

METHOD FOR ANALYSING THE HEAT RECOVERY POTENTIAL OF THERMOPROCESSING EQUIPMENT

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ABSTRACT

The importance of a heat recovery capability on thermoprocessing systems for increased cost-efficiency is rising steadily, along with energy costs. Accordingly, an increasing number of methods for recovering heat from exhaust gas have emerged in recent years, and the processes in question are being applied to ever more fields. For a plant-related assessment of diverse exhaust gas heat recovery solutions from an economic and/or process engineering viewpoint, it is imperative to analyze the plant-specific energy flows and to present the resulting findings in a clear-cut manner. On the basis of this analysis it is then possible to perform a plant-related evaluation and selection of the most suitable heat recovery method.

The present article describes a method that facilitates an economic assessment of the heat recovery potential of thermoprocessing equipment while also permitting a comparison of the plant-specific savings potentials for a given heat recovery process. To this end, a thermodynamic model is first developed to analyze the plant-specific energy flows; this model then enables us to compute these energy flows on the basis of process data. It is further shown how the results are presented in a clearly structured fashion to serve as the basis for further investigation. Next, it is explained how suitable heat recovery solutions can be selected for various equipment results and how the associated savings potentials can be determined. The method is applied, by way of example, to the dataset of an industrial furnace and the results obtained are discussed.

It is shown that the method permits an economic evaluation of diverse heat recovery solutions for different plants in day-to-day operation. In addition, the data analysis provides a capability to detect defective equipment components and unidentified energy flows.

INTRODUCTION

The iron and steel industry ranks among the most energy-intensive economic sectors in Germany, its annual demand of

800 PJ accounting for close to 5.7% of the country's primary energy consumption (approx. 14,000 PJ p.a.) [1]. This is in part because steelmaking involves diverse thermal processes requiring high temperatures.

Process and equipment engineering innovations in the field of industrial furnaces (e.g., optimized furnace operating regimes, improved burner settings, etc.), but above all the utilization of the associated process gases within the relevant plants, can help operators to cut this energy demand.

The focus here lies mainly on "waste energy" recovery measures, since these have already yielded enormous advances in energy efficiency in recent years. Technologies providing primary energy savings are given particular preference in this context. However, where they are no longer realizable for process or technical reasons, solutions aimed at alternative energy utilization modes (e.g., by generation of electricity) are likewise gaining increasing attention.

The major portion of such rejected ("waste") energies consists of energy exiting the furnace with the hot flue gases. In thermoprocessing equipment, where flue gases are generally obtained in ample quantities and at high temperatures, reclaiming this energy will prove economically beneficial in many cases. On most furnaces, energy recovery is already implemented to some degree via diverse built-in systems (e.g., air preheaters) today. To the plant operator, it is of particular interest to know how much heat can be additionally recovered from an existing system and/or how much of it still escapes unused through the flue-gas stacks.

In order to gain an overview of the amounts of energy rejected via the flue gas from thermal processes (looking at industrial furnaces as an example) and of the potentials inherent in today's technology options for energy recovery, plant-specific process data must be analyzed. To this end, an energy balance for a given installation (in our example, a single furnace) is established and all incoming and outgoing energy flows are

computed and graphically represented to depict the current situation of a given furnace.

Relying on these results, it is then possible to develop and present tailor-made solutions for existing plants in which energy recovery would provide an economically viable power and cost-saving option adapted to the company's specific boundary conditions. To facilitate decision-making, different waste heat utilization concepts are compared in general terms for each furnace and then evaluated based on the results regarding energy flows.

NOMENCLATURE

A	[a]	Payback period
$E_{a,in}$	[J]	Input of external energy
$E_{a,out}$	[J]	Output of external energy
E	[€]	Savings potential
H	[J]	Enthalpy
h	[J]	Mass-specific enthalpy
h_v	[J]	Mass-specific enthalpy of the substance v
H_u	[J/kg]	Net calorific value of the substance supplied
I	[€]	Price of investment
L_{min}	[--]	Minimum air demand
λ	[--]	Stoichiometric ratio
m_{D}^{VD}	[kg]	Steam mass produced by evaporator
\dot{m}	[kg/s]	Mass flow
\dot{m}_A	[kg/s]	Flue gas mass flow
\dot{m}'_A	[kg/s]	Altered flue gas mass flow
\dot{m}_{BL}	[kg/s]	Combustion air mass flow
\dot{m}'_{BL}	[kg/s]	Altered combustion air mass flow
\dot{m}_{BL}^{st}	[kg/s]	Stoichiometric combustion air mass flow
\dot{m}_{BS}	[kg/s]	Fuel mass flow
\dot{m}_{KL}	[kg/s]	Cooling air mass flow
\dot{m}_v	[kg/s]	Mass flow of substances v participating in the system
M_v	[kg/mol]	Molar mass of the substance v
p	[Pa]	System pressure
Q	[J/s]	Heat flux
\dot{Q}_V^O	[J/s]	Loss heat flux at furnace through insulation
\dot{Q}_V^R	[J/s]	Loss heat flux at HRS through insulation
T	[°C]	Temperature of a given incoming / outgoing substance v
$\varphi_{O_2}^{gem}$	[--]	Measured air oxygen concentration
φ_{O_2}	[--]	Air oxygen concentration
V	[m ³]	Volume
\dot{V}_v	[m ³ /s]	Volume flow of the substance v
$\dot{V}_{O_2}^L$	[m ³ /s]	Volume flow of oxygen in the excess air
W_{Diss}	[J]	Dissipation work
x	[h/a]	Operating hours per year
y	[€/kg]	Value of steam

MODELING THE ENERGY FLOWS OF A THERMO-PROCESSING PLANT

In this section we shall initially define the system and its system boundaries while also explaining the assumptions made, given that the latter have a substantial impact on results. The enthalpy and heat fluxes of the system as well as the system boundaries will be examined as outlined in Fig. 1. Accordingly, the overall system shall be divided into the two coupled subsystems "furnace" and "HRS" (heat recovery system).

These two shall be viewed as open systems, given the various incoming and outgoing substance flows traversing their boundaries, and the boundaries themselves shall be deemed diathermal since a major part of the enthalpy is lost as heat, especially in the case of the furnace system. Moreover, a simple flow process that should be considered quasi-steady takes place

throughout the system at a constant pressure p , i.e., no work is done on the system and hence we can write:

$$V dp = dW_{Diss} = 0 \quad (1)$$

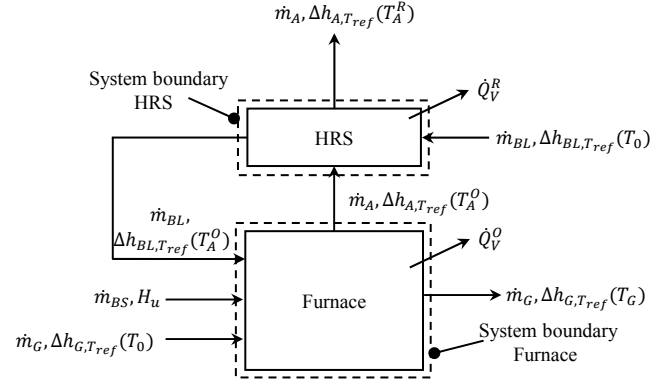


Figure 1 Schematic energy balance of a rolling mill furnace

Moreover, the changes in incoming and outgoing external energy $E_{a,in}$ and $E_{a,out}$, respectively, shall be ignored as they are negligible:

$$dE_{a,in} = dE_{a,out} = 0 \quad (2)$$

Under these conditions and given the relation $H = m h(T)$, wherein m is the mass stored in the system and $h(T)$ is the mass-specific standardized enthalpy according to [6] at temperature T , the equation of the first law of thermodynamics for open systems, duly time-differentiated, takes the following simplified form:

$$\frac{dH}{dt} = \dot{Q} + \dot{m}_{in} \cdot h(T_{in}) - \dot{m}_{out} \cdot h(T_{out}) \quad (3)$$

The indices *in* and *out* refer to the incoming and outgoing mode. As the flow process is considered to be steady-state, all derivatives with respect to time of our status and process variables are zero, hence the following expression applies:

$$\dot{m}_{in} = \dot{m}_{out} = \dot{m} \quad (4)$$

Accordingly, Eq. 3 is simplified to:

$$0 = \dot{Q} + \dot{m} \cdot [h(T_{in}) - h(T_{out})] \quad (5)$$

However, as is evident from Fig. 1, we are dealing with more than one substance flow through the system, so Eq. 5 must be supplemented accordingly. Taking into account the substances $v \in S = \{A; BL; BS; G\}$ participating in the system, the expanded Eq. 5 takes this form:

$$0 = \dot{Q} + \sum_{v \in S} \dot{m}_v \cdot [h_v(T_{in}) - h_v(T_{out})] \quad (6)$$

with

$$h_{A,in} = 0, h_{BS,in} = 0 \text{ and } h_{BL,out} = 0$$

In this expression, A denotes the flue gas, BL the combustion air and G the product being heated. The temperature-related standard enthalpies $h_v(T)$ can be obtained from tabulated data sheets, cf., for example, [6]. Using the enthalpy difference

$\Delta h_{v,T_{ref}}(T) = h_v(T) - h_v(T_{ref})$ and the selected reference temperature $T_{ref}=0^\circ\text{C}$, Eq. 6 can be rewritten as:

$$0 = \dot{Q} + \sum_{v \in S} \dot{m}_v \cdot [\Delta h_{v,T_{ref}}(T_{in}) - \Delta h_{v,T_{ref}}(T_{out})] \quad (7)$$

For the „furnace“ subsystem as shown in Fig. 1, taking into account

$$\dot{m}_A = \dot{m}_{BL} + \dot{m}_{BS} \quad (8)$$

and $v \in S$, we can rewrite the balance equation 7 as

$$\begin{aligned} 0 = & \dot{m}_{BS} \cdot [H_u - \Delta h_{A,T_{ref}}(T_A^O)] + \\ & \dot{m}_{BL} \cdot [\Delta h_{BL,T_{ref}}(T_{BL}) - \Delta h_{A,T_{ref}}(T_A^O)] + \\ & \dot{m}_G \cdot [\Delta h_{G,T_{ref}}(T_0) - \Delta h_{G,T_{ref}}(T_G)] - \dot{Q}_V^O \end{aligned} \quad (9)$$

with the furnace losses $\dot{Q} = \dot{Q}_V^O$ and the net calorific value H_u of the fuel, cf. [5], [6]. The same method can be used for the heat recovery system. In analogy to Eq. 9, we obtain the following:

$$\begin{aligned} 0 = & \dot{m}_{BL} \cdot [\Delta h_{BL,T_{ref}}(T_0) - \Delta h_{BL,T_{ref}}(T_{BL})] + \\ & (\dot{m}_{BL} + \dot{m}_{BS}) \cdot [\Delta h_{A,T_{ref}}(T_A^O) - \Delta h_{A,T_{ref}}(T_A^R)] - \dot{Q}_V^R \end{aligned} \quad (10)$$

The above equations can be employed to calculate the enthalpy flows but need to be adapted, if and as appropriate, to the specific situation of the given thermoprocessing system.

DATA REPRESENTATION AND EXPLORATION

In modern furnaces it is common practice to measure the volume or mass flows of the fuels and combustion air, as well as the furnace and flue gas temperatures. Enthalpy differences can thus be easily calculated from these data and can be presented in the form of an energy flow diagram ("Sankey" diagram). To this end, the enthalpy flows of each substance $v \in S = \{A; BL; BS; G\}$ are integrated, on the basis of Eq. 6, over a time interval $[0, t]$:

$$0 = \dot{Q} + \int_0^t \sum_{v \in S} [\dot{m}_v(\tau) \cdot [\Delta h_{v,T_{ref}}(T(\tau))]] d\tau \quad (11)$$

The above is visualized in normalized format in Sankey diagrams by depicting the percentage shares of the individual enthalpy differences ΔH_v^{ges} in relation to the externally introduced fuel enthalpy ΔH_{BS}^{ges} :

$$\Delta h_v^{proz} = \frac{\Delta H_v^{ges}}{\dot{m}_{BS} H_u} \cdot 100\% \quad (12)$$

The insulation losses \dot{Q}_V^O are determined by rearranging the balance equation 10, given that no measured values of this variable are available. The heat losses across the recuperator, \dot{Q}_V^R , are so low as to be negligible due to the design of the heat exchangers.

For an analysis of the enthalpy flows of the furnace and HRS, one should be able, on principle, to obtain the following quantitative and temperature data via flow meters and thermocouples as input for the calculations:

- Mass flows:

As a general rule, all mass flows entering the system from outside should be measured. This specifically includes the fuel mass flow \dot{m}_{BS} , the product mass flow \dot{m}_G and the combustion air mass flow \dot{m}_{BL} .

- Temperatures:

As a general rule, the temperatures upstream and downstream of the HRS and the furnace are to be measured, i.e., the flue gas temperature T_A^O on the furnace exit side, the flue gas temperature T_A^R and combustion air temperature T_{BL} downstream of the HRS, and the product temperature T_G . The temperature T_0 always corresponds to the ambient temperature, i.e., $T_0 \approx 20^\circ\text{C}$.

For further mass and temperature values that may be required to determine the enthalpies, reference should be made to the respective individual balances. Apart from the mass flows and temperatures, it is necessary to know the enthalpy values for the substances present in the overall system as well as the calorific value of the fuel. The enthalpies for the gases in S and for the product being heated can be taken from tabulated data sheets for various temperatures, cf., for example, [6].

FUNDAMENTALS OF ANALYZING FLUE GAS ENTHALPY

Apart from the magnitude of the prevailing flue gas enthalpy, we must take a look, first and foremost, at its quality. This involves an examination in terms of

- the temperature level;
- the amounts of flue gas emitted; and
- its availability over time.

Thus, for instance, if flue gas is present in large quantities but at low temperature, heat recovery may be just as inefficient as reclaiming the enthalpy from small amounts of very hot flue gas.

Similarly, if a flue gas enthalpy that is basically of high quality (i.e., large volumes of flue gas at high temperature) is available only at specific times, this will typically impair the economic efficiency of the heat recovery. Looking at the individual enthalpy differences of the furnaces, one must therefore determine the frequency distribution of the available flue gas quantity as well as its temperature over the time interval considered in order to obtain a revealing overview of the overall situation at the furnaces.

EXEMPLARY CALCULATION OF A FURNACE'S ENTHALPY FLOWS – WITH A BALANCING ERROR

The furnace described here by way of example is already equipped with a central heat recovery system relying on recuperative heat transfer. The system consists of two recuperators connected in series which shall be considered as one unit (HRS) for purposes of the enthalpy calculation. This can be done because no additional substance flows capable of affecting the calculation enter or leave between the two heat exchangers.

Accordingly, the flue gas temperature at the HRS inlet corresponds to the measured temperature T_A^O of the flue gas exiting the furnace, while the outlet temperature is equal to the temperature measured downstream of the second recuperator, T_A^R . Similarly, the combustion air inlet temperature is the ambient

temperature T_0 and the outlet temperature is equal to the temperature measured on the exit side of the second recuperator, T_{BL} . An analysis of the enthalpy flows described yields the energy flow diagrams shown in Fig. 2.

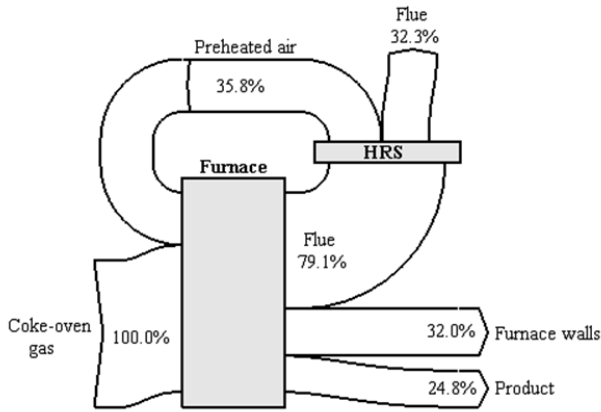


Figure 2 Energy flow diagram of a furnace with defective heat recovery

What can be noted here is an evidently large balancing error for the recuperative HRS. Since a functional test of the thermocouples supplying the relevant data for the balancing calculations showed these to work properly, it was suspected that the recuperator tube walls were damaged. When leaks through these tube walls occur, a part of the original and measured combustion air intake \dot{m}_{BL} fed to the recuperator tubes by a combustion air fan will be drawn into the flue-gas stack due to the prevailing pressure conditions (overpressure in the tubes on the combustion air side, underpressure on the flue gas side due to the natural draft of the flue-gas stack) and will exit the system together with the flue gas flow \dot{m}_A , i.e., the measured combustion air mass \dot{m}_{BL} supplied to the recuperator by the combustion air fan is no longer constant throughout the system and will differ from the amount entering the furnace. The flue gas mass emitted from the furnace will vary accordingly.

To verify the above hypothesis, we rely on the readings from the O_2 measurement and use these to recalculate the enthalpies within the system. This measurement takes place in the flue-gas stack at the furnace exit. It can be used to quantitate the oxygen content, and hence the actual air content ("excess air") in the flue gas. However, the chemical composition of the flue gas must be known for this purpose; it depends on the chemical composition of the fuel employed. The above will initially be explained in the following section.

ALTERNATIVE CALCULATION OF FLUE GAS ENTHALPY FLOWS BASED ON OXYGEN SENSORS

Flue gas constitutes the reaction product resulting from the burning of fuel gas with combustion air. For this combustion to occur in the first place, a certain mixing ratio of the two components – i.e., fuel and oxygen – must be set beforehand by the burners. The flue gas composition will thus depend primarily on the composition of the fuel employed and of the fuel-oxygen mixing ratio (oxygen being supplied in the form of air) [4].

If the combustion process is stoichiometric, i.e., if the oxygen supply is adjusted so that (in theory) each fuel molecule will react fully with oxygen and the combustion proceeds without any oxygen shortage or surplus, we obtain the flue gas compositions shown in Fig. 3 from coke gas and mixed gas, respectively, with quantities stated in kilograms per m^3 of fuel.

	CO ₂ [kg]	H ₂ O [kg]	SO ₂ [kg]	O ₂ [kg]	N ₂ [kg]
Mixed Gas [m ³]	0.86	0.21	≤ 0.01	0	1.89
Coke Gas [m ³]	1.03	0.83	≤ 0.01	0	4.05

Figure 3 Composition of flue gas from stoichiometric ($\lambda=1$) combustion

In a stoichiometric combustion the air/fuel ratio λ has the value $\lambda = 1$. However, most furnaces are typically operated with more oxygen (e.g., an "excess of air", $\lambda \approx 1,05$) than would be necessary, in theory, to achieve complete combustion. Accordingly, the flue gas mass flow will usually contain a residual portion of unused air. The assumed situation on our specimen furnace is illustrated in Fig. 4.

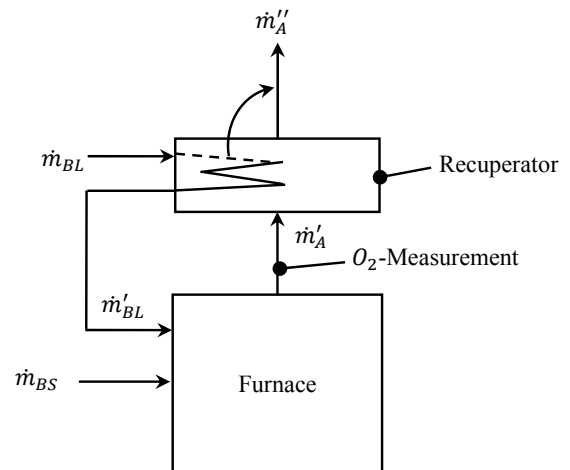


Figure 4 Energy flow diagram of a furnace with defect

The flue gas mass flow \dot{m}_A had originally been computed according to Eq. 8 from the total fuel mass flow \dot{m}_{BS} supplied and the combustion air mass flow \dot{m}_{BL} introduced into the system. Since there is reason to suspect a defect on the heat recovery system that causes part of the combustion air to be drawn into the flue-gas stack under the prevailing pressure conditions, the furnace only receives the combustion air flow \dot{m}'_{BL} diminished by the combustion air leakage flow $\dot{m}'_{BL}{}^{Leck}$, hence the flue gas mass flow actually reaching the HRS is reduced to:

$$\dot{m}'_A = \dot{m}'_{BL} + \dot{m}_{BS} \tag{13}$$

The magnitude of the flow rates $\dot{m}'_{BL}{}^{Leck}$ and \dot{m}'_{BL} is not measured, so these values are unknown. However, the excess air still present in the furnace after combustion can be determined from the readings of the O_2 sensors fitted at the inlet of the flue-gas stack. Using these results, the real flue gas volume flow \dot{V}'_A can be newly calculated. The sensors detect the oxygen portion in the flue gas flow, i.e., the "excess air" left in the furnace after combustion. The flue gas flow itself is determined

as the sum of the volume flows of the individual flue gas components (Fig. 3). This is expressed as:

$$\varphi_{O_2}^{gem} = \frac{\dot{V}_{O_2}^L}{\sum_{v \in G} \dot{V}_v} \quad (14)$$

Using the ideal gas law we can write Eq. 14 as a function of the mass flows according to [5], which, by rearranging terms and taking into account the known composition of air ($\varphi_{O_2}^L \approx 21\%$, $\varphi_{N_2}^L \approx 79\%$), enables us to calculate the actual excess air reaching the HRS inlet:

$$\dot{m}_L = \frac{\varphi_{O_2}^L \sum_{v \in G} \frac{\dot{m}_v}{M_v}}{\frac{0,21}{M_{O_2}} - \varphi_{O_2}^L \left(\frac{0,79}{M_{N_2}} + \frac{0,21}{M_{O_2}} \right)} \quad (15)$$

From this value we can then determine the actually incoming flue gas mass flow from the sum of the stoichiometric combustion air portion (i.e., the combustion air flow ensuring, in theory, a complete reaction of the fuel with oxygen), the air surplus detected by the sensors, and the quantity of fuel supplied:

$$\dot{m}_A = \dot{m}_{BL}^{st} + \dot{m}_L + \dot{m}_{BS} \quad (16)$$

where

$$\dot{m}_{BL}^{st} = \dot{m}_{BS} \cdot \lambda \cdot L_{min}$$

The stoichiometric combustion air flow can be determined via the air/fuel ratio $\lambda \approx 1.05$ and the minimum air demand $L_{min} = 11.1$ (for coke gas) as a function of the fuel. The following equation is obtained:

$$\dot{m}'_A = \dot{m}_L + \dot{m}_{BS} \cdot [\lambda \cdot L_{min} + 1] \quad (17)$$

The Sankey diagram reflecting the flue gas mass flow thus recalculated via the O_2 measurement is shown in Fig. 5.

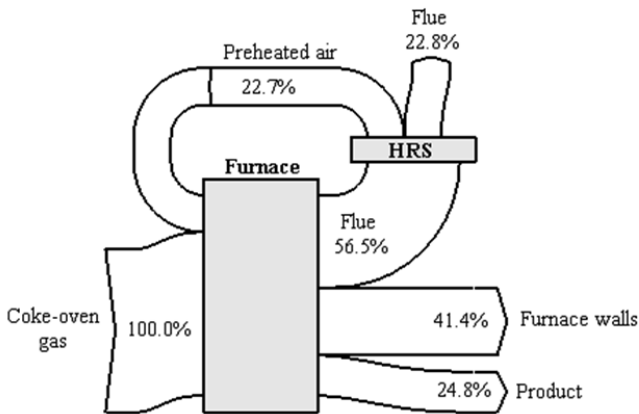


Figure 5 Energy flow diagram based on O_2 sensor data

It can be seen that the flue gas enthalpy newly computed from the O_2 sensor data is over 40% lower than the one originally calculated and depicted in Fig. 2. It follows that a large portion of the combustion air flow measured ahead of the recuperator inlet gets lost on its way to the O_2 measurement. Fur-

thermore, it emerges that the highest potential for energy savings does not reside in the plant's heat recovery system. Rather, it is evident from Fig. 5 that the primary energy consumption can be reduced directly by optimizing the furnace lining and minimizing heat riding areas at the level of the furnace housing.

If we assume that the entire combustion air flow leaks into the flue gas stack through the recuperator tube walls, we obtain the frequency distribution of the flue gas mass and temperature levels shown in Fig. 6. It is evident from the diagram that although large flue gas quantities are emitted, these tend to be of rather low temperature on average.

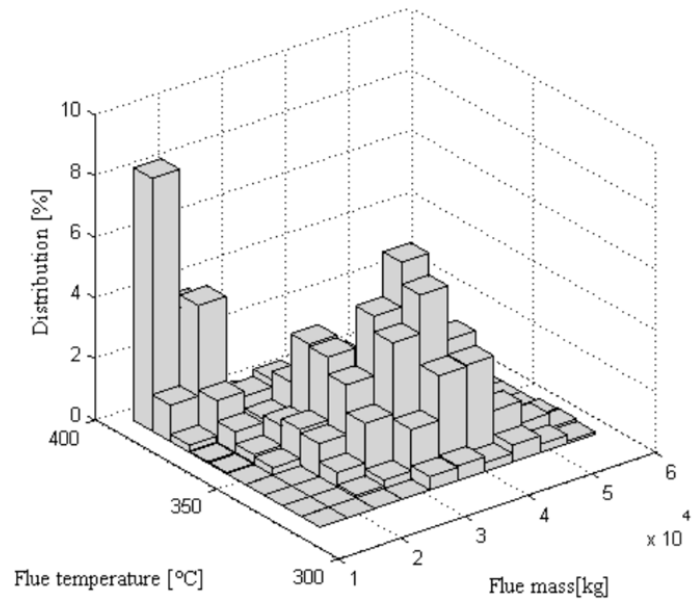


Figure 6 Histogram of flue gas mass flows at the respective temperatures on the specimen furnace

The method described above can be used on any thermo-processing system and may serve as a basis for evaluating potential designs.

REQUIREMENTS ON HEAT RECOVERY SYSTEM DESIGN

On principle, the main criterion in HRS design is that the system should be economically beneficial. In examining the options, it is therefore recommended to proceed in the following order:

1. Improvement of existing heat recovery solutions through adaptation of diverse process parameters (low investment cost): These include, e.g., reducing the furnace temperature during standstill periods, minimizing furnace temperatures in the stack inlet zone, or raising the combustion air temperature. It should be checked beforehand at this point whether a potential for improving the plant's energy efficiency still exists, as such measures will typically have been integrated already.

2. Addition of new HRS components to existing systems (medium investment cost): If the efficiency of the recovery system can no longer be increased through a selective modifica-

tion of process parameters, suitable add-on measures should be examined (e.g., use of a better air preheater, installation of an additional waste heat boiler if a connecting line to a steam network is available nearby).

3. Realization of an entirely new system (high investment cost): Where the use of add-on equipment is not feasible, it should be examined whether a new system can be put in place in an economically viable manner. If so, the preference will normally be on technologies enabling recoverable energy to be returned into the process from which it originated, as this approach will boost plant efficiency and thereby reduce operating costs directly. Moreover, if the energy is otherwise utilized in upstream or downstream processes it will always be necessary to synchronize one system's energy supply with the other's energy demand.

Based on the data obtained from the analysis, the economic benefit of individual projected solutions can be assessed in accordance with the above guidelines. Both saving potentials and payback periods can be calculated, although this aspect merits separate discussion.

CALCULATION OF SAVING POTENTIALS

For saving energy and thus boosting the cost efficiency of a thermoprocessing system, two basic options exist:

1) Utilization of heat energy within system boundaries, i.e., the energy contained in the flue gas is returned as useful energy into the process from which it originated without any particular time delay (e.g., by preheating combustion air); or

2) utilization across system boundaries, i.e., the recovered energy is supplied to other production processes with a (normally substantial) time delay (e.g., heat is used to generate electricity).

To this end, the following options can, on principle, be considered:

- a) Use of a higher combustion air temperature and hence, an increased pre-heat level in the case of recuperators;
- b) installation of regenerative burners;
- c) installation of a waste heat boiler system.

On some furnaces, investing in a waste heat boiler system (item c) may be worthwhile. Its maximum theoretical steam output depends on the efficiency of the evaporator. Since this efficiency is influenced, above all, by the flue gas mass supplied and its temperature, the characteristic performance diagram of the evaporator stack must be known for an assessment of the saving potential. This characteristic diagram indicates the evaporator's maximum theoretical efficiency at various load conditions, i.e., different flue gas flow rates and temperatures. Where the diagram lacks data for a given operating point, the value can be approximated by linear interpolation, with a rate of approx. 45-55 % (the maximum achievable in practice) to be considered the outside limit. Based on the efficiency figure, the amount of steam which can be theoretically produced can be calculated thus:

$$m_D^{VD} = \int_0^t \eta_{VD}^{th}(\tau) \cdot \frac{\dot{m}_A(\tau) \Delta h_{A,Tref}(T_A^R(\tau))}{\Delta h_{D,Tref}(T_D(\tau))} d\tau \quad (18)$$

Assuming that the steam is to be fed into a 12 bar steam line, the specific enthalpy is defined to be a constant $\Delta h_{D,Tref}(T_D(\tau)) \approx 2783 \frac{kJ}{kgK}$ (steam in saturated state at $p = 12$ bar). The data for computing the theoretically achievable steam output can be derived using the above calculation principles.

By linear extrapolation of this data to one year (which is assumed to consist of x operating hours) and assuming a steam value of approx. EUR y per kilogram of steam, we can then calculate the saving potential as:

$$E = m_D^{VD} \cdot x \cdot y \quad (19)$$

For an industry-typical investment price I of a waste heat boiler system (depending on the size range), the payback period A can thus be calculated as

$$A = E/I \quad (20)$$

For other solutions, the payback period can be computed in analogy to the above example.

CONCLUSION

The method presented herein shows a way to evaluate the heat recovery potential of thermoprocessing systems from an economic benefit viewpoint. The thermodynamic model for an analysis of plant-specific energy flows is found to be useful.

The model allows evaluating the particular energy flows via the Sankey diagrams, whereby the fields of highest saving potentials get traceable (e. g. furnace walls or HRS). Additionally the analysis of histograms of flue gas mass flows at the respective temperatures allows the rating of energy flow usability respective the different concepts of heat recovering. The results may serve as a basis for further investigations. Depending on the heat recovery method, the corresponding saving potentials can be calculated.

An example applied to the data set of an industrial furnace illustrates the basic approach. Moreover, it is evident from this example how the data analysis provides a tool for detecting plant component failures and concealed energy flows. The method is very well suited for assessing thermoprocessing systems and their economic efficiency.

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