Original Paper

Modelling of combustion and heat transfer in glass furnaces

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A mathematical model for the three-dimensional turbulent flow, combustion and radiative heat transfer for a glass furnace is described. Submodels for turbulence ($k - \varepsilon$), combustion and zone models for radiation are presented. Results for a high-temperature gas-fired glass furnace combustion chamber are given. Temperature, velocity and fuel/air mixing profiles are obtained. Heat-flux distributions to the glass melt are discussed. Also simple radiation models like the well-stirred and plug-flow zone models are used, results are compared with the complete radiation model. With the well-stirred model the effect of increasing gas emissivity could be quantified. Spectral effects in gas radiation are shown to be important. Including this in the well-stirred zone model showed that increasing roof emissivity can improve the energy efficiency of a high-temperature gas-fired glass furnace.

Modell der Verbrennung und des Wärmeüberganges in Glasschmelzöfen

Es wird ein mathematisches Modell für die dreidimensionale turbulente Strömung, die Verbrennung und den Wärmeübergang durch Strahlung in einem Glasschmelzofen beschrieben. Teilmodelle für die Turbulenz ($k - \varepsilon$) sowie Verbrennungs- und Zonenmodelle für die Strahlung werden vorgestellt. Für eine mit hoher Temperatur beaufschlagte gasbeheizte Brennkammer eines Glasschmelzofens werden die Temperatur-, Geschwindigkeits- und Brennstoff/Luft-Gemischprofile beschrieben. Es werden auch einfache Strahlungsmodelle, wie solche für vollständige Durchmischung und Kolbenströmung, angewendet und die Ergebnisse mit den im Gesamtmodell erhaltenen verglichen. Mit dem Modell vollständiger Durchmischung konnte der Einfluß der zunehmenden Gasemission quantitativ bestimmt werden. Ferner wurde gezeigt, daß spektrale Effekte bei der Gasstrahlung eine bedeutsame Rolle spielen und daß dies – auf das Modell vollständiger Durchmischung angewendet – bei ansteigendem Emissionsvermögen der Decke den thermischen Wirkungsgrad eines gasbeheizten Hochtemperatur-Glasschmelzofens verbessern kann.

1. Introduction

Heat transfer from a flame in a glass furnace is important for the efficient operation of a glass melter. Energy savings and an optimization of the process to obtain high glass qualities depend on the heat input from the flames to the melt, as has been discussed by Trier [1] and Trier and Voss [2]. For a good description of the heat transfer a modelling of the combustion and three-dimensional (3-D) flow coupled to a radiative heat – transfer model is required.

The use of advanced mathematical simulation models for glass furnaces have been recently reported by Gosman et al. [3], Carvalho et al. [4 and 5], Ungan and Viskanta [6 and 7]. They describe the heat transfer from the flame. However, Ungan and Viskanta [8] also give a modelling of the 3-D flow in the glass melt. A similar 3-D glass-melt model has been reported by Simonis et al. [9], whereas Horvath and Hilbig [10] reported on a 2-D model coupled to a simplified flame-radiation model.

In this paper a model of 3-D flow, combustion and heat transfer in the combustion chamber of a glass furnace will be presented. Comparison will be made with simple radiation models. Using the latter the effect of spectral band radiation of the flame gases will be discussed.

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2. Modelling

2.1. Flow and combustion model

The flow model of the combustion chamber uses a numerical method to solve the continuity equation and the three-dimensional Navier-Stokes equations for a steady state case. The effect of turbulence on the flow has also to be modelled. This has been done using a standard $k_e - \varepsilon$ model with wall functions [11]. As a result a system of six coupled equations is obtained from which the three velocity components can be found, and the two variables of the turbulence model, the turbulent kinetic energy k_e and the dissipation of the turbulent energy ε. The general transport equation for each of these variables can be written as:

$$\operatorname{div} (\varrho\, v\, \phi) = \operatorname{div} (D_\phi \operatorname{grad} (\phi)) + S_\phi \qquad (1)$$

where ϱ = density and D_ϕ = diffusivity. Here ϕ stands successively for the three velocity components of the vector v and for k and ε. With $\phi = 1$ one has the continuity equation. Solutions of the equations are obtained numerically. Using a fine grid the discretized equations are solved by an iterative procedure. The first term giving the convective effects (div ($\varrho\, v\, \phi$)), the second the diffusion effects (div (D_ϕ grad (ϕ))) and the last (S_ϕ) is a source term. The energy (enthalpy) equation has also to be solved to obtain the temperature field. This equation has the same form as equation (1). The source term in it comes from the enthalpy of combustion and radiative energy exchange.

When modelling the turbulent combustion in the natural gas/air mixture the following assumptions have been made:

a) The reaction is expressed as

fuel + s oxidant $\rightarrow$ (s + 1) products

where s is the stoichiometric mass ratio oxidant/fuel.

b) The reaction is infinitely fast, consequently the reaction rate is determined by the turbulent mixing of fuel and oxidant.

c) The fuel and oxidant may coexist at the same place, but at different times (intermittency). This is due to concentration fluctuations caused by turbulence. They are accounted for by means of a probability density function. Here the probability density function is represented by a combination of two Dirac delta functions and by a corresponding square wave temporal distribution.

In the model the following five species are considered: fuel, O_2, N_2, H_2O, CO_2. The composition of a fluid element can be expressed in terms of a mixture fraction f. The mixture fraction gives the concentration of the fuel or of oxygen relative to their initial concentrations. For stoichiometric conditions in the present case $f = 0.0612$, whereas $f = 0.0$ for air and $f = 1.0$ for the fuel. Applying the above-mentioned assumptions one can find a set of two coupled transport equations again of the form of equation (1) for the mixture fraction f and for the function representing the concentration fluctuation g as derived by Khalil [12]. For further details it is referred to the earlier papers [13 to 15].

2.2. Radiative heat transfer models

In order to evaluate the source term in the energy equation the radiative heat exchange must be modelled. In this study the well-known Hottel zone method was used [16]. One of the main reasons for choosing this method is the fact that in the present study the heat flux to the relatively cold and strongly absorbing glass surface is an important quantity. For that case the zone method gives very reliable results.

In the Hottel zone method one divides the combustion space in a number of equally sized volume zones (parallelepipeds), surrounded by surface zones along the glass bath and roof. The three-dimensional space has been divided in the model in 10 by 2 by 2 volume zones (flame length, width, height direction, respectively). The modelling consisted of half the combustion chamber, using symmetry conditions along the mid plane through the flame.

In the Hottel zone method one has to evaluate the radiation exchange factors:

$$\overline{s_i s_j},\ \overline{s_i g_k} \text{ and } \overline{g_m g_k}\,.$$

They represent the surface to surface, the surface to gas volume and volume to volume exchanges, respectively. With these factors evaluated beforehand and a calculated temperature field one finds for the net heat exchanges for surface zones:

$$Q_{n,j} = \sum_{i=1}^{N} \overline{s_i s_j}\, q_i^- + \sum_{k=1}^{K} \overline{s_i g_k}\, E_k - A_j\, q_j^-$$

with q_i^- and q_j^- = leaving fluxes from areas A_j, E_k = emitted flux from volume zone k, and for volume zones:

$$Q_{n,k} = \sum_{j=1}^{N} \overline{s_j g_k}\, q_j^- + \sum_{m=1}^{K} \overline{g_m g_k}\, E_m - 4\, k_k\, V_k\, E_k$$

with k_k = absorption coefficient of volume V_k.

Especially the calculation of the $\overline{g_m g_k}$ factors is cumbersome as they contain a six-fold space integration. Therefore, it was made use of a recently published algorithm from Siddal [17] for the $\overline{g_m g_k}$ factors. In this case there are 40 volume zones and 88 surface zones. Up to now the full zone method was only applied for one gray gas with a constant absorption coefficient k. For the latter a value of $k = 0.08\ \text{m}^{-1}$ has been taken as a good average value for the whole space. This has been done on basis of a literature review for natural gas flames [15], in this reference it has also been shown that the results are not very sensitive to k in the range of 0.06 to 0.10.

Simplified zone models common in engineering practice were also used. The well-stirred and plug-flow models, described by Hottel and Sarofin [16] were employed. Essentially the well-stirred model consists of only one volume zone averaging over the whole combustion chamber and two surface zones: glass surface and roof. The average gas temperature of the volume zone can be obtained by using an enthalpy balance over the furnace together with the zone-radiation model. In the plug-flow model a number of volume zones in the axial flame direction are used. An enthalpy balance is made over each zone, assuming a certain combustion profile in the axial direction. In the plug-flow model radiation exchange in the axial direction is neglected, the furnace is considered to be long.

2.3. Spectral effects in gaseous flames

In the complete 3-D zone method for radiative calculations the gas has been assumed to be a gray gas. However, for a natural gas flame one has mainly radiation from the combustion products' components H_2O and CO_2. This results in typical band radiation from these hot gases. In one part of the spectrum there will be a high emission/absorption, in other parts there will be a window with a high transmission. This should be taken into account for a radiative heat transfer calculation.

Recently it has been reported [18 to 23] that in high-temperature gas-fired furnaces the interaction between spectral effects in the gas and the gray roof of a furnace should be taken into account. It has been shown that increasing the emissivity of the roof leads to a better energy efficiency of the furnace. Such spectral effects can easily be studied with the simplified zone models. In this study use has been made of a 15-band gas radiation model in the well-stirred furnace zone method. The effect of increasing the emissivity of the roof material from 0.3 to 0.95 has been studied.

3. Furnace configuration and conditions

The glass melting furnace modelled is a regenerative side-fired furnace, consisting of six burner compartments, three of which are cyclically being fired. The furnace considered here is fired with natural gas and the combustion air is preheated in the regenerator. In this study one combustion chamber compartment is modelled corresponding to one burner as shown in figure 1 and the stationary situation in this chamber is simulated. Also the burner belonging to one compartment has been studied, using a laboratory and a mathematical model. The burner used consists of a primary air nozzle and two gas nozzles injecting obliquely into an axial stream of preheated secondary air.

In the standard situation studied a natural gas flow of 0.0596 kg/s of fuel per inlet gas nozzle and an air excess of 10 % with a net calorific value of the natural gas of 44.6 MJ/kg have been taken. The secondary air temperature is 1423 K. Due to the high secondary air temperature the flame temperature at adiabatic conditions for this case is 2573 K. Dissociation of combustion products has been taken into account [15]. The glass surface temperature is assumed to be 1773 K. The dimensions of the combustion chamber as shown in figure 1 are: length 7.3 m, height 2.4 m and width of one complete compartment 3.3 m.

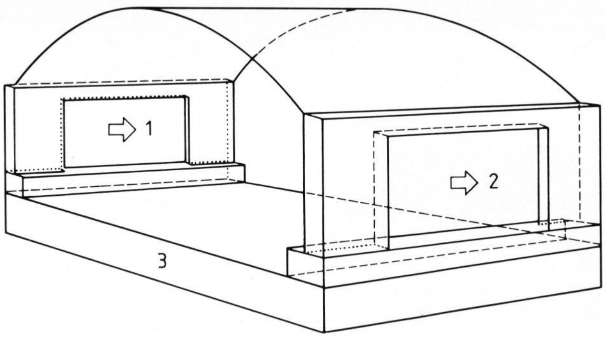


Figure 1. Perspective view of a combustion chamber. 1: inlet gas/air mixture from burner, 2: outlet flue gas, 3: glass bath.

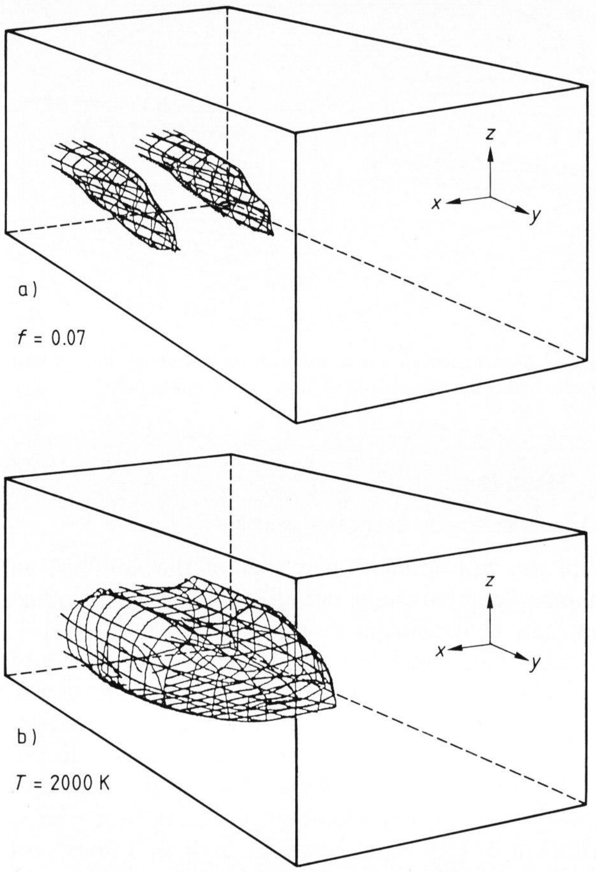
4. Results

4.1. Results for complete model

With the full simulation model of the combustion chamber compartment details of the flow, combustion and heat transfer can be obtained [13 to 15]. Whether the flame is short and hot or longer and cooler depends strongly on the inlet conditions, it was found that the initial turbulent viscosity μ_t is particularly important here. The turbulent viscosity in the combustion chamber is influenced by the flow and mixing in the burner. An intense mixing with a large turbulent length scale leads to high μ_t values and consequently to short and hot flames. In this way the flame configuration, and thus, the heat flux to the glass can be influenced strongly. For short flames the average heat load of the glass can be up to 19 % higher than for longer flames. However, such a short and hot flame will be unfavorable in terms of the production of NO_x in the flame.

Some results for a long flame (low turbulent viscosity at inlet port) will now be given. A picture of a long flame can be seen in figure 2a, here the surface of a constant mixture fraction $f = 0.07$, corresponding to an air factor of 0.87, is shown and in figure 2b the 2000 K surface is shown. The prescribed flow profiles at the combustion chamber are for the case of a precombustion of 44 % in the burner. From the computations a 96 % burnout of the fuel at the outlet is found for the long flame, complete burnout will occur in the passage to the regenerator. For the short flame a 100 % burnout has been found.

For glass melting furnaces the heat transfer to the glass is an important point to study. For the calculated case of a short, hot flame as average heat flows for half of the compartment is found: inlet enthalpy 3.80 MW (2.66 from the fuel and 1.14 MW from the preheated secondary air), to glass bath 1.51 MW (99 % of this by radiation and only 1 % by means of convection), wall losses 0.10 MW and flue gas heat loss 2.20 MW. Of course a large part of the last (1.14 MW, that is 52 %) will be recycled through the secondary air from the regenerator. The efficiency of the furnace, related to total fuel input, is 42 % in this case.



Figures 2a and b. Flame pictures in the combustion chamber. Surface of constant mixture fraction $f = 0.07$, showing presence of fuel ($f = 0.0612$ stoichiometric); b) surface of constant temperature ($T = 2000$ K).

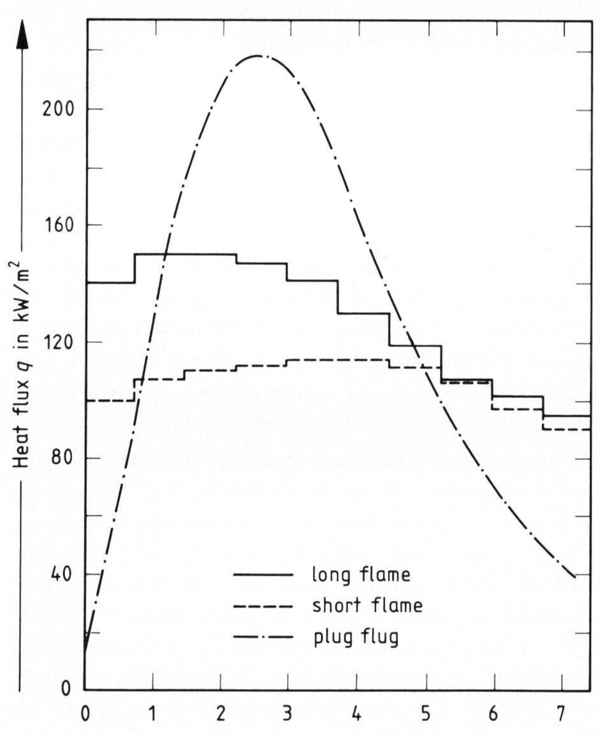


Figure 3. Distribution of heat fluxes to glass along flame length.

Table 1. Comparison of the simplified models with the full model for long and short flames

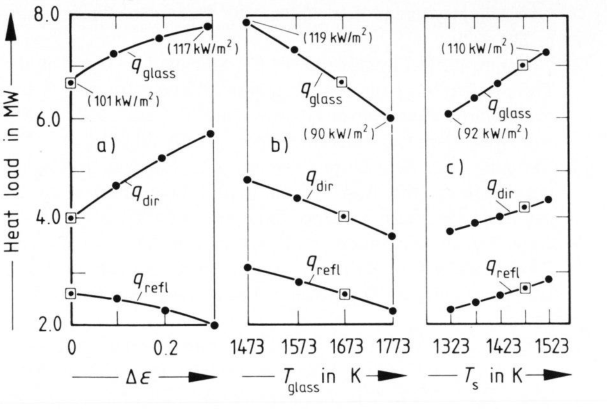
	heat loads to glass in kW/m²		
	well-stirred model	plug-flow model	full model
burnout 90 % (long flame)	104	112	104
burnout 100 % (short flame)	114	137	126

For a long, relatively cool flame with the same heat input the heat flow to the glass bath is 1.25 MW and the flue gas heat flow is 2.46 MW, where the flue gas still contains unburned fuel. Combustion chamber efficiency is now 35 %.

4.2. Results for simplified radiation models

It is interesting to compare the heat fluxes to the glass bath obtained with the various radiation models that are used in the present study. The well-stirred furnace model using a 100 and a 96 % burnout, respectively, to correspond with the short and long flame configuration of the complete model, gives an average heat load of the glass of 114 and 104 kW/m² respectively (table 1). The plug-flow model used with a precombustion of 44 %, equal to the precombustion in the complete model, and a burnout of 100 and 96 %, respectively, gives an average glass heat load of 137 and 112 kW/m², respectively. The complete model, with a precombustion of 44 % and with a short, hot flame configuration in which the burnout is 100 %, gives an average heat load of the glass of 126 kW/m². For a longer and cooler flame, where the burnout is 96 %, the average glass heat load is 104 kW/m². This would indicate that, with respect to the average radiative heat transfer, the well-stirred furnace model gives very good predictions for a long-flame configuration when taking incomplete combustion into account and the plug-flow model gives reasonable predictions for the short-flame case.

However, the distribution of the heat flux to the glass found with the complete model (using Hottel's zone method) differs from the distribution found using the plug-flow model, whereas the well-stirred model only predicts the average flux and can not give a flux distribution at all. Figure 3 shows the flux distributions for short and long flames as well as found from the plug-flow model. It is clear that the complete model gives a much more uniform heat flux. The plug-flow model shows unrealistic high values for the peak flux. The main cause for this is that in the plug-flow model no radiative exchange in the flame direction, arising from the hot flame gases is taken into account. This assumption works well for very long and low furnaces, but in the present case it



Figures 4a to c. Direct (q_{dir}), reflected (q_{refl}) and total fluxes (q_{glass}) for varying flame emissivity ($\Delta\varepsilon$) (figure a), glass temperature (figure b) and secondary air temperature (figure c).

predicts a heat-flux distribution which is too peaked about half-way of the furnace. The ratio of the highest heat flux to the lowest one is 1.4 for a long flame and 2.0 for a short flame. For the plug-flow model this ratio is found to be 3.3 for the 96 % burnout case.

The reliability of the well-stirred furnace model for long-flame configurations makes it possible to investigate the influence of several parameters with relatively little computational effort, but still with sufficient accuracy, and enables the establishment of trends in the behavior of radiative exchange in glass melting furnaces. Figures 4a to c show the results of a number of sensitivity studies which were made using the well-stirred furnace model.

The effects of flame emissivity, glass temperature and secondary air temperature have been studied. The resulting heat flows to the glass are given in figures 4a to c, they show very clear trends. Noteworthy is the effect of flame emissivity. Figure 4a shows the effect of an increase in the flame emissivity with a value $\Delta\varepsilon$ to simulate the effect of additives (oil injection) in the natural gas, the emissivity for the gas itself being 0.2. This gives a higher heat flow to the glass. However, it must be considered that a higher heat flow leads to a lower flue-gas temperature. As the flue gases are used to preheat the secondary air this means a reduction of the secondary air temperature. This compensates part of the effect. Taking this into account it has been calculated that an increase ($\Delta\varepsilon$) of 0.30 in ε results in a 4.5 % lower gas consumption, keeping the heat load of the glass the same (104 kW/m^2).

4.3. Spectral effects on radiative transfer

As the well-stirred radiation model gives a good prediction of the average heat transfer in a glass furnace this was used to study the spectral effects. Use has been made of a 15-band radiation spectrum of the flue gases as reported by Elliston et al. [19] and Alexander et al. [24] and given in figure 5. In the

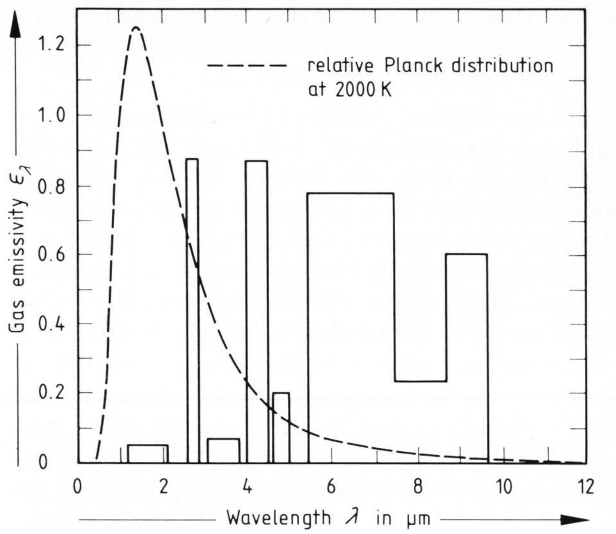


Figure 5. Band spectrum of gas emissivity (ε_λ) and Planck distribution at 2000 K.

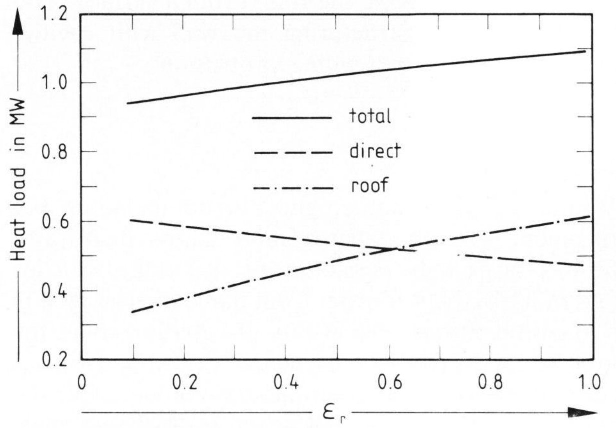


Figure 6. Effects of roof emissivity (ε_r) on heat flux to glass bath.

same figure the Planck spectrum at a temperature of 2000 K is shown. This represents the case of gray radiation at flue-gas temperature of the relatively high-temperature glass furnace considered here. It is clear that a large part of the radiation is in the visible and near-infrared range. In that range the gas absorption shows some peaks and many windows. The refractory materials being of a white or light gray color will have a low emissivity in that range. For a specific sample an emissivity for 2000 K of 0.4 was measured.

Using this 15-band model one can calculate for each band the radiative transfer at an assumed flue gas temperature T_{fg} by the one-zone method. The total energy transfer summed over the spectrum (15 bands) has been used in an enthalpy balance to satisfy the first law of thermodynamics. If this has not been satisfied a new guess of T_{fg} has been made and by a few iterations the value of T_{fg} to satisfy the enthalpy balance has been found. Figure 6 gives the results for the heat flow to the melt as a function of the emissivity ε_r of the roof. An increase of total heat flow to the glass melt of 8 % has been found by increasing the roof emissivity from 0.4 to 1.

One can understand this by considering the direct and roof heat flows to the bath separately. The roof contribution goes up with ε_r because the gray emissivity of the roof transforms the radiation from band to a continuously emitted radiation. With low ε_r values there is mainly reflection in the same bands as the gas radiation. This will then be partly reabsorbed in the gas and does not reach the melt. For higher ε_r values roof radiation in the wavelength regions of the windows will be fully transmitted to the melt. The roof radiation to melt increases considerably. With increasing ε_r values at constant gas load a lower flue gas temperature satisfying the enthalpy balance has been found. The lower value of T_{fg} gives a lower direct gas radiation to the melt. However, the net result is positive as figure 6 shows. In practical applications one can increase ε_r by special coatings [21 to 23] or by giving the roof a rough surface. This can be done by structuring the wall with cavities and/or outwardly extending protrusions.

5. Conclusions

With a complete mathematical model including 3-D turbulent flow, combustion and radiative heat transfer it is shown to be possible to predict flame behavior and radiative heat transfer from flame to glass melt in a gas-fired furnace. Heat-flux distributions over the melt surface have been predicted. A comparison has been made with simplified radiation models. For the average heat-flux calculation the well-stirred single zone model gives a good prediction compared to the complete model, especially for long flames. A plug-flow model also gives a reasonable prediction of average heat load, for short flames, but largely overpredicts local peak heat fluxes to the glass melt. With a well-stirred gray gas model it has been shown that increasing gas emissivity from 0.2 to 0.5 results in a 4.5 % lower gas consumption at a constant heat load to the melt.

Spectral effects in gas radiation have been included in the well-stirred zone model by a 15-band approximation. It has been shown that increasing the roof emissivity from 0.4 to 1 increased the heat flow to the melt by 8 % at a gas temperature of about 2000 K. It can be concluded that applying high emissive coatings or rough structured roof walls can increase glass furnace efficiency when gas firing is applied.

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