COMPUTER SIMULATION OF THE PUSH-TYPE SLAB REHEATING FURNACE

by

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B.Sc. (Eng), Northeast Institute of Technology, 1982

A THESIS SUBMITTED IN PARTIAL FULFILMENT OF
THE REQUIREMENTS FOR THE DEGREE OF
MASTER OF APPLIED SCIENCE

in

THE FACULTY OF GRADUATE STUDIES
Department of Metallurgical Engineering

We accept this thesis as conforming
to the required standard

THE UNIVERSITY OF BRITISH COLUMBIA
DECEMBER 1986

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ABSTRACT

A mathematical heat-transfer model for the slab reheating furnace has been developed. Radiation in the furnace chamber was calculated using the zone method, with the gas temperature distribution being assumed, and heat transfer in the slab was determined using a finite-difference approximation of two-dimensional transient conduction. These individual calculations were coupled to allow prediction of the temperature profiles in, and heat flux to, refractory walls and slabs at any point inside the furnace.

The emissive/absorptive characteristics of the gas mixture within the furnace chamber were simulated with a clear-plus-two-gray-gas model which simulated the real gas behaviour to within 5%. For the calculation of radiative exchanges, the furnace chamber was subdivided into 432 isothermal zones, and radiative exchange factors to slab surfaces were evaluated rather than relying on empirical or experimental estimations as in previous studies. An iterative technique was devised in order to combine the radiative and slab heat conduction calculations. For the purpose of identifying the mechanism of skidmark formation, the region of skidrail/slab contact was examined in detail by introducing a radiation shielding factor to account for the presence of the skid structure.

The gas temperature distribution inside the furnace chamber was found to have a significant influence on the heat flux to the slab surface. Nonuniform gas temperature transverse to the push direction causes an uneven transverse slab temperature distribution and subsequent rolling problems. Higher gas temperatures near the sidewall refractory were
shown to cause serious distortion of the transverse heat-flux distribution.

The heating practice for the hot charging of slabs was simulated by the model in order to improve the process from the standpoint of energy conservation and slab temperature uniformity. Model predictions have shown that the fuel input could be reduced substantially near the slab entrance where the port to the chimney is located, thus maximizing the residence time of the combustion products. Alternatively the throughput of the furnace can be increased if the fuel input remains the same as for charging cold slabs. The extent of increase in production rate can be determined by the off-line computer model.

The model was used to predict the thermal behaviour of slabs for various thicknesses, steel grades and push rates. The results consistently indicated that the selection of an appropriate push rate is crucial to the final temperature distribution.

The study of the mechanism of skidmark formation showed that the radiation shielding effect of the skidrail was the dominant factor, accounting for 90% of the heating deficit around the slab/skidrail contact region. Computer simulation of the possible measures that could be taken to alleviate the skidmark formation has indicated that reducing the height and width of the skidrail improved radiative heat transfer in the contact region. Coating highly reflective materials on the exterior surface of the skidrail to increase reflectivity from 0.3 to 0.8, could enhance heat transfer locally around the the skidrail by about 25% - 30% when the skidrail temperature is lower than the slab bottom temperature.
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NOMENCLATURE

A  area of an element (m^2)
a  thermal diffusivity (m^2/s)
\(a_{g,i}\)  gas absorptivity weighting coefficient
C  shielding factor of the skidrail.
CR  contact resistance (J/(m^2K))
\(C_p\)  specific heat (J/kg°C)
D  diameter of the skidpipe (m).
d  width of the skidrail (m).
E  radiative emission from a blackbody (W/m^2)
e  natural logarithm 2.7183
\(e\)  net radiative exchange coefficient
F  view factor
\(\overline{G_{i,j}}\)  total exchange coefficient between two gas zones (m²).
\(\overline{G_{i,j}^S}\)  total exchange coefficient between a gas zone i and a surface zone j (m²).
\(\overline{\Gamma_{i,j}}\)  directed exchange coefficient for a gas zone i to a a surface zone j (m²)
\(\Gamma_{i,j}^S\)  direct exchange coefficient for a gas zone i to a surface zone j (m²).
H  total incident heat flux to a surface(W/m²)
h  height of the skidrail (m)
h  convective heat transfer coefficient (W/m²K)
k  extinction coefficient of a gray gas (kPa.m)^{-1}
L  path length (m)
M  number of node divisions in the slab thickness direction.
ND  number of node divisions in the slab transverse direction.
N  number of zones
\(N\)  number of gray gas components
\( n \) \quad \text{unit normal vector}

\( P \) \quad \text{Pressure (Pa)}

\( p \) \quad \text{partial pressure (Pa)}

\( \text{Pr} \) \quad \text{Prandtl number } (\nu/a)

\( Q \) \quad \text{quantity of heat flow (W)}

\( q \) \quad \text{heat flux (W/m}^2\text{)}

\( R \) \quad \text{thermal resistance (Km}^2/\text{W)}

\( \text{Re} \) \quad \text{Reynolds number } (\text{UD}/\nu)

\( r_{g,i} \) \quad \text{gas emissivity weighting coefficient}

\( S_{i,j} \) \quad \text{directed exchange coefficient between surface zone } i \text{ and gas zone } j \text{ (m}^2\text{)}

\( \dot{S}_{i,j} \) \quad \text{direct exchange coefficient between surface } i \text{ and } j \text{ (m}^2\text{)}.

\( \ddot{S}_{i,j} \) \quad \text{total exchange coefficient between surface } i \text{ and } j \text{ (m}^2\text{)}.

\( s \) \quad \text{distance between two skidrails (m)}

\( T \) \quad \text{temperature (°C)}

\( t \) \quad \text{time (s)}

\( U \) \quad \text{over-all heat transfer coefficient (W/m}^2\text{K)}

\( V \) \quad \text{volume of a gas element (m}^3\text{)}

\( W \) \quad \text{total leaving radiative flux from a surface (W/m}^2\text{)}

\( x \) \quad \text{fraction of the coke oven gas in the fuel mixture for the reheating furnace}

\( x \) \quad \text{coordinate for the slab width direction (m)}

\( y \) \quad \text{coordinate for the slab thickness direction (m)}

\text{Subscript}

\( b \) \quad \text{bottom furnace chamber}

\( c \) \quad \text{centre of an element}
CG subscript pertaining to the experimental approach to determine $\phi$
en subscript pertaining to environment
F furnace chamber
g gas in the furnace chamber
H slab bottom surface
i subscript pertaining to an element $i$
j subscript pertaining to an element $j$
n subscript pertaining to gray gas component
r refractory side wall
s slab surface
t top half of furnace chamber
to subscript pertaining to the total amount
T the exterior surface of the skidrail
w subscript pertaining to the refractory wall
wa subscript pertaining to the cooling water

Greek Alphabet

$\alpha$ absorptivity.
$\beta$ relative dimension of slab/skidrail contact zone
$\delta$ thickness of the slab (m)
$\delta$ relative error.
$\nu$ kinematic viscosity (m$^2$/s)
$\epsilon$ emissivity.
$\rho$ reflectivity of a surface.
$\theta$ angle between the unit normal vector of an element and a line connecting it to another spatial element (degree)
\( \lambda \) thermal conductivity (W/mK)

\( \phi \) effective radiative exchange coefficients.

\( \tau \) transmissivity.
ACKNOWLEDGEMENT

I would like to express my sincere gratitude to Drs. J.K. Brimacombe and P.V. Barr for their guidance and understanding throughout the course of this study. I am also grateful to the National Science and Engineering Research Council and the People's Republic of China for providing financial support for this research.

It would be a mistake not to mention the friendship and encouragement I received from my friends, fellow students, the Faculty and technical staff. Their enthusiasm will become the memory I cherish in my future.

Last but not least, I would like to express my appreciation to Ms. Savithri for her patience in typing my thesis and Hemaguptha's excellent engineering drawing of most figures.
1. INTRODUCTION

Despite considerable efforts to direct roll continuously cast slabs, the reheating furnace is still essential to the contemporary iron and steel plant. Due primarily to the presence of defects from the continuous casting process, which must be removed before subsequent processing, no hot strip mill in the world has yet achieved one hundred percent direct hot rolling and thus eliminated the reheating process. At present, the detection and removal of these defects require that the slab be cooled, thus creating a major barrier to the achievement of direct rolling. Currently, two approaches toward eliminating this obstacle are being pursued. One is to develop mechanical automatic inspection and conditioning systems to enable detection and removal of defects while the slab is in the hot state\(^{(1)}\), while the second method relies upon mathematical models to define the mechanisms of defect formation and thus identify corrective measures. Both methods will take time to be fully successful and will involve large capital expenditure. Therefore, the reheating furnace is likely to remain an indispensable part of the steel hot working process for decades to come.

A typical three-zone push-type reheating furnace is shown in Fig. 1.1. Slabs are mechanically pushed through the reheating furnace and are elevated to a temperature level suitable for rolling. Skidpipes are constructed at the centreline of the furnace to support the slabs during the heating process. The furnace chamber is heated by burners located in various positions within the furnace, thus forming a high temperature field in the enclosure. Heat transfer within the reheating furnace chamber is very complicated, involving radiation, convection and conduction, with radiation being the dominant mode in the operating temperature range of the furnace\(^{(2)}\). Since all reheating furnaces burn hydrocarbon fuels, CO\(_2\) and H\(_2\)O in the products of combustion are the primary sources of gas radiation. The basic paths of heat transfer inside the reheating furnace are shown in Fig. 1.1 and the
FURNACE LONGITUDINAL DIRECTION

Primary zone

Heating zone

Soaking zone

ENTRANCE

SKIDRAIL

SLAB

SOLID HEARTH

TANGENTIAL BURNER

ROOF BURNERS

Note:

1. Gas to Slab Surface (Radiation + Convection)
2. Refractory Wall to Slab Surface (Radiation)
3. Gas to Refractory Wall (Radiation + Convection)
4. Refractory Wall to Environment (Convection + Radiation)
5. Slab to Refractory (Radiation)

Fig. 1.1 Basic heat transfer paths and processes in the reheating furnace.
heating environment for a slab being heated inside the furnace is shown in Fig. 1.2. Primary radiative transfer occurs from furnace gases to slab surfaces and to the bounding refractory wall while secondary exchanges occur between the slab and refractory wall surfaces as well as between areas on the refractory. Convective heat transfer occurs from furnace gas to the refractory wall and furnace gas to the slab surfaces. Heat transferred to the slab surface is then conducted to the interior of the slab. The water-cooled skids cause temperature depressions on the slab bottom surface due to conduction between the slab and the contacting skids and radiative shadowing by the skids resulting in visible skidmarks.

The operation of the reheating furnace is closely linked to that of the other processes in an integrated rolling mill. For instance the hot charging of slabs from the continuous casting machine or delays in the rolling schedule exert a very significant influence on the reheating process. Fig. 1.3 shows the reheating furnace in relationship to other processes in a typical hot strip mill where the slabs may originate either from a slabbing mill or a continuous caster. The latter is providing a progressively larger proportion of slabs for hot rolling.

The reheating process received little attention until the early 1970s, prior to which only a few papers were published on the mechanism of reheating slabs in a furnace chamber. The reheating furnace was regarded as an essential, but not critical, process in the steel hot working plant. As long as the furnace could provide sufficiently hot slabs so as to meet the rolling schedule, the furnace operation was considered acceptable. The operation of reheating furnaces was mainly based on furnace heat balances combined with the experience of operators. With progressively more stringent quality requirements and the increasing cost of energy, the reheating furnace was identified as a key process affecting product quality and fuel consumption in the hot strip mill. Since most mechanical properties of the slabs are strongly dependent upon temperature, rolling problems, such as gauge control in the roughing operation or thermal stress build-up due to non-uniform
Fig. 1.2 Heat transfer environment inside the reheating furnace
Fig. 1.3 The relationship between the reheating furnaces and the other relevant processes.
temperature distribution inside the slab are related to the thermal history of the slab, and hence to the reheating furnace. The reheating furnace is also the major consumer of non-electrical energy in the hot strip mill. Consequently, attention began to focus on the reheating operation and various attempts were made (1),(5–8) to better understand the mechanism of reheating.

Numerous mathematical models have been developed since the early 1970s, and some plants have adopted these models to monitor and control the operation of their reheating furnace (2),(9–18). However, these models tend to oversimplify the complex heat exchange processes inside the furnace chamber, particularly, gas radiation to the slab surface and heat transfer in the contact region between the skidrail and the slab. Gas and refractory wall radiation to the slab surface was often represented by ambiguous 'effective' heat-transfer coefficients and good agreement with the measured slab temperature was achieved only by adjusting effective coefficients. Most of these measurements referred to the average slab temperature and a detailed picture of heat transfer inside the furnace chamber other than conduction in the slab remained unclear. Many questions, such as the influence of the gas temperature distribution transverse to the direction of slab movement on the slab heating process could not be answered by such simplistic mathematical models. Uneven gas temperature distribution, induced by fuel firing conditions and burner arrangement, and the depression of slab temperature adjacent to the skids are thought to be detrimental to the quality of the rolled product. Therefore, the necessity to develop more sophisticated models becomes apparent.

The present study attempts to fill the gaps in previous works by developing a more sophisticated mathematical model. The model utilizes the geometry and operational parameters from the Stelco Lake Erie Works reheating furnace. In order to cope with the complexity of the radiative exchange in the furnace chamber filled with an absorptive–emissive gas, the zone method (19),(20) was employed to calculate the radiative exchanges among gas,
refractory wall surfaces and slabs. The results provide boundary conditions for a two-dimensional transient heat-conduction model to predict temperature profiles within the slab. The model significantly increases predictive capacities, facilitates and analysis of the process and provides guidance for design of the reheating furnace.

The inputs to the current model are the production rate and the gas temperature profile inside the chamber. In order to proceed beyond this point, gas flow patterns would have to be obtained, which is beyond the scope of this project. However the model could eventually be extended so as to require only the input of two operational parameters: fuel firing rate and the slab push rate, both of which are readily available.
2. LITERATURE REVIEW

The purpose of the reheating furnace is to supply slabs which are adequately heated for deformation in a rolling mill. Important operational objectives are the maximization of the tonnage through the furnace/mill system and efficiency of heating. Yet these objectives are difficult to achieve because of the many operating variables involved (i.e., fuel firing rate, slab push rate, furnace gas temperature etc.). These factors interact inside the furnace chamber, and their relationships cannot be known until the mechanism of heating is well understood. Therefore, the need to develop a heat-transfer model becomes obvious. As was mentioned in Chapter 1, numerous papers have been published on the mathematical modelling of reheating furnace mostly in the last fifteen years. Owing to progressively more stringent requirements for slab quality, a number of studies also have appeared on the skidmark phenomenon (21–26).

2.1 General Modelling of a Reheating Furnace

A basic objective in the development of any mathematical model of the reheating furnace is to better understand the temperature response of a slab being heated. Such a model can be adopted subsequently for various purposes such as on-line monitoring, computer control, energy saving, and so on. Despite diversified forms developed for different purposes, existing models can be classified into three categories according to the input parameters required.
2.1.1 Slab Surface Temperature as an Input Parameter

Avoiding the complicated heat exchanges occurring within the furnace enclosure, Hollander (2),(17) and Fitzgerald (16) proceeded directly to the slab conduction problem by assuming the relationship between surface temperature and time of a slab being heated in a furnace. Neglecting edge effects in the slab, the problem was reduced to that of semi-infinite conduction in one dimension which is governed by the following equation (2)

\[ \rho C_p \frac{\partial T}{\partial t} = \frac{\partial}{\partial y} \left( \lambda \frac{\partial T}{\partial y} \right) \]  \hspace{1cm} (2.1)

\[ \frac{\partial T}{\partial x} = 0 \]  \hspace{1cm} (2.2)

The initial condition depended upon the preceding slab process and was taken to be

\[ T(y,0) = T_0(y) \]  \hspace{1cm} (2.3)

The boundary conditions before the solid hearth were assumed to be:

\[ (T)_{y=0} = f_1(t), \quad (T)_{y=\delta} = f_2(t) \]  \hspace{1cm} (2.4)

while at the solid hearth (refer to Fig 1.1), the adiabatic condition was imposed

\[ \left( \frac{\partial T}{\partial y} \right)_{y=0} = 0 \]  \hspace{1cm} (2.5)

where \( y \) denotes the coordinate in the slab thickness direction and while \( f_1(t) \) and \( f_2(t) \) represent the temperature-time relationship for both top and bottom slab surfaces. The explicit finite-difference method was utilized (16) to obtain solutions for Eq. (2.1). The discretization steps, \( \Delta y \) and \( \Delta t \), were restricted by the stability criterion \( \Delta y^2 / \Delta t > 0.5 \) (27),(28).

Because \( f_1(t) \) and \( f_2(t) \) were difficult to express in analytical form and were entirely dependent upon the situation to be simulated, Hollander argued that they could be
determined iteratively depending on furnace length, throughput and the required total heat content of the slab at the exit. Once \( f_1(t) \) and \( f_2(t) \) had been defined, the temperature distribution through the slab thickness was obtained using the finite-difference approximation. The net heat flux to the slab surface at each axial point was calculated as follows

\[
q_{\text{surface}} = -\lambda \left( \frac{\partial T}{\partial y} \right)_{y=0} = \lambda \left( \frac{T_j - T_{j-1}}{\Delta y} \right)
\]

where \( \lambda \) is the thermal conductivity of the slab, \( T_j \) is the surface temperature and \( T_{j-1} \) is the temperature of the node adjacent to the slab surface. An empirical model was used to estimate the required fuel firing rate and the fuel distribution at the different positions inside the furnace. Typical model results are shown in Fig 2.1\(^{(2)}\).

These relatively simple, semi-empirical models have predicted the fundamental features of the slab temperature response, such as the centreline temperature. The method was also applied to on-line computer control of a reheating furnace\(^{(2,13,17)}\).

Since these models relied upon knowledge of the slab surface temperature response and ignored the heat transfer inside the furnace chamber, it was impossible to link the slab temperature field to the operational parameters, such as the gas temperature and its distribution. In addition, edge effects and the transverse\(\dagger\) slab temperature distribution, which are believed to be mainly responsible for many rolling problems \(^{(25,29)}\), were neglected.

### 2.1.2 Global Furnace Temperature as an Input Parameter

\(\dagger\) Transverse refers to the direction in the plane of the slab normal to the motion of the slabs in the furnace. Since the slabs move through the furnace sideways, the transverse direction in the furnace is actually the rolling direction after the slabs are discharged.
Fig. 2.1  Temperature response and heat flux distribution inside the reheating furnace (from Hollander)
This is the most commonly encountered reheating furnace model in the literature. It has been widely used for on-line monitoring of slab temperature (11),(12),(14),(30), computer control of the furnace (5),(31),(32) and optimization of furnace performance (5),(9).

Rather than requiring the surface temperature of the slab to be specified, heat transfer from furnace gases and refractory walls was utilized for the boundary conditions at the top and bottom slab surfaces. All the radiative heat transfer to the slab surface, regardless of whether from the furnace gases or refractory walls, was expressed in a single parameter , which is determined by furnace geometry and operational variables.

Collin(30) introduced a configuration factor φ and calculated the heat transfer to the slab surface as follows

\[ q_{st} = \phi_{st} \sigma \left( T_g^4 - T_s^4 \right) + h_t (T_{gt} - T_s) \]  \hspace{1cm} (2.7)

\[ q_{sb} = \phi_{sb} \sigma \left( T_g^4 - T_s^4 \right) + h_t (T_{gb} - T_s) \]  \hspace{1cm} (2.8)

where the subscripts s and g represent slab surface and gases respectively, while t and b represent the top and bottom slab surface. \( \phi_{st} \) and \( \phi_{sb} \) were assumed to vary longitudinally but to be constant at each axial position in the furnace. The methods used to estimate \( \phi \) were not reported by Collin(30), but reportedly the results given in Hottel and MacAdams(33) were applied. Other papers(15),(16),(34) also cited Hottel and McAdams for the gas-to-slab surface radiative exchange. According to them, the net heat flux from a gray gas at \( T_g \) to a gray heat sink at \( T_1 \) is

\[ q_{g \rightarrow 1} = \phi \sigma \left( T_g^4 - T_1^4 \right) \]  \hspace{1cm} (2.9)
where $\phi$ is

$$
\phi = \frac{A_T}{\epsilon_g + \frac{C}{\epsilon_1} - 1}
$$

and

$$
A_T = A_1 + A_R
$$

$$
C = \frac{A_1}{A_R}
$$

$A_1$ is the area of the sink (in a reheating furnace it refers to the slab surface); $A_R$ is the area of the refractory surface and $C$ is the ratio of the area of the heat sink to the area of refractory surface. $\epsilon_1$ and $\epsilon_g$ are the emissivities of the sink and the gas respectively. Although Eq. (2.10) provides a simple formulation, the net heat flux is based on very restrictive assumptions:

(i) The gas and flame in the furnace chamber can be assigned a single mean temperature $T_g$.

(ii) The gas is gray.

(iii) The surface of the heat sink is gray and can be assigned a single temperature.

(iv) External losses through the furnace walls are negligible and internal convection to refractory walls of area $A_R$ is negligible.

(v) The disposition of sink surface and refractory wall is such that from any point on the walls, the view-factor to sink surface is the same as from any other point. This is referred to as the 'speckled' wall condition.
Yoshisuke (15) and Ishida (5) defined the radiative heat transfer inside the furnace chamber with an equation similar to Eq. (2.8)

$$q = \phi_{CG} \sigma ( T_F^4 - T_s^4 )$$  \hspace{1cm} (2.13)

where $\phi_{CG}$ was determined experimentally using a heat-resistant data logger to record the temperature of a slab passing through the reheating furnace (Fig. 2.2). The heat transfer into the slab surface was calculated from the measured temperature gradient across the slab surface and $\phi_{CG}$ was determined from Eq. (2.13). The values of $\phi_{CG}$ thus calculated were only valid for the particular conditions and geometry represented in the experiment.

Another approach to characterize the radiative exchanges inside the furnace chamber was proposed by Fitzgerald (16). The heat flux to the slab surface was considered to originate from three sources: radiation from the refractory wall, radiation from the flames (or emitting combustion products), and convection from the furnace gas. However, convection has been shown to contribute less than 5% of the total heat at typical furnace temperatures(2). The net heat transfer rate to the slab was expressed by:

$$q = h_c A_s ( T_g - T_s ) + \sigma A_w F_{ws} ( T_w )^4$$

$$+ \alpha_s A_s \sigma ( \epsilon_g T_g^4 ) - \sigma A_s \epsilon_s ( T_s )^4$$  \hspace{1cm} (2.14)

Due to the length of the reheating furnace, the gas and surface temperatures will vary significantly with axial position. Therefore the furnace was considered to consist of several transverse surface and gas 'zones' of uniform temperature (not equal). The evaluation of $F_{ws}$ was based on purely geometrical view factors (in the clear medium) while $\alpha_s$ was the absorptivity of the slab surface. A similar treatment of the radiative exchange inside the furnace chamber was adopted by Ford(21).
Instrument Container
(Data collecting Capsule + Insulation device)

After removal and recovery, It is sent to the data recorder

Fig. 2.2 An experimental approach to evaluate $CG$ (from Yoshisuka)
Models in this class have provided predictions that agree well with the measured average temperature of the slab. Because of their simplicity and flexibility, these models have adequately fulfilled tasks such as temperature monitoring, or computer control of the furnace. In these applications, only the prediction of the rising trend of slab temperature is required.

However, the absorptive–emissive gases within the reheating furnace chamber exhibit quite different behaviour than that of the assumed gray gases. An emissive–absorptive real gas has discontinuous bands of emission and absorption while the behaviour of a gray gas is just the opposite. Moreover, the gas temperatures within the furnace vary in three dimensions while for the models it has been assumed that the gas temperature changes only in the longitudinal direction of the furnace.

The most difficult problem with these models has been the effective heat-transfer coefficient, which is a function of many furnace parameters, such as the geometry of the furnace, gas temperature distribution, slab dimensions and so on. A theoretical basis to link this key parameter to different operating conditions of the furnace has not been developed and, worse, it is not feasible to characterize all the situations empirically. Values of $\phi$ determined experimentally (5),(15) are valid only for the particular furnace and the operational conditions under which the experiments were conducted. Owing to complicating factors such as the skidrails in contact with the bottom surface of the slab or the influence of the furnace side wall, the transverse heat-flux distribution is likely to be quite uneven, particularly when side burners are installed (29). The transverse temperature distribution (down the length of the slab) has been shown to be critical to the final product quality (29). One-dimensional heat-transfer models (the variables depend only on the longitudinal dimension of the furnace) are incapable of addressing these problems.
Ambiguity also exists for the definition of the furnace temperature used to evaluate the radiative exchanges inside the furnace chamber. The value chosen might refer to the temperature balanced among gas, refractory wall and slab temperatures or it could be the gas temperature alone. The 'furnace temperatures' were normally obtained from the readings of thermocouples or radiative pyrometers mounted in different positions on the roof of the furnace; these temperature sensors were subjected to radiation from gases, refractory wall and slabs. The temperatures thus measured represent a balance among refractory wall, gas and slabs.

2.1.3 Application of the Zone Method to the Reheating Furnace

Hottel et al published a series of pioneering works (19),(20),(35),(36) on the development of the zone method for the computer modelling of the heat transfer inside an enclosure. The zone method allows the presence of an emissive-absorptive medium in the enclosure and a temperature gradient in the medium. However, owing to the relatively slow calculation speed of the computers at that time, the new method was not very popular until the late 1960s and the early 1970s. Hottel and Sarofim (35) again summarized the application of the zone method to the calculation of radiation from non-luminous flames while Pieri (37) extended the zone method to allow for concentration gradients in the medium. Johnson and Beer(38) developed a mathematical model for incorporating luminous flames into the zone method. These studies have demonstrated that the zone method is mathematically a reliable technique in spite of its complexity.

Patankar(39) critically summarized and compared the different methodologies for the modelling of furnaces. He reviewed different stages in the development of the mathematical modelling of furnaces and concluded that before the 1970s, a furnace was normally treated as a uniform-temperature enclosure (so-called zero-dimensional model); then one-dimensional analysis was employed for long furnaces of modest width. Recent developments in computers
and numerical methods have made it possible to make two- or three-dimensional analyses of furnaces, with steadily increasing realism and refinement.

Modelling of reheating furnaces has, to the author's knowledge, remained at the stage of one-dimensional analysis of radiative exchanges within the furnace chamber. Despite the demonstrated advantages of the zone method, remarkably few studies using this technique for the reheating furnace are available in the literature. The model developed by Veslocki and Smith (14) has been the most thorough application of the zone method to the reheating furnace. The furnace chamber was discretized into a series of nodes (they are equivalent to "zones") but the slab was considered to be only one dimensional in the thickness direction, thus posing a one-dimensional conduction problem. Fig. 2.3 shows the nodal system in a given furnace zone. Node temperatures were calculated from heat balances. For example, the top surface of each slab was assumed to receive heat by radiation from the roof surface nodes, from nodes in the circulating gas, and from nodes in the flame. In addition, the top slab surface receives heat by convection from the gas flowing directly over it. The heat received from the furnace chamber was transferred to the interior of the slab by conduction.

The main results from this model are:

(i). The longitudinal gas temperature profile has a significant effect on required furnace energy input, and on the temperature gradients in the slab.

(ii). In response to a delay, the firing strategy should be not only to reduce fuel firing rate during the delay, but to continue to use reduced firing rates following the delay.

(iii). A slab temperature estimator suitable for on-line computer control was developed from the model.
Fig. 2.3 A nodal system of a furnace zone (from Veslocki and Smith)
However, quantitative details of radiative exchanges between each component within the reheating furnace were not given, nor was the detail of gas and flame radiation. The application of the model would be limited by the failure to account for the furnace sidewalls and the skid system. Because the effects of refractory side wall and skidrails were not considered in the model, the heat flux variation in the furnace width direction was neglected. As pointed out before, the assumption of uniform transverse gas temperature is likely to be contrary to resultant reality and the resulting nonuniform heating.

Fitzgerald\textsuperscript{(16)} and Fontana\textsuperscript{(32)} also applied the zone method to predict radiative exchanges inside the reheating furnace. Since both studies adopted a one-dimensional model of the furnace chamber and neglected the influence of the refractory side wall and the skidrail, they could not amplify the results of Veslocki and Smith\textsuperscript{(14)}. In both studies, the mean beam length method was applied to characterize gas radiation to the slab surface. However, the mean beam length approach is only valid for gas of uniform temperature in an enclosure.

2.2 Hot Charging of Slabs

It is to be expected that the hot charging of slabs would decrease the furnace fuel consumption per ton of slab, because the energy content of each slab from the preceding process is delivered to the furnace. However, this simple and effective process of conserving energy has not been widely adopted yet owing to technical problems \textsuperscript{(1),(8),(40)}, such as slab defects as well as management problems in the overall mill, eg, the prompt transportation of hot slabs from the primary rolling process to the entrance of the reheating furnace.

In order to maintain slab quality, inspection and conditioning of the slabs are required normally before they are transferred for subsequent hot working. Forrest and
Wilson\textsuperscript{(41)} have summarized the surface defects usually found in continuously cast slabs (Fig 2.4). The presence of longitudinal midface, or longitudinal corner cracks, can be visibly detected, but entrapped slag and transverse broadface cracks are more difficult to find.

The conveyance of hot slabs to the entrance of a reheating furnace is another problem arising in hot charging. An insulated vehicle is being developed in Japan\textsuperscript{(8)} in order to retain the energy of the hot slabs during transit.

The potential benefits offered by hot charging can only be realized if proper heating strategies are available. However, heating strategies for hot charging have yet to be established owing to the newness of the process.

2.3 Previous Studies on the Formation of Skidmarks

As previously mentioned, conduction around the slab/skidpipe contact surface and the shadowing effect of skidpipes beneath the slab bottom surface cause distortion of the local heat flux, thus forming regions of temperature depression known as skidmarks. The generation of skidmarks in a reheating furnace produces unfavourable effects in the subsequent rolling process. This region gives rise to different deformation properties in the final rolling process and therefore makes accurate gauge control difficult. However, since skidpipes are an indispensable supporting structure for the current push-type reheating furnaces, the generation of skidmarks is irrevocably associated with the process.

Ford \textit{et al} \textsuperscript{(21)} suggest that the mechanism of skidmark formation involves:

\begin{enumerate}
\item Conduction across the slab/skid interface.
\item Radiative shadowing of the slab by the supporting skid structure.
\end{enumerate}
1. Longitudinal Midface Crack
2. Star Cracks
3. Longitudinal Corner Crack
4. Transverse Crack (4a) Corner Crack (4b)
5. Slag Patches Surface Defects

Fig. 2.4 Common defects in the continuously cast slabs.
It was concluded that the most significant factor in the formation of skidmarks was the radiation shadowing. In the model, carbon dioxide and water vapour were approximated by a gray gas in the calculation of radiative flux to the slab and skidpipe in the reheat zone. In the soaking zone of the furnace (Fig 1.1), blackbody radiation was assumed for the slab surface. An explicit finite-difference method was employed to solve the temperature distribution in the slab and skidpipe. Although the 'gray gas' and the 'blackbody' assumptions were not realistic, some important and useful results have been generated by the model.

(i) Thickers slabs require greater soaking time to even out the temperature difference in the slab and reduce the skidmarks.

(ii) The inlet temperature of cooling-water in the range considered (25-100 °C) had little effect on skidmarks, thus indicating that skidmark formation was strongly influenced by radiation shielding of the skidpipe and very little by the direct conductive heat loss to the cooling water.

Howell et al (22),(23) carried out an investigation of the thermal contact conductance between the skidpipe and the slab and concluded that its value varies from 0.1 to 5.0 kW/m²·°C⁻¹ according to the contact pressure and temperature in the ranges 0.1-1.0 MPa and 150-1000 °C respectively. Weaver and Barraclough (25) combined the computer modelling of skidrail geometry with experiments to study the configuration of skidrails and concluded that a teardrop-shaped design would be the best for minimizing skidmarks. Roth et al (24) employed a one-dimensional model to simulate skidmark temperatures. They also showed radiative shadowing to be predominant in the formation of skidmarks, with heat conduction through the wear bar and rail having only a minor influence. Temperature predictions in the region of the skids are shown in Fig. 2.5.
Fig. 2.5  Temperature distribution in the skidmark region (from Roth et al)
In general, past studies have concentrated on the effect of skidpipe design on the generation of skidmarks and have achieved useful results (eg, the identification of optimal geometrical skidrail shapes). However, the formation of skidmarks will later be shown to be closely related to the operation of the furnace as well as to skidrail design. Moreover, the one-dimensional description of radiative exchanges inside the furnace chamber cannot accurately characterize heat transfer around the skidmark regions, which includes the effects of refractory side wall and nonuniform transverse gas temperatures. The radiative exchanges among the exterior surface of the skidrail, the slab bottom surface and the furnace chamber must also be studied in detail in order to link the temperature of the skidmark to the furnace operating parameters.
3. OBJECTIVES OF THE PRESENT WORK

Although the previous mathematical heat-transfer models of the reheating furnace have fulfilled some tasks such as computer control and prediction of trends in slab temperature, many inadequacies have been identified in the previous chapter. More stringent requirements for slab heating quality and further improvements in furnace operation cannot be achieved by a one-dimensional model of the furnace chamber. A more detailed, quantitative description of the heat-transfer processes occurring inside the furnace is necessary.

The objective of the present study is to formulate a sophisticated mathematical heat-transfer model of the reheating furnace, capable of providing detailed predictions of slab and refractory wall temperatures. Since the influence of the furnace refractory side wall and the nonuniform transverse gas temperature distribution in the process are to be taken into account, a three-dimensional zone-type model of the furnace chamber has been adopted to calculate the radiative exchanges. The combustion gases inside the chamber, which are major participants in the radiative exchanges and have often been inaccurately simulated as gray gases by previous models, have been treated as 'real' gases and characterized by discontinuous absorption bands. Skidrails were also incorporated into the model geometry in order to identify their effects on slab heating. The furnace chamber heat-transfer model was combined with a two-dimensional unsteady conduction model in order to predict the temperature of slabs being heated in the furnace.

Gas temperature, slab push rate, slab initial temperature distribution and steel grade are the required inputs to the model. To proceed beyond this point would require complete knowledge of the gas flow pattern inside the furnace chamber, due to the coupling of the heat-transfer problem to the flow field. However, the gas flow field is currently unavailable.
and would be exceedingly difficult to obtain; Therefore the gas temperature profiles had to be assumed as input to the model. Any extension of the present study require knowledge of the gas flow pattern inside the chamber so that the assumption of the gas temperature profile can be removed.

The geometry and operational parameters for the model calculations are taken from the reheating furnace at the Lake Erie Works (LEW) of Stelco. Although the model is applied to simulate the thermal behaviour of the Stelco furnace, the modelling approach is general and the results generated from the model can be applied to other slab reheating furnaces, since the Stelco furnace is typical of a three-zone type reheating furnace.

The model was applied to investigate the following:

(i) The effects of varying furnace operating parameters, such as both longitudinal and transverse gas temperature distributions, steel slab grade and slab push rate etc, on slab exit temperatures and skidmark formation.

(ii) The mechanism of skidmark formation and identification of measures to eliminate or reduce it.

(iii) The effects of various heating strategies for hot-charged slabs so as to minimize fuel consumption and improve slab temperature uniformity.
4. REAL GAS TREATMENT FOR THE ZONE METHOD

Gas radiation plays a dominant role in the thermal behaviour (20),(36),(43) of furnaces. Since most reheating furnaces are fired with hydrocarbon fuels, carbon dioxide and water vapour in the gaseous combustion products constitute the major source of radiation within the furnace chamber.

In the Stelco LEW reheating furnace, which is fired by natural gas and a mixture of natural gas and coke-oven gas under pre-mix conditions, the contribution of particulates to the radiative emission is unlikely to be significant and therefore has been ignored in the present study. Since the furnace gases are mixed by high momentum jets from combustion burners, a well-stirred chamber assumption has been adopted. In order to characterize the emissive and absorptive behaviour of the combustion gas in the Stelco reheating furnace, a mixture of three gray gases was assumed to represent the radiative behaviour of the real gas, which exhibits discontinuous emission and absorption over specific wavelength bands. The approach described below has been applied in a three-dimensional heat-transfer model of the reheating furnace chamber.

4.1 Method of Treatment of Real Gas Emissivity and Absorptivity

Absorption and emission of radiation by a real gas occurs only over specific bands of the wavelength spectrum. Thus the calculation of gas radiative exchange is rendered extremely difficult. In order to overcome this complication real gas emission/absorption can be simulated by the weighted summation of a sufficient number of gray gases, so that the mathematical formulation which characterizes a gray gas can be applied to the real gas. As described by Hottel and Sarofim\(^{(20)}\), the total gas emissivity then can be approximated by
the following equation:

\[ e_g = \sum_{i=0}^{N} r_{g,i} (1 - \exp(-k_{i}pL)) \quad (4.1) \]

with the restriction that

\[ \sum_{i=0}^{N} r_{g,i} = 1.0 \quad r_{g,i} > 0 \quad (4.2) \]

where \( k_{i} \) are the extinction coefficients for the \( N \) gray gas components and \( r_{g,i} \) are the weighting coefficients. Although in theory \( k_{i} \) and \( r_{g,i} \) are both functions of gas temperature, in practice it has been found that \( k_{i} \) are weak functions of gas temperature so that all temperature dependence can be carried by the weighting coefficients\(^{(20)}\). After selecting a characteristic gas temperature, which lies in the range of the furnace gas temperature, a set of values for \( r_{g,i} \) and \( k_{i} \) can be obtained by fitting the real total emissivity data to the form of Eq. (4.1). The change of real gas emissivity with gas temperature is reflected in \( r_{g,i} \).

A similar treatment can be applied to the absorptivity of a real gas, except that the absorptivity depends on both gas temperature and emitting surface temperature \( T_s \). The relationship between absorptivity and emissivity, according to Hottel and Sarofim\(^{(20)}\), is given by the following equation:

\[ a_g(T_s, T_g, pL) = \left( \frac{T_g}{T_s} \right)^{0.65} \left[ e_g(T_s, pL, \frac{T_s}{T_g}) \right] \quad (4.3) \]

and \( a_g \) is also expressible as follows

\[ a_g = \sum_{i=0}^{N} a_{g,i} (T_g, T_s)[1 - \exp(-k_{i}pL)] \quad (4.4) \]

The determination of \( a_{g,i} \) is similar to that of \( r_{g,i} \); except \( a_{g,i} \) is a function not only of absorbing gas temperature, \( T_g \), but also of the the emitting surface temperature, \( T_s \).
4.2 Application of the Method to the Real Gas in a Reheating Furnace Chamber

At the Lake Erie Works of Stelco, the fuel fired in the reheating furnace consist of the following:

- **Soak zone**
  - Natural gas

- **Heating zone**
  - Natural gas or mixture of natural gas and coke oven gas

- **Primary zone**
  - Natural gas or mixture of natural gas and coke oven gas

The composition of typical coke oven gas at Stelco is shown in Table 4.1 (private communication with Stelco). Because \( \text{CO}_2 \) and \( \text{H}_2\text{O} \) are the major components affecting the emissivity and absorptivity of furnace gas, their respective partial pressures, which, in turn, depend upon the molar composition of the combustion products, must be calculated from stoichiometry. For this purpose, complete combustion has been assumed inside the furnace chamber since the excess air is 10%.

For the stoichiometric calculations, the fraction of coke oven gas was taken to be \( x \) and the natural gas to be \((1-x)\). Assuming complete combustion and excess air of 10%, the stoichiometric equation for the natural gas (\( \text{CH}_4 \)) combustion is

\[
(1-x)(\text{CH}_4 + 2.20_2 + 8.272\text{N}_2) \rightarrow (1-x)(\text{CO}_2 + 2\text{H}_2\text{O} + 8.272\text{N}_2 + 0.2\text{O}_2) \quad (4.5)
\]
Table 4.1 Typical Composition of the Coke Oven Gas

<table>
<thead>
<tr>
<th>Gas</th>
<th>Molar fraction</th>
</tr>
</thead>
<tbody>
<tr>
<td>H\textsubscript{2}</td>
<td>0.513</td>
</tr>
<tr>
<td>CH\textsubscript{4}</td>
<td>0.234</td>
</tr>
<tr>
<td>H\textsubscript{2}O</td>
<td>0.055</td>
</tr>
<tr>
<td>N\textsubscript{2}</td>
<td>0.100</td>
</tr>
<tr>
<td>CO</td>
<td>0.052</td>
</tr>
<tr>
<td>CO\textsubscript{2}</td>
<td>0.018</td>
</tr>
<tr>
<td>C\textsubscript{2}H\textsubscript{4}</td>
<td>0.019</td>
</tr>
<tr>
<td>O\textsubscript{2}</td>
<td>0.007</td>
</tr>
<tr>
<td>C\textsubscript{2}H\textsubscript{6}</td>
<td>0.002</td>
</tr>
</tbody>
</table>

Similar stoichiometric equations apply to the coke-oven gas combustion:

\[ \text{H}_2 + 0.55\text{O}_2 + 2.068\text{N}_2 \rightarrow \text{H}_2\text{O} + 2.068\text{N}_2 + 0.05\text{O}_2 \] (4.6)

\[ \text{CH}_4 + 2.2\text{O}_2 + 8.272\text{N}_2 \rightarrow \text{CO}_2 + 2\text{H}_2\text{O} + 8.272\text{N}_2 + 0.2\text{O}_2 \] (4.7)

\[ \text{CO} + 0.55\text{O}_2 + 1.88\text{N}_2 \rightarrow \text{CO}_2 + 1.88\text{N}_2 + 0.05\text{O}_2 \] (4.8)

\[ \text{C}_2\text{H}_4 + 3.3\text{O}_2 + 12.4\text{N}_2 \rightarrow 2\text{CO}_2 + 2\text{H}_2\text{O} + 12.4\text{N}_2 + 0.3\text{O}_2 \] (4.9)

\[ \text{C}_2\text{H}_6 + 3.85\text{O}_2 + 14.5\text{N}_2 \rightarrow 2\text{CO}_2 + 3\text{H}_2\text{O} + 14.5\text{N}_2 + 0.35\text{O}_2 \] (4.10)

The total moles in the final combustion product are:

\[ 0.055x + 0.100x + 0.018x + 0.007x + 1.025x + 0.328x + 3.359x + 0.08145x + (1-x) + 2(1-x) + 8.272(1-x) + 0.2(1-x) = 11.472 - 6.5x \] (4.11)
The moles of CO$_2$ are

$$(1-x) + 0.0018x + 0.328x = 1 - 0.654x$$ (4.12)

The moles number of H$_2$O are

$$2(1-x) + 0.055x + 1.025x = 2 - 0.92x$$ (4.13)

Therefore, the final molar fraction of the two emissive-absorptive gases in the combustion products can be expressed as:

$$\text{CO}_2 = \frac{(1.0 - 0.654x)}{11.472 - 6.5x}$$ (4.14)

$$\text{H}_2\text{O} = \frac{(2.0 - 0.92x)}{11.472 - 6.4986x}$$ (4.15)

Both the volume fraction of CO$_2$ and H$_2$O are functions of the fraction x. Table 4.2 shows the change of molar composition of CO$_2$ and H$_2$O with the mixing parameter x. Because the mixing parameter in the reheating furnace seldom goes higher than 0.4, %H$_2$O/%CO$_2$ was found basically to be 2.0. This is the ratio the present calculations are based upon.

The partial pressure of CO$_2$ and H$_2$O can be determined if the total pressure distribution inside the furnace chamber is given. According to previous measurements (16), the variation in total pressure inside the pusher-type reheating furnace chamber is less than 1% and the total pressure of the chamber can be assumed to be atmospheric. Therefore, the ratio of partial pressure of CO$_2$ to H$_2$O in the final combustion product can be taken as

$$\frac{P_{\text{H}_2\text{O}}}{P_{\text{CO}_2}} = 2.0$$ (4.16)
Table 4.2 The Change of Composition of CO\textsubscript{2} and H\textsubscript{2}O with the Mixing Ratio.

<table>
<thead>
<tr>
<th>Coke oven gas proportion</th>
<th>CO\textsubscript{2}%</th>
<th>H\textsubscript{2}O%</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0 (pure natural gas)</td>
<td>8.72</td>
<td>17.44</td>
</tr>
<tr>
<td>0.1</td>
<td>8.64</td>
<td>17.63</td>
</tr>
<tr>
<td>0.3</td>
<td>8.44</td>
<td>17.63</td>
</tr>
<tr>
<td>0.4</td>
<td>8.32</td>
<td>18.40</td>
</tr>
<tr>
<td>0.6</td>
<td>8.02</td>
<td>19.12</td>
</tr>
<tr>
<td>0.9</td>
<td>7.32</td>
<td>20.84</td>
</tr>
<tr>
<td>1.0 (pure coke oven gas)</td>
<td>6.96</td>
<td>21.72</td>
</tr>
</tbody>
</table>

throughout the furnace chamber.

Based on the above calculated partial pressure and the known gas temperature range of the furnace, the emissivities of both carbon dioxide and water vapour can be obtained from Fig. 6.9 and Fig. 6.11 in Hottel & Sarofim\textsuperscript{(20)}. However, because the atmosphere of the reheating furnace is a mixture of not only CO\textsubscript{2} and H\textsubscript{2}O but also radiatively inert gas components, the latter interfere with the emissivity of this real gas. In addition, some of the CO\textsubscript{2} and H\textsubscript{2}O absorption bands overlap. For these reasons, the real emissivity of the combustion products in a furnace is actually smaller than the direct sum of emissivities of CO\textsubscript{2} and H\textsubscript{2}O. A correction factor has been introduced as suggested by Hottel and Sarofim\textsuperscript{(20)}:

\[ e_g = e_{CO_2} + e_{H_2O} - \Delta e \]  \hspace{1cm} (4.17)

Three gray gases were assumed to simulate the emissivity of the real gas in the reheating furnace as mentioned previously. One of the three gases is a clear gas characterized by an extinction coefficient \( k_0 = 0.0 \). A computer program was developed to fit
Eqs. (4.1) and (4.4) to within 5% using a typical gas temperature $T_c$ of $1300^\circ$C for the calculation of the extinction coefficient $k_1$. The output of the program was the desired extinction coefficient $k_1$ and the temperature dependent $r_{g,i}$ and $a_{g,i}$. The flowchart of the computer program is shown in Appendix I. The program was written for general purposes and could be used to fit any number of gray gases.

The extinction coefficients of the three gray gases calculated from the computer program are:

$$k_0 = 0, \quad k_1 = 1.499 \times 10^{-2} \text{(kPa.m)}^{-1}, \quad k_2 = 2.168 \times 10^{-1} \text{(kPa.m)}^{-1}$$

An exponential form, $r_{g,i} = A_i \exp(-B_i T)$, was utilized to correlate the variation of $r_{g,i}$ with gas temperature, as shown in Fig. 4.1. The values of $A_i$ and $B_i$ are listed in Table 4.3. As indicated in Eqs. (4.3) and (4.4), the weighting coefficients for the absorptivity $a_{g,i}$ are not only a function of the gas temperature $T_g$, but also of the surface temperature $T_s$. For the sake of convenience, in using the zone method, a linear equation was proposed to correlate the change of the weighting coefficient $a_{g,i}$ with the surface temperature at a specified gas temperature as follows:

$$a_{g,i} = C_i T_s + D_i$$  \hspace{1cm} (4.18)

where $C_i$ and $D_i$ are the slope and intercept respectively, and $T_s$ is the surface temperature. The slopes and intercepts for these linear equations are expressed in Table 4.4.

Fig 4.2 shows a comparison between the linear interpolation and the real values of absorptivity at a gas temperature of $1000^\circ$C. Excellent agreement was found.

From the results shown in Table 4.3 and Table 4.4, the following relationship was
Fig. 4.1 $r_{g,i}$ change with temperature
Fig. 4.2 Linear interpolation of $a_{g,i}$
Table 4.3 Coefficients of $A_i$ and $B_i$ for $r_{g,i}$

<table>
<thead>
<tr>
<th>i</th>
<th>$A_i$</th>
<th>$B_i$</th>
<th>Correlation coefficient</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.3601</td>
<td>4.06x10^-4</td>
<td>0.995</td>
</tr>
<tr>
<td>1</td>
<td>0.4338</td>
<td>-2.79x10^-4</td>
<td>0.919</td>
</tr>
<tr>
<td>2</td>
<td>0.5001</td>
<td>-1.349x10^-4</td>
<td>0.995</td>
</tr>
</tbody>
</table>

Table 4.4 Linear Correlation for the Weighting Coefficient $a_{g,i}$

<table>
<thead>
<tr>
<th>Gas Temp.</th>
<th>$a_{g,0}$</th>
<th>$a_{g,1}$</th>
<th>$a_{g,2}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>°C</td>
<td>$C_0$</td>
<td>$D_0$</td>
<td>$C_1$</td>
</tr>
<tr>
<td>900</td>
<td>0.00038</td>
<td>0.18126</td>
<td>-0.00026</td>
</tr>
<tr>
<td>1000</td>
<td>0.00039</td>
<td>0.14458</td>
<td>-0.00028</td>
</tr>
<tr>
<td>1100</td>
<td>0.00041</td>
<td>0.11224</td>
<td>-0.00029</td>
</tr>
<tr>
<td>1200</td>
<td>0.00042</td>
<td>0.08337</td>
<td>-0.00031</td>
</tr>
<tr>
<td>1300</td>
<td>0.00043</td>
<td>0.05727</td>
<td>-0.00032</td>
</tr>
<tr>
<td>1400</td>
<td>0.00043</td>
<td>0.03339</td>
<td>-0.00033</td>
</tr>
</tbody>
</table>

found

$$r_{g,i}(T_g) = a_{g,i}(T_s, T_g) \text{ when } T_g = T_s$$

(4.19)

which indicates that Kirchoff's Law was satisfied by the above treatment of the real gas model.

When the emissivity simulated by the three-grey-gas model is compared with the real gas emissivity (Fig. 4.3), less than 5% error is observed. The results generated by the current gas model were incorporated into the heat-transfer model to calculate radiative exchanges between the furnace gas and its bounding surfaces within the furnace chamber.
Fig. 4.1 Change of $r_{g,i}$ with temperature
5. METHODOLOGY

Heat transfer inside a reheating furnace chamber can be divided into two distinct parts, the first being the radiative and convective exchanges occurring among the furnace gases, the bounding refractory wall and the slab surface (Fig. 1.2), and the second being heat conduction inside the slab and through the refractory wall. Since radiative and convective heat transfer provide the boundary conditions for slab and refractory wall conduction, the two heat-transfer processes are interrelated.

The radiative exchange is the dominant mode of heat transfer inside the furnace chamber because the temperature level is well above 900°C. The convective heat transfer accounts for less than 5% of the total. In order to calculate radiative exchanges in the furnace chamber, a zone-type heat-transfer model was developed. In its ultimate form, the zone method requires knowledge of the combustion and gas flow patterns inside the furnace chamber, thus presenting a formidable obstacle to its application. Due to the irregular geometry of the furnace and the complicated burner orientation, the gas flow inside a reheating furnace is likely to be complex and data is unavailable. Lacking this data, energy balances on gas zones are impossible and for this reason gas temperature profiles have been assumed for the current study.

Four points distinguish this investigation from previous studies of reheating furnace heat transfer:

(i) Gas temperature gradients, in both longitudinal and transverse directions of the furnace chamber, can be accommodated.

(ii) The influence of refractory side walls was taken into account.
The emissive/absorptive characteristics of the furnace gas were closely approximated using a clear-plus-two-gray-gas model rather than a single gray gas.

The energy interchanges near the slab/skidrail contact region were calculated in detail using a separate conduction heat-transfer model which was integrated into the furnace chamber heat-transfer model.

5.1 Radiative Heat Transfer Inside the Furnace Chamber

An introduction to the zone method is provided below. For a detailed description, one is directed to reference (20). The initial step in the application of the method to a furnace chamber is to subdivide the furnace enclosure and emitting-absorbing gas mixture into a sufficient number of surface zones and volume zones, respectively, so that each may be assumed to be isothermal with uniform radiative properties. The next step is to evaluate the radiative exchanges between each zone pair for each gray gas component. The radiative flux between two zones in an enclosure containing a clear gas plus two gray gases is the summation of the independent contributions from each gray gas i (with absorption coefficients $k_i$), weighted in proportion to the absorptivity coefficients $a_{g,i}$ evaluated at the temperature of the emitting zone. To simplify the nomenclature it is to be understood that the following formulation is based on a single gray emission band ($k=\text{constant}$).

5.1.1 Direct Exchange Coefficients

For the pair of surface zones $A_i$ and $A_j$ shown in Fig. 5.1 the radiative energy
Fig. 5.1  Geometry for radiative exchange between two surfaces
emitted by surface element $dA_i$ which impinges upon $dA_j$ directly is

$$Q_{A_i \rightarrow A_j} = \int_{A_i} \int_{A_j} E_i \cos \theta_i \cos \theta_j \frac{\tau_i \rightarrow_j dA_i dA_j}{\pi r_{ij}}$$

$$= \overline{s_{i,j}} E_i$$  \hspace{1cm} (5.1)

where $\overline{s_{i,j}}$ is the direct exchange coefficient between surface $i$ and surface $j$. For a gray gas

$$\overline{s_{i,j}} = \overline{s_{j,i}} \text{ (reciprocity)} \hspace{1cm} (5.2)$$

From Eq. 4.1 the transmissivity $\tau_i \rightarrow_j$ is given by

$$\tau_i \rightarrow_j = \exp(- \int kpdl)$$  \hspace{1cm} (5.3)

The net direct radiative exchange between $A_i$ and $A_j$ is therefore

$$Q_{A_i \leftrightarrow A_j} = \overline{s_{i,j}} \left( E_i - E_j \right)$$  \hspace{1cm} (5.4)

A similar result can be obtained for gas–surface direct exchange,

$$Q_{G_i \leftrightarrow A_j} = \overline{g_{i,j}} \left( E_{gi} - E_j \right)$$  \hspace{1cm} (5.5)

where the direct exchange coefficient between a gas zone $i$ and a surface zone $j$ is defined as

$$\overline{g_{i,j}} = \frac{1}{\pi} \int_{v_i} \int_{s_j} \frac{4k^2 \cos \theta_i}{r_{ij}^3} e^{- \int kpdl} dv_idA_j$$  \hspace{1cm} (5.6)

The direct exchange coefficients will be functions of both chamber geometry and partial pressure distribution of the gas.
Energy conservation requires that for surface zones

\[
\sum_{j=1}^{N_s} \overline{s_i s_j} + \sum_{j=1}^{N_g} \overline{s_i g_j} = A_i \tag{5.7}
\]

and for gas zones

\[
\sum_{j=1}^{N_s} \overline{g_i s_j} + \sum_{j=1}^{N_g} \overline{g_i g_j} = 4k\Delta V_i \tag{5.8}
\]

where \(N_s\) and \(N_g\) denote the total number of surface and gas zones respectively.

In evaluating the direct exchange coefficients from Eq. (5.1) and Eq. (5.6), the values of the multiple integrations are extremely difficult to obtain and are impossible to express in analytical format (except for a few very simple geometries). Therefore an approximate approach has been taken involving the uniform subdivision method \(^{(52)}\) in which the zone pairs were further subdivided into subzones and the summation over these subzones was performed according to the definition of multiple integrations.

### 5.1.2 Total Exchange Coefficients

The direct exchange coefficients consider only the radiation originating from a zone \(i\) which directly impinges upon another zone \(j\). Total exchange coefficients, which account for both the direct radiation from \(i\) to \(j\) and the radiation originating from \(i\) and reaching \(j\) through single or multiple reflections in the enclosure, are required.

As shown in Fig. 5.2, a radiative balance on a surface zone \(A_i\) requires that:

\[
A_i W_i = A_i (\epsilon_i E_i + \rho_i H_i) \tag{5.9a}
\]

\[
A_i W_i = A_i \epsilon_i E_i + \rho_i \left( \sum_{j=1}^{N_g} \overline{g_i s_j} E_{b_j} + \sum_{j=1}^{N_s} \overline{s_i s_j} W_j \right) \tag{5.9b}
\]
Fig. 5.2 Radiative energy balance for a surface zone.
where \( W_i \) is the total leaving radiative flux from a surface, \( H_i \) is the total incident radiative flux and \( \rho_i \) and \( \varepsilon_i \) are respectively the reflectivity and emissivity of the surface \( i \). When applied to each surface zone, Eq. (5.9b) provides a set of \( N_s \) simultaneous equations which may be rearranged into the more useful form

\[
\sum_{j=1}^{N} \left( \frac{A_i}{\rho_i} - \delta_{ij} \frac{A_i}{\rho_i} \right) W_i = \frac{A_i}{\rho_i} \varepsilon_i E_i \sum_{j=1}^{N} g_{ij} E_{bj} \]

(5.10)

where \( \delta_{ij} \) is defined by

\[
\delta_{ij} = \begin{cases} 
1 & i=j \\
0 & i \neq j
\end{cases}
\]

If all zone temperatures are known, the set of simultaneous linear equations (5.10) can be solved for the \( N_s \) leaving flux densities, from which the net zone radiative flux follows readily.

The net heat flux between \( j \) and \( i \) can be evaluated from

\[
Q_{j \rightarrow i} = Q_{j \rightarrow i} - Q_{i \rightarrow j} = S_{i j} (E_j - E_i)
\]

(5.11)

where \( S_{i j} \) is the total exchange coefficient term having the dimensions of area. The total exchange coefficients can be derived from Eq. (5.10) resulting in

\[
S_{i j} = \frac{A_i}{\rho_i} \left( W_i - \delta_{ij} \varepsilon_j \right)
\]

(5.12)

\[
S_{i j} = \frac{A_i}{\rho_i} \left( W_i - \delta_{ij} \varepsilon_j \right)
\]

(5.13)

\[
G_{i j} = \frac{A_i}{\rho_i} \left( W_i - \delta_{ij} \varepsilon_j \right)
\]

(5.14)

where \( W_i \) is the leaving flux density at the surface \( j \) per unit emissive power of zone \( i \).
Again, since a gray component is being considered, reciprocity applies and

\[ \mathbf{S}_{ij} = \mathbf{S}_{ji}, \quad \mathbf{G}_{ij} = \mathbf{G}_{ji}, \quad \mathbf{G}_{j} = \mathbf{G}_{j} \]  

(5.15)

Since the radiative energy emitting from a zone must equal the summation of its net radiative exchanges with all the other zones in the enclosure, the total exchange coefficient for a surface zone must satisfy the condition

\[ \sum_{j=1}^{N_s} \mathbf{S}_{ij} + \sum_{j=1}^{N_g} \mathbf{S}_{ij} \mathbf{G}_{j} = A_i \varepsilon_i \]  

(5.16)

\((i=1, 2, \ldots, N_s)\)

while, for a gas zone

\[ \sum_{j=1}^{N_s} \mathbf{S}_{j} \mathbf{G}_{i} + \sum_{j=1}^{N_g} \mathbf{G}_{j} \mathbf{G}_{j} = 4k \Delta V_i \varepsilon_g \]  

(5.17)

\((i=1, 2, \ldots, N_g)\)

5.1.3 Energy Balance For a Surface Zone

Energy conservation for a surface zone \(i\) in an enclosure filled with a gray gas requires that the summation of radiation received from gas zones and surface zones plus convection from the adjacent gas must equal its rate of radiative emission plus the rate of
conduction away from the surface. This result can be expressed as

$$\sum_j \left( \begin{array}{c} \frac{Q_{S_j}}{J_i} E_{g_j} \\ \frac{S_{S_j}}{J_i} E_{l_j} \end{array} \right) + \sum_j \left( \begin{array}{c} \frac{S_{S_i}}{J_i} E_{l_i} \end{array} \right) + h_i A_i (T_g - T_i)$$

$$= Q_{\text{net},i} + e_i A_i E_i$$  \hspace{1cm} (5.18)

where $Q_{\text{net},i}$ includes wall heat loss and heat enthalpy changes with respect to time (unsteady term). For the steady-state condition, the unsteady term can be dropped. Eq. (5.18) yields as many simultaneous non-linear equations as there are unknown zone temperatures. The flux distribution follows directly from the solution of Eq. (5.18) for the unknown zone temperatures.

However, all real emitting/absorbing gases exhibit a variation in absorption coefficient with wavelength. To account for this, the directed exchange coefficients $\bar{S}_{i_{j}}$ and $\bar{S}_{j_{i}}$ (with the direction of arrow in the direction of the radiation flux) are introduced and the net energy exchange between two surface zones using the clear-plus-two-gray-gas emissivity model is

$$Q_{A_{i} \leftrightarrow A_{j}} = E_{s,i} S_{i_{j}} - S_{j_{i}} E_{s,j}$$  \hspace{1cm} (5.19)

with

$$\bar{S}_{i_{j}} = \sum_n a_{s,n}(T_i)(\bar{S}_{i_{j}})_{n}$$  \hspace{1cm} (5.20)

$$\bar{S}_{j_{i}} = \sum_n a_{s,n}(T_j)(\bar{S}_{j_{i}})_{n}$$
Similarly, the net heat exchange between a gas zone and a surface zone is

\[
Q_{G_i} \leftrightarrow S_j = \overrightarrow{G_{i,j}} E_{g,i} - \overrightarrow{S_{j,i}} E_{s,j}
\]  

(5.21)

with

\[
\overrightarrow{G_{i,j}} = \sum_n r_{g,n} (T_g)(\overrightarrow{S_{j,i}})_n
\]

\[
\overrightarrow{S_{j,i}} = \sum_n a_{s,n} (T_j)(\overrightarrow{G_{i,j}})_n
\]  

(5.22)

The subscript \( n \) denotes the three components of gray gases being used to simulate the radiative behaviour of the real reheating furnace gas.

The energy balance on a surface zone \( A_i \) of unknown temperature is given by

\[
\sum_j \overrightarrow{S_{j,i}} E_{s,j} + \sum_j (\overrightarrow{G_{j,i}} E_{g,j}) + h_i A_i (T_{gk} - T_{s,i}) = Q_{net,i} + \epsilon_i A_i E_{s,i}
\]  

(5.23)

where \( T_{gk} \) is the temperature of the gas contiguous to \( A_i \), and \( Q_{net,i} \) includes useful flux, such as heat loss through conduction or a transient term if any. The solution to the set of nonlinear simultaneous equations Eqs. (5.23) results in the final temperature field.

5.2 Zoning of the Reheating Furnace

As shown in Fig. 5.3, the Stelco LEW furnace is about 32 m by 11 m by 2–3 m. If the entire furnace was to be considered simultaneously, a computationally unmanageable number of zones would result. Fortunately, due to the characteristics of the reheating furnace geometry, the opening (Fig. 5.3) between furnace sub-chambers is sufficiently narrow.
Fig. 5.3  Reheating furnace showing division into sub-chambers (Schematic)
to provide a barrier to the radiative interaction between them. Radiation originating in one sub-chamber which passes through the transitional section is unlikely to be reflected back, both because the area of opening between furnace sub-chambers is small compared with the whole chamber and because the gas in the chamber is highly absorptive. Thus for the evaluation of direct exchange and total exchange coefficients, the reheating furnace was considered as 5 separate enclosed sub-chambers by introducing fictitious surfaces at the connection area (Fig. 5.3). A typical sub-chamber is shown in Fig. 5.4. The radiative exchange between each adjacent sub-chamber is assumed to be only that which directly passes through the opening. In order to examine the above assumption (that radiation passing through the opening has a rare chance of being reflected back), an estimation of the equivalent effective emissivity of the fictitious surface was calculated. An effective emissivity approaching unity implies that the majority of the radiation impinging upon this opening will be absorbed by the next sub-chamber. From the geometry of the reheating furnace at Stelco, the effective emissivity $\epsilon_f$ was calculated to be approximated 0.984 based on the following equation

$$
\epsilon_f = \frac{1}{(1-\epsilon)\left(\frac{A_f}{A_c}\right)+1}
$$

(5.24)

where $\epsilon$ is the emissivity of the interior wall (0.5), $A_f$ is the area of fictitious surface and $A_c$ is the total area of the interior refractory wall of the sub-chamber. The magnitude of emissivity indicates that the fictitious surface is close to a blackbody. It justifies the assumption that the radiative interaction between each adjacent sub-chamber is only in the form of direct radiation through the opening.

The division of the entire furnace chamber into different sub-chambers has two advantages:
Fig. 5.4  Typical sub-chamber showing the fictitious surface
(i) It simplifies the calculation of direct and total exchange coefficients in such a large furnace, since their calculations are being restricted to a small sub-chamber.

(ii) since furnace sub-chambers resemble each other in geometrical configuration, a general computer program could be employed for the calculation of these coefficients.

Each furnace sub-chamber (including gas) was discretized into a number of zones. In order to characterize the heat flux and temperature variation in both the longitudinal and transverse directions of the furnace, the gas volume, slab surface and roof refractory walls were all divided into a series of zones in these two directions. To investigate the effect of the side walls, both left and right refractory side walls were divided into a series of zones along the furnace longitudinal direction. In the transverse direction, four gas zones and four slab zones were created to account for any nonuniformity of temperature.

For zone numbering, maximum use was made of symmetry with respect to the longitudinal furnace axis. A typical numbering sequence on a slab surface in the first sub-chamber is shown in Fig. 5.5a. Each pair of symmetrical surface zones differ by 20. Exchange coefficients calculated for one half of the furnace chamber were applied to the other half. The zone numbering sequences for the side refractory wall (Fig. 5.5b), the gas (Fig. 5.5c), and the furnace roof zones (Fig. 5.5d) are also shown. The number of zones for each furnace sub-chamber is listed in Table 5.1.

5.3 Direct Exchange Coefficient and Total Exchange Coefficient Calculation

One of the major tasks in the application of the zone method is the evaluation of the direct exchange coefficients between zones. The importance of accurate calculation of these coefficients is vital to the ultimate prediction of the temperature fields and the heat-flux distribution in the furnace enclosure. A unique nonuniform division method was
Fig. 5.5(a)  Zone numbering sequence for the slab surface

Fig. 5.5(b)  Zone numbering sequence for the furnace side wall
Fig. 5.5(c)  Zone numbering sequence for the gas zone

Fig. 5.5(d)  Zone numbering sequence for the furnace roof
Table 5.1 Summary of Sub-Chamber Zoning

<table>
<thead>
<tr>
<th>Subchamber</th>
<th>Gas zone</th>
<th>Surface Zone</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>32</td>
<td>88</td>
<td>120</td>
</tr>
<tr>
<td>2</td>
<td>24</td>
<td>60</td>
<td>84</td>
</tr>
<tr>
<td>3</td>
<td>32</td>
<td>88</td>
<td>120</td>
</tr>
<tr>
<td>4</td>
<td>12</td>
<td>36</td>
<td>48</td>
</tr>
<tr>
<td>5</td>
<td>16</td>
<td>44</td>
<td>60</td>
</tr>
</tbody>
</table>

introduced to calculate the gas-to-surface direct exchange coefficients. The evaluation of total exchange coefficients between zones in a medium having a large extinction coefficient was simulated by a pure diffusion process\(^\text{(20), (48)}\).

5.3.1 Calculation of Direct Exchange Coefficients

Since analytical solutions are not available, the evaluation of the direct exchange coefficients for the furnace chamber using Eq. 5.1 and Eq. 5.6 requires the use of approximate numerical methods. Owing to the large computational effort required for even a coarsely zoned furnace chamber, it is imperative to carry out all calculations efficiently.

The direct radiative exchange between two zones separated by a large distance will be relatively small due to the intervening absorptive medium. Since the angles \(\theta_i\) and \(\theta_j\) (Fig 5.1) will vary slightly within the geometrical domain of both zones, the direct exchange coefficients for widely separated zones were approximated by

\[
\bar{s}_{ij} = \left( \frac{\cos \theta_i}{\pi r_i^2} \right) \left( \frac{\cos \theta_j}{\pi r_j^2} \right) \exp(-kp r_c) \Delta A_i \Delta A_j
\]

\[
\bar{g}_{ij} = \left( \frac{k(\cos \theta_i)}{\pi r_i^2} \right) \left( \frac{\Delta v_i}{\Delta A_i} \right) \exp(-kp r_c)
\]
where \( r_c \) is the centre to centre distance between zones.

Radiative exchange between zones increases rapidly with decreasing zone separation. Thus closely spaced zones were discretized into a series of subzones, each sufficiently small that \( \theta_i \) and \( \theta_j \) could be considered constant over the subzones. According to the definition of multiple integration, the direct exchange coefficients could be approximated by

\[
\bar{s}_{i,j} = \sum_i \sum_j \frac{\cos \theta_i \cos \theta_j}{\pi (r_c)_{ij}} \exp(-kp(r_c)_{ij}) \Delta A_i \Delta A_j \quad (5.27)
\]

\[
\bar{g}_{i,j} = \sum_i \sum_j \frac{k \cos \theta_i \Delta V_i \Delta A_j}{\pi (r_c)_{ij}} \exp(-kp(r_c)_{ij}) \quad (5.28)
\]

A criterion for zone subdivision was developed. An estimate of the relative importance of radiative exchange between any two zones can be determined by the ratio of their approximate direct exchange coefficient (calculated from Eq.(5.25) or Eq.(5.26)) to the summation of all direct exchange coefficients to the zone (equal to the area of the surface zone as per Eq. (5.7)). If the ratio of a zone pair was above a prescribed value (1%), further subdivision was carried out. Fig. 5.6 illustrates the method of the subdivision of a surface zone pair.

Since gas elements closest to any surface will contribute disproportionately to the surface irradiation, a non-uniform division method was devised to account for this effect. Thus instead of uniform divisions in the whole gas zone, those gas elements closest to the surface were more finely divided. Fig 5.7 illustrates the method utilized. The accuracy of the direct exchange calculation was established from Eq. (5.7) and by defining the error \( \delta \) according to

\[
\delta = \frac{\frac{N_s}{\sum_{i=1}^N \bar{s}_{i,j}} + \frac{N_g}{\sum_{j=1}^N \bar{g}_{i,j}}}{A_i} - A_i \times 100\% \quad (5.29)
\]

For \( \delta > 5\% \) additional subdivision was carried out. However, for the second gray gas
Fig. 5.6  Subdivision of a surface zone pair
Fig. 5.7 Non-uniform subdivision of gas zones used for the calculation of $g_{j_i}$
\( k_2 = 2.168 \times 10^{-1} \text{ (kPa.m)}^{-1} \), the diffusive method (20),(48) was used to simulate the radiative exchange (Section 5.3.2).

The code T1.CHAMBER written in FORTRAN IV was employed to calculate the direct exchange coefficients.

5.3.2 Calculation of the Total Radiative Exchange Coefficient.

Total exchange coefficients were calculated with Eqs. (5.10) to (5.14). From the left hand side of Eq. (5.10), a general transfer matrix can be written as

\[
D = \begin{bmatrix}
\overline{s_1} - A_1/\rho_1 & \overline{s_{12}} \\
\overline{s_{21}} & \overline{s_{2}} - A_2/\rho_2 \\
\vdots & \vdots \\
\overline{s_{n1}} & \overline{s_{n2}} & \cdots & \overline{s_{nn}} - A_n/\rho_n
\end{bmatrix}
\]

(5.30)

A unique property was found in the transfer matrix, which can lead to savings in computational time. Since any convex surface zone \( i \) (typical of the furnace zones) cannot
directly irradiate itself

\[ \bar{s}_{s_1} = 0.0 \]  \hspace{1cm} (5.31)

\[(i=1, 2, 3, \ldots \ldots \ldots \ldots \ldots \ldots N_s)\]

the diagonal elements in the transfer matrix become

\[ \bar{s}_{s_1} - A_i/\rho_i = -A_i/\rho_i \]  \hspace{1cm} (5.32)

Utilizing Eq. (5.7) and noting that

\[ \rho_i = 1 - \epsilon_i, \quad 0 < \epsilon_i < 1, \quad |A_i/\rho_i| > F_i \]

yield the following result

\[ \sum_{j=1}^{N_s} |\bar{s}_{s_j}| < |\bar{s}_{s_i} - A_i/\rho_i| \]  \hspace{1cm} (5.33)

Thus the transfer matrix is a typical diagonal-priority matrix, which is very stable. For this reason a decomposition method was employed to solve Eq. (5.10) for the \( W_i \) and iterative improvement of the result was unnecessary.

Using a similar technique as described by Eq. (5.29), the reliability of the calculation of total exchange coefficients was checked and the results found to be accurate to better than 5%.

When the medium is highly absorptive, the subdivision method becomes excessively demanding of computation time. For the second gray gas \( (k_2=2.168\times10^{-1}\text{(kPa.m)}^{-1}) \), when a beam of radiation impinges upon it and one measures the radiative intensity at 1 m from the source, only 0.3247% of the original radiation will be transmitted through the gas.
It has been shown \((20)\) that when the product of the centre-to-centre distance between the zones and extinction coefficient \(k\) is greater than 3.0, radiation can be approximated as a diffusion process and adjacent gas zones will interact radiatively with the surface. All other radiation will be blocked by the absorbing medium.

Applying the diffusion method developed by Sarofim and Hottel\((20)\) to the second gray component, the net radiative transfer between an adjacent gas zone and a surface zone was calculated from the following

\[
q_{\text{net}} = \frac{(GS)_2}{A} (E_g - E_s) \tag{5.34}
\]

\[
q_{\text{net}} = \frac{4}{3k} \left( \frac{dE_g}{dx} \right)_\text{bulk} = \frac{8}{3kd} (E_g - E_s) \tag{5.35}
\]

Comparing Eq. (5.34) to Eq. (5.35),

\[(GS)_2 = \frac{8A}{3kD} \tag{5.36}\]

where \(A\) is the interfacial area, \(D\) is the centre-to-centre distance between the gas zone and the surface zone, \(k\) is the extinction coefficient, and \((GS)_2\) is the total exchange coefficient between the gas zone and its contiguous surface zones.

The total exchange coefficients were evaluated with the code COTEC.

5.3.3 Energy Balance on the Slab Surface and Refractory Wall

Since the model assigns the gas temperature distribution in the furnace chamber, the gas flow pattern is not required. The interior refractory wall surface temperature and the slab surface temperature were obtained by applying an energy balance at each surface zone. Net heat transfer at a typical refractory wall zone \(i\) is the radiation resulting from the
zones of gas and the other refractory wall and the slab surface, convection from the adjacent gas and conduction through the refractory wall (assumed to be one dimensional). In steady state this can be expressed by

\[ \sum_j \overrightarrow{G_{ij}} E_{gj} + \sum_j \overrightarrow{S_{ij}} E_{sj} - A_i \varepsilon_i E_{s,i} + A_i h_i(T_{gk} - T_{si}) \]

\[ = U_i A_i (T_i - T_{en}) \] (5.37)

where \( h_i \) is the local convective heat transfer coefficient, \( U_i \) is the thermal conductance of the refractory wall and \( T_{en} \) is the temperature of the surroundings. The calculation of \( U_i \) depends on the structure of the refractory walls and their corresponding thermal conductivities. The design of the refractory wall together with the relevant material properties for Stelco LEW furnace are shown in Fig. 5.8. The calculated values for the conductances were \( U_T = 1.207 \, \text{W/m}^2\text{K} \) (top wall) and \( U_B = 0.725 \, \text{W/m}^2\text{K} \) (bottom wall).

Convection to the refractory wall was calculated using \( h_i = 7.80 \, \text{W/(K.m)}^2 \) obtained from Glinkov. Since convection is likely to account for less than 5% of the total heat transfer, an established estimation of the convective heat transfer coefficient \( h_i \) should result in negligible error.

The furnace chamber was subdivided into 234 refractory wall zones (of which 116 are in the top half), resulting in 234 non-linear simultaneous equations to be solved for zone temperature and, subsequently, zone heat flux.

Zones on the slab surface are subjected to the same heating conditions as the refractory wall zones (Fig. 5.9). The net energy incident on the exposed slab surface is
Top Half Furnace

Unit of dimension: mm

Bottom Half Furnace

Thermal Conductivity (W/ m.K)

- 0.198
- 0.44
- 1.31
- 0.1784

Fig. 5.8 Structure of furnace refractory walls
Fig. 5.9 Heat transfer for the slab in the reheating furnace
conducted into the slab and the skidrail system. For an exposed slab surface zone

\[
\sum_j G_{\text{S},j} E_{\text{S},j} + \sum_j S_{\text{S},j} E_{\text{S},j} - A_{i} \varepsilon_{i} E_{\text{S},i} + A_{i} h_{i} (T_{g,k} - T_{s,i})
\]

\[
= - \lambda_{i} A_{i} \left( \frac{\partial T}{\partial y} \right)_{y=\text{surface}}
\]

(5.38)

\(i=\text{number of slab surface zone}\)

The right hand side of Eq. (5.38) represents the heat being conducted into the slab. \(\lambda_{i}\) is the local slab thermal conductivity. The situation at the slab/skidrail contact region is more complex and will be discussed in Section 5.4.

Since the temperature distribution in the slabs is unknown and related to the radiative transfer from the furnace chamber by Eq. (5.38), the radiative and conductive transfers are coupled heat-transfer phenomena. Thus the radiative equations have to be solved in conjunction with the two-dimensional conduction problem. The coupling of the equations from the zone method and from the two-dimensional conduction model will be presented in Section 5.4.

In performing an energy balance over the refractory wall and slab surface zones, the emissivity of the refractory wall and the slab, respectively, are required. However, information on the emissivity of the refractory wall, particularly its dependence on temperature, is rather scarce\(^{(48),(49)}\). Therefore a value of 0.5, which is commonly adopted for industrial application\(^{(48)}\), was selected. According to Masashi\(^{(46)}\), the slab emissivity does not vary much with temperature and 0.8 is a good estimate.
5.4 The Slab Conduction Model

To describe heat flow within the slab as it moves through the furnace, an unsteady state two-dimensional conduction model was applied. Heat conduction was considered in the through-thickness (y) and width (x) directions only, since conduction in the slab longitudinal direction can be reasonably ignored. The governing equation is

$$\rho C_p \left( \frac{\partial T}{\partial t} \right) = \frac{\partial}{\partial x} \left( \lambda \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left( \lambda \frac{\partial T}{\partial y} \right)$$

(5.39)

The initial temperature distribution depends on the thermal history of the slab from the previous process (e.g. hot charging from a continuous casting machine). If the slab is inspected and conditioned at room temperature, its initial temperature will be at the ambient value. Three types of boundary conditions have been identified for this slab conduction problem:

(i) The Exposed Slab Surface

The top slab surface and the exposed area of the slab bottom surface exchange energy directly with the furnace chamber, with the slab surface temperature gradient given by Eq. 5.38. In order to link the slab conduction equation to the furnace radiative exchange, a radiative exchange coefficient

$$e_i = \sum_j \frac{G_{S_i j} E_{g j}}{A_i} + \sum_j \frac{S_{j i} E_{S j}}{A_i} - A_i \epsilon_i E_{S_i} - A_i (T_{s k} - T_i)$$

(5.40)

was introduced into the model. $T_{s k}$ is then the 'characteristic' temperature of the chamber gas. Substituting Eq. (5.40) into Eq. (5.38), the boundary condition can be written in a
familiar form:

$$- \lambda \left( \frac{\partial T}{\partial y} \right)_{y=\text{boundary}} = e_i \left[ T_{s_k} - T_i^* \right] + h_i \left( T_g - T_i \right)$$  \hspace{1cm} (5.41)

The radiative heat-transfer coefficient serves to connect the chamber zone model to the slab conduction model. Eq. 5.41 can be employed to determine the minimum time step for the explicit finite-difference used to calculate the temperature distribution in the slab. Although $e_i$ was evaluated for the geometrical centre of each slab surface zone, the radiative flux at any other points on the zone was obtained using a linear interpolation between $e_i$ and the neighbouring zones. Thus the radiative flux for any point on the slab surface could be calculated and the boundary conditions established.

Under given operating conditions in the reheating furnace, the radiative exchange coefficients $e_i$ are influenced by the slab zone temperature as well as by geometrical position in the furnace chamber. The sensitivity to the slab surface temperature was examined by introducing a temperature perturbation of 50 °C. The variation of $e_i$ was typically found to be less than 1%, thus indicating little sensitivity to changes in the slab surface temperature. Since the variation of surface temperature between adjacent zones is small, linear interpolation between zones will be unlikely to introduce significant error.

(ii) The slab/skidrail contact region

In this vicinity, radiation and convection from the furnace environment are partially blocked by the skid structure, thus reducing heat transfer from the chamber. This effect is compounded by conduction at the skid/slab contact area, resulting in localized depression of slab temperature. Fig. 5.10 depicts the situation in detail. The problem was analyzed by introducing the fictitious surface AB, through which all radiation coming from the gas and the refractory wall to the slab bottom must pass. The slab bottom surface, the exterior
H: Bottom slab surface
T: Exterior surface of skidrail
F: Fictitious surface representing furnace chamber
s: distance between centreline of two adjacent skidrails
h: height of the skidrail
d: width of skidrail

Fig. 5.10    Contact region between the skidrail and the slab (detail)
surface of the skidrail and the fictitious surface form a radiative enclosure (assumed to be infinite in the direction perpendicular to the page). If all the surfaces are assumed to be diffuse and to have uniform properties and temperatures, and furthermore, gas emission within this small region is ignored, the view factor from the chamber to the slab can be expressed analytically (the derivation is shown in Appendix II)

\[
\phi = \left( \sqrt{(h-d/2)^2 + (s-d)^2} - (h-d/2)/s \right)
\]

(5.42)

Based on values of \( s=1700(\text{mm}) \), \( d=301(\text{mm}) \), \( h=475(\text{mm}) \) from the Stelco LEW furnace the view factor was calculated to be \( \phi = 0.65 \).

The slab surface around the skidmark region receives heat directly from the furnace chamber as well as heat reflected and radiated from the exterior surface of the skidrails. If the exterior surface of the skidrail is assumed to be in radiative equilibrium, an electrical analogue shown in Fig. 5.11, can be established. The assumption of radiative equilibrium for the skidrail is reasonable because the convective heat transfer from the furnace gas to surfaces of the skidrail is of the same magnitude as that from the interior skidpipe wall to the cooling water.

Introducing subscripts \( F, T, H \) to indicate the fictitious surface, the exterior surface of the skidrail and the slab bottom surface respectively, it can be seen that

\[
F_F \to T = 1.0 - F_F \to H
\]

(5.43)

The total resistance expression

\[
R_{to} = 1.0/(A_F F_F \to H + 0.5 A_F F_F \to T)
\]
Fig. 5.11 Radiative network among skidpipe, slab surface and the fictitious surface
can be rearranged in the form

$$A_F R_{to} = \frac{1.0}{(F \rightarrow H + 0.5 F \rightarrow T)}$$  \hspace{1cm} (5.44)

The shielding factor of the skidrail can be expressed by

$$C = F \rightarrow H + \frac{(1.0 - F \rightarrow H)}{2.0}$$  \hspace{1cm} (5.45)

The first term of Eq. 5.45 gives the direct radiation from the gas and the refractory wall to the slab surface, while the second term represents the reflected and reradiated components from the exterior wall of the skidrail. The shielding factor of the skidrail for the Stelco LEW reheating furnace was calculated to be $C=0.83$.

As previously noted, the slab will also lose energy to the cooler skidpipe due to conduction at the contact area. Values for contact resistance between skidrail and slab were obtained from Ford $^{(21)}$.

$$CR = \begin{cases} 
1.4 & 0 < T_{slab} < 120^\circ C \\
1.4 + 3.2/20.0(T_{slab} - 120.0) & 120 \leq T_{slab} \leq 200 \\
4.4 - 4.2/300.0(T_{slab} - 200.0) & 200 < T_{slab} \leq 500^\circ C \\
0.2 & 
\end{cases} \hspace{1cm} (5.46)$$

The Dittus–Boelter equation$^{(45)}$ was utilized to determine the convective heat-transfer coefficient between the cooling water and the interior tube.

$$h = 0.023{Re^*}^{0.8} \frac{Pr^*}{D} \lambda/D$$  \hspace{1cm} (5.47)

and the average conductivity of the pipe blanket was taken to be 0.623 W/mK.
The overall heat transfer coefficient to the skid-pipes was calculated assuming steady state

\[ U_c = \frac{1}{\left( \frac{1}{CR} + \frac{\Delta x}{\lambda} + \frac{1}{h} \right)} \]  

(5.48)

where \( \Delta x \) is the thickness of the insulating jacket, and \( \lambda \) is its average conductivity.

In characterizing the total conductance from the furnace chamber to the slab bottom surface in Eq. (5.45), uniform irradiation has been assumed for all the surfaces. In reality, this assumption does not hold. The slab zone around the contact region will be subjected to the most severe shadowing effect of the skidpipe. The shadowing of the skid structure will decrease as the distance from the surface zone to the slab/skidrail interface increases. Therefore, the shielding factor suggested in Eq. (5.45) was applied only for the zone near the slab/skidrail interface. Coupling Eq. (5.42) with Eq. (5.48), the boundary condition for a zone in the contact region is

\[ -\lambda \left( \frac{\partial T}{\partial y} \right) = q_{\text{radiative}} + q_{\text{convective}} - q_{\text{conductive}} \]

\[ = C(1-\beta) q_{\text{net},i} - \beta U_c (T- T_{wa}) \]  

(5.49)

where \( \beta \), the parameter characterizing the dimension of the contacting area, is given by

\[ \beta = \text{width of contacting zone/node size } \Delta x \]  

(5.50)

\( U_c \) is the overall heat-transfer coefficient for the heat lost to the cooling water and can be evaluated from Eq. (5.46) to Eq. (5.48), and includes the contact resistance, the conductive resistance of the pipe blanket and the convective resistance of the cooling water. Eq. (5.38) can be used to evaluate \( q_{\text{net},i} \). The term \( C \) is the shielding factor due to the presence of the skidpipe.
(iii) The side surface of the slab

Since the slab thickness is typically less than 6% of the slab length, heat transfer to the end surface will not be a significant factor in determining the overall temperature distribution. Heat transfer to the end surface of the slab can be expressed by

\[ q = v \sigma \left[ \left( \frac{T_r}{100} \right)^{4} - \left( \frac{T_s}{100} \right)^{4} \right] + h(T_g - T_s) \]  \hspace{1cm} (5.51)

where \( v \) is the radiative exchange factor and \( \sigma \) is the Stefan–Boltzmann constant. Owing to its proximity, the end of the slab was assumed to interact radiatively with the refractory side wall, resulting in the analogous electrical circuit shown in Fig. 5.12, from which

\[ q_r = \sigma \left( T_r - T_s \right)^{4} \left( \frac{1}{F_S \rightarrow r} + \frac{1 - e_r}{e_r A_r A_s} + \frac{1 - e_s}{e_s} \right) \]  \hspace{1cm} (5.52)

AB=0.24 m; DC=3.8 m and the perpendicular distance between AB and DC is 1.15 m (refer to Fig. 5.12). According to the cross-string method \(^{43}\)

\[ F_{AB \rightarrow CD} = F_{S \rightarrow R} = \frac{AC + BD - (AD + BC)}{2AB} = 0.84 \]  \hspace{1cm} (5.53)

The boundary condition for the ends of the slab is therefore found to be

\[ -\lambda \frac{\partial T}{\partial x} \bigg|_{x=L} = 0.665 \sigma \left( T_r^{4} - T_s^{4} \right) + h(T_g - T_s) \]  \hspace{1cm} (5.54)

where \( T_r \) represents the average interior refractory wall temperature.

5.4.1 Heat Conduction Inside the Slab

By assembling Eq. (5.39) and Eq. (5.54), the complete slab conduction equation and its corresponding boundary conditions can be defined. An explicit finite-difference model was employed to compute the temperature distribution in the slab. The longitudinal
Fig. 5.12 Radiative network for the slab end surface and the furnace side wall.
cross-section of the slab (transverse to the push direction) was divided into a two-dimensional nodal mesh, as shown in Fig. 5.13, with 7 nodes through the slab thickness and 30 nodes in the slab longitudinal direction. Since the different boundary conditions for the slab surface, and the slab conduction equation, required different minimum time steps, a compromise time step was selected in order to maintain the stability of the explicit finite-difference calculation.

5.4.2 Coupling of the Zone Method with the Heat-Conduction Model.

As seen in Eq. (5.38), the slab conduction problem and the radiative chamber problem are phenomena coupled by the shared boundary condition. Thus the two individual models have to be solved together in order to obtain the temperature field and heat-flux distribution. As mentioned earlier, Eq. (5.41) and Eq. (5.49) provide the connection between the zone method in the chamber model and the finite-difference scheme in the slab conduction model. The common parameter $e_i$ determined for the chamber model was incorporated into the conduction model.

Since in the zone method, the directed exchange coefficients are functions of the zone temperatures, which are to be obtained from the solution procedure, an initial assumed temperature profile is required in order to start the chamber calculation. Energy balances are not required for the gas zones, since the gas temperature distribution was assigned. The solution of the energy balance (Eq. 5.37) for the refractory wall zones and slab surface zones (Eq. 5.38), when combined with the conduction model of the slab, provide a new estimate for the temperature distribution. This new distribution was compared with the previous distribution and the procedure repeated until satisfactory convergence was obtained. From a guess of the initial temperature profile, generally 8 to 10 iterations were required. A detailed outlined of the procedures is as follows:
SYMMETRICAL LINE OF SLAB

Fig. 5.13  Nodal division of the slab (transverse to the furnace longitudinal direction)
(i) Calculate the direct exchange coefficients for the furnace chamber.

(ii) Assume temperatures for all refractory wall and slab surface zones.

(iii) Evaluate total exchange coefficients in the furnace chamber. (Eq. (5.10) to Eq. (5.14) and Eq. (5.36))

(iv) Evaluate directed exchange coefficients from the known zone temperatures.

(v) Solve energy balance (Eq. (5.37)) for the refractory wall temperature distribution using UBC library routine QNEWT (59).

(vi) Calculate the radiative coefficients and obtain heat flux on the slab surface zone.

(vii) Using the calculated boundary conditions, initiate the two-dimensional unsteady-state conduction model for the slab.

(viii) Compare the calculated temperature profile with the assumed profile and, if the maximum temperature deviation exceeds 10°C, adopt the new profile and return to (iv). If not, calculation procedure is stopped.

The flow chart of the procedure is shown in Appendix III.

As has been described, the heat-transfer model consists of three sub-models, i.e., the clear-plus-two-gray gas emissive/absorptive model, the zone model of the furnace chamber and the two-dimensional transient conduction model of the slab. The interaction among the sub-models is illustrated by the flow chart in Appendix III.
5.5 Sensitivity Analysis of the Nodal Division in the Conduction Model

As mentioned earlier, heat transfer within the slab was calculated using an explicit finite-difference method to obtain the slab temperature distribution. The sensitivity of the resultant temperature predictions to the grid spacing was investigated in order to minimize computing cost. A general code COND2 was written to simulate the two-dimensional unsteady conduction problem in a rectangular system with multi-mode heat transfer at the boundary, i.e. Fig. 5.13. (M denotes the number of thickness nodes (Y axis); ND the number of longitudinal nodes (X axis)).

A sample rectangle with dimensions of AD=0.24(m) and AB=4.35(m), has been selected. Surfaces AB, CD and AD are considered to have combined modes of radiative and convective heat transfer. An adiabatic condition was imposed on the side BC to simulate symmetrical heating. Nodes E and F are conductive boundary conditions. The thermophysical properties of the slab are temperature dependent.

Evidently, the sensitivity of the predicted temperature will depend upon two variables: M and ND. In order to reveal their relationships, M was first varied and ND held constant then ND was varied and M held constant. Fig 5.14 shows the predicted temperature profile across the slab thickness when M varies from 5 to 10 (ND=25). The calculated temperature profile for M=8 and M=10 differ by less than 3% and even for M=5 the results are satisfactory. With M=6, ND was varied from 25 to 50 and the temperature distribution along the length of the slab was found to converge satisfactorily (Fig. 5.15); the difference between ND=40 and ND=50 is less than 3%. The situation when both M and ND are varied is depicted in Fig. 5.16, in which three temperature contours have been plotted corresponding to three values of nodal division. These results indicate that the predicted temperature distribution is relatively insensitive to the number of nodes, the smoother contours for fine grids being due to the improved curve fitting possible.
Fig. 5.14 Sensitivity of internal slab temperature to the node spacing in the thickness direction.
Fig. 5.15 Sensitivity of slab surface temperature to the node spacing in the longitudinal direction.

Legend

\begin{itemize}
\item ND=25
\item ND=40
\item ND=50
\end{itemize}
Fig. 5.16 Isotherm contours with respect to three nodal divisions.
with additional points. In order to obtain satisfactory temperature contours, a network of \( M = 7 \) by \( ND = 30 \) node points was selected.
6. RESULTS AND DISCUSSION

The heat-transfer model developed for the reheating furnace was used to investigate three aspects of furnace behaviour:

(i) To predict the effects of different operational parameters, such as slab size, gas temperature distribution, push rate and steel grade, on the temperature distribution in the slab and refractory walls.

(ii) To calculate the nonuniform heat-flux and temperature distributions around the slab/skidrail contact region in detail and to identify possible measures to alleviate the skidmark effect.

(iii) To develop improved heating strategies for the hot charging of slabs.

The effect of varying each of the identified furnace operating parameters was established by holding the remaining parameters constant in the model. Unless otherwise specified, the standard slab selected for the calculation was medium-carbon (0.23%) steel and 4.35 m wide by 0.24 m thick. Although the 0.23% carbon steel is chosen as a sample grade of steel for the calculation, common carbon steels are expected to exhibit similar heating behaviour under the same heating conditions, since they have very similar thermal properties.

6.1 General Thermal Behaviour of the Reheating Furnace

6.1.1 Slab Temperature and Refractory Wall Temperature Response

By assuming gas temperature profiles, the temperature response of the slab and the refractory walls were calculated by the heat-transfer model. Figure 6.1 shows results obtained for a pushing rate of 0.0034 m/sec. (200 T/hr. for slabs charged in tandem).
Fig. 6.1  Predicted slab temperature profiles longitudinally in the reheating furnace
Since the level of gas temperatures in most reheating furnaces is similar, the range of the assumed distribution for the current calculations is based on the values of previous reports and the measurements of suction temperature thermocouples from Stelco. The gas temperature of both top and bottom chambers was assumed to be uniform in the transverse direction and to vary longitudinally, as shown in Fig. 6.1, from 900°C to 1400°C. Under these conditions, the slabs are heated from room temperature to a predicted exit (drop-off) temperature of 1180 - 1200°C, a value well above the austenizing temperature for the steel. As expected, the slab centreline temperature is lower than the surface temperature until the slab reaches the soaking zone, with the maximum temperature difference (230°C) occurring at the centre of the heating zone (about 20 m into the furnace). The lag in centreline temperature is mainly due to the low conductivity of the steel.

Isotherm contours in a slab offer a useful visual form for a two-dimensional temperature distribution in the slab cross section. In particular, a detailed temperature distribution around the slab/skidrail contact region will be clearly depicted. Figure 6.2 shows temperature contours at three different axial furnace positions (25.2 m, 27.5 m, 32.0 m). Although at the entrance to the soaking zone (25.25 m), the slab surface temperature is significantly higher than at the centreline, the influence of the skids is relatively minor. As the slab progresses through the soaking zone the localized slab temperature depression adjacent to the skids becomes more apparent. At 27.5 m, the average temperature of the skidmarks is 25°C lower than the unaffected surface, which increases to 50°C at the exit. In the soaking zone, the centreline temperature quickly approaches the surface temperature. These results are indicative of the fact that, before the soaking zone, the major heat sink in the slab is its center, but once the slab enters the soaking zone, the slab/skidrail contact

† As was pointed out before, these values are not gas temperatures but are "balanced" temperatures among gas, refractory walls and slabs in the furnace chamber.
‡ Centreline refers to the geometric centre of the slab cross-section
Note: Isotherm Values are $T \times 10^{-3}$ (°C)

Fig. 6.2 Predicted slab temperature contours at three longitudinal positions
region becomes the main heat sink and the skidmark effect grows quickly until the slab is discharged.

Predictions for the top and side furnace refractory wall surface temperatures are shown in Fig. 6.3. The refractory wall surface temperature stabilizes at 250–270 °C lower than the local gas temperature. The drop in the refractory side wall temperature is due to the fact that the furnace geometry is relatively unfavourable for the radiative heat transfer to the side wall compared with that to the roof surface. In addition, the refractory side wall near the exit end of the furnace loses heat to the surroundings through the furnace discharge door.

6.1.2 Heat-Flux Distribution to the Slab Surface

Model predictions for the net surface heat flux (Fig. 6.4) indicate that the maximum slab heating rate occurs around the midpoint of the heating zone, about 20 m into the furnace. In the soaking zone, the radiative heat flux is much reduced, partly due to the locally high slab temperature. These results are in agreement with other studies\(^{(2),(15)}\). Of more significance, however, are the transverse heat-flux distributions shown in Fig. 6.5, since these are believed to exert a significant impact on the rolling process\(^{(29)}\). The transverse heat-flux distribution can be seen to exhibit nonuniformity in contrast to the gas temperature which was specified to be uniform in the transverse direction. Near the furnace entrance (10.67 m), the heat-flux profile is concave due to the influence of the refractory side wall where the slab surface receives more radiative energy. However, as the slab approaches the furnace discharge (28 m into the furnace), the heat flux profiles becomes convex, since the maximum heat flux occurs in the middle of the furnace. This effect results from the slab end temperatures becoming greater than those in the central part of the furnace so that the net radiative heat received is proportionally decreased.
Fig. 6.3  Predicted refractory wall temperatures
Fig. 6.4  Predicted heat flux to top slab surface as a function of axial position in reheating furnace
Fig. 6.5  Predictions of heat flux to slab across width of the reheating furnace.
The importance of direct radiation to the slab surface from the refractory wall, relative to direct radiation from the gas, is shown in Fig. 6.6. At the furnace entrance, gas radiation provides the dominant contribution to the slab surface heat flux, while near to the furnace exit, the refractory wall provides about 58% of the total radiation. This efficiency of surface/surface radiative exchange, relative to gas/surface exchange, is the basic reason for moving to open radiative tube (ORT) furnace design\(^{(29)}\). These furnaces rely heavily on radiation from the refractory wall.

6.1.3 The Effect of Push Rate on the Slab Temperature Distribution

Varying the rate of slab throughput (push rate) is a common plant practice. Fig. 6.7 shows a comparison of the centreline temperature for two push rates: 0.0034 m/s (200 T/hr.) and 0.004 m/s (235 T/hr.); the remaining parameters were held constant. The centreline temperature (at furnace exit) is predicted to be 80°C lower for the higher push rate. The same effect is seen in temperature contours (Fig. 6.8) in the discharged slabs under the two production rates although the surface temperatures do not differ much. The higher production rate also exacerbates the process of skidmark formation, which is not desirable for rolling. These results demonstrate that the major obstacle to increasing the productivity of the furnace is the poor conductivity of the slab being heated. Since increasing the productivity implies reducing the residence time of the slab inside the reheating furnace, the energy received at the surface does not have adequate time to conduct into the centre of the slab, resulting in a large temperature difference between the centre and surface of the slab. An alternative process, electrical induction heating, might alleviate this situation since the magnetic field used induces eddy current inside the slab.
Fig. 6.6 A comparison of direct radiation to the slab surface from the chamber gas and refractory
Fig. 6.7 Comparison of the slab centreline temperature for two slab push rates
Note: Isotherm Values are $T \times 10^{-3}$ (°C)

(a) Push rate: 0.004 m/sec.

(b) Push rate: 0.0034 m/sec.

Fig. 6.8 Slab temperature contours at furnace exit for two push rates
6.1.4 The Effect of Slab Dimension on the Temperature Distribution

Increasing the slab thickness, while holding the other operational parameters constant, has a similar effect to increasing the push rate. Figure 6.9 shows predicted temperature contours at the furnace exit, for two different thickness slabs, 240 mm and 300 mm. The centreline temperature of the 240 mm slab at exit can be seen to be about 130°C higher than that of the 300 mm slab, while the temperature difference between centreline and surface also tends to be slightly less. This could reduce subsequent problems in the final gauge control of the rolled products. One solution for the push-type reheating furnace to heat thick slabs is to increase the residence time of the slabs in the furnace chamber by lowering the push rate of the furnace. Another approach is to increase the gas temperature of the furnace chamber by increasing fuel firing rates. However, there is a limit to the latter since the poor conductivity of the slabs being heated is likely to lead to a larger temperature difference between the surface and centre of the slabs. Both approaches can be simulated by the off-line model discussed later in Sec. 6.1.6. It is strongly advisable that thick slabs be arranged in groups to facilitate easier control of the reheating furnace. If slabs of different thicknesses are mixed in the furnace chamber, a compromise has to be made to ensure that overheating of thin slabs and underheating of thick slabs do not occur. This heating strategy is very difficult to achieve and has to be carefully simulated with the off-line model.

6.1.5 The Effect of Steel Grades on the Slab Heating Process

It is to be expected that different grades of steel might exhibit somewhat different temperature response curves as a result of differences in their thermophysical properties. Since most common carbon steels have very similar thermophysical properties\(^{(50)}\), their responses under the same heating conditions will show little difference. However, alloy steels
Note: Isotherm Values are $T \times 10^{-3}$ (°C)

(a) Slab thickness = 0.30(m)

(b) Slab thickness = 0.24(m)

Fig. 6.9 The effect of slab thickness on the slab temperature at furnace exit.
can have quite significantly different thermophysical properties from mild carbon steel, as shown in Fig. 6.10. The thermal diffusivity of the alloy steel (3.5% Ni, 1.0% Cr–Mo) is almost 35–40% lower than the common carbon steel. A comparison of temperature prediction for carbon steel (0.23%C) and alloy steel (3.5% Ni, 1% Cr–Mo) is given in Fig. 6.11. At the furnace exit, the surface temperature of alloy steel was found to be about 100°C lower than that of the carbon steel. While the disparity between the centreline temperatures was even greater, roughly about 152°C. These discrepancies cannot be eliminated entirely by altering the heating condition (e.g. increased firing) since the phenomenon originates from the properties of the material being heated. Since the alloy steel exhibits (generally) lower thermal diffusivity values (Fig. 6.10) than carbon steel, energy entering the alloy slab will have more tendency to accumulate near the surface. In order to obtain satisfactory slab temperature uniformity at the furnace exit, additional soaking time will be required for the alloy steel. The computer model can be used to study the different push rates for the heating of alloy steel. The calculated slab exit temperature can be compared and a proper push rate can be selected based on the discharge temperature requirement set by the rolling process.

6.1.6 Off-line Computer Control Model

As the previous sections have demonstrated, the model is capable of simulating the effects of furnace operating parameters on the slab heating process. Clearly the model could be used off-line for computer control because it is capable of predicting desired operating parameters in the form of a data bank for different slab conditions. An example is given below to illustrate the steps in using an off-line model.

The requirement from the rolling mill and information from the slabbing mill consist of the following:
Fig. 6.10  Thermal diffusivities of two steels (from reference(50)).
Fig. 6.11  Predicted longitudinal slab temperature profiles for two steels.
Grade of slab: 0.23% C, carbon steel

Slab dimension: 4.35x0.24x1.0 m³

Slab charging temperature: Room temperature 25–30°C.

Required slab exit temperature (average): 1200 °C.

Maximum allowable difference between the centre temperature and the average slab temperature: 50°C

Maximum allowable difference between skidmark temperatures and the average slab bottom surface temperature: 50°C.

The desired operating parameters, which are controlled variables in the terminology of process control, are the required gas temperature profile and the push rate. These variables were obtained iteratively running the program "ENERGY" (flow chart shown in Appendix III). Suitable operating conditions were selected based on satisfactory exit slab temperature. The desired operating parameters are push rate (0.0034m/s) and the desired top and bottom gas temperature profiles shown in Fig. 6.1. It must be emphasized that the model predictions are conditional upon control of fuel firing rates so that the specified gas temperature profiles are obtained.

A data bank of results from the off-line simulation (for various sets of operating variables) would then be stored for later retrieval by the on-line process computer. In the on-line control, as long as the requirements from the rolling process and the information from slabbing mill are input into the process computer, the process computer will automatically fit the situation on hand into a certain category and choose the appropriate operational parameters from the memory and execute the necessary commands.
6.1.7 Influence of Gas Temperature on the Slab Heat Flux and Temperature Profiles

Within the furnace chamber, the primary source of radiation is the furnace gas and therefore, the heat-flux distributions will be strongly influenced by the gas temperature distribution. Figure 6.12 provides a comparison of net radiative heat flux distributions for the top slab surface for two specified gas temperature profiles. Lower gas temperatures at the furnace entrance result in significantly lower net heat fluxes to the slab surface. It is also interesting to note that gas temperature effects on heat flux are localized, the reason being that the emitting gas is highly absorptive of its own radiation. Thus nonuniform gas emission will not penetrate more than a metre or so. At the slab exit end, the heat flux for the lower gas profile is higher since the slab surface temperature is lower.

All the above calculations are for uniform gas temperatures in the transverse direction of the furnace. However, this is unlikely to be the case in reality, especially when side burners are installed. Then severely nonuniform transverse gas temperature profiles may result in correspondingly nonuniform transverse slab temperatures. The model was used to investigate this problem. At the furnace longitudinal position of 15.22 m, the maximum gas temperature difference of 100°C was assigned in the cross-furnace direction. If the higher gas temperature region is located near the refractory side wall, the transverse irradiation-flux distribution at this longitudinal position is shown in Fig. 6.13(a). The flux is severely distorted with the maximum value close to the refractory side wall. As was shown in Fig. 6.5, the transverse net-flux distribution to the slab surface is not uniform across the furnace, even under transverse isothermal conditions. This is due to the presence of the side wall, which contributes more radiative heat to the slab surface closest to it. This effect would be aggravated in the presence of high gas temperature near the refractory side wall because again the slab area adjacent to the refractory side wall would receive more heat than that in the central part of the furnace. Figure 6.14 shows temperature contours of a discharged slab heated under two gas temperature fields: one with uniform transverse
Fig. 6.12  Predicted longitudinal slab surface heat flux profiles for two gas temperature profiles.
Fig. 6.13(a) Effect of the non-uniform transverse gas temperature on the transverse irradiation flux to the slab surface.
Fig. 6.13(b). Effect of the non-uniform transverse gas temperature on the transverse irradiation flux to the slab surface.
temperature distribution and the other with a locally nonuniform distribution. The temperature of the skidmark near the centre of the furnace is about 10°C lower in the case of the nonuniform transverse gas temperature than its counterpart heated under a uniform gas temperature.

When higher temperature gas is located near the central part of the furnace, at the same longitudinal position, the effects of the side wall on the cross-furnace irradiation flux is somewhat suppressed, since the gas radiation makes up for the reduced radiation coming from the refractory side wall. Figure 6.13(b) shows the transverse irradiation flux variation when the gas temperature at the furnace centre is 100°C higher than that near the refractory side wall at the longitudinal position x=15.22(m). Though the flux still exhibits a slight central minimum, a more even irradiation flux distribution is found, compared with the case in which high gas temperatures are concentrated near the refractory side wall.

It should be pointed out that the above calculations are limited to a small scale nonuniformity in the transverse gas temperature. If the gas temperature shifts towards either the centre or the side of the furnace on a large scale, the consequences for slab heating would be more visible and severe.

Based on the above results, it is understood that precautions have to be taken to prevent hot gas from circulating near the refractory side wall, because this causes unfavourable heating of the slab, such as a large temperature drop along the slab length, overheating near the slab end and a more severe skidmark effect. In the design of side burners for diffusion flames, the centre of the fuel heat release should be in the centre of the furnace.
Note: Isotherm Values are $T \times 10^{-3}$ (°C)

(a) High gas temperature close to side wall at longitudinal position

$x = 15.22$ (m)

(b) Uniform transverse gas temperature

Fig. 6.14 Effect of non-uniform transverse gas temperature on a slab temperature contours at the furnace exit.
6.2 SKIDMARK EFFECT

The occurrence of skidmarks in the heating of slabs brings many undesirable effects to the rolling process, particularly in the roughing operation. The skidmark is a region of nonuniform temperature and nonuniform ductility in the slab which results in nonuniform deformation in the rolling process. Thus, width variation results from nonuniform spread and nonuniform edging in rougher/edger mills and requires extra width allowance that results in 1% yield loss. There is also a positive gauge variation at each skidmark. The immediate problem that also is believed to result from skidmarks is the transfer bar "head-end turn-down" in the roughing mill. Turn-down results in rapid deterioration of table rolls, aprons and supporting structure, owing to the impact of the turned-down end in the early roughing process. This deterioration will, in turn, result in costly repairs.

The computer simulation model considers mechanisms of formation of skidmarks, which include the following:

(i) The shielding effect of the skidrail, which prevents radiation from reaching the contact area between the slab bottom surface and the skidrail.

(ii) The conductive heat loss to the water-cooling skidpipes.

(iii) The radiative exchange between the exterior surface of the skidrail and the slab bottom surface.

The computer model was applied to analyze the formation of skidmarks during the reheating process at the Stelco Lake Erie Works. Heat transfer around the slab/skidrail contact was described previously in Chapter 5.
The skidmark is a region of the slab having a locally low temperature. However, the skidmark not only affects the temperature distribution of a slab at the contact area, but it also causes distortion of the temperature distribution of the slab as a whole, as was seen in Fig. 6.2(c). It is seen that the skidmark effect extends to the top surface of the slab. The temperature of the central region of the slab is also low, due to the presence of the skidpipe, since the heat received on the slab bottom surface is distributed between the slab/skidrail contact region and the central part of the slab because both are heat sinks. As shown in Fig. 6.15, the irradiation flux distribution across the length of the slab bottom is very uneven, and in particular, deep troughs are observed around the skidmark region. This serious distortion indicates that the skidmark region receives much less heat than the normal exposed slab surface, which has great impact on the transverse temperature distribution (slab rolling direction). Figure 6.16 shows the temperature distribution along the slab length direction of a discharged slab. The troughs represent skidmark temperatures and they are roughly 50°C lower than the average slab temperature.

An investigation with the computer model has been performed to determine the relative importance of the skidrail radiative shielding effect and the conductive heat loss from the slab to the skidpipe, as shown in Fig. 6.17. It was found that the reduction of slab surface radiation due to the presence of the skidrail is the primary cause of the skidmark and is far more important than the conductive loss to the skidpipe, which accounts for less than 1% of the total surface radiation. However, the conductive heat loss does assume more importance at the slab discharge due to the higher slab temperature. Thus, in order to reduce the severity of skidmarks, emphasis has to be placed on the radiative shielding of the skidrail using highly insulating material and consideration given to the geometrical size of the skidpipe by narrowing the dimension of the contact area between skidrail and slab. Use of a highly insulating material not only reduces the conductive heat loss, but more importantly, reduces the necessary size of the skidrail and
Fig. 6.15 Irradiation flux across the slab bottom surface (furnace axial position = 24.2 m)
Fig. 6.16

Longitudinal slab bottom temperature distribution at the furnace exit
Fig. 6.17 Comparison between the conductive heat loss and the radiative heat loss at the skid/slab contact region.
increases its exterior surface temperature, which constitutes a major part of the radiative shielding loss. This will become clearer when coating with reflective materials is discussed.

Based on the computer predictions, it is recommended that the following measures be considered to alleviate the skidmark effect:

(1) **Reduce the geometrical size of the skidrail**

The shadowing effect of the skidpipe can be improved dramatically if the geometrical size of the skidrail is reduced, allowing more radiation to reach the bottom surface of the slab. The current structure and geometrical dimensions of the contacting region between the longitudinal skidrail and the slab are shown in Fig. 5.10. The width of the longitudinal skidrail is 301 mm and the height (h) is 475 mm; the distance between two skidrails (s) is 1700 mm. About 18% of the bottom area of the slab is covered by the skidrail, where it does not have access to the outside radiation. The area bounded by the slab/skidrail contact line is also affected by the presence of the skidrail. All the outside radiation has to pass through the surface AB in Fig. 5.10 to reach the bottom surface of the slab. Ignoring the influence of cross-furnace skidrails, the proportion of radiation reaching the slab bottom surface relative to the incident outside radiation has been calculated as a function of the width of skidrail, W, and is listed in Table 6.1.

From Table 6.1, the proportion of incident radiative heat transfer from the furnace chamber to the slab bottom surface can be increased noticeably by narrowing the skidrail width W. Based on the geometry of the skidrail in the Stelco LEW Reheating Furnace, the computer predicted ratio, PR, is 66.7%. If the width of the current skidrail is reduced to 170 (mm), the values of PR will be increased by 3.3%, from the original 66.7% to 70.0%. However, the reduction of the width of the skidrail is often limited by insulation requirements. Better insulation and high thermal capacity materials are necessary if the
Table 6.1 Computer Prediction of the Shielding Effect of Skidrail as a Function of Its Width W

<table>
<thead>
<tr>
<th>W/AB</th>
<th>Width of skidrail (mm)</th>
<th>PR %</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.44</td>
<td>748.00</td>
<td>50.4</td>
</tr>
<tr>
<td>0.40</td>
<td>678.00</td>
<td>52.6</td>
</tr>
<tr>
<td>0.35</td>
<td>595.00</td>
<td>55.4</td>
</tr>
<tr>
<td>0.20</td>
<td>340.00</td>
<td>63.5</td>
</tr>
<tr>
<td>0.18</td>
<td>305.00</td>
<td>65.2</td>
</tr>
<tr>
<td>0.15</td>
<td>255.00</td>
<td>67.0</td>
</tr>
<tr>
<td>0.10</td>
<td>170.00</td>
<td>70.0</td>
</tr>
<tr>
<td>0.05</td>
<td>85.00</td>
<td>72.9</td>
</tr>
</tbody>
</table>

AB: Distance between skidrails
PR: percentage of radiation incident on plane AB which reaches bottom surface of slab.

According to the computer predictions, another potential way to alleviate the shielding effect of the skidrail is to reduce its height. Table 6.2 lists the proportion of incident heat reaching the slab bottom surface as a function of the height of the skidrail. It is seen that reducing the height of the skidrail offers more potential to reduce the shielding effect than changing its width. In addition, variation of the height of the skidrail is not as restricted by the thickness of insulation layer. The current skidrail uses double tubes inside the skidrail and the height is 475 mm. From the computer results, about 30% of radiation is directly blocked by this height.

Since the current system uses double tubes (each of diameter 145 mm) inside the skidrail, potential exists for converting the double-tube skidrail to single-tube which will increase direct radiation from the furnace atmosphere significantly. This change would reduce...
Table 6.2 Computer Prediction of the Shielding Effect of Skidrail as a Function of its Height H

<table>
<thead>
<tr>
<th>Height (mm)</th>
<th>PR %</th>
</tr>
</thead>
<tbody>
<tr>
<td>500.0</td>
<td>65.6</td>
</tr>
<tr>
<td>475.0</td>
<td>66.7</td>
</tr>
<tr>
<td>450.0</td>
<td>67.7</td>
</tr>
<tr>
<td>425.0</td>
<td>68.8</td>
</tr>
<tr>
<td>400.0</td>
<td>69.9</td>
</tr>
<tr>
<td>375.0</td>
<td>71.1</td>
</tr>
<tr>
<td>300.0</td>
<td>74.7</td>
</tr>
<tr>
<td>250.0</td>
<td>77.2</td>
</tr>
<tr>
<td>200.0</td>
<td>80.0</td>
</tr>
<tr>
<td>175.0</td>
<td>81.1</td>
</tr>
<tr>
<td>150.0</td>
<td>82.5</td>
</tr>
<tr>
<td>100.0</td>
<td>85.3</td>
</tr>
<tr>
<td>50.0</td>
<td>88.2</td>
</tr>
</tbody>
</table>

the height of skidrail by almost half which implies that the net gain of direct radiation would be increased from the current 66.7% to 77.2%. This is a significant improvement in the heat-transfer conditions.

Changing from the double-tube to single-tube skidrail system is technically feasible since it only requires the modification of the exit and entrance of the cooling water. However, better insulation is required for the single-tube skidpipe to eliminate the case in which constant vaporization of cooling water could lead to a "burn-down" accident of the skidrail system. Moreover, stronger materials are preferred for the skidpipes in order to support the load of slabs in the furnace.
(2) Provide adequate heat to the bottom surface of the slab

Since the skidmark originates at the bottom surface of the slab, insufficient heat transfer from the bottom furnace chamber (to ensure equal temperature rise between top and bottom of the slab) will aggravate the skidmark effect. Increasing surface temperature difference between the top and bottom of the slab results in the "head-end turn down" phenomenon described at the beginning of Section 6.2. As was observed from the temperature contours of a slab moving toward the exit of the furnace, Fig. 6.2, the low temperature region near the skidmark propagates to the top surface.

Figure 6.18 compares the skidmark temperature and the centreline temperature of a slab. The results clearly indicate that the slab centreline is cooler during reheating until the slab is near the furnace exit. The reason for the difference is that heat received on the bottom surface not only has to reach the skidmark region, but also has to conduct to the cooler central part of the slab. Therefore, more heat should be input to the bottom furnace chamber.

(3) Provide a more even transverse heat-flux distribution

Most rolling problems arise from a nonuniform temperature distribution in the transverse direction in the furnace (rolling direction of slabs). However, the variation of transverse gas temperature and heat flux has been neglected by previous models which have been based on a one-dimensional gas temperature distribution. From the computer simulation results shown in Fig. 6.13(a), the transverse irradiation flux distribution will be distorted severely if the higher gas temperature concentrates near the refractory side wall. Then the slab end will be subjected to a much higher intensity of radiation than the central area. The temperature of the central area is inevitably more severely depressed, and this in turn,
Fig. 6.18  Comparison of the skidmark and centreline temperature of a slab.
enhances the spread of the low temperature region near the skidmark and leads to an adverse heating situation. In the contour shown in Fig. 6.14, corresponding to the case of higher gas temperatures concentrating near the refractory side wall, the temperature of the skidmark near the centre of the slab is found to be more depressed than near the end. Therefore, a long flame (shifting the main heat release to the central part of the furnace) is preferred if side burners are used. This will have significant impact on the skidmark effect.

(4) Coat the skidrail with highly reflective materials

Because the efficacy of coating highly reflective material on the skidpipes to enhance radiation to the slab bottom surface is debatable, a computer simulation was performed to investigate the effect. A radiative network was established among the slab bottom surface, the exterior surface of the skidrail and the furnace chamber represented by the fictitious surface AB, as was shown in Fig. 5.10. A new radiative electrical analogue for the three components is shown in Fig. 6.19, which removes the assumption made in Chapter 5 that the skidrail is in radiative equilibrium. Of primary interest is net radiative heat flux to the slab bottom surface. Performing an energy balance on each node in the electrical analogue:

For the slab bottom surface

\[
\frac{E_s - W_h}{R_4} + \frac{W_t - W_h}{R_5} + \frac{E_h - W_h}{R_2} = 0 \quad (6.1)
\]

For the exterior surface of the skidrail

\[
\frac{E_t - W_t}{R_1} + \frac{E_s - W_t}{R_3} + \frac{W_h - W_t}{R_5} = 0 \quad (6.2)
\]

Solving Eqs. (6.1) and (6.2) \(W_t\) and \(W_h\) in terms of \(E_s\), \(E_h\) and \(E_t\), the net heat flux to
Fig. 6.19  Radiative network among the slab bottom surface, the skidrail and the furnace bottom chamber.
the slab surface is:

\[ Q_{\text{net} \, \text{to slab}} = \frac{W_h - E_h}{R_2} \]  

(6.3)

Two cases have been studied to determine the effect of modifying the reflectivity of the skidrail exterior surface on the net heat flux to the slab bottom surface. The results are shown in Fig. 6.20. If the temperature of the skidrail, \( T_t \), is higher than the slab bottom surface temperature, \( T_h \), no substantial increase in the net heat flux to the slab bottom surface is observed, which indicates that there is no necessity to coat new material on to the skidrail surface. However, if \( T_t \) is lower than \( T_h \), increasing the reflectivity of the skidrail from 0.1 to 0.9, raises the net heat flux from 32.4 kW/m² to 46.9 kW/m². Often, due to insufficient insulation, the temperature of the skidrail surface is lower than that of the slab bottom surface. Therefore it is recommended to apply high-reflectivity coating materials.

(5) Improve the design of the skidrail

The essence of improving skidrail design is to alter its shapes and configuration to increase the view factor from the furnace chamber to the area of the slab bottom adjacent to the skidrail. When the reflectivity of the exterior surface of the skidrail is fixed, modifying the shape of the skidrail to increase this view factor, either in the case of \( T_t > T_h \) or \( T_h > T_t \), will significantly enhance the radiative heat flux impinging on the slab bottom surface, as shown in Fig. 6.21. Thus, this is a very effective way to reduce the skidmark and confirms the results of previous workers. In a recent study by Lee (25), it has been confirmed experimentally that a teardrop-shaped skidrail is the best design for minimizing skidmarks. The present configuration of skidrail in the Stelco reheating furnace is
Fig. 6.20  Effect of reflectivity of skidrail on the net heat flux to the bottom surface.
Fig. 6.21  Effect of view factor from furnace chamber to skidmark area around the skidmark region.
rather unfavourable with respect to the radiative heat reaching the slab bottom surface, since the skidrail is too wide and too high (double tubes) and has a rectangular shape. Clearly there is considerable scope for improvement in the configuration of skidrail in the LEW reheating furnace. A proposed design of configuration is shown in Fig. 6.22 (sketch).

It should be emphasized that the temperature of the exterior surface of the skidrail, \( T_t \), plays a very significant role in the net heat flux to the slab bottom surface. When \( T_t > T_h \) (bottom slab surface temperature), under any conditions, a substantially higher heat flux is observed than in the case where \( T_h > T_t \). The temperature \( T_t \) depends mainly upon the insulation of the skidrail and increases with better insulation. Because the direct conductive heat loss to the skidpipe is minor compared with the radiative shielding effect, the main function of improving insulation of the skidrail is to increase its surface temperature.

Based on the model prediction, a possible design of an alternative skidrail is shown in Fig. 6.23. The skidrail is offset slightly in the reheating furnace, allowing the area which originally contacts the skidrail to be exposed to the radiative environment so that the temperature distribution in the slab bottom surface will be more even. Implementation of the offset is recommended at the end of the heating zone, where the formation of skidmarks becomes most apparent (as was reported in Fig. 6.2) and the heat flux to the slab surface remains high.

In conclusion, to alleviate the skidmark effect, it is equally important to pay attention to both the design of the skidrail and the appropriate operation of the reheating furnace. A good design of skidrail but improper operation of furnace will also bring severe skidmark problems.
Fig. 6.22  A proposed design of the skidrail system to reduce the skidmark
Fig. 6.23 Possible alternative rail design to reduce skidmark formation.
6.3 Hot Charging Practice

Many advantages accrue from the hot charging of slabs:

(i) Increase the throughput of the reheating furnace. It is quite common that the reheating furnace is the production-constraint process in the rolling mill. If the production rate of the reheating furnace can be increased, the production rate of the whole mill will be increased as well.

(ii) Reduce the residence time of the slab in the reheating furnace and therefore reduce oxidation and scale thickness on the slab surface.

(iii) Reduce energy consumption per ton of slab in the reheating furnaces.

(iv) Reduce the skidmark effect and thermal stress build up in the slab due to a shallower temperature profile.

However, hot charging is not a straightforward practice since there are many problems involved, as was discussed in Chapter 2. In the implementation of hot charging, the corresponding heating strategies and their effect on the thermal behaviour of the slab remain unclear. This section focuses on investigating the thermal effect of hot charging practice using the computer model.

Compared with cold charging slabs, hot charging brings more energy content from the previous process into the reheating furnace. The direct consequence is that the hot charged slab does not require as much heat from the furnace to reach the required rolling temperature. The effective utilization of this physical heat becomes a crucial problem. Evidently, there are two alternatives available. One is to lower the gas temperature of the furnace chamber, so that less fuel input is required and energy conservation is achieved.
The other approach is to increase the production rate, so that the total fuel consumed per unit ton of slab is decreased.

Figure 6.24 shows the top surface temperature response of a hot charged slab compared with a cold charged slab under two different gas temperature profiles. The slab hot charging temperature has been taken as 600 °C, as is commonly encountered in the Stelco reheating furnace. The heating strategy is set such that, for the heating of hot charged slabs, the gas temperature is substantially lowered at the slab entrance side, and slightly reduced at the slab discharge end. The justifications for such a heating strategy are as follows:

(i) In order to maximize the furnace efficiency, the combustion products (or furnace gases) are expected to have as long a residence time as possible to exchange heat with the stock being heated. The fuel input at the furnace discharge end will have longer residence time than that fired at the slab entrance side, where the port of the chimney is located. Therefore, maintaining higher gas temperature at the slab discharge end will increase the time for the thermal exchange between hot gas and the slab.

(ii) Since the hot charged slab has virtually been "preheated", no immediate direct heat is required at the furnace entrance. The fuel input could be proportionally decreased.

Figure 6.24 shows that, even though the furnace gas temperature for the hot charging slab is substantially decreased (200°C lower than the corresponding cold charged slab at the furnace entrance), the exit top slab surface temperature of the hot charged slab remains 20–30 °C higher than that of the cold charged slab.
Fig. 6.24  Predicted slab top surface temperature response for hot and cold charging.
As was discussed in Section 6.1, for the cold charged slab, one of the main heat sinks is the centre of the slab; and heat conduction from the surface to the centre is slow, which directly limits the production rate of the furnace. However, for the hot charged slab, the centre is no longer an important heat sink. From Fig. 6.25, it is observed that the disparity between the centreline and the surface temperature of the hot charged slab is less severe than its counterpart for the cold charged slab. The reason is that the heating rate for the hot charged slab is lower and smoother, because it is already in a high or medium temperature range. Figure 6.26 compares the net heat flux along a furnace between a hot charging and a cold charging case. It shows that the shape of the net heat flux variation remains basically the same, but the area under the hot charging curve is significantly less than for cold charging. This is consistent with the fact that a hot charged slab requires less heat than a cold charged slab inside the furnace to reach the same temperature.

Temperature contours in a discharged cold and hot charged slab are provided in Fig. 6.27. The hot and cold slab heating strategies are the same as those shown in Fig. 6.24. The temperature distribution in the hot charged slab tends to be much more homogeneous than that in the cold charged slab. The difference between the skidmark temperature and the average temperature of the rest of the slab is less than 30°C, which is an improvement for supplying a slab of nearly uniform ductility for rolling.

The production rate was kept constant in the above calculation. However, as was mentioned before, the production rate could be increased for hot charging if the gas temperature level is the same as for cold charging. A rise in production rate is equivalent to a fuel saving. Figure 6.28 compares the discharged slab temperature contours in a cold charged slab and in a hot charged slab under the same gas temperature profile but with different productivities, 200 T/hr and 300 T/hr respectively. The overall slab discharge temperature level is slightly lower for the hot charged slab but the severity of the
Fig. 6.25  Predicted top and centreline temperature response for hot and cold charging
Fig. 6.26  Comparison of longitudinal heat flux to the slab surface for cold and hot charging.
Note: Isotherm values are \( T \times 10^{-3} \) (°C)

(a) Charging temp. = 600°C, productivity = 200T/hr. under improved heating strategy (shown in Fig. 6.24)

(b) Charging temp. = 25°C, productivity = 200T/hr.

Fig. 6.27 Predicted slab temperature contours at the furnace exit for hot and cold charging
Note: Isotherm Values are $T \times 10^{-3}$ ($^\circ$C)

(a) Charging temperature = 600 $^\circ$C, productivity = 300 T/hr.

(b) Charging temperature = 25 $^\circ$C, productivity = 200 T/hr.

Fig. 6.28 Comparison of two predicted slab temperature contours at furnace exit for two productivity.
skidmark effect is basically unchanged.

Thus the off-line computer model is capable of simulating a range of throughputs and hot charging temperatures, and finally of providing the most desirable production rate for a given set of conditions.
7. CONCLUSIONS

In striving towards the objective of understanding the thermal behaviour of reheating furnaces and the formation of skidmarks in slabs, a comprehensive computer simulation model has been developed capable of predicting the temperature and heat-flux distribution to the slab and the refractory wall for various operating conditions. The computer simulation model predicts the following:

(1) The characteristic heat-flux distribution to the slab surface is determined largely by the gas temperature profile in the furnace. The investigation of transverse heat flux, which could not be undertaken by previous one-dimensional furnace chamber models, shows that a nonuniform gas temperature distribution across the furnace chamber is mainly responsible for the distortion of heat flux in this direction. In particular, when the high temperature gases are concentrated near the refractory side wall, a severe distortion of transverse heat flux is likely to occur, which will lead to adverse heating of the slab. Therefore, the centre of heat release should be shifted toward the central part of the furnace.

(2) The formation of skidmarks has many important characteristics:

(a) The skidmark effect is not merely a local phenomenon. Its influence penetrates to the top surface of the slab and the entire slab temperature is more or less affected. The study clearly shows that the radiative shielding of skidrails is a dominant factor causing the heat deficit near the slab/skidrail contact region. In order to alleviate the skidmark effect, emphasis should be directed to reducing the skidrail size and improving the configuration.

(b) The effectiveness of coating reflective materials on the exterior surface of
the skidrail to reduce skidmarks depends on the temperature difference between the local bottom slab surface, $T_h$, and the exterior surface of the skidrail, $T_t$. If $T_h > T_t$, an increase of reflectivity of the skidrail would be favourable to reduce the skidmark effect. If $T_t > T_h$, increasing the reflectivity does not enhance the heat transfer in this region. Generally the skidrail is cooler than the slab bottom surface temperature, thus coating of high reflectivity materials would alleviate the skidmark effect.

(c) The benefit from improved insulation for the skidrail system is primarily due to increase the exterior surface temperature of the skidrail. This reduces the absorption of heat originating from the slab bottom surface by the skidrail. The reduction in heat loss across the slab/skidrail contact is of secondary importance.

(d) The modification of the current skidrail geometry to reduce the shielding effect to a minimum is crucial to alleviate the present severe skidmark problem. An alternative skidrail design and a single-pipe skidrail system have been proposed based on the results of the computer simulation.

(3) A comparison between the radiative contributions from the gas and from the refractory wall to the slab surface has indicated that the latter plays a significant role in the heat flux to the slab surface. The model results suggest that a more uniform transverse heat-flux distribution across the furnace should result from the use of open radiation tube. 

(4) For the case of hot charging slabs, the required gas temperature at the charge end can be lowered significantly compared with that for cold charging slabs; and fuel savings can be achieved by this heating strategy.

(5) From the slab heating profiles, it has been found that the heating rate, or the
production rate, is limited by heat conduction from the surface to the centre of the slab.

The computer simulation provides a flexible off-line model to monitor the thermal behaviour of a reheating furnace. Further work is needed to combine the current model with a model of gas flow and combustion inside the furnace chamber. Optimization of the model, and provision of on-line computer control of the reheating processes are other areas for future work.
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Appendix I  COMPUTER FLOW CHART FOR THE REAL GAS TREATMENT

START

Select $T_k$

- INPUT $e_{g,i}$, $(pL)_i$
  $i=1, 2, \ldots, n$

linear regression of

\[
\ln(e_{g,i} - \sum r_{g,i}) = \ln r_{g,1} - k_1pL_i
\]

iteration of linear regression

\[
\ln(e_{g,i} - r_{g,i} - a_1e^{-k_1pL_i}) = \ln r_{g,i} - k_2pL_i
\]

Repeat n times if there are n terms of exponential form

Note. It is an iterative process to select an optimum $\sum_{i} r_{g,i}$

INPUT $(pL)_K$, $T_g$, $e_{g,j}$

$j=1, 2, \ldots, m$, $k \geq 2$
At $T_j$, solve $r_{g,1}(T_j)$, $r_{g,2}(T_j)$ by an algebra equation.

CONTINUE

linear regression to fit $r_{g,1}(T_j)$, $r_{g,2}(T_j)$, ...

$I = 1, N$

$J = 1, M$

$A_1(T_i, T_j, pL_{k})$

$A_2(T_i, T_j, pL_{k1})$

linear algebra equation

solving for $a_k(T_S, T_G)$, $a_k(T_S, T_G)$

END
Appendix II. Calculation of the View Factor from the Furnace Chamber to the Slab Bottom Surface

\[ \text{Arc } AC = \frac{\pi}{4} \]
\[ CD = \sqrt{(h-(d/2))^2 + (s-d)^2} \]
\[ EB = AD \]
\[ AD = \left( \frac{\pi}{4}d \right) + CD \]

According to symmetry:
\[ EB = AD \]
\[ AE = \left( \frac{\pi}{4}d \right) + (h - d/2) \]
\[ BD = AE \]

According to string method, the view factor from the bottom chamber to the slab bottom surface is:
\[ F_{AB-ED} = \frac{(AD + EB - AE - BD)}{(2xAB)} \]

Finally
\[ \phi = \frac{\sqrt{(h-(d/2))^2 + (s-d)^2 - (h - d/2)}}{s} \]
Appendix III. FLOW CHART OF THE CODE "ENERGY"

1. **GAS EMISSIVITY AND ABSORPTIVITY TREATMENT**
2. **SELECT DIFFERENT \( k_j \) AND \( a_j \) AT VARIOUS TEMPERATURE**
3. **INPUT THE GEOMETRY OF THE FURNACE CHAMBER AND PHYSICAL PROPERTIES**
   - **DIRECT EXCHANGE COEFFICIENTS CALCULATION**
   - **TOTAL EXCHANGE COEFFICIENTS CALCULATION**
4. **INPUT GAS TEMPERATURE AND ASSUMED REFRACTORY WALL AND SLAB TEMPERATURE**
5. **CALCULATE \( S_S^j \), \( G_S^j \)**

The flow chart outlines the process for calculating the energy distribution and exchange coefficients in a furnace chamber, considering the geometrical and physical properties of the system.
PERFORM ENERGY BALANCE ON REFRACCTORY WALL

SOLVE THE TEMP. DISTRIBUTION OF THE REFRACCTORY WALL USING QUASI-NEWTONIAN TECHNIQUE

OBTAIN HEAT FLUX ON THE SLAB SURFACE ZONE

2-D UNSTEADY CONDUCTION MODEL

COMPARE THE CALCULATED TEMPERATURE WITH THE ASSUMING PROFILE

YES

THE FINAL SOLUTION CONVERGES

STOP

NO