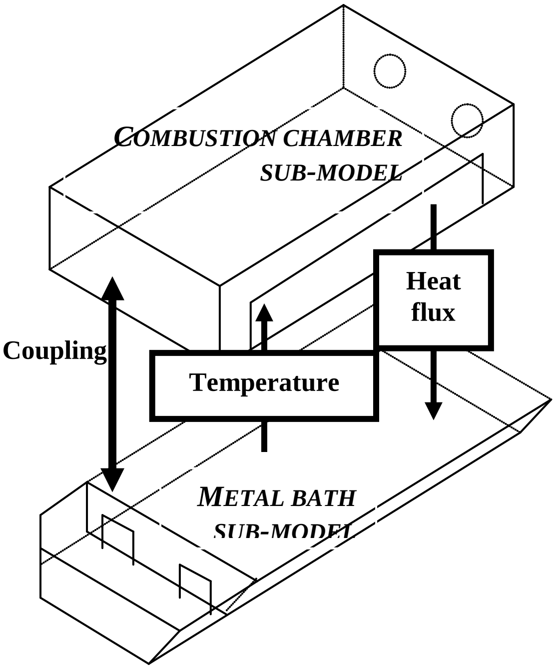

Figure 2 The global model of the sidewell furnace

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ONE-GAS-ZONE MODEL

In this model, the gas is assigned an average radiation temperature (T_g), and it takes a value between the adiabatic flame temperature ($T_{ad.fl.}$) and the gas exit temperature (T_e) based on a weighting factor (a_T):

$$T_g = a_T\, T_e + (1 - a_T)\, T_{ad.fl.} \tag{1}$$

When a_T=1, the gas is well mixed and the gas exit temperature becomes equal to the radiating gas temperature. If a_T=0, there is no mixing and the gas radiates at the adiabatic flame temperature which, of course, is not possible. The weighting factor depends on the furnace design and the operating conditions; therefore, it is furnace specific. In the present study, a_T was taken as 0.75 which gives reasonable predictions for the sidewell furnaces.

Total heat input to the furnace (Q_{in}) is given by (fuel is natural gas which was considered as methane):

$$Q_{in} = m_f \; \Delta H_{fuel} + m_f \; C_{p,f} \; (T_{fuel} - T_{amb}) + m_{air} \; C_{p,air} \; (T_{air} - T_{amb}) \tag{2}$$

The ambient temperature T_{amb} is also used as the reference temperature. The air flow rate is determined from the percent excess air and the stoichiometry of the combustion reaction. The adiabatic flame temperature is calculated from:

$$Q_{in} = m_{prod} \; C_{p,prod} \; (T_{ad.fl.} - T_{amb}) \tag{3}$$

The overall energy balance for the gas in the enclosure is:

$$Q_{in} = Q_{ref} + Q_{dir} + Q_{exit} \tag{4}$$

Q_{ref} is the heat transferred to the refractory surface by convection ($Q_{ref,con}$) and radiation ($Q_{ref,rad}$); Q_{dir} is the heat transferred directly from gas to the metal surface by radiation ($Q_{dir,rad}$) and convection ($Q_{dir,con}$); and Q_{exit} is the heat loss at the chimney due to the combustion products leaving the system. The energy balance for the refractory surface is given by:

$$Q_{ref} = Q_{ind} + Q_{loss} \tag{5}$$

The heat absorbed by the refractory is partly lost through the wall (Q_{loss}), and the rest is transferred to the metal surface by radiation (Q_{ind}). The transient effects were not taken into account for the parametric study of the combustion chamber.

The model solves the above equations simultaneously to calculate the variables given in Figure 3. Total heat transfer to the metal surface Q_{met} is the sum of direct and indirect components:

$$Q_{met} = Q_{ind} + Q_{dir} \tag{6}$$

The overall energy balance for the furnace requires that:

$$Q_{in} = Q_{met} + Q_{exit} + Q_{loss} \tag{7}$$

The thermal efficiency of the furnace is defined as the total heat transferred to metal surface divided by the total heat input:

$$\eta = Q_{met} \; / \; Q_{in} \tag{8}$$

Simulations have been carried out to determine the effects of different parameters on furnace performance based on the above model[38]. The values for the Base Case are given in Table 1. The results are presented in Tables 2 and 3 and discussed in the next section.

The heat balance on the refractory (Equation 5) can be written in terms of overall interchange factors as follows:

$$(GS1R + GS1H)\,\sigma\,(T_g^4 - T_{s_1}^4) = [3.2]$$

$$(S_1 \; S_2 \; g)\,\sigma\,(T_{s_1}^4 - T_{s_2}^4) + UA_1 (T_{s_1} - T_{amb}) \tag{9}$$

Table 1 Description of the Model Parameters for the Combustion Chamber, Values for the Base Case, and Definitions of Some of the Variables Given in Tables 2 and 3

- Mass flow rate of fuel (m_f) : 0.1 kg/s
- Heat of combustion (ΔH_{fuel}) : 40 MJ/kg of CH_4
- Chamber dimensions : 7 × 5.8 × 1.5 m
- Emissivity of refractory (ε_r) : 0.6
- Emissivity of metal surface (ε_m) : 0.6
- Overall heat transfer coefficient (U) : 2 W/m^2 K (for heat loss between the inside surface temperature of the refractory and the ambient temperature)
- Convective heat transfer coefficient ($h_{c,m}$) : 20 W/m^2 K (for metal surface)
- Convective heat transfer coefficient ($h_{c,r}$) : 10 W/m^2 K (for refractory surface)
- Maximum refractory temperature ($T_{ref,m}$) : 1100°C
- Metal surface temperature (T_{met}) : 750°C
- Inlet air temperature (T_{air}) : 27°C
- Inlet fuel temperature (T_{fuel}) : 27°C
- % excess air (%xair) : 10%
- Burner power (Q_{bur}) : $m_f \times \Delta H_{fuel}$ (MW)
- Percent direct radiative transfer to metal, $\%Q_{dir,rad}/Q_{dir}$
- Percent total radiative transfer to metal, $\%(Q_{dir,rad}+Q_{ind})/Q_{met}$
- Percent direct component of total heat transfer to metal, $\%Q_{dir}/Q_{met}$
- Percent chimney loss based on total heat input, $\%Q_{exit}/Q_{in}$
- Percent heat loss to surroundings based on total heat input, $\%Q_{loss}/Q_{in}$
- Percent heat loss to surroundings based on heat transfer to refractories, $\%Q_{loss}/Q_{ref}$
- Percent convective transfer to refractories, $\%Q_{ref,con}/Q_{ref}$
- Gas radiation temperature (T_g)
- Refractory temperature (T_{ref})
- Exit gas temperature (T_e)
- Adiabatic flame temperature ($T_{ad.fl.}$)
- Heat flux on the metal surface (q_{met}) : Q_{met}/(metal surface area) (W/m^2)
- Percent difference from Base Case: $\%(Q_{met(Case\ i)}-Q_{met(Base\ Case)})/\ Q_{met(Base\ Case)}$
  A negative value indicates a lower heat transfer to the metal surface, and a positive value shows a higher heat transfer to the metal surface.

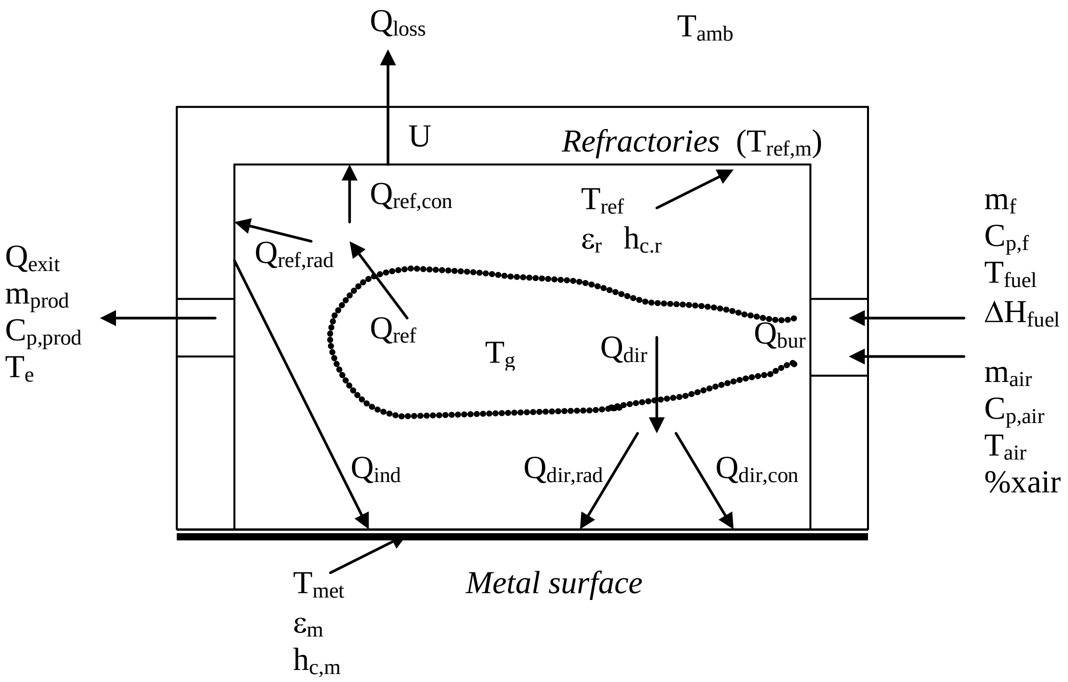

Figure 3 Schematic Representation of the Combustion Chamber and Model Parameters (Definitions are Given in Table 1)

Similarly, the heat balance for the gas is given by:

$$Q_{in} = (GS1R + GS1H)\,\sigma\,(T_g^4 - T_{s_1}^4) + (GS2R + GS2H)\,\sigma\,(T_g^4 - T_{s_2}^4) + m_{prod}\ C_{p.prod}\ (T_e - T_{amb}) \qquad (10)$$

GS1R, GS2R, GS1H, GS2H are the total interchange factors for radiation (R) and Convection (H) for surfaces 1 and 2. Number 1 indicates the refractory surface and number 2 indicates the metal surface.

RESULTS AND DISCUSSIONS

The simulation results are presented in Tables 2 and 3. Each series of simulations is separated by darker solid lines. The refractory temperature T_{ref} is the average value for the entire refractory surface. The temperature $T_{ref,m}$ is the maximum set value and, in plants, it is measured by a thermocouple placed in the roof of the combustion chamber (the roof is hotter than the rest of the refractory). In the model, T_{ref} is constrained to an upper limit at 50°C below $T_{ref,m}$. This accounts for the temperature distribution on refractory surface. If the refractory exceeds the maximum temperature, the fuel flow rate is reduced until the refractory temperature is reduced to allowable maximum level. The overall energy balance is checked after the calculations are done; the error in above simulations is less than 0.2%. This error can be reduced by simply decreasing the error tolerance.

In Cases 2 to 4, the effect of convective heat transfer coefficient is studied (see Figure 4). For the Base Case, the coefficient for the metal surface is taken as 20 W/m^{2}K. Values varying between 15 to 20 W/m^{2}K were found from different correlations. Since the burner is usually directed towards the metal surface, the higher value was chosen. It is seen in these cases that a significant increase in the coefficients yields only a marginal increase in heat transfer to metal. It should also be noted that it is not easy to achieve such increases in the convective heat transfer coefficients without making considerable changes in design.

Cases 5 and 6 examine the effect of percent excess air (see Figure 5). When excess air is reduced to zero, 13% increase in heat transfer to metal is obtained. In practice, excess air is always needed to achieve complete combustion in the chamber. Too much excess air reduces the heat transfer (12.3% decrease for an increase from 10 to 20% excess air). Evidently, the excess air should be reduced as much as possible.

Case 7 looks at the effect of metal surface temperature (see Figure 6). When it is increased by 50°C, the heat transfer to metal decreases by 5.7%. This is due to the decrease in the temperature difference between the gas and metal surface which is the driving force for the heat transfer. It also shows the importance of maintaining a well-mixed liquid metal bath with as little vertical temperature gradient as possible so as to yield a lower surface temperature.

Cases 8 and 9 show the effect of increasing the surface emissivities (see Figure 7). Increase in metal surface emissivity has an important impact on heat transfer to metal (6% for an increase from 0.6 to 0.8). The refractory emissivity has a marginal effect. The same increase in refractory emissivity (an increase from 0.6 to 0.8) results in only 0.1% increase in heat transfer to metal. It is very difficult to change these variables in practice.

In Cases 10 to 12, the fuel flow rate is increased (see Figure 8). The heat transfer to the metal surface increases considerably (31.2%, 56.1% and 76.2% for 50%, 100% and 150% increase in fuel flow rates, respectively), however the thermal efficiency of the furnace decreases.

In Case 13, the air temperature is 627°C instead of 27°C (see Figure 9). The heat transfer to the metal as well as the furnace thermal efficiency increase significantly. This shows the importance of preheating the combustion air which is the case with regenerative burners. Such augmentation in heat transfer is achieved due to the increase in the gas temperature. The refractory temperature increases along with the gas temperature and may reach the maximum limit which, in turn, restricts the fuel flow rate.

Cases 14 and 15 show the effect of overall heat transfer coefficient through the refractory wall (see Figure 10). Doubling the coefficient reduces the heat transfer to metal by 6.7%. Obviously, the walls should be insulated as much as possible to reduce this heat loss. If there were no heat loss, the heat transfer to metal would increase by the same percentage.

In Tables 2 and 3, the temperatures and the components of heat transfer (in percentages) are presented in detail. The results indicate that direct transfer from the gas to the metal is in the order of 60 %. The remaining 40% comes via the refractories indirectly. This shows that both components are significant. Also the radiation accounts for ~75-80% of direct transfer and ~85-90% of total transfer to the metal surface. Heat loss through the refractory walls is about 3-4% of the total heat input. Wall heat loss as percent of the total heat transfer (convection and radiation) to the refractory surfaces is in the order of 10-14%. This value is similar to the heat transfer from the gas to the refractory surfaces by convection as expected. Figure 11 shows the heat transfer rates based on a burner input (energy input) of 100% as the basis for a typical condition. Depending on the operating conditions, these percentages change.

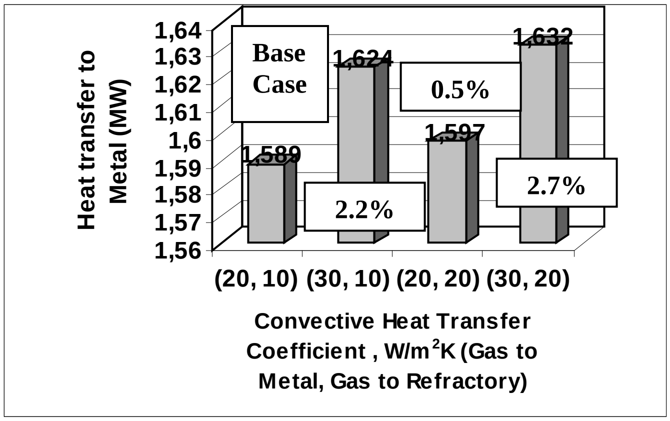

Figure 4 Effect of Convective Heat Transfer Coefficient on Heat Transfer to Metal (Numbers in Parentheses Indicate the Coefficients: First One Gas to Metal, Second One Gas to Refractory)

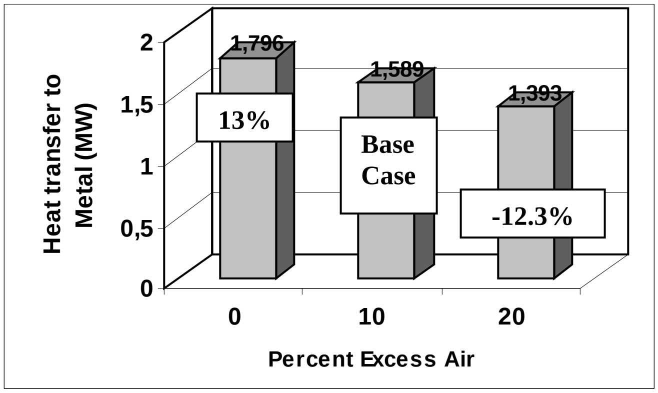

Figure 5 Effect of Percent Excess Air on Heat Transfer to Metal

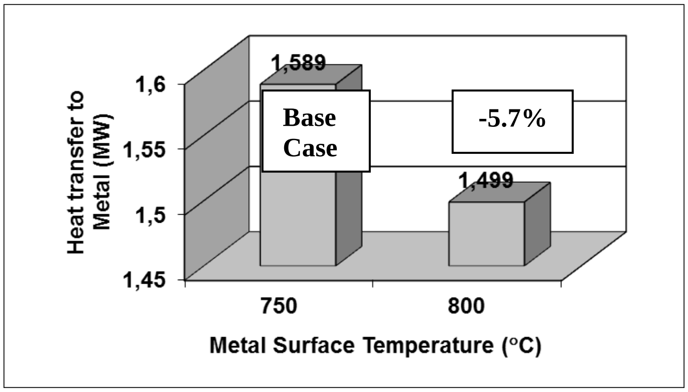

Figure 6 Effect of Metal Surface Temperature on Heat Transfer to Metal

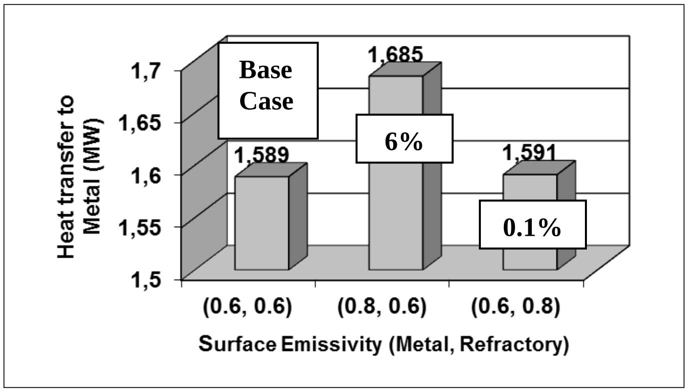

Figure 7 Effect of Surface Emissivity on Heat Transfer to Metal (Numbers in Parentheses Indicate the Emissivities: First One is the Metal Surface Emissivity, Second One is the Refractory Surface Emissivity)

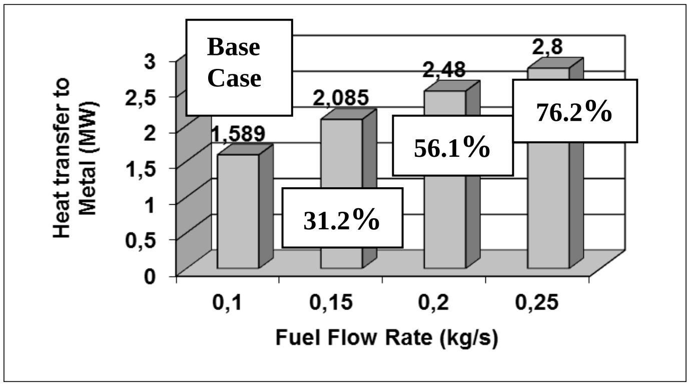

Figure 8 Effect of Fuel Flow Rate on Heat Transfer to Metal

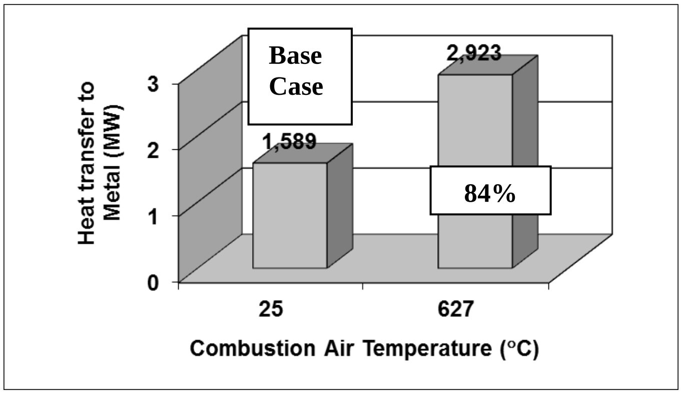

Figure 9 Effect of Combustion Air Temperature (i.e. Preheating Air) on Heat Transfer to Metal

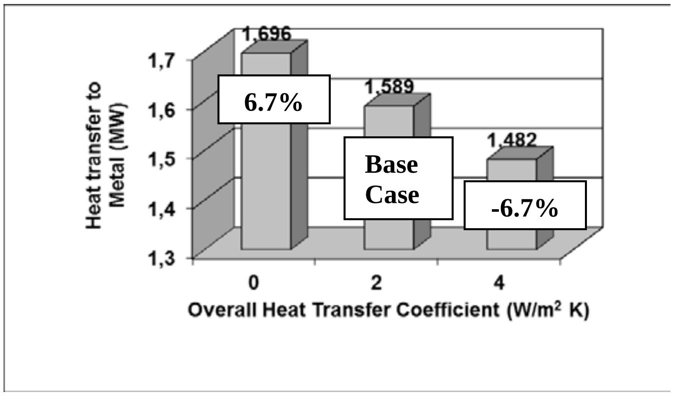

Figure 10 Effect of Overall Heat Transfer Coefficient through the Refractory on Heat Transfer to Metal

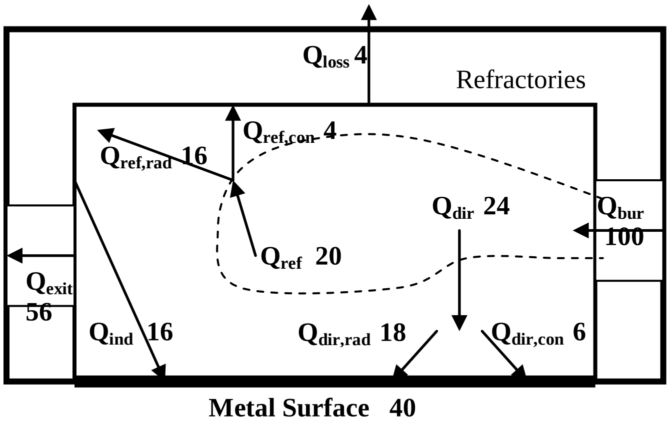

Figure 11 Various Heat Transfer Rates in the Combustion Chamber Based on 100% as the Heat Input by the Burner for a Typical Operation (These Percentages Vary Depending on the Conditions).

CONCLUSIONS

A one-gas-zone model has been developed to calculate the heat transfer in the combustion chamber of the sidewell furnace. The model has been used to determine the effects of various parameters on the heat transfer to the liquid metal bath surface.

1. Increasing emissivity of the metal increases the heat transfer. However, it is very difficult to vary the emissivity. Certainly, higher convection means better heat transfer, but, again, it is not easy to realize this without significant changes in design (burner position and design).
2. Varying surface emissivity and convection on refractories has a marginal effect. It is also very difficult to modify these parameters.
3. Metal surface temperature has a direct impact on heat transfer. The lower the surface temperature is, the higher the temperature difference between the gas and the surface temperatures is, and, consequently, the higher the heat transfer rate is. Mixing in the liquid bath reduces the vertical temperature gradient and the surface temperature. Therefore, it is important to promote conditions which favor mixing in the metal bath.
4. It is also important to insulate the walls and to reduce the excess air (and air infiltration) as much as possible to decrease losses from the combustion chamber.
5. The heat transfer to the metal surface can be increased significantly by increasing the fuel flow rate. However, this increase is realized at the expense of the furnace efficiency. That is, of the additional heat input, a higher portion is lost through the chimney.
6. The parameter that appears to have the greatest impact is the inlet air temperature. This shows the importance of preheating the combustion air which results in higher gas temperatures and, consequently, better heat transfer rates. Recuperation of heat also increases the furnace efficiency. Higher gas temperature means, of course, higher refractory temperature which may attain the maximum value easily. With air preheating, it is important to set the maximum refractory temperature as high as possible.

The model accounts for all the important phenomena. It is also flexible for use in determining the effects of various parameters on furnace performance.

Table 2 Results of the simulation: Temperatures and Heat Fluxes

Case no.	Difference in specifications	Q_{met} (MW)	% difference from Base Case	q_{met} (W/m^2)	η(%)	T_g (°C)	T_{ref} (°C)
1	Base	1.589	0.0	39070	39.7	1012	909
2	$h_{c,m}$=30	1.624	+2.2	39940	40.6	1004	903
3	$h_{c,r}$=20	1.597	+0.5	39280	39.9	1010	913
4	$h_{c,m}$=30 $h_{c,r}$=20	1.632	+2.7	40140	40.8	1001	907
5	%xair=0	1.796	+13.0	44180	44.9	1034	927
6	%xair=20	1.393	-12.3	34260	34.8	990	891
7	T_{met}=800	1.499	-5.7	36870	37.5	1033	936
8	ε_m=0.8	1.685	+6.0	41430	42.1	991	876
9	ε_r=0.8	1.591	+0.1	39120	39.8	1011	909
10	m_f=0.15	2.085	+31.2	51270	34.7	1070	948
11	m_f=0.20	2.480	+56.1	60990	31.0	1113	977
12	m_f=0.25	2.800	+76.2	68860	28.0	1144	999
13	T_{air}=627	2.923	+84.0	71890	53.4	1156	1007
14	U=0	1.696	+6.7	41700	42.4	1019	924
15	U=4	1.482	-6.7	36440	37.0	1006	892

Table 3 Results of the simulation: Components of Heat Transfer

Case no.	Difference in specifications	%$Q_{dir,rad}/Q_{dir}$ (dirct radiationto metal)	%$(Q_{dir,rad}+Q_{ind})/Q_{met}$ (total radiation to metal)	%Q_{dir}/Q_{met} (direct component of the total tr. to metal)	%Q_{exit}/Q_{in} (chimney loss)	%Q_{loss}/Q_{in} (heat loss to env. based on total heat input)	%Q_{loss}/Q_{ref} (heat loss to env. based on heat tr. to refractories)	%$Q_{ref,con}/Q_{ref}$ (convection to refrac.)
1	Base	77.9	86.4	61.5	57.0	3.5	14.3	11.0
2	$h_{c,m}$=30	69.9	80.7	64.1	56.1	3.5	13.3	11.1
3	$h_{c,r}$=20	77.8	86.6	60.4	56.7	3.5	14.5	19.8
4	$h_{c,m}$=30 $h_{c,r}$=20	69.6	81.6	63.0	55.9	3.5	13.5	20.0
5	%xair=0	78.9	87.0	61.7	51.7	3.6	12.8	10.3
6	%xair=20	76.9	85.8	61.4	61.9	3.4	16.0	11.7
7	T_{met}=800	79.1	87.3	60.9	59.1	3.6	15.7	10.5
8	ε_m=0.8	81.7	88.2	64.4	54.9	3.4	12.4	12.5
9	ε_r=0.8	74.8	86.5	53.8	56.9	3.5	16.3	9.31
10	m_f=0.15	79.0	87.3	60.7	62.9	2.4	11.5	10.1
11	m_f=0.20	79.8	87.9	60.2	67.3	1.9	10.1	9.5
12	m_f=0.25	80.4	88.3	59.8	70.6	1.5	9.2	9.0
13	T_{air}=627	80.6	88.4	59.7	43.9	2.8	8.9	8.9
14	U=0	78.0	87.0	59.4	57.6	0.0	0.0	10.8
15	U=4	77.8	85.8	63.9	56.4	6.8	28.9	11.2

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