

Control of radiant tubes in an indirect-fired strip annealing furnace for improved efficiency

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Abstract—The possibilities for improving the efficiency of an indirect-fired strip annealing furnace with radiant tubes are explored. First, the influence of the fuel/air-ratio on the efficiency is analyzed by means of linearized relationships. Second, the optimum operating range of an individual radiant tube and a new control strategy for the fuel supply are derived. Both analyses are based on the balance model of a single radiant tube. The input data were taken from an experimental measurement campaign of an industrial furnace.

Throughput of steel	45,7 t/h
Nominal heating power	10 MW
Strip dimensions	
Thickness	0.35 – 1.2 mm
Width	800 – 1 640 mm
Strip velocities	max. 180 m/min
Number of radiant tubes	62
Number of control zones	3

TABLE I

NOMINAL PARAMETERS OF THE INDIRECT-FIRED FURNACE.

I. INTRODUCTION

The manufacturing and processing of steel consumes large amounts of fossil fuels, which are at the same time important cost drivers. Because of rising energy prices, the incentive to increase the efficiency by new equipment or improvement of the existing equipment remains unbroken.

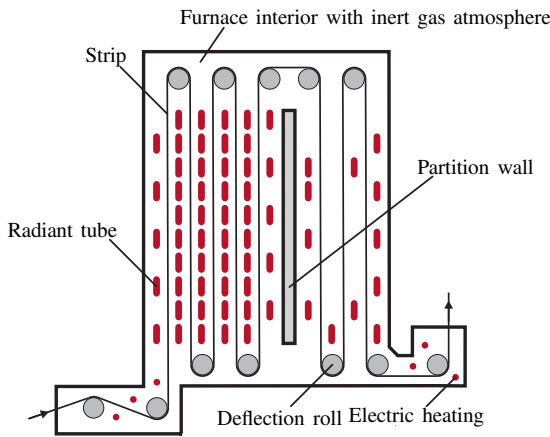


Fig. 1. Indirect-fired furnace.

In the current paper, potentials for improving the efficiency of an indirect-fired furnace, which is shown in Fig. 1, are identified and quantitatively analyzed. The focus of research are control-related possibilities, which do not require changes of the (existing) equipment. The considered furnace is part of a hot-dip galvanizing line of voestalpine stahl GmbH and is used for continuous annealing of steel strip. Some nominal parameters of the considered furnace are tabulated in Table I.

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The processed strip is conveyed through the furnace along a meander-shaped path by means of deflection rolls. To shield the strip from oxidation, an inert gas atmosphere is realized within the furnace. The strip is heated by radiant tubes which are grouped into individual control zones. All radiant tubes of a control zone are supplied with the same amounts of fuel and combustion air. As shown in Fig. 2, the main components of a radiant tube are the burner, the tube, and the local recuperator for preheating the combustion air. Moreover,

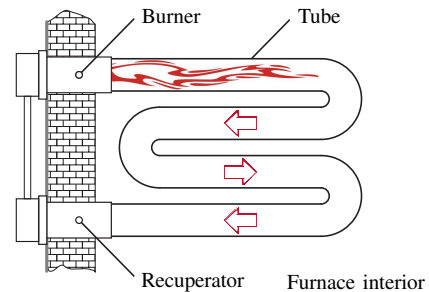


Fig. 2. W-shaped radiant tube.

the burners are operated in a continuous control mode meaning that the fuel supply to the burners can be adjusted continuously between a minimum and maximum value. The supply of combustion air depends on the fuel supply and on the excess air coefficient [1]. In the current furnace control strategy, the excess air coefficient is selected according to a user-defined setpoint curve that is parameterized with the fuel flow, see Fig. 3. For small fuel flows, an extra amount of combustion air is supplied to avoid local overheating of the radiant tube, which might otherwise occur due to the too low flow velocity.

A comprehensive overview of possibilities of energy savings in fuel-fired heating processes is given in [2]. According to this analysis, the highest potential for maximum efficiency lies in the improvement of the combustion equipment and the

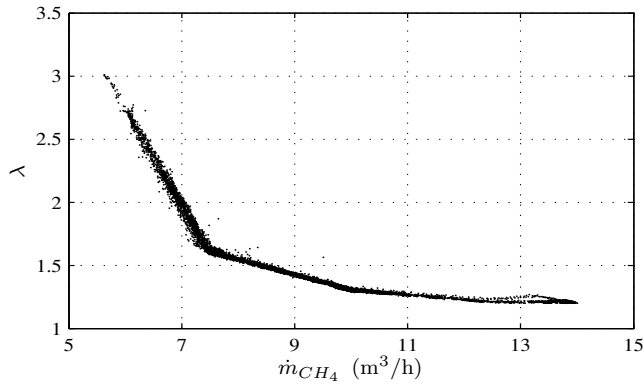


Fig. 3. Measured setpoint curve for the excess air coefficient λ as a function of the fuel flow \dot{m}_{CH_4} .

utilization of the energy lost with the exhaust gas. Apart from the desired increase in efficiency, another important objective is the reduction of NO_x emissions.

Basically, radiant tube systems can be categorized as recirculating and non-recirculating [3], [4]. These two systems differ in their temperature uniformity, their ability to recover energy contained in the exhaust gas, and their NO_x emissions. For this reason, the two systems have different potentials for increasing the efficiency.

Generally, the temperature uniformity of a radiant tube is an important criterion for its lifetime because local overheating can cause tube failures due to high thermal stress or by burning holes into the tube. Moreover, uniform tube temperatures are desirable to achieve a homogeneous and accurate temperature field in the strip [3], [5]. In non-recirculating radiant tubes, the flame typically extends over the first tube leg (cf. Fig. 2). The temperature uniformity is thus rather poor in this type of radiant tube. Recirculating systems achieve a more uniform temperature by circulation of the exhaust gas.

In general, a (local) recuperator or a regenerator is used to improve the efficiency of a radiant tube by utilizing the exhaust gas [3], [4], [5]. The main idea of both heat recovery systems is to use the energy of the exhaust gas for preheating the combustion air. Thus, less fuel is needed since less energy is required for heating the combustion air. Typical recuperators are counterflow heat exchangers, where the hot flue gas transfers heat to the combustion air. Regenerators, however, transfer heat to the combustion air by means of intermediate storage in a porous storage block. The advantage of the regenerator is the hotter temperature of the combustion air.

As outlined in [6], [7], [8], another possibility to increase the efficiency is the usage of oxygen-enriched air or pure oxygen. In this case, less or no nitrogen has to be heated. Moreover, combustion of pure oxygen avoids NO_x emissions due to the absence of nitrogen.

In general, a higher combustion temperature leads to higher NO_x emissions. Reduction of these emissions and simultaneous realization of a highly efficient combustion can be achieved by the so-called flameless oxidation [9].

These approaches for improving the efficiency of indirect-fired furnaces require more or less significant changes of the hardware. However, an improvement in efficiency can also be achieved by a modern control strategy for the desired strip temperature [10], [11], [12]. If the strip temperature is controlled more accurately, the production rate can be increased by higher line speed. Moreover, the desired final temperature can be reduced, i. e., the safety margin between the controlled final strip temperature and the temperature at least required for subsequent process steps can be smaller.

In contrast to the published approaches, which often require costly revamping, this paper explores the potential for increasing the efficiency by improved control strategies for the radiant tubes. Mainly the influences of the excess air coefficient and the operating range of the radiant tubes on the fuel consumption are analyzed. A basic assumption of the current analysis is that the heat input from the radiant tubes into the furnace is invariant with respect to the proposed changes of the control strategy. Hence, there is only a change inside the radiant tube whereas all other temperatures and heat flows within the furnace chamber (outside of the radiant tubes) remain unchanged. For this analysis it is thus sufficient to consider a single radiant tube.

II. BALANCE MODEL OF A RADIANT TUBE

In the considered furnace, the burners are operated with natural gas, which is more or less pure methane (CH_4). The heat released by the combustion process inside a single W-shaped radiant tube (cf. Fig. 2) is either transferred through thermal conduction in the wall of the radiant tube into the furnace chamber or it is lost in the form of sensible heat of the exhaust gas leaving the radiant tube after the recuperator. The flue gas consists mainly of carbon dioxide, oxygen, water, and nitrogen and leaves the radiant tube after passing the recuperator. Abbreviations for the flue gas components are summarized in the set $S_\nu = \{CO_2, O_2, H_2O, N_2\}$. As indicated in Fig. 4, the radiant tube is considered as a closed system with input and output quantities.

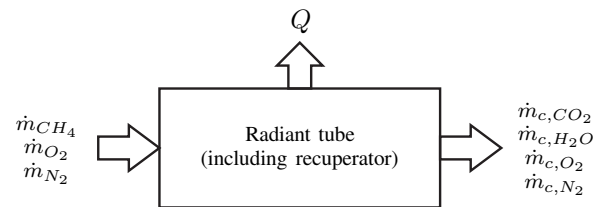
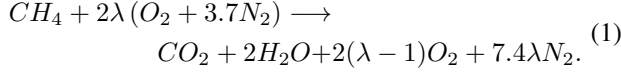


Fig. 4. Flow diagram of a radiant tube.

The input quantities are the fuel flow \dot{m}_{CH_4} and the mass flow of oxygen \dot{m}_{O_2} and nitrogen \dot{m}_{N_2} , i. e., the combustion air. Furthermore, the output quantities are the mass flow of exhaust gas \dot{m}_c and the heat input Q into the wall of the radiant tube. This heat input Q is transferred through the wall by means of thermal conduction and is of major interest for this analysis.

The combustion inside the radiant tube is controlled to be fuel-lean (excess air coefficient $\lambda > 1$), which justifies the

assumption of a complete combustion. Hence, there is no natural gas left in the exhaust gas. Since the time constants of the combustion are considerably smaller than those of the remaining system, i. e., the heat conduction through the wall of the radiant tube, the combustion within the radiant tube is considered as instantaneous. Therefore, the corresponding reaction equation reads as



Consequently, the incoming mass flows are coupled by

$$\dot{m}_{O_2} = 2\lambda \frac{M_{O_2}}{M_{CH_4}} \dot{m}_{CH_4} \quad (2a)$$

$$\dot{m}_{N_2} = 7.4\lambda \frac{M_{N_2}}{M_{CH_4}} \dot{m}_{CH_4}, \quad (2b)$$

where M_κ is the molar mass of the component κ with $\kappa \in S_\kappa = \{CH_4, O_2, N_2\}$. Based on (1), the outgoing mass flows are defined as

$$\dot{m}_{c,CO_2} = \frac{M_{CO_2}}{M_{CH_4}} \dot{m}_{CH_4} \quad (3a)$$

$$\dot{m}_{c,H_2O} = 2 \frac{M_{H_2O}}{M_{CH_4}} \dot{m}_{CH_4} \quad (3b)$$

$$\dot{m}_{c,O_2} = 2(\lambda - 1) \frac{M_{O_2}}{M_{CH_4}} \dot{m}_{CH_4} \quad (3c)$$

$$\dot{m}_{c,N_2} = 7.4\lambda \frac{M_{N_2}}{M_{CH_4}} \dot{m}_{CH_4} = \dot{m}_{N_2}. \quad (3d)$$

Finally, the heat input Q follows from the energy balance [13] in the form

$$Q = \underbrace{\sum_{\kappa \in S_\kappa} \frac{\dot{m}_\kappa}{M_\kappa} H_\kappa^0 - \sum_{\nu \in S_\nu} \frac{\dot{m}_{c,\nu}}{M_\nu} H_\nu^0}_{\text{Latent heat}} + \underbrace{\sum_{\kappa \in S_\kappa} \frac{\dot{m}_\kappa}{M_\kappa} h_\kappa(T_\kappa) - \sum_{\nu \in S_\nu} \frac{\dot{m}_{c,\nu}}{M_\nu} h_\nu(T_c)}_{\text{Sensible heat}} \quad (4)$$

where $h_j(T)$ and H_j^0 with $j \in S_\kappa \cup S_\nu$ describe the specific enthalpy per mol and the enthalpy of reaction per mol, respectively. H_j^0 can be calculated from tabulated values [14], [15]. T_{CH_4} and $T_{O_2} = T_{N_2}$ are the temperatures of the fuel and the combustion air, respectively. Furthermore, T_c describes the temperature of the exhaust gas that leaves the radiant tube after the recuperator.

In normal operation, the quantities \dot{m}_{CH_4} , \dot{m}_{O_2} , \dot{m}_{N_2} and thus $\dot{m}_c = \sum_{\nu \in S_\nu} \dot{m}_{c,\nu} = \dot{m}_{CH_4} + \dot{m}_{O_2} + \dot{m}_{N_2}$ are known. Furthermore, the temperatures of the supplied fuel and combustion air are also available. However, the temperature T_c of the exhaust gas after the recuperator would also need to be known for an exact calculation of Q , cf. (4). Lacking a permanent measurement of the exhaust gas temperature T_c , this temperature was temporarily measured in an experimental measurement campaign. In the experiment, T_c was measured by means of a thermocouple directly after the recuperator. Hence, the heat input Q , which could not be

measured directly, can be determined at least for the duration of the measurement campaign. Note that in normal operation, i. e., without a measurement of T_c , the heat input Q cannot be precisely computed by the current mathematical model because there is no analytical description for the exhaust gas temperature.

III. INFLUENCE OF THE FUEL-AIR RATIO

In the next step, the balance model (4) of an individual radiant tube is used to analyze the influence of the excess air coefficient λ on the efficiency and the fuel supply. Since the fuel flow \dot{m}_{CH_4} appears as a factor in all terms on the right-hand side of (4), the specific heat input

$$\hat{Q} = \frac{Q}{\dot{m}_{CH_4}} = h_L - h_S(\lambda, T_c) \quad (5a)$$

with the abbreviations

$$h_L = \frac{1}{M_{CH_4}} (H_{CH_4}^0 + 2H_{O_2}^0 - H_{CO_2}^0 - 2H_{H_2O}^0) \quad (5b)$$

and

$$h_S(\lambda, T_c) = \frac{1}{M_{CH_4}} \sum_{\nu \in S_\nu} \zeta_\nu(\lambda) h_\nu(T_c) - \frac{1}{M_{CH_4}} \sum_{\kappa \in S_\kappa} \zeta_\kappa(\lambda) h_\kappa(T_\kappa) \quad (5c)$$

can be introduced. Here, the quantities ζ_ν and ζ_κ are determined by stoichiometry in the form $(\nu, \zeta_\nu) \in \{(CO_2, 1), (H_2O, 2), (O_2, 2(\lambda - 1)), (N_2, 7.4\lambda)\}$ and $(\kappa, \zeta_\kappa) \in \{(CH_4, 1), (O_2, 2\lambda), (N_2, 7.4\lambda)\}$.

The total efficiency of a single radiant tube is defined as

$$\eta = \frac{Q}{\dot{m}_{CH_4} h_L} = \frac{\hat{Q}}{h_L}, \quad (6)$$

were Q is the heat input according to (4) and $\dot{m}_{CH_4} h_L$ denotes the primary energy input from the fuel. It follows from (5a) that \hat{Q} is determined by the excess air coefficient λ and the exhaust gas temperature T_c . The current control strategy of the radiant tube selects λ according to the stored setpoint curve $\lambda(\dot{m}_{CH_4})$ of Fig. 3. The inlet temperatures T_κ with $\kappa \in S_\kappa$ are assumed to be constant. In general, the exhaust gas temperature T_c depends on \dot{m}_{CH_4} , λ , the flow conditions inside the radiant tube and the recuperator, and their temperature states. Hence, there is a weak influence of the furnace on T_c by the temperature state of the radiant tube wall. In the following, it is examined whether this weak influence can be neglected in the model without compromising the accuracy. For this purpose, the sensitivity of \hat{Q} with respect to λ and T_c is analyzed. Consider the linearization of (5a)

$$\Delta \hat{Q} = -(S_\lambda \Delta \lambda + S_T \Delta T_c) \quad (7a)$$

with

$$S_\lambda(\lambda, T_c) = \frac{\partial h_S(\lambda, T_c)}{\partial \lambda} > 0 \quad (7b)$$

$$S_T(\lambda, T_c) = \frac{\partial h_S(\lambda, T_c)}{\partial T_c} > 0 \quad (7c)$$

for small variations $\Delta\lambda$ and ΔT_c from the operating point defined by λ and T_c . The inequalities (7b) and (7c) follow directly from an evaluation of (5c).

In the next step, the influence of both terms on the right-hand side of (7a) is will be investigated. S_λ and S_T can be easily calculated for every operating point. However, typical values for the variations $\Delta\lambda$ and ΔT_c are basically unknown. Therefore, the standard deviations of the measurement data λ and T_c are used as realistic values for $\Delta\lambda$ and ΔT_c . These standard deviations are calculated separately for time slots with a length of about 1min. The time slots has been chosen small enough so that within each time slot the changes of the signals are small. Hence, it is justified to use the linearized relation (7a) with S_λ and S_T evaluated at the respective operating point of each time slot (temporal mean value). By substituting these values into (7a), the relative influence

$$-\frac{S_\lambda \Delta\lambda}{\Delta\hat{Q}} = 1 + \frac{S_T \Delta T_c}{\Delta\hat{Q}}$$

of the excess air coefficient λ on \hat{Q} can be derived, see Fig. 5. Clearly, Fig. 5 shows the dominant influence of λ on \hat{Q} , i. e., the larger part of the variability of \hat{Q} can be explained by the variability of λ . On average, the relative influence is

$$\text{mean}_t \left\{ -\frac{S_\lambda \Delta\lambda}{\Delta\hat{Q}} \right\} = 91\%.$$

This justifies to analyze the efficiency of the radiant tube as a function of λ only, i. e., the (unknown) variation of the exhaust gas temperature T_c will be ignored.

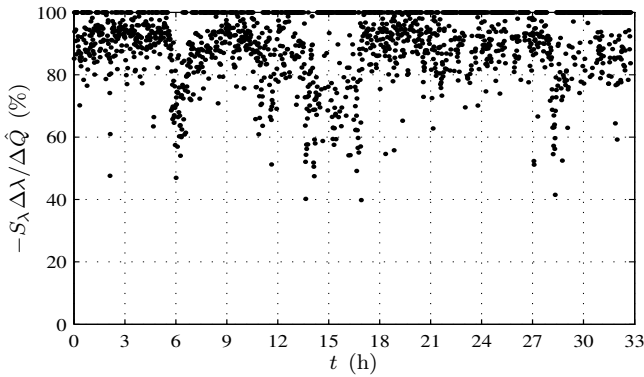


Fig. 5. Influence of the excess air coefficient λ on \hat{Q} .

In the following, the influence of λ on the efficiency η and thus on the fuel flow \dot{m}_{CH_4} is computed by means of (4), (5a), and (6). As it is expected, Fig. 6 shows that a reduction of λ has a noteworthy positive effect on the efficiency η .

A key assumption for the next steps is that, if the excess air varies, the heat input Q remains unchanged by adjusting the fuel flow \dot{m}_{CH_4} . Thus, the necessary change $\Delta\dot{m}_{CH_4}$ of the fuel supply for a unchanged heat input Q can be calculated from (cf. (5a))

$$\frac{Q}{\dot{m}_{CH_4} + \Delta\dot{m}_{CH_4}} = h_L - h_S(\lambda + \Delta\lambda, T_c) \quad (8)$$

$$= (\eta + \Delta\eta)h_L.$$

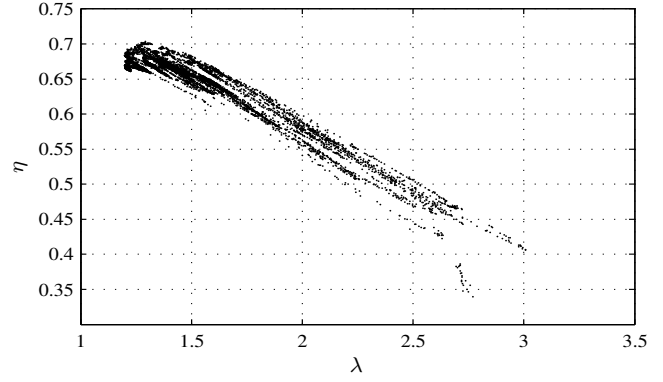


Fig. 6. Efficiency η of the radiant tube.

Here, the above result that ΔT_c does not have a significant impact on \hat{Q} is essentially used. Next, elimination of the (unchanged) heat input Q in (5a) and (8) and the approximation

$$1 + \frac{\Delta\eta}{\eta} = \frac{1}{1 + \frac{\Delta\dot{m}_{CH_4}}{\dot{m}_{CH_4}}} \approx 1 - \frac{\Delta\dot{m}_{CH_4}}{\dot{m}_{CH_4}},$$

which is valid for $|\Delta\dot{m}_{CH_4}| \ll \dot{m}_{CH_4}$, yields the relative change

$$\frac{\Delta\dot{m}_{CH_4}}{\dot{m}_{CH_4}} = \frac{S_\lambda(\lambda, T_c)\Delta\lambda}{h_L - h_S(\lambda, T_c)} = \frac{S_\lambda(\lambda, T_c)\lambda}{h_L - h_S(\lambda, T_c)} \frac{\Delta\lambda}{\lambda} = -\frac{\Delta\eta}{\eta} \quad (9)$$

for the fuel supply. S_λ is defined in (7b). Note that h_S increases with λ for a constant temperature T_c according to (5c). Consequently, the elasticity

$$\frac{\frac{\Delta\eta}{\eta}}{\frac{\Delta\lambda}{\lambda}} = -\frac{S_\lambda(\lambda, T_c)\lambda}{h_L - h_S(\lambda, T_c)} \quad (10)$$

due to (9) progressively decreases with λ . The denominator of this expression would even disappear for (very) high values of λ , i. e., the whole energy, which is supplied by the fuel, would only be used for heating up the exhaust gas. Since $Q \gg 0$ for the entire measurement campaign, such an extreme situation did not occur.

Results from (9) are shown in Fig. 7 for the available measurement data and $\Delta\lambda/\lambda = -0.01$. For a reduction of the excess air by 1%, Fig. 7 shows the expected fuel savings. Clearly, the reduced fuel consumption is equivalent to an improvement of the efficiency η , cf. (9). For typical values of λ , Fig. 7 shows that the relative change in fuel consumption is in the range $\Delta\dot{m}_{CH_4}/\dot{m}_{CH_4} \in [-1.43\%, -0.4\%]$. Fig. 7 essentially shows the elasticity of η with respect to λ , cf. (10). The larger λ the stronger the positive effect of reducing λ by 1%. For large values of λ , a reduction by 1% can even cause an efficiency improvement (relative fuel saving) of more than 1%.

The above results indicate that high relative fuel savings can be expected for high values of the excess air λ . In absolute numbers the achievable fuel savings are not that high because high values of λ only occur in part-load operation, i. e., at low values of \dot{m}_{CH_4} . If the radiant tube

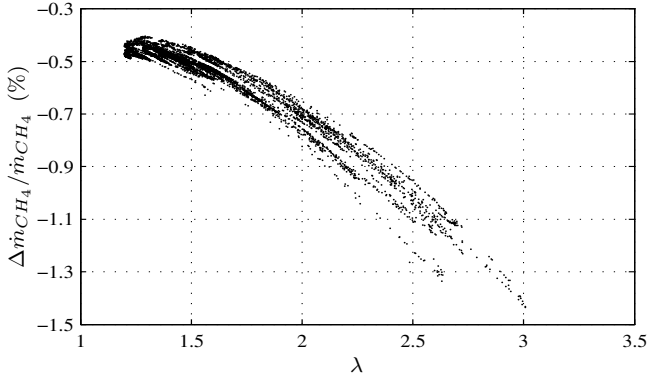


Fig. 7. Relative change in fuel supply according to a reduction of the excess air coefficient λ by 1%.

had been operated during the measurement campaign with excess air coefficients reduced by $-\Delta\lambda/\lambda = 0.01$, the fuel consumption would have been reduced by

$$-\frac{\int \Delta\dot{m}_{CH_4} dt}{\int \dot{m}_{CH_4} dt} = -\frac{\int \frac{S_\lambda(\lambda, T_c)\lambda}{h_L - h_S(\lambda, T_c)} \dot{m}_{CH_4} dt}{\int \dot{m}_{CH_4} dt} \underbrace{\frac{\Delta\lambda}{\lambda}}_{=-0.01} = 0.52\%.$$

It should be stressed that the presented results have been determined from linearized relationships. Therefore, they only apply to small deviations from the baseline scenario.

IV. CONTROL STRATEGY

First, the efficiency η of an individual radiant tube and the mass flow of fuel \dot{m}_{CH_4} are analyzed for the stored setpoint curve $\lambda(\dot{m}_{CH_4})$ from Fig. 3. Second, an optimized control strategy is proposed to increase the efficiency of the indirect-fired furnace.

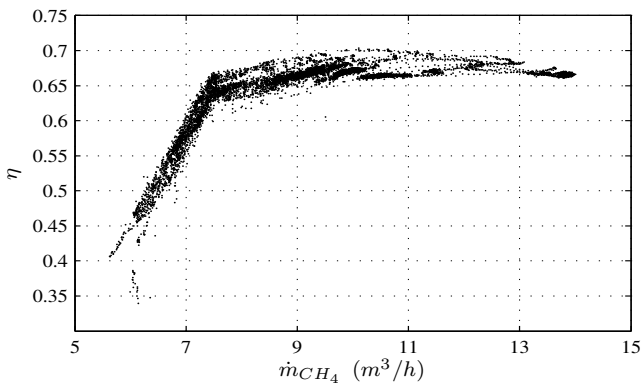


Fig. 8. Efficiency η of a radiant tube as a function of the fuel flow \dot{m}_{CH_4} .

Based on measurement data, Fig. 8 shows the efficiency η as a function of the fuel flow \dot{m}_{CH_4} according to (6). To achieve a high efficiency, it is therefore recommendable to operate the radiant tube in the upper load range. The drastic reduction in efficiency below $\dot{m}_{CH_4} = 7.5 \text{ m}^3/\text{h}$ is due to the stored setpoint curve for the excess air coefficient λ (cf. Fig. 3). As shown in Fig. 3, the supply of combustion air in

part-load operation is significantly higher than in full-load operation. The essential reason for the significant loss is therefore that more combustion air has to be heated by less fuel, i. e., the combustion air cools the radiant tube. The loss in efficiency may be reduced by modification of the stored setpoint curve.

With the current setpoint curve for the excess air coefficient, it is reasonable to operate the radiant tube above the limit of $\dot{m}_{CH_4} = 7.5 \text{ m}^3/\text{h}$. This might be realized by a better load distribution among the control zones of the furnace.

In the current configuration, the furnace interior is divided into individual control zones, where all radiant tubes of a control zone are supplied with the same amount of fuel. Let

$$Q_d = NQ(\dot{m}_{CH_4}) \quad (11)$$

be the desired heat input to a control zone. N describes the total number of radiant tubes in the considered control zone and $Q(\dot{m}_{CH_4})$ is the heat input of an individual radiant tube according to (4). The relationship between Q and the fuel flow \dot{m}_{CH_4} is shown in Fig. 9.

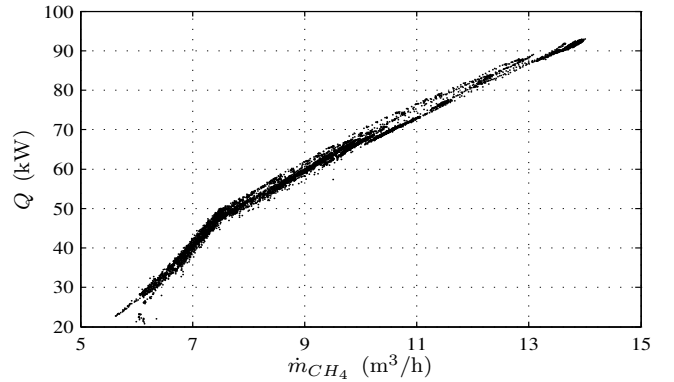


Fig. 9. Heat input Q of an individual radiant tube.

According to (11), the desired heat input Q_d is realized by a synchronous variation of the fuel supply for all radiant tubes. It is thus not possible to operate the radiant tubes always in the upper load range, i. e., with high efficiency. For this reason, a control strategy that improves the efficiency is proposed. The main idea is that a required heat input Q_d is realized only by some radiant tubes that are operated in full-load operation while the others are turned off completely. Formally, this strategy leads to a discrete optimization problem [16] with the objective to increase the efficiency η , i. e.,

$$\text{maximize}_{n \in \{1, 2, \dots, N\}} \eta \left(Q^{-1} \left(\frac{Q_d}{n} \right) \right) \quad (12a)$$

$$\text{subject to } \dot{m}_{CH_4}^- \leq Q^{-1} \left(\frac{Q_d}{n} \right) \leq \dot{m}_{CH_4}^+ \quad (12b)$$

where $Q(\dot{m}_{CH_4}) = Q_d/n$ is the characteristic curve that is obtained by smoothing the data shown in Fig. 9. By analogy, $\eta(\dot{m}_{CH_4})$ is found from Fig. 8. The optimization variable n denotes the discrete number of active radiant tubes in

the considered control zone, i. e., $N - n$ radiant tubes are switched off. Due to the transient interaction between the radiant tubes and the moving strip, it is not critical which of the N radiant tubes are switched off. The constants $\dot{m}_{CH_4}^-$ and $\dot{m}_{CH_4}^+$ represent the lower and upper bound, respectively, of the fuel supply of a single radiant tube.

Due to the small number N of radiant tubes per control zone, this problem can be easily solved iteratively. The expected fuel savings for the considered control zone is thus given by

$$n^* Q^{-1} \left(\frac{Q_d}{n^*} \right) - N Q^{-1} \left(\frac{Q_d}{N} \right),$$

where n^* denotes the optimal number of active radiant tubes according to (12). Note that (12) is a nominal formulation that may require additional measures to avoid too high switching frequencies of individual radiant tubes.

By full analogy, the potential savings for other control zones can be found. For the analyzed measurement campaign, this curbs the total fuel consumption by 7.3%. This result shows that there is a considerable potential for increasing the efficiency and the profitability of the furnace. Note, that this efficiency improvement requires only changes in the control strategy, i. e., the hardware remains unchanged.

V. CONCLUSIONS

The main conclusions from this work can be summarized as follows:

- 1) The balance model of a radiant tube has been developed for the analysis of the efficiency of an indirect-fired strip annealing furnace. The heat input into the furnace chamber through the wall of the radiant tube and its efficiency can thus be determined. Based on a linearized relationship, the influence of the fuel-air ratio on the efficiency can be analyzed by means of plant data from an experimental measurement campaign.
- 2) The improvement in efficiency of the radiant tube is equivalent to a fuel saving according to $\Delta\eta/\eta = -\Delta\dot{m}_{CH_4}/\dot{m}_{CH_4}$.
- 3) A relative reduction of the excess air coefficient λ by 1% results in fuel savings in the range of $-\Delta\dot{m}_{CH_4}/\dot{m}_{CH_4} \in [0.40\%, 1.43\%]$, depending on the respective operating point and the stored setpoint curve $\lambda(\dot{m}_{CH_4})$ for the excess air coefficient. For the operating conditions during the considered measurement period, a reduction of λ by 1% would have led to fuel savings of 0.52%.
- 4) To achieve a high efficiency of the radiant tubes, and thus of the indirect-fired furnace, it is essential to operate the burners above the critical point $\dot{m}_{CH_4} = 7.5\text{m}^3/\text{h}$. This critical load level has been found by taking into consideration the current setpoint curve $\lambda(\dot{m}_{CH_4})$. Operation below this point leads to a drastic loss of efficiency, i. e., then the radiant tube should be completely turned off.
- 5) In order to avoid fuel flow rates below $\dot{m}_{CH_4} = 7.5\text{m}^3/\text{h}$ and thus to ensure a high efficiency, an

optimized control strategy for the load distribution among the radiant tubes has been proposed. For this strategy, the required heat input Q_d is realized only by radiant tubes that are operated in full-load operation while the others are turned off. For the entire furnace, this control strategy would have led to fuel savings of 7.3%.

- 6) These significant efficiency improvements can be realized just by small modifications of the control strategy. The existing hardware remains thus unchanged.

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