A COUPLED NUMERICAL STUDY OF SLAB TEMPERATURE AND GAS TEMPERATURE IN THE WALKING-BEAM TYPE SLAB REHEATING FURNACE

Chia-Tsung Hsieh1*, Mei-Jiau Huang1, Shih-Tuen Lee1, Chao-Hua Wang2

1Department of Mechanical Engineering, National Taiwan University
2China Steel Corporation, Kaohsiung, Taiwan, R.O.C.

* E-mail: sgt0714@yahoo.com.tw, cthsieh@scvs.tpc.edu.tw

ABSTRACT

In the present study, a three-dimensional simulation is performed for the turbulent reactive flow and radiative heat transfer in the walking-beam type slab reheating furnace by STAR-CD software. The geometric model takes care of all components of the furnace, including the burners, the walking beam system with skid buttons, the slab, the dam and the down-take etc. The study employs the high-Reynolds-number k-ε turbulence model based on Favre-averaged governing equations. The pre-assumed PDF model associated with the fast chemistry assumption and a single diffusivity is used to account for turbulent combustion. The absorption coefficient of the gases mixture is calculated by WSGGM (weighted-sum-of-gray-gases model). The discrete ordinates radiation model is adopted to calculate the radiative heat transfer. The temperature distributions of the slab and the gas mixture are obtained through a coupled calculation. The slab is modeled as a laminar flow having a very high viscosity and thus moving at a nearly constant speed. No radiation is concerned within the slab. To obtain a steady solution, the walking beams are assumed fixed in the furnace. The simulation results agree with the measurements very well. The difference between the predicted heating efficiency and the measured one of the furnace is only 3.7%. The prediction errors at six temperature-monitored points are all under 10%, except the one in the heating zone which appears to be 13%. The simulation results successfully predict the surface temperature and the heat flux distributions on the slab in the reheating furnace, and discover, for the first time, the influence of the walking beam system with skid button on the skid mark. The result shows that radiative shielding is the main cause of the skid mark. The heat loss from the slab through the skid button to the cooling system worsens the skid mark. Consequently, the temperatures of the slab above the static beams are lower than those temperatures measured in the middle and on side surfaces of the slab. This imposes unfavorable influences on the following rolling process.

Keywords: reheating furnace, skid mark, radiative shielding.

NOMENCLATURE

\[ a_{e,i}, b_{e,i}, k_{e,i} \] the coefficients for the \( i \)th gray-gas component
\[ c_p \] specific heat
\[ C_D \] the efficient for the PDF model
\[ C_{1e}, C_{2e}, C_{1i}, C_{2i} \] constants in the k-ε model
\[ D_m \] mass diffusivity
\[ D_m^\tau \] the thermal diffusivity associated with the species \( m \)
\[ f \] mixture fraction
\[ f^* \] mixture fraction variance
\[ F_{b,j} \] total energy flux
\[ g_i \] gravity acceleration
\[ G \] the total incident radiative intensity
\[ h \] static enthalpy
\[ h_t \] thermal enthalpy
\[ H_m \] the heat of formation
\[ I \] radiation intensity
\[ k \] thermal conductivity
\[ k \] turbulent kinetic energy
\[ L \] optical path
\[ n \] the unit surface vector directed outwards
\[ P \] pressure
\[ P \] the probability density function
\[ r \] position vector
\[ R \] the universal gas constant
\[ S_{\dot{e}} \] the mean rate of strain tensor
\[ s_e \] source term
\[ T \] temperature
\[ t \] time
\[ u_i \] the \( i \)th velocity component
\[ u' \] fluctuation velocity
\[ \Delta V \] cell volume
\[ V_{m,j} \] the \( j \)th component of the diffusion velocity of the \( m \)th constituent

* Chia-Tsung Hsieh serves in National San-chung Commercial and Industrial Vocational High School
\( X \) mole fraction
\( x \) the moving direction of the slabs
\( x \) Cartesian coordinate
\( Y \) mass fraction
\( y \) the height direction in the furnace
\( z \) the transverse direction in the furnace
\( \alpha \) The radiation absorption coefficient
\( \varepsilon \) turbulent kinetic energy dissipation rate
\( \varepsilon \) the emissivity
\( \gamma \) the reflectivity
\( \delta \) Kronecker delta
\( \kappa \) absorption coefficient
\( \mu \) molecular viscosity
\( \mu_t \) turbulent viscosity
\( \rho \) density
\( \sigma \) Stefan-Boltzman constant
\( \sigma_s, \sigma_e, \sigma_z \) turbulent Prandtl number
\( \tau \) total stress tensor
\( \Psi \) any physical property
\( \Omega \) solid angle

Subscripts
\( i,j,k \) indices
\( m \) the species m
\( t \) time or turbulence
\( w \) wall

Superscripts
\( \sim \) mass-weighted average quantity
\( \cdot \) time average quantity
\( ' \) fluctuation quantity in Reynolds averaging
\( " \) fluctuation quantity in Frave averaging

1. INTRODUCTION

Before the slab is processed into a plate, it is necessary to heat the slab in the reheating furnace above the recrystallization temperature for subsequent rolling process. Furthermore, it is significant to understand that the slab temperature and heat flux are both important factors when conducting quality control of the finished products of steel plates. In the reheating furnace, burning fuels cause a high temperature turbulent flow and heat of combustion is transmitted in the form of radiation into the slab through the refractory wall and high-temperature gases. The walking-beam system, for moving and supporting the slab, will block some radiation, and heat will be transmitted from the slab to the cooling system through the contact area between the slab and the walking-beam system. Uneven temperature distributions, i.e. skid marks, are thus caused.

In order to improve skid marks, to enhance the efficiency of the reheating furnace, and to decrease pollution, research has been conducted. Ford et al. [1] established the heat transfer model between the slab and cooling water, and suggested the temperature of cooling water be raised to lower the heat loss to the cooling system. Zongyu et al. [2] calculated the radiative exchange by the zone method, and observed the transient thermal response of the slab in the pusher-type reheating furnace. The result indicates the influence of the radiation shielding effect is much greater than the heat loss problem, and the radiation of the refractory walls causes uneven heating in the transverse direction. Chapman et al. [3] solved the burned-gas velocity and temperature fields based on the \( k-\varepsilon \) turbulence model, the probability density function (PDF) combustion model, and the gray-gas assumption, and so on, to analyze one-dimensional and two-dimensional direct-fired furnaces. Zhang et al. [4] applied the moment closure method combustion model, and set up a more detailed three-dimensional analytical model. Uede et al. [5] utilized the commercial software, STAR-CD, to simulate the reheating furnace and explore the influence of the geometrical shape of the reheating furnace and the interval of burners on the distribution of high-temperature gases and on the creation rate of NOx. Kim et al. [6,7] utilized the Fluent software to investigate high-temperature gases and velocity field, and obtained the temperature distributions of the slab and skid buttons. Kim et al. [6] presumed the surface temperature and the emissivity of the slab and skid buttons, and attempted to capture the temperature field of the gases and the net heat transfer rate on the surfaces of the slab. The later was then employed to predict the temperature distribution inside the slab[7]. Skid marks were thus computed. Tang et al. [8] applied the commercial software, Phoenics, to conduct research on a pusher-type reheating furnace for the sake of understanding what kind of a flow field and where in the reheating furnace can easily cause the scale accumulation. Kim[9] only took into consideration the outer shape of the reheating furnace and discussed the heating process of the slab in different heating zones.

Measurements of surface temperatures of the slab can be employed to specify the boundary conditions and to verify the accuracy of the calculations. However due to the limit of measuring equipments, positions measured are limited and the error involved with high-temperature measurements turns out to be large. When slab surface temperatures are measured, thermocouple can be affected by radiation and convection of high-temperature gases. When temperatures inside the slab are measured, the contact thermal resistance between the thermocouple and the slab can also affect the measuring result. Wikstrom et al. [10] utilized the inverse method, and calculated surface temperatures of the slab with measured temperatures inside the slab. Hsieh et al. [11] applied the software, STAR-CD, to simulate a more detailed reheating furnace, which includes all burners and the walking-beam system that supports the slab. The measured surface temperatures of the slab are used as boundary conditions and the
radiation shielding effect of the walking-beam system was studied.

Above all, all the previous investigations of a slab-reheating furnace presumed slab temperatures, calculated gas temperature, and obtained the heat transfer information of the slab. On the other hand, the simulation of temperature distribution inside the slab is done based on the condition that gas temperature is known or the heat flux at the slab surfaces is known. Iteration is thus required in order to get a more accurate solution. In the study, we instead attempt to perform a coupled simulation in which neither the slab surface temperature nor the slab surface heat flux is pre-known. The complicated temperature and heat flux distributions on the slab surfaces will be solved simultaneously. A detailed, qualitative as well as quantitative, study of the skid marks is thus targeted. This paper is arranged as follows. The geometry of the reheating furnace under investigation and its operating conditions are introduced in section 2. The mathematical models employed are summarized in section 3. The simulation results and discussions are given in section 4. The conclusion is made in section 5.

### 2. Furnace Structure and Operating Conditions

#### 2.1 Furnace Structure

The furnace is divided into four zones along the moving direction (x) of the slabs, namely the preheating, heating, soaking, and screen zones as shown in Fig.1. The furnace under investigation has a dimension of 29.9m × 8.2m × 4.6m. It is symmetric with respect to the plane z=0. Therefore, only half of the furnace is simulated for the purpose of reducing the computational time. The dams separating the zones are connected to the furnace wall as well as the symmetrical plane but have a breach in the middle as illustrated in Fig.2. There are four sets of axial burners in the upper zones, each having three burners distributed along the transverse (z) direction in a half furnace. Ten side burners are there in the lower zones. The down-take outlets exist on both sides of the furnace, below and adjacent to the slab entrance as shown in Fig.1b. The down-take outlet in the simulation model is 5m extended downward in order to prevent the occurrence of a reverse flow so that the wasted gases can flow out smoothly. In the furnace, the slab is supported by two static beams and moved by two walking beams. The static beams are in contact with the slab by skid buttons. Skid buttons are made of heat-resistant steel and welded on the cooling pipe as shown in Fig.3.

The constructed cells used for the finite volume method are mostly 10cm×10cm×10cm cubes. Cells near the furnace wall are adjusted to fit the actual shape of the reheating furnace. To capture the influence of the skid buttons more accurately, the cells of the slab above the skid buttons are refined. Skid buttons are shaped into rectangular columns. Each static (walking) beam is mounted by 10 (9) pillars. Cooling pipes and pillars are coated by castable. The wall thickness of the cooling pipes is small compared to that of the castable and thus ignored in the present study. A constant temperature (the temperature of the cooling water) is imposed at the walls of the cooling pipes inside the beams as well as the pillars. The cross sections of the walking and static beams are designed to be like an oval runway as also shown in Fig.3, similar to that commonly used in a real furnace. Finally, there are six (upper) and nine (side) air entrance holes in a real burner. For simplicity, it is assumed that the fuel entrance hole is surrounded by just four air entrance holes as shown in Fig.3c.

Consequently, a total of 1,236,237 cells are generated for the whole furnace, among which 15.1% of the cells are associated with the slab, 25.9% are associated with the solid components, and the rest is associated with the gas mixture.

The temperature distributions of the slab and the gas mixture are obtained through a coupled calculation. In order to obtain a steady solution, it is assumed the walking beams are fixed and the slab is continuously pushed into the furnace. In fact, the slab is modelled as a laminar flow having a very large viscosity. The outer surfaces of the slab exchange heat with the surroundings by convection as well as radiation. Velocity and temperature at the interface between the slab and the gas mixture are determined based on the continuity conditions of momentum and energy. No radiation is considered nonetheless inside the slab.

---

![Fig. 1. An illustration of the walking-beam-type slab-reheating furnace under investigation. P1-P6 are measured points. (a). A three-dimensional computational model of a half furnace. (b).](image-url)
Fig. 2. An illustration of the arrangement of relevant components in the furnace.

Fig. 3. Part of the numerical model constructed for the walking-beam system. The two beams with skid buttons are static beams and the other two are walking beams (a). A cross-section showing the slab, the static beam, the skid button, and the cooling system (b). The cells near the air and fuel entrance holes of a burner (c).

2.2 Operating Conditions

The fuel used in this study is the coke oven gas, which components and reaction equation are as follows:

\[
\begin{align*}
0.5715H_2 + 0.2335CH_4 + 0.0705CO + 0.0269C_2H_4 \\
+ 0.074N_2 + 0.0014O_2 + (0.8687 - 0.0014)(O_2 + 3.76N_2) \\
+ 0.0222CO_2 \rightarrow 1.0923H_2O + 0.38CO_2 + 3.335N_2
\end{align*}
\]

The coke oven gas enters with a flow rate of 8868 m³/hr at 300 K. The airflow rate is 81282 m³/hr at 700 K. The side burners have a swirling number of 0.2. The outlet pressure is 20 Pa gauge. The slab is made of low carbon steel, with a density of 7854 kg/m³, and each piece has a dimension of 1.8 m × 0.27 m × 3.6 m. Its viscosity is set as large as \(5 \times 10^{-5}\) kg/ms to assume a laminar flow having a constant speed of 0.0025 m/s. The thermal conductivity and the specific heat of the slab are functions of temperature as shown in Fig. 4. The materials of the walking-beam system and dams are castable with a thermal conductivity \(k\) of 0.7 W/mK, a density \(\rho\) of 2000 kg/m³, and a specific heat \(c_p\) of 800 J/kgK. The temperature of the cooling water is 309.5 K. The radiation absorption coefficient \(\alpha\) of the outer slab surfaces (the surfaces between the furnace gas and the slab) is assumed to be 0.85. The slab is opaque inside however. That is, the absorption coefficient \(\kappa\) inside the slab is zero, and all inner slab surfaces have zero emissivity. Consequently, no radiation is considered inside the slab. All the other solid boundaries inside the furnace are assumed to be blackbody, i.e. \(\alpha = 1\). Finally, the wall, the roof, and the floor of the furnace are all assumed to be adiabatic.

Fig. 4. (a) The thermal conductivity and (b) the specific heat of the slab.

3. MATHEMATICAL MODELS

3.1 Turbulence model

The Favre-averaged equations of the continuity, momentum, energy equation are employed and shown below:

\[
\frac{\partial \bar{\rho}}{\partial t} + \frac{\partial (\bar{\rho} \bar{u}_j)}{\partial x_j} = 0
\]

\[
\frac{\partial \bar{\rho} \bar{u}_j}{\partial t} + \frac{\partial (\bar{\rho} \bar{u}_j \bar{u}_j - \tau_{ij})}{\partial x_j} = -\frac{\partial \bar{p}}{\partial x_j}
\]

\[
\frac{\partial \bar{h} \bar{u}_j}{\partial t} + \frac{\partial (\bar{h} \bar{u}_j + F_{h,j})}{\partial x_j} = \frac{\partial \bar{P}}{\partial t} + \bar{u}_i \frac{\partial \bar{P}}{\partial x_j} + \tau_{ij} \frac{\partial \bar{u}_j}{\partial x_j} + s_h
\]

where \(\bar{\cdot}\) represents the Favre-average, \(x_j\) is the Cartesian coordinate, \(\bar{u}_j\) is the velocity vector, and \(\mu\) is the molecular viscosity. The total stress tensor \(\tau_{ij}\) is defined as

\[
\tau_{ij} = 2 \mu S_{ij} - \frac{2}{3} \mu \frac{\partial \bar{u}_k}{\partial x_k} \delta_{ij} - \bar{p} \delta_{ij} \frac{\partial \bar{u}_j}{\partial x_j}
\]

in which the Reynolds stress is assumed to be

\[
\frac{\partial \bar{p} \bar{u}_i}{\partial x_j} = 2 \mu S_{ij} - \frac{2}{3} \mu \left( \frac{\partial \bar{u}_k}{\partial x_k} + \frac{\partial \bar{u}_j}{\partial x_j} \right) \delta_{ij}
\]
and \( S_y \) is the mean rate of strain tensor,
\[
S_y = \frac{1}{2} \left( \overline{\partial u_i} \overline{\partial x_j} + \overline{\partial u_j} \overline{\partial x_i} \right)
\]  
\( \text{(6)} \)

Moreover, in Eq.(3), \( h \) is the static enthalpy defined as \( h = h_i + \sum Y_{m} H_{m} \), where \( h_i \) is the thermal enthalpy, and \( Y_m \) and \( H_m \) are the mass fraction and the heat of formation of the \( m \)th constituent. The source term \( s_h \) in Eq.(3) arises from the radiation heat transfer. Finally \( F_{h,j} \) is the total energy flux defined as
\[
F_{h,j} = -k \frac{\partial T}{\partial x_j} + \overline{\rho u_j h'} + \sum_{m} h_m \rho V_{m,j}
\]  
\( \text{(7)} \)

where the turbulent diffusion is evaluated by
\[
\overline{\rho u_j h'} = \frac{\mu_t}{\sigma_k} \frac{\partial h}{\partial x_j}
\]  
\( \text{(8)} \)

The symbol \( V_{m,j} \) represents the \( j \) component of the diffusion velocity of the \( m \)th constituent, namely
\[
V_m = -\frac{1}{X_m} D_m \nabla X_m - \frac{D_m^T}{\rho X_m} \nabla \ln T
\]  
\( \text{(9)} \)

where \( X_m \) is the mole fraction, \( D_m \) is the mass diffusivity and \( D_m^T \) is the thermal diffusivity associated with the species \( m \).

The turbulent model employed herein is the high-Reynolds-number \( k-\varepsilon \) model. The turbulent kinetic energy (\( k \)) and its dissipation rate (\( \varepsilon \)) are governed by
\[
\frac{\partial \overline{p k}}{\partial t} + \frac{\partial}{\partial x_j} \left[ \overline{p u_j} \overline{k} - (\mu + \frac{H_1}{\sigma_k}) \frac{\partial \overline{k}}{\partial x_j} \right] = \mu_t (P + \rho b) - \frac{2}{3} \left( \mu + \frac{H_1}{\sigma_k} \right) \frac{\partial \overline{u_i}}{\partial x_i} \overline{\partial u_j} \overline{\partial x_j}
\]  
\( \text{(10)} \)

and
\[
\frac{\partial \overline{p} \varepsilon}{\partial t} + \frac{\partial}{\partial x_j} \left[ \overline{p u_j} \overline{\varepsilon} - (\mu + \frac{H_1}{\sigma_k}) \frac{\partial \overline{\varepsilon}}{\partial x_j} \right] = C_{\varepsilon} \frac{\overline{\varepsilon}}{k} \left[ \mu_t P - \frac{2}{3} \left( \mu + \frac{H_1}{\sigma_k} \right) \frac{\partial \overline{u_i}}{\partial x_i} \overline{\partial u_j} \overline{\partial x_j} \right] + C_{\varepsilon} \frac{\overline{\varepsilon}^2}{k} + C_{\varepsilon} \overline{p e} \frac{\partial \overline{u_i}}{\partial x_i}
\]  
\( \text{(11)} \)

with
\[
\mu_t = C_{\mu} \frac{\overline{p} \varepsilon^2}{\varepsilon}
\]  
\( \text{(12)} \)

\[
P = S_{\theta} \frac{\overline{\partial u_i}}{\partial x_j}
\]  
\( \text{(13)} \)

\[
P_{\theta} = \frac{g}{\sigma_\theta} \frac{1}{\overline{\overline{P}}} \frac{\overline{\partial T}}{\partial x_j}
\]  
\( \text{(14)} \)

where \( g \) is the gravity acceleration and \( \sigma_\theta \) is the turbulent Prandtl number. The values of model parameters used are: \( C_{\mu} = 0.09 \), \( \sigma_\theta = 1.0 \), \( \sigma_k = 1.22 \), \( \sigma_\varepsilon = 0.9 \), \( C_{\varepsilon} = 1.44 \), \( C_{f} = 1.92 \), \( C_{k} = -0.33 \), and \( C_{\varepsilon} = 1.44 \) for \( P_b > 0 \) and \( C_{\varepsilon} = 0 \) for \( P_b \leq 0 \).

### 3.2 Turbulent Combustion Model

The fast chemistry and a single diffusivity are assumed in the present study [12, 13]. Because all burners have a same fuel and a same inlet temperature, the single-fuel PPDF model (Prescribed Probability Density Function) is applied herein [14]. In the PPDF model, the mean and variance of the mixture fraction, \( f \), are first obtained, by solving
\[
\frac{\partial \overline{p f}}{\partial t} + \frac{\partial}{\partial x_j} \left[ \overline{p u_j} f - (\rho D_f + \frac{\mu}{\sigma_\varepsilon}) \frac{\partial \overline{f}}{\partial x_j} \right] = 0
\]  
\( \text{(15)} \)

\[
\frac{\partial \overline{p f^2}}{\partial t} + \frac{\partial}{\partial x_j} \left[ \overline{p u_j} f^2 - (\rho D_f + \frac{\mu}{\sigma_\varepsilon}) \frac{\partial \overline{f^2}}{\partial x_j} \right] = 2 \frac{\mu}{\sigma_\varepsilon} \frac{\partial \overline{f}}{\partial x_j} + C_{\varepsilon} \overline{p f} \frac{\partial \overline{u_i}}{\partial x_i}
\]  
\( \text{(16)} \)

where \( C_{\varepsilon} = 2.0, \sigma_\varepsilon = 0.9, \) and \( D_f = 3.004 \times 10^{-5} \) are employed. Any physical property \( \Psi \) (except the density) is then computed as
\[
\Psi = \frac{1}{0} \Psi' f P(f) df
\]  
\( \text{(17)} \)

where \( \Psi' (f) \) is the adiabatic equilibrium value of \( \Psi \).

The prescribed PDF is a \( \beta \) function defined as follows:
\[
P(f) = \frac{f^{\alpha - 1} (1 - f)^{b - 1}}{\int_a f^{\alpha - 1} (1 - f)^{b - 1} df}
\]  
\( \text{(18)} \)

where
\[
a = \frac{\int_a f^{\alpha - 1} (1 - f)^{b - 1} df}{\int_a f^{\alpha - 1} (1 - f)^{b - 1} df}
\]  
\( \text{(19)} \)

Finally, the ideal gas law is used to calculate the local density, that is
\[
\rho = \frac{P}{RT \left( \sum_{m} \frac{Y_m}{M_m} \right)}
\]  
\( \text{(20)} \)

### 3.3 Radiative Heat Transfer

In this study, the discrete ordinates method is employed to solve the radiative heat transfer equation [15]:
\[
(\Omega \cdot \nabla) I_r(\Omega) + \kappa(r) I(r, \Omega) = \kappa(r) I_0(\Omega)
\]  
\( \text{(21)} \)

where \( I \) is the radiation intensity, \( I_0 \) is the blackbody intensity, \( \Omega \) is the solid angle, and \( \kappa \) is the absorption
coefficient. The gas is assumed non-scattering. The boundary condition for the radiative heat transfer equation is
\[ I(r_e, \Omega) = \varepsilon(r_e) I_0(r_e, \Omega) + \frac{\gamma(r_e)}{\pi} \int_{-\Omega}^{0} n \cdot \Omega' |I(r_e, \Omega')| d\Omega', \quad n \cdot \Omega > 0 \quad (22) \]
where \( \varepsilon(r_e) \) is the emissivity of the wall and \( \gamma(r_e) = 1 - \varepsilon(r_e) \) is the reflectivity. Once the radiation intensity field is obtained, the energy source term can be calculated by
\[ s_s = \kappa(r)[G(r) - 4\sigma T^4(r)] \quad (23) \]
where
\[ G(r) = \int_{4\pi} I(r, \Omega) d\Omega \quad (24) \]

### 3.4 The Absorption Coefficient of Radiation

In this study, the weighted-sum-of-gray-gases model (WSGGM) [16] is adopted. We only consider the radiations of CO, CO\(_2\), and H\(_2\)O. First of all, the emissivity of CO\(_2\) and H\(_2\)O suggested in [17] is accepted, that is
\[ \varepsilon_{CO_2,H_2O} = \frac{3}{7} \kappa_{g_i} (T) \left(1 - \exp \left(-k_{g_i} \left(P_{CO_2} + P_{H_2O}\right) L\right)\right) \quad (25) \]
\[ a_{g_i}(T) = \sum_{j=0}^{3} b_{g_i,j} T^j \quad (26) \]
where \( P_{CO_2} \) and \( P_{H_2O} \) are the partial pressures of CO\(_2\) and H\(_2\)O. The associated coefficients \( k_{g_i} \) and \( b_{g_i,j} \) are listed in Table 1. Furthermore, the optical path \( L \) is taken to be
\[ L \equiv 0.6A^{1/3} \quad (27) \]
where \( A^{1/3} \) is the cell volume. The absorption coefficient of CO\(_2\) and H\(_2\)O is then computed as
\[ \kappa_{CO_2,H_2O} = \frac{1}{L} \ln \left(\frac{1}{1 - \varepsilon_{CO_2,H_2O}}\right) \quad (28) \]

Next, the empirical relation suggested by Maracino and Lentini [18] for the absorption coefficient of CO is used. The absorption coefficient of CO is determined by
\[ \log_{10} \kappa_{CO} = 4.6398 \cdot 10^{-4} T^7 - 5.805 \cdot 10^{-5} T^6 + 2.5489 \cdot 10^{-12} T^4 - 3.8439 \cdot 10^{-10} T^2 \quad (29) \]
\[ \kappa_{CO} = k_{CO} P_{CO} \quad (30) \]
with \( P_{CO} \) being the partial pressure of CO. Finally, the total absorption coefficient, \( \kappa_{total} \), is
\[ \kappa_{total} = \kappa_{CO_2,H_2O} \left(T, P_{CO_2}, P_{H_2O}\right) + \kappa_{CO} \left(T, P_{CO}\right) \quad (31) \]

### Table 1. The coefficients used for evaluating the emissivity of CO\(_2\) and H\(_2\)O (Smith et al. [17])

<table>
<thead>
<tr>
<th>( i )</th>
<th>( k_{g_i} )</th>
<th>( b_{g_i,1} \times 10^3 )</th>
<th>( b_{g_i,2} \times 10^3 )</th>
<th>( b_{g_i,3} \times 10^3 )</th>
<th>( b_{g_i,4} \times 10^3 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.4201</td>
<td>6.508</td>
<td>-5.551</td>
<td>3.029</td>
<td>-5.353</td>
</tr>
<tr>
<td>2</td>
<td>6.516</td>
<td>-0.2504</td>
<td>6.112</td>
<td>-3.882</td>
<td>6.528</td>
</tr>
<tr>
<td>3</td>
<td>131.9</td>
<td>2.718</td>
<td>-3.118</td>
<td>1.221</td>
<td>-1.612</td>
</tr>
</tbody>
</table>

### 4. RESULTS AND DISCUSSIONS

#### 4.1 Verification of the Computations

In this section, we first compare the calculated results with the experimental measurements for the sake of verifying the accuracy of the simulations. Listed in Table 2 are the calculated and measured temperatures at six selected locations inside the furnace. The first three checking points are located in the upper zones and the other three in the lower zones. The calculated ones are obtained by averaging the gas temperature in a small neighborhood of each selected location (about a cubic region of \( 30 \times 30 \times 30 \) cm\(^3\)). The results agree very well. The maximum error occurs at the fifth checking point, which is found very close to a side burner. A larger measurement error is conjectured due to the influence of radiation on the thermocouple.

The heating efficiency of the reheating furnace is examined now. The combustion energy is found to be 18135 kW and the sensible energy of the preheated air is 2370 kW. The slab enters into the furnace at 300K and leaves with an averaged surface temperature of 1308K. Assuming the temperature inside the interior of the slab is also uniform at 1308K, the total heat absorbed is thus 13169 kW, corresponding to a heating efficiency of 64.2%. The total heat transfer rate (into the slab) is found to be 12486 kW (see Table 3) in the numerical furnace. The heating efficiency is about 60.9%. The difference between the simulated and measured heat transfer rate is 683 kW, about 3.3% of the total input energy. The reason why the heating efficiency of the real furnace is higher may be explained by an overestimation of the average slab temperature at exit due to the uniformity assumption.

The results of measured and calculated temperatures along the central lines as well as lines above the static beams on the upper and lower surfaces of the slab are shown as Fig. 5. Figure 5(a) shows the calculations on the lower surface of the slab agree well with the measurements, except those bumps that appear near side burners. The disagreement is significant on the upper surface of the slab in the preheating zone. The measured temperatures are found closer to the gas temperatures. When the temperatures on the upper surface of the slab are measured, the thermocouples are just placed on the top of the slab and exposed to the high-temperature gas. Measurements must be contaminated by the radiation exchange between the high-temperature gas and the thermocouple.
Consequently, the measured temperatures are closer to the gas temperature.

In short, the temperatures at all checking points, the heating efficiency of the furnace, and surface temperatures of the slab are all examined acceptably accurate, so the accuracy of the simulation is verified.

Table 2. Comparison of the measured and calculated temperatures at six selected locations, three (1-3) in the upper zones and three (4-6) in the lower zones.

<table>
<thead>
<tr>
<th>Check Point</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measurement</td>
<td>1265</td>
<td>1503</td>
<td>1459</td>
<td>1334</td>
<td>1479</td>
<td>1444</td>
</tr>
<tr>
<td>Calculation</td>
<td>1244</td>
<td>1438</td>
<td>1366</td>
<td>1238</td>
<td>1296</td>
<td>1373</td>
</tr>
<tr>
<td>Relative error</td>
<td>1.7%</td>
<td>4.4%</td>
<td>6.4%</td>
<td>7.2%</td>
<td>12.4%</td>
<td>4.9%</td>
</tr>
</tbody>
</table>

Fig. 5. The temperature distributions along the central lines (a) and lines above the static beams (b) on the upper and lower surfaces of the slab.

4.2 Analysis of Heat Transfer

We first show the temperature and heat flux distributions on the slab surfaces in Fig. 6. As expected, the distributions are highly non-uniform due to the complexity of the furnace. The calculated results show radiation is responsible for, up to 97%, the heat transfer to the slab (see Table 3). In the upper zones, the gas temperature is always higher than the slab temperature. Heat flux increases when gas temperature rises and the maximum is measured near the burners in the upper heating zone. Later the heat flux becomes smaller in the soaking zone and screen zone mainly because there the temperature of the slab has been increased to certain extent.

The heat flux distribution on the lower surface is strongly influenced by the existence the skid buttons and the beams. First, the contact area between the slab and the skid buttons on the static beams is blocked from radiation completely. The static beams are only 58mm away from the slab. Figure 6 (d) shows radiation shielding causes the heat flux in the lower surface above the static beams much lower, and so are the temperatures. The walking beams are 158mm away from the slab and have weaker influence on the heating of the slab but the heat flux in the lower surface above the walking beam is still lower than those along the central line of the lower surface. It indicates that the radiation shielding effect of the walking-beam system is very apparent. The local maxima in Fig. 6(d) correspond to the locations of the side burners. In remark, along the transverse (z) direction, the temperature is the lowest on the lower surface right above the static beams. The second lowest appears above the walking beams. The temperature at the center of the slab was the third lowest while temperatures at edges were the highest.

Table 3. The heat transfer rates into the slab through the top, bottom, and the side surfaces.

<table>
<thead>
<tr>
<th></th>
<th>Upper zones</th>
<th>Lower zones</th>
<th>Total rate (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Preheating zone</td>
<td>2318.5</td>
<td>132.6</td>
<td>2213.9</td>
</tr>
<tr>
<td>Heating zone</td>
<td>2220.2</td>
<td>41.1</td>
<td>1979</td>
</tr>
<tr>
<td>Soaking zone</td>
<td>1041.7</td>
<td>10.6</td>
<td>1103.6</td>
</tr>
<tr>
<td>Screen zone</td>
<td>416.1</td>
<td>12.4</td>
<td>267.4</td>
</tr>
<tr>
<td>Skid button</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Total rate</td>
<td>5996.5</td>
<td>196.7</td>
<td>5563.9</td>
</tr>
</tbody>
</table>

(48.0%) (1.6%) (44.6%) (1.1%)
4.3 Skid Mark

An uneven temperature distribution inside the slab can cause thermal stresses, which in turn possibly cause the slab curved or even fractured during the subsequent rolling process. Understanding the cause of the uneven distribution can help improving the quality control. Figure 7 shows the cross-sectional temperature distributions of the slab at several locations. It is seen that heat is transferred to the slab from all directions. But the radiation is obviously blocked by the static beams and skid buttons conduct also some heat from the slab to the cooling water. The former is so serious that two local minima are resulted inside the slab, instead of at the skid buttons. Moreover, because the side burners are located on the furnace wall and fuels are burned less there, the temperature of the slab near the furnace wall is lower than that on the other side.

To highlight the heat transfer direction at the skid buttons, the temperature difference between the slab and the skid button are compared in Fig. 8. It is shown that the temperature of the skid button in the preheating zone is higher than that of the slab. With the rising of the slab temperature, the temperatures of the skid buttons soon become lower than that of the slab. Some heat loss through the conduction of the skid buttons to the cooling water is generated and becomes one of the reasons that form skid marks.

![Image](73x253 to 206x273)

Fig. 7. The temperature distributions of the slab at x=28.25m, 28.85m, 28.85m, 29.15m, 29.45m, and 29.75m from the top to the bottom.

![Image](239x256 to 273x345)

Fig. 8. The temperature difference distribution of the slab and the skids.

5. CONCLUSION

This study has successfully built up a three-dimensional numerical model for the walking-beam-type slab-reheating furnace. Relevant components of the furnace, such as the real geometry of the furnace, the burners, the slab, the beams and their pillars, the dams, the down-take outlet, etc., have for the first time all been implemented and solved in a coupled way. The slab is assumed to be a high-viscosity fluid, moving at a constant speed. The accuracy of this numerical model is tested reasonably well. The relative error is controlled under 10%.

Under the condition that furnace wall is adiabatic, the calculational results indicate that the radiation shielding effect is mainly caused by the walking-beam system and is responsible for the skid marks. The temperature of the skid buttons is higher than that of the slab only in the preheating zone; it is lower than that of the slab elsewhere. The skid buttons not only precludes the slab from absorbing radiative flux, but also conduct heat to the cooling water, consequently worsening the skid mark.

ACKNOWLEDGEMENT

The support of this work by the National Science Council, Taiwan, under the contract NSC 95-2212-E-002-336-MY2 and that by the China Steel Corporation under the contract 96T6D0012E are both gratefully acknowledged.

REFERENCES


