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A Mathematical Model of a Horizontal Direct-Fired Strip Annealing Furnace

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Abstract: A tractable mathematical model of a horizontal direct-fired furnace is presented. The first-principles model captures the essential non-linearities of the furnace and includes submodels for the flue gas, the rolls, the wall, and the strip. These submodels are interconnected by the heat transfer mechanisms conduction, convection, and radiation. For both the identification of unknown parameters and the validation of the developed model, measurement data from the considered furnace is used. The model is computationally inexpensive and thus suitable for model-based control and real-time optimization.

Keywords: Steel industry, direct-fired strip annealing furnace, mathematical model, Galerkin method, variable strip emissivity, leakage air flows, control

1. INTRODUCTION

In the steel industry, strip annealing furnaces are used for the heat treatment of steel products. In this paper, a horizontal direct-fired counter-flow strip annealing furnace for stainless steel strips is analyzed. In order to achieve the desired product quality, the strip has to be heated to a predefined target temperature. In particular in transient operating situations, strip temperature control is a challenging task. For this reason, efficient model-based control concepts are increasingly used (Banerjee et al., 2004).

In the current paper, a simple mathematical furnace model is developed that is computationally inexpensive and that captures the essential non-linearities of the furnace. This model can serve as a basis for model-based control and real-time optimization.

The furnace considered in this analysis is outlined in Fig. 1. It consists of four chambers, each with a length of approximately 20 m. Between the chambers, water-cooled rolls are installed for conveying the strip through the furnace. In chamber 1, the strip is preheated by means of the hot flue gas from the combustion of fuel in the chambers 2, 3 and 4, the so-called heating chambers. They are divided into several heating zones, each equipped with a set of burners. In each heating zone, the combustion is controlled to be fuel lean. The resulting flue gas thus contains no harmful products. Due to a pressure gradient, the flue gas streams in the opposite direction of the strip movement and leaves the furnace through a funnel. Inside the furnace, the strip is heated by radiation and convection. To keep the heat losses through the furnace wall as small as possible, it consists of several layers of insulation.

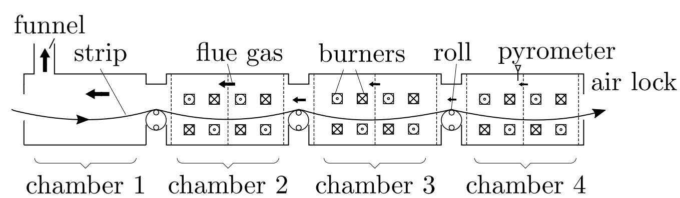

Fig. 1. Horizontal direct-fired annealing furnace.

In literature, many different approaches can be found for modeling a strip annealing furnace. One possible approach is to use numerical methods like computational fluid dynamics (Depree et al., 2010). Such models are characterized by a high accuracy in terms of modeling the dynamical furnace behavior. However, the mathematical complexity is usually very high and these models are therefore not suitable for real-time control.

In Yoshitani (1993), a semi-analytical model based on physical principles and measured system dynamics is derived. The model consists of a dynamical and a static part. The dynamical model, which is most important for control, is derived based on heat balance equations whereas the statical model is based on a curve-fitting-approximation of the strip temperature. This approach leads to a simple model that can be executed in real-time. However, the physical interpretation of the occurring parameters is difficult. Moreover, the proposed model does not include the flue gas.

Another possibility to ensure real-time capability is to use linearized models, see, e.g., (Depree et al., 2010). Linearized state space models typically neglect the non-linear effects of the material parameters and of the radiative heat transfer.

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In Depree et al. (2010), the finite difference method is used for the spatial discretization of the heat conduction problem. An alternative approach is the *weighted residual method* (Bathe, 2002; Fletcher, 1984), which is explained in more detail in Sec. 2.2 and Sec. 2.3.

In Strommer et al. (2014), a mathematical model of a direct-fired vertical annealing furnace is presented. However, this model cannot be directly transferred to the furnace considered here for the following reasons:

- Since the rolls in the considered furnace are water-cooled, they have a significant influence on the strip temperature. For this reason, they are modeled in more detail than in Strommer et al. (2014).
- The contact area between the strip and the rolls is a function of the wrap angle. Between the rolls, the strip is modeled as a catenary.
- The combustion is assumed to take place within the furnace chamber and not right after the nozzle. Moreover, the combustion is controlled to be fuel-lean.
- To capture the dynamic behavior of the furnace wall, different trial functions are used.
- Due to the scale formation inside the furnace, the surface conditions of the strip change while travelling through the furnace. Thus, the strip emissivity depends on the spatial position.
- An additional air input at the air lock of the furnace affects the composition of the flue gas inside the furnace. This air input is modeled by a *Couette*-flow (Munson et al., 2009).

The paper is structured as follows: In Section 2, the mathematical model of the considered furnace is developed. The model finally consists of submodels for the flue gas, the rolls, the wall, and the strip. The thermal interaction of these submodels is described by conduction, convection, and radiation. In Section 3, parameters are identified that concern the unknown strip emissivity and the additional air flow into the furnace. In Section 4, the accuracy of the model is verified by means of measurement data from a real plant.

2. MATHEMATICAL MODEL

2.1 Flue gas

In the considered furnace, all burners of a single heating zone are supplied with the same mass flow of fuel in the form of CH_4. The required combustion air depends on the mass flow of fuel and the excess air coefficient λ. Since the combustion of fuel is controlled to be fuel-lean, i.e. $\lambda > 1$, the flue gas only contains carbon dioxide, hydrogen, oxygen, and nitrogen. Abbreviations of these components are summarized in the set $S_\nu = \{CO_2, H_2O, O_2, N_2\}$.

For modeling the flue gas, the furnace is divided into N_v volume zones, where each volume zone i is considered as a well-stirred reactor with a constant flue gas temperature $T_{v,i}$. In view of a computationally efficient model for the strip temperature, the flue gas dynamics are neglected, i.e. they are considered quasi stationary. This assumption can be verified by means of the singular perturbation theory (Kokotovic et al., 1999).

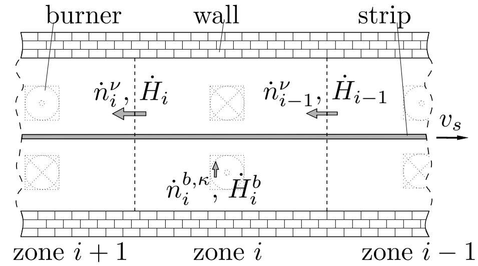

Fig. 2. Molar and enthalpy flows of a volume zone i.

For the following, let

$$\dot{n}^\nu = \dot{M}^\nu / \overline{M}^\nu, \tag{1}$$

with the mass flow $\dot{M}^\nu$ and the molar mass $\overline{M}^\nu$, be the molar flow of a component $\nu \in S_\nu \cup S_\kappa$, where $S_\kappa = \{CH_4, N_2, O_2\}$. In Fig. 2, the molar and enthalpy flows of an individual volume zone i are shown. Here, $\dot{n}_{i-1}^\nu$ and $\dot{n}_i^\nu$ denote the molar flows of a component $\nu \in S_\nu$ that enter and leave the volume zone i, respectively. The additional molar flow $\dot{n}_i^{b,\kappa}$, $\kappa \in S_\kappa$, enters the volume zone through the burners, where the relations $\dot{n}_i^{b,O_2} = 2\lambda \dot{n}_i^{b,CH_4}$ and $\dot{n}_i^{b,N_2} = 7.52\lambda \dot{n}_i^{b,CH_4}$ holds. Extra air, i.e. $\dot{n}_0^{O_2}$ and $\dot{n}_0^{N_2}$, enters the volume zone 1 through the air lock, see Fig. 1. The stationary reaction equation for the volume zone i reads as

$$\begin{aligned}&\dot{n}_{i-1}^{CO_2} CO_2 + \dot{n}_{i-1}^{H_2O} H_2O + \dot{n}_{i-1}^{O_2} O_2 + \dot{n}_{i-1}^{N_2} N_2 + \\ &\dot{n}_i^{b,CH_4} \left(CH_4 + 2\lambda_i O_2 + 7.52\lambda_i N_2\right) \xrightarrow{\lambda_i > 1} \dot{n}_i^{CO_2} CO_2 + \\ &\qquad \dot{n}_i^{H_2O} H_2O + \dot{n}_i^{O_2} O_2 + \dot{n}_i^{N_2} N_2,\end{aligned} \tag{2}$$

with the unknown molar flows $\dot{n}_{i-1}^\nu$ and $\dot{n}_i^\nu$, $\nu \in S_\nu$. By using simple mole balances, the molar flow $\dot{n}_i^\nu$ can thus be determined as a function of the unknown molar flow $\dot{n}_{i-1}^\nu$ and the known molar flows $\dot{n}_i^{b,\kappa}$, $\kappa \in S_\kappa$. If all volume zones $i = 1, \ldots, N_v$ are considered, a linear set of equations for determining the $4N_v$ unknowns $\dot{n}_i^\nu \in S_\nu$ is obtained.

For determining the flue gas temperatures, the stationary enthalpy balance, cf. Fig. 2,

$$0 = \dot{H}_{i-1} + \dot{H}_i^b - \dot{H}_i + \dot{Q}_{v,i} \tag{3}$$

for each volume zone $i \in \{1, \ldots, N_v\}$ is considered. Here, $\dot{H}_{i-1}$ denotes the incoming enthalpy flow from the upstream volume zone $i-1$, and $\dot{H}_i^b$ is the incoming enthalpy flow associated with the fuel and combustion air. Moreover, $\dot{H}_i$ represents the outgoing enthalpy flow and $\dot{Q}_{v,i}$ summarizes the net radiative and convective heat flows from the environment to the flue gas.

Let $h^\nu(T)$ be the specific enthalpy of a component $\nu \in S_\nu \cup S_\kappa$, see Poling et al. (2004). An enthalpy flow $\dot{H}(T)$ can therefore be defined as $\dot{H}(T) = \sum_{\nu \in S_\nu \cup S_\kappa} \dot{n}^\nu \overline{M}^\nu h^\nu(T)$. Thus, the non-linear set of algebraic equations for determining the flue gas temperatures $T_{v,i}$, cf. (3), results in

$$\begin{aligned}0 = &\sum_{\nu \in S_\nu} \dot{n}_{i-1}^\nu \overline{M}^\nu h^\nu(T_{v,i-1}) + \sum_{\kappa \in S_\kappa} \dot{n}_i^{b,\kappa} \overline{M}^\kappa h^\kappa(T_i^{b,\kappa}) \\ &- \sum_{\nu \in S_\nu} \dot{n}_i^\nu \overline{M}^\nu h^\nu(T_{v,i}) + \dot{Q}_{v,i}, \; i = 1, \ldots, N_v,\end{aligned} \tag{4}$$

where $T_i^{b,\kappa}$, $\kappa \in S_\kappa$, represents the known temperatures of the fuel and the combustion air.