APPENDIX E

Paper V

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Modelling of the finned convection section of a biomass boiler

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Abstract

Fouling of the convection section of a boiler has an effect on the efficiency of thermochemical biomass conversion. In this paper, a modelling for the heat transfer of a fouled fin is discussed. The temperature of the base of the fouled fin is assumed to be equal to the surface temperature of the wall and the difference of the heat transfer rate is ±2% when compared to the results of finite element method calculations. A calculating procedure for simulating the thermal performance of the entire convection section is introduced. It was established that the calculations describe the effects of changes in the operation conditions with satisfactory accuracy. This was verified by measurements at three biomass boiler plants.

Keywords: Deposit; Extended surface; Finite element method; Fouling; Heat transfer

1. Introduction

There is an increasing consensus that the activities of man are contributing to the greenhouse effect, hence resulting in an unfavourable impact on the environment. In the production of energy, the portion of wood used fuels has grown and will grow because of their largely CO_2 neutrality. In addition to technical performance, cleanliness of combustion has become an important factor. Therefore, the emphasis of modelling work has recently been focused on combustion processes. In the convection section of a boiler, there are no significant chemical reactions, and the studies mainly concentrate on heat transfer, fluid dynamics or eventual corrosion caused by fouling. Wood fired boilers, which heat large single buildings or small district heating networks, are often grate type boilers. The operating range of the convection section is usually quite large. The use of moist wood fuel often requires an increase in excess air. For that reason, the highest volume flow rate of flue gas through the convection section occurs at peak load, minimum outdoor temperature and maximum moisture of fuel. Boiler plants also need to operate at partial load, often at $10,\ldots,20\%$ of the peak load. Therefore, the smallest gas flow rate occurs at minimum load, maximum outdoor temperature and minimum moisture of fuel. Because of energy economy, the final temperature of the flue gas should not be excessively

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Nomenclature

a	interfin distance, m
A	surface area, m^2
d	diameter, m
h	specific enthalpy, J kg^{-1}
l	height of fin, m
m	auxiliary variable
$\dot{m}$	mass flow, kg s^{-1}
n	number of elements
Nu	Nusselt number
P	circumference, m
p	auxiliary variable
R	heat resistance, $m^2\,kW^{-1}$
s	thickness, m
T	temperature, °C
U	flue gas wetted perimeter, m
x_d	dimensionless length

Greeks

α	heat transfer coefficient, W $m^{-2}\,K^{-1}$
Φ	heat transfer rate, W m^{-1}
λ	heat conductivity, W $m^{-1}\,K^{-1}$
θ	temperature difference, °C

Subscripts

b	base
cs	cross-section
cd	conduction, fouling
cv	convection
dg	deposit, gas side
dw	deposit, water side
f	fin
fg	flue gas
f, x	fin, x-direction
f, y	fin, y-direction
g	gas, gas side
h	hydraulic
p	plate
sd	surface, fouling
sg	surface, gas
sw	surface, water
t	fully developed turbulent flow
w	water

Superscripts

ep	end of plate
f	fin
p	plate

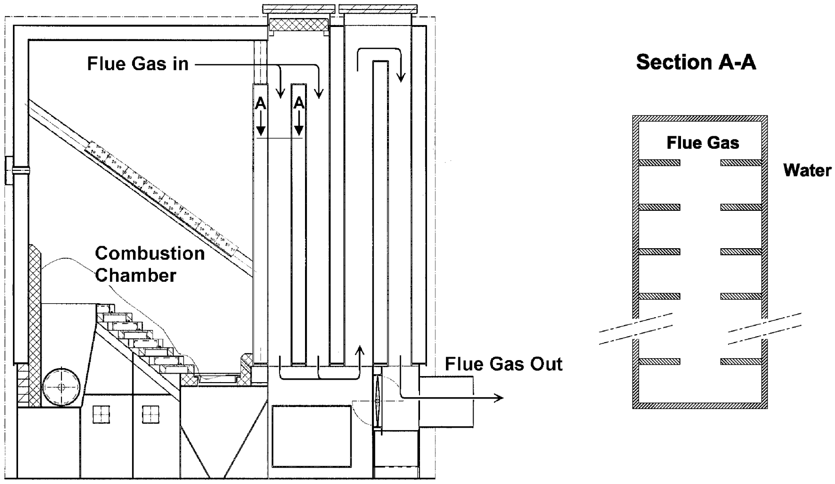


Fig. 1. The boiler Arimax BIO of Thermia Ltd. and a section of a flue gas duct in the convection section.

high at the high flow rates. On the other hand, the risk of condensing water vapour increases if the temperature is low at small flow rates. Simulation of the convection section is needed to predict either possible energy losses or operational risks. Knowledge of the behaviour of the convection section is required also for choosing realistic input parameters for the CFD (computerized fluid dynamics) calculations of the combustion chamber [1]. The model should, however, be sufficiently easy to use and economically feasible to serve in practical design work.

Commonly, the convective heat exchange section is of rectangular duct construction with an array of longitudinal rectangular fins for the flue gas side and a smooth outside surface for the water side, see Fig. 1. During the past eight decades, many researchers have considered the thermal performance of extended surfaces. Laor and Kalman [2] and Razelos and Kakatsios [3] have extensively summarised the previous works in the context of defining the optimum dimensions of different fins. In the literature, the calculation for a finned surface is usually presented for a clean fin [4–7]. Fouling from a gaseous stream has, however, a significant effect on the energy efficiency of the boiler [8]. The heat transfer surfaces of the convection section are inclined to fouling because of the fly ash and soot on the flue gas side and because of the boiler scale on the water side. Fouling reduces the heat transfer because the thermal conductivities of ash, soot and boiler scale are lower than that of steel. Therefore, a simplified model of the heat transfer rate for a fin element with fouling layers is introduced.

2. Modelling

The initial data for modelling are the geometry of the finned surface (see Fig. 2) and the mass flow rate, initial temperature and composition of the flue gas and the temperature of the boiler water.

The heat transfer of a single fin can be derived from the steady state heat balances in Eqs. (1) and (2) on the assumption that the heat conduction in the fouling layer is negligible in the direction parallel to the fin (x-direction).

$$\phi_x^{\mathrm{f}} = \phi_{x+\mathrm{d}x}^{\mathrm{f}} + \mathrm{d}\phi_{\mathrm{cd}}^{\mathrm{f}} \tag{1}$$

$$\mathrm{d}\phi_{\mathrm{cd}}^{\mathrm{f}} = \mathrm{d}\phi_{\mathrm{cv}}^{\mathrm{f}} = (P + 4s_{\mathrm{dg}})\mathrm{d}x(T_{\mathrm{g}} - T_{\mathrm{f}})/(1/\alpha_{\mathrm{g}} + s_{\mathrm{dg}}/\lambda_{\mathrm{dg}}) \tag{2}$$

A change in the heat transfer rate inside the fin can be calculated with Eq. (3) on the assumption that the thermal conductivities and cross-sections of the fin and the fouling layer are constant, and further still, that the temperature gradient is positive and the conduction is a diminishing function in the x-direction

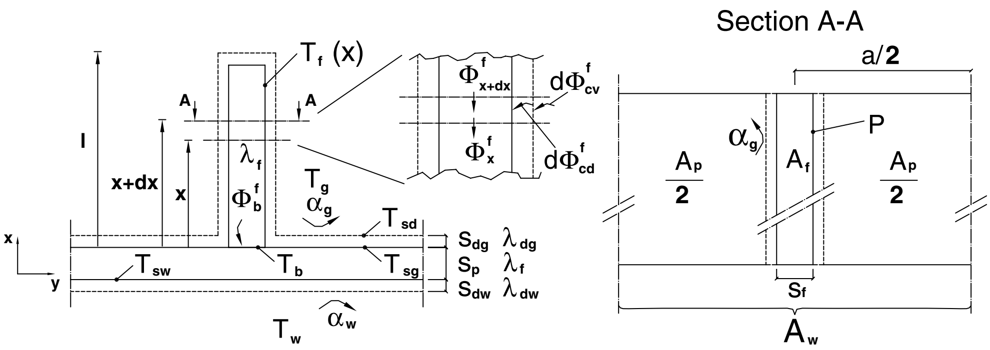


Fig. 2. The structure and nodes of the fin element.

$$\frac{d\phi_x^f}{dx} = -\lambda_f A_f \frac{d^2 T_f}{dx^2} \tag{3}$$

By notation

$$m^2 = (P + 4s_{dg})/[\lambda_f A_f (1/\alpha_g + s_{dg}/\lambda_{dg})] \tag{4}$$

and

$$\theta = T_g - T_f \tag{5}$$

The differential equation describing the temperature change along the fin is

$$\frac{d^2\theta}{dx^2} - m^2\theta = 0 \tag{6}$$

with the boundary conditions that the heat transfer coefficient is constant from the tip to the base of the fin

$$-\lambda_f \frac{d\theta}{dx}\bigg|_{x=l} = \theta|_{x=l}/(1/\alpha_g + s_{dg}/\lambda_{dg}) \tag{7}$$

and that the base temperature is known

$$\theta|_{x=0} = \theta_b = T_g - T_b \tag{8}$$

and by notation

$$p = 1/[\lambda_f (1/\alpha_g + s_{dg}/\lambda_{dg})] \tag{9}$$

The rate of heat transfer of the fin can be solved as

$$\phi_b^f = \lambda_f m A_f \frac{\sinh ml + (p/m)\cosh ml}{\cosh ml + (p/m)\sinh ml}(T_g - T_b) \tag{10}$$

The authors do not have an accurate analytical method to evaluate the heat transfer rate to and from the plate. However, two different limit assumptions are available. The thermal conductivity of the fin + plate element can be assumed to be either infinite or zero in the direction of the plate (y-direction). It follows from the former that the base temperature of the fin is equal to the surface temperature of the plate ($T_{sg} = T_b$), see Eqs. (11) and (12). In the latter case ($\lambda_{f,y} = 0$) all the heat of the fin is transferred through the fin base area A_f of the plate. Then, the fin and the place on the plate where the fin is attached must be treated as a separate heat transfer element from the rest of the plate, see Eqs. (13) and (14). The accurate solution is somewhere between these two ultimate assumptions because the thermal conductivity in the homogenous material is equal in the x and y directions ($\lambda_{f,y} = \lambda_{f,x}$). In the following, the results calculated according to these two assumptions are compared with the results from the commercial ANSYS program, which uses a numerical two dimensional finite element method [9]. In all of these three methods, both the thickness of the fouling layer and the heat transfer coefficient are assumed to be constant throughout the entire element (plate + fin).

From the assumption $\lambda_{f,y} = \infty$, it follows that $T_{sg} = T_b$. Therefore, the heat transfer rate to the plate is

$$\phi^p = A_p(T_g - T_b)/(1/\alpha_g + s_{dg}/\lambda_{dg}) \tag{11}$$

It also follows that the surface temperature of the water side T_{sw} is uniform, and hence, the heat transfer of the entire element is:

$$\phi_b^f + \phi^p = (A_f + A_p)(T_b - T_w)/(1/\alpha_w + s_{dw}/\lambda_{dw} + s_p/\lambda_f) \tag{12}$$

According to the assumption $\lambda_{f,y} = 0$, the heat transfer rate of the plate is

$$\phi^p = A_p(T_g - T_w)/(1/\alpha_g + s_{dg}/\lambda_{dg} + s_p/\lambda_p + s_{dw}/\lambda_{dw} + 1/\alpha_w) \tag{13}$$

and all the heat of the fin is transferred through the fin base area A_f of the plate

$$\lambda_f m A_f \frac{\sinh ml + (p/m)\cosh ml}{\cosh ml + (p/m)\sinh ml}(T_g - T_b) = \frac{A_f(T_b - T_w)}{\left(\frac{s_p}{\lambda_f} + \frac{s_{dw}}{\lambda_{dw}} + \frac{1}{\alpha_w}\right)} \tag{14}$$

The comparison of the results of the assumptions with those of the FEM approach is exemplified in Fig. 3 with typical initial values of the biomass boiler. The heat transfer surfaces are assumed to be clean.

With the assumption that $\lambda_{f,y} = \infty$ (plate at uniform temperature), the heat transfer rate is closer to the result of the FEM calculation than when assuming that $\lambda_{f,y} = 0$ (fin and plate separately), see the bars

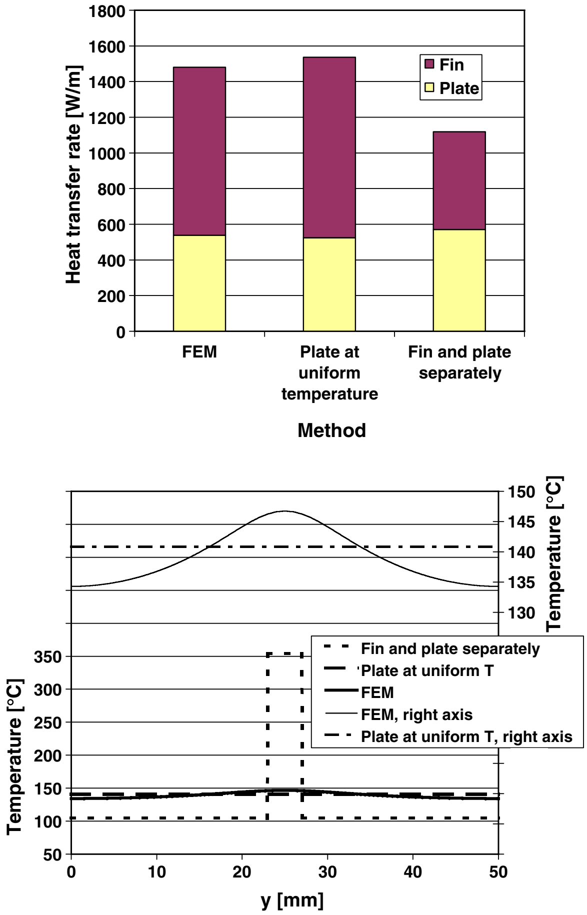
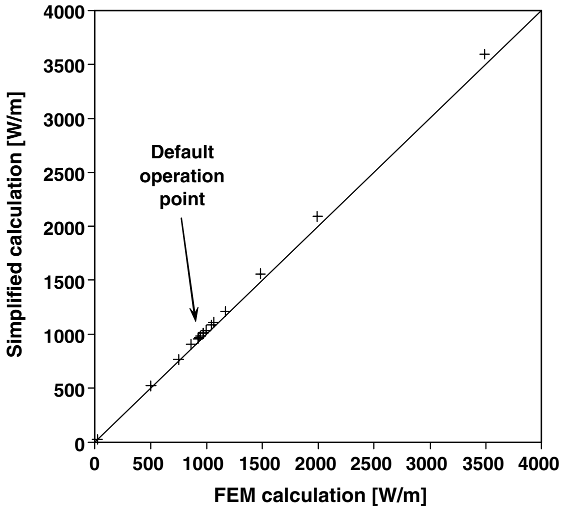


Fig. 3. A comparison of the heat transfer rate (above) and the corresponding surface temperatures T_{sw} (below) between the three methods. Initial data: $\alpha_g = 25\ \mathrm{W\,m^{-2}\,K^{-1}}$, $\alpha_w = 500\ \mathrm{W\,m^{-2}\,K^{-1}}$, $T_g = 600\ °C$, $T_w = 80\ °C$, $\lambda_f = 52\ \mathrm{W\,m^{-1}\,K^{-1}}$, $l = 50$ mm, $a = 50$ mm ($y = 0 \rightarrow 50$ mm), $s_f = 4$ mm, $s_p = 6$ mm, $s_{dg} = s_{dw} = 0$ mm.



		Default operation point			
α_g	10	25	50	100	W/m^{2}K
α_w	300	500	1500		W/m^{2}K
T_g	100	500	1500		°C
T_w	70	90	110		°C
s_{dg}	0	1	2		mm
s_{dw}		0	1		mm
l		50	25		mm
s_f	2	4			mm
a		50			mm

Fig. 4. A comparison between the heat transfer rates according to the FEM calculation and the assumption $\lambda_{f,y} = \infty$ (simplified calculation, plate at uniform temperature) when changing initial parameters.

on the upper portions. The latter gives the smallest rate of heat transfer, especially for the fin. With the assumption that $\lambda_{f,y} = 0$, the surface temperature T_{sw} rises unrealistically high at the place of the fin, see the lower graph. This decreases the heat transfer rate of the fin. Elsewhere, the temperature T_{sw} is lower than with the other methods. Further, with the assumption that $\lambda_{f,y} = \infty$, the temperature T_{sw} is constant and between the extreme values of the FEM calculation, which is shown in detail on the right axis. Because the base temperature of the fin is lower than with the FEM calculation, the heat transfer rate of the fin is higher.

Having proved to be more accurate against the FEM calculations, the assumption of $\lambda_{f,y} = \infty$ is evaluated in a more versatile way by changing initial data, see Fig. 4. The initial parameters are changed one by one from the default operation point to the assumed extremes occurring in the convection section of the biomass boiler.

The accuracy is fairly good. The FEM calculation gives $2,\ldots,6\%$ lower heat transfer rates than the assumption $\lambda_{f,y} = \infty$ in all cases.

Fig. 5 shows the heat transfer rate and the temperature field of the element (fin + plate) with a thick layer of deposit.

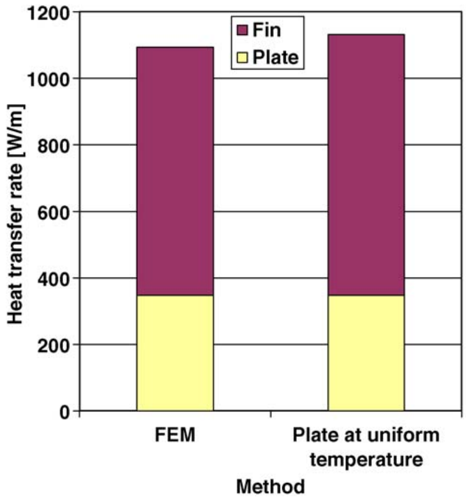
With the fouled fin, the assumption $\lambda_{f,y} = \infty$ gives a 3.6% higher heat transfer rate than the FEM calculation (1132–1093 W m^{-1}). With the clean fin, the difference is 3.7% (1536–1481 W m^{-1}, respectively), see Fig. 3. The case of a fouled fin in Fig. 5 is similar to the case of a clean fin in Fig. 3 as far as the fluid flows and geometry are concerned. By comparing the upper portions of Figs. 3 and 5, it can be seen that in this example, the fouling layer reduces the heat transfer rate by 26% with both methods.

3. Simulation

A calculating procedure for simulating the thermal performance of the entire convection section is developed. In the procedure, the passage of the convection section can be divided into an optional number of computational cells in the direction of the flue gas flow. The calculation algorithm is based on the simultaneous numerical solution of the enthalpy change and the total heat transfer rate of the finned plate elements and the end walls

$$-\dot{m}_{fg}\frac{\partial h}{\partial z} = n(\phi_b^f + \phi^p) + \phi^{ep} \tag{15}$$

The heat transfer rate of the end walls is calculated analogously with that of the plate. The heat transfer coefficient of the water side is assumed constant because it is usually one order of magnitude higher than that



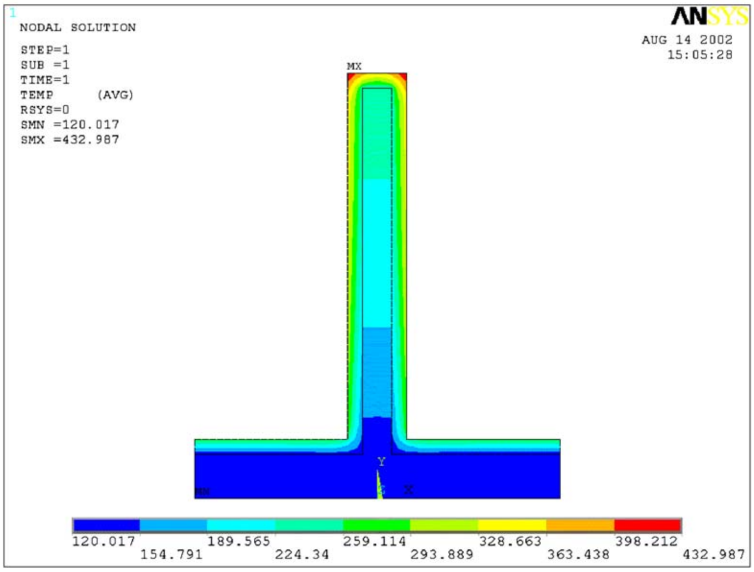


Fig. 5. A comparison of the heat transfer rate of a fouled fin between FEM and the assumption $\lambda_{f,y} = \infty$ (above) and the corresponding temperature field using the FEM calculation (below). The data for fouling is $s_{dg} = 2$ mm, $s_{dw} = 0$ mm, $\lambda_{dg} = 0.12$ W K^{-1} m^{-1}, other initial data is similar to the data in Fig. 3.

of the flue gas and, thus, is a minor source of error, as can be seen in Fig. 6. The heat transfer coefficient of the flue gas side is calculated by using the common correlation of Petukhov, Kirillov and Popov for the Nu number of smooth round tubes [10]. This is applied to the finned rectangular cross-section geometry by replacing the diameter with the hydraulic diameter of $d_h = 4A_{cs}/U$. The friction factor is calculated with the formula proposed by Zigrang and Sylvester [11]. The heat transfer properties and the radiative heat transfer of the flue gas are calculated according to its elementary composition [12,13]. The effect of the entrance region is calculated with Eq. (16) [6,10].

$$\overline{Nu}/Nu_t = 1 + C/x_d^m \tag{16}$$

The configuration of the entrance region is taken into account by the coefficient C and the exponent m. Because the configuration of the boiler type under review differs from those reported in the literature [5–7,10,14], the simulation is adjusted to measurements by choosing appropriate values for C and m. The properties of the

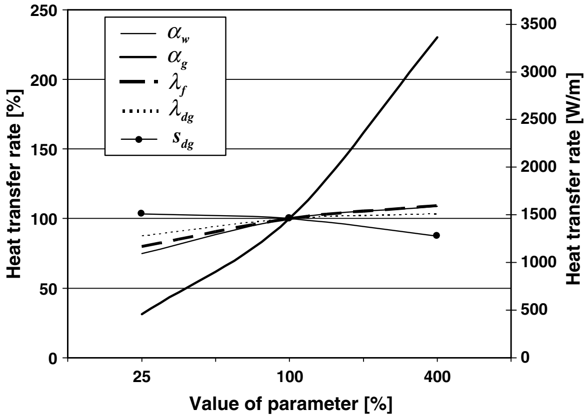


Fig. 6. The effect of some parameters on the heat transfer rate of an element (plate + fin). The assumed typical values at point 100% are $\alpha_g = 25$ W m^{-2} K^{-1}, $\alpha_w = 500$ W m^{-2} K^{-1}, $\lambda_f = 52$ W m^{-1} K^{-1}, $\lambda_{dg} = 0.1$ W m^{-1} K^{-1}, $s_{dg} = 0.25$ mm, $s_{dw} = 0$ mm.

flue gas are calculated at the mean temperature of the cell or of the boundary layer, where needed. The effect of mixed convection is considered according to the method proposed by Aicher and Martin [15]. The heat transfer coefficient is assumed to be constant in the entire cell, and the temperature of the ambient flue gas is assumed to be uniform at the whole cross-section (= at the plate and along the height of the fin).

Fig. 6 shows the sensitivities of the heat transfer rate of the element (plate + fin) to changes in the values of some key parameters when they are changed one by one from one fourth to four fold of the assumed typical value.

The heat transfer coefficient of the gas side has the strongest influence on the heat transfer rate, and the thermal conductivities have the weakest influence. However, the thickness of the fouling layer grows in the course of time, increasing the fuel consumption. Typically, the operation staff can only observe the flue gas exhaust temperature. With the simulation, it is possible to calculate the connection between the flue gas temperature, the loss in heat transfer rate and the thickness of the deposit, see Fig. 7. This helps estimate the appropriate cleaning periods, taking into account the fuel consumption and the cleaning costs. The initial data in Fig. 7 is that of the boiler Arimax BIO 300 operating at full load, see Section 4.

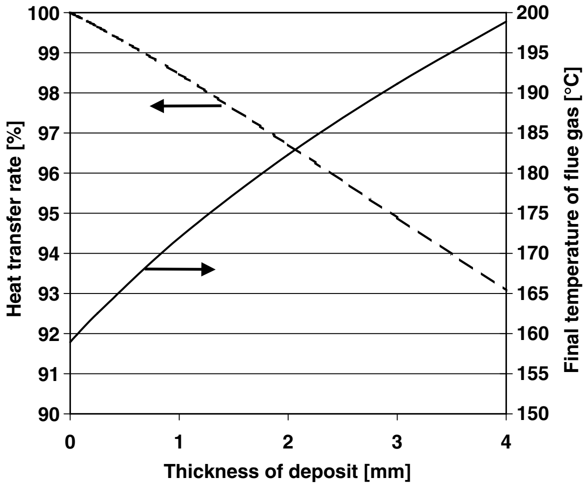


Fig. 7. The relative heat transfer rate of the convection section and the final temperature of flue gas as a function of thickness of the deposit.

4. Experiments

Verification measurements were conducted at the laboratory of VTT Processes in Jyväskylä on an Arimax BIO 300 boiler of Thermia Ltd. The measurements at VTT were performed on a test rig made according to the standard SFS-EN 303-5. Temperatures were measured after the combustion chamber and after the first, second and third (final) passes with K-type thermoelements. The concentrations of O_2, CO_2, total hydrocarbons (on line gas chromatograph) and NO of the flue gas were measured continuously. All measurements were logged at 15 s intervals. All measurement devices were calibrated according to the VTT quality standard. The calculation results were also compared with field measurements taken in an earlier research project on an Arimax BIO 500 boiler at the Pori Forest Institute in Kullaa [16] and with measurements taken on an Arimax BIO 1000 boiler at the Sports Institute of Finland in Vierumäki [1]. The boiler used at VTT was new, and the convection sections of the boilers in Kullaa and Vierumäki were cleaned just before the measurements.

In Fig. 8, the measured final flue gas temperatures of the passes of a convection section are compared to those calculated with the simulation procedure. In the calculations, the measured flue gas initial temperature and mass flow rate have been used as input data. The measurement results are mean values of a longer time period during steady state operation of the boiler.

The results of the boiler at 500 kW_{th} are shown only for the first pass. There was a regulating damper between the second and third pass, and the deviation of the mass flow rate was not known, see Fig. 1. The mean value of the magnitudes of the relative differences between the calculated and measured flue gas temperatures is 4.6%. The expanded relative uncertainty in the measurement of flue gas temperature is estimated to be ±5%. It is based on a standard uncertainty multiplied by a coverage factor of $k = 2$, which provides a confidence level of approximately 95%. The main contribution to the uncertainty is evaluated to be the uneven temperature profile in the flue gas duct. The temperature was measured only at one position of each cross-section.

5. Discussion

The presented calculation method includes several simplifying assumptions. Firstly, some of them have a minor influence on the result. The thermal conductivity of the plate-fin element is assumed to be infinite in the y direction. This leads to uniform base and plate temperatures. The FEM calculation leads to 2–6% lower values of the heat transfer rate in a wide range of initial data. Hence, by multiplying the result by

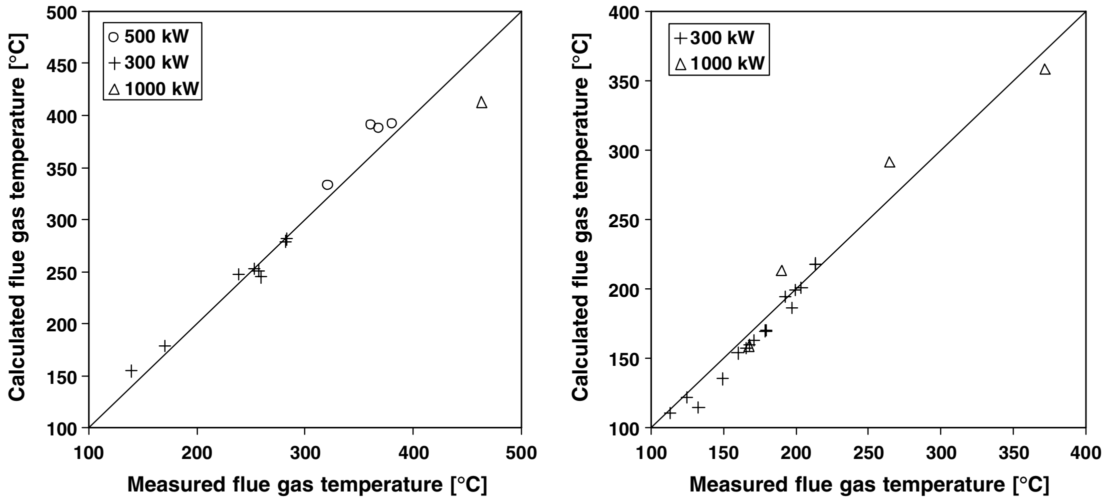


Fig. 8. A comparison between measured and calculated final flue gas temperatures of the first pass (on the left) and other passes (on the right) of the convection section. In the calculations $s_{dg} = s_{dw} = 0$ mm.

the constant factor of 0.96, the error is always below ±2% (compared to the FEM calculation), which is sufficient for practical dimensioning and developing. The thermal conductivity of an element is an input value in the model. Changing it from the default value of 52 $W K^{-1} m^{-1}$ to 13 and 208 $W K^{-1} m^{-1}$ leads to changes of −20% and 9% in the heat transfer rate, respectively, see Fig. 6. The conductivity is assumed to be independent of temperature, which is a usual idealizing assumption [2,3,18]. According to Ref. [13], the conductivity is 10–15% smaller at the tip temperature than at the base temperature. The convective heat transfer coefficient of the water side is assumed to be constant, and it is an input value in the model, too. The impact of changing it from the default value of $\alpha_w = 500 W m^{-2} K^{-1}$ to 125 and 2000 $W m^{-2} K^{-1}$ is –25% and 9% in the heat transfer rate, respectively. The first value is unrealistically low for heat transfer from a hot surface to water. A range of $300, \ldots, 600 W m^{-2} K^{-1}$ is reported to be appropriate for free convection [19]. With this more probable range, the changes of the heat transfer rate are –10% and 3%, respectively. Thus, the changes anticipated in the water side heat transfer coefficient have only a small effect on the heat transfer rate, see Fig. 6.

Secondly, the assumptions and the initial data concerning fouling are both doubtful and may have a significant effect on the result, see Figs. 3–6. Fouling is modelled in Eqs. (2), (4), (7), (9) and (11)–(14) by means of its thickness and thermal conductivity. The effect of fouling is often known only for its effect on the total heat transfer rate. In such case, fouling is described as a heat transfer resistance (=fouling factor R). However, in an illustrative simulation program, it would be easier for the user to perceive the mean thickness than the heat resistance value of the fouling. Hesselgreaves prefers the use of conductivity and thickness for these practical reasons. In this approach, it is possible to take simultaneously into account the demands for an acceptable decrease in heat transfer and increase in pressure drop [20]. Some information about the thermal conductivity of fouling in boilers is shown in Table 1. The thermal conductivity of boiler scale components on the water side is greater than that of the ash on the flue gas side, though a wide range of values is given for both ash and actual boiler scale. The assumptions of constant thickness of a fouling layer and its negligible heat conduction in the direction parallel to the fin, Eq. (1), are theoretical. The former may cause a significant error but the latter probably does not because it is one order of magnitude lower than the conductivity of the fin, at the minimum.

If there is only information about the heat resistance available, the model is easy to modify. In any case, the designer must decide about the anticipated decrease in heat output due to fouling. The simulation helps to evaluate it.

Thirdly, the heat transfer coefficient of the flue gas side was found difficult to estimate, having a most significant effect on the results. Yapici and co-workers have reported an uncertainty of 10.9% in the Nusselt number for laboratory tests of pin finned surfaces at the comparable Re number range [17]. Especially with the in situ measurements (500 and 1000 kW_{th}), the uncertainty may be even higher. Radiation of the emitting gas components CO_2 and H_2O has been included in the model, but radiation between the fin and the plate is excluded. The effect of gas radiation was found to be 3–10% of the total heat transfer because the temperature of the flue gas and the concentrations of the emitting components are rather low and the effective thickness of the gas layer is small.

The convective heat transfer coefficient is assumed to be constant along the height of the fin. Mokheimer has found a marked improvement of the fin efficiency under natural convection by using a variable heat

Table 1
Thermal conductivities of fouling

Material	Conductivity λ ($W K^{-1} m^{-1}$)	Source
Ash (coal)	$0.02, \ldots, 1.9$	[7]
Ash (oil)	$0.05, \ldots, 1.9$	[7]
Calcium carbonate	2.9	[5]
	2.19	[21]
Calcium sulphate	2.3	[5]
	0.74	[21]
Ferrous oxide (magnetite)	2.9	[5,22]
Boiler scale	$0.081, \ldots, 2.2$	[7]
	2.0	[23]

transfer coefficient [24]. Naik et al. have measured three to four fold values for the local Nusselt number at the tip of the fin compared to that at the base of it in turbulent forced convection if there is a clearance above the fin [25]. However, the interfin distance in the considered convection section is large, 3.75 fold compared to the test arrangement of Ref. [25] and more than double that compared to the optimal value calculated by the empirical correlation proposed in Ref. [25]. Therefore, the change in the local value of *Nu* probably is smaller. The fin spacing in the convection section is large because of cleaning reasons. The deposit is typically mechanically removed, and the array of fins must not make it difficult.

The convective heat transfer coefficient changes in the direction of the flow because of the developing flow conditions and the changes in the properties of the gas. Laor and Kalman state that the local heat transfer coefficient of a longitudinal fin depends on the location along the flow path in turbulent forced convection [2]. Naik et al. describe a significant improvement in the heat transfer coefficient in the middle of a long rectangular fin compared to that at its end [25]. Here, the correlation of the entrance region of the tube without an extended surface is applied. The experimental in situ arrangement allowed measuring only the inlet and final temperatures of the flue gas for each pass of the convection section. Thus, the effects of the entrance region could not be studied. The calculation results have been adjusted by the measurements by choosing appropriate values for the coefficients C and m in the correlation. With the selected values of the coefficients, a sufficient accuracy was achieved for all boilers tested, but it cannot be assumed that those values are applicable for other boilers. The simulation of different types of boilers requires further measurements and fitting the calculation results to them. After this, changes in operating conditions can be described with a reasonable accuracy, as can be concluded from Fig. 8.

6. Conclusions

The aim of this study was to develop a calculation procedure for designing and simulation of the convection section of a biomass boiler. Heat transfer was modelled on the finned surface with a fouling layer. In a few details, this modification differs from the equations for the clean finned surface. The assumption of uniform base temperature was compared to the results from the finite element method using the calculation program ANSYS. The verification measurements were conducted at the laboratory of VTT Processes on a 300 kW_{th} boiler. The calculation results were also compared to field measurements on one 500 and one 1000 kW_{th} boiler. The main discoveries of this study are:

- The simplified method of heat conduction calculations on the finned surface with fouling layer achieves a degree of error below ±2% compared with the FEM calculations within the operating range of biomass boilers. In this method, the temperature distribution of the wall is considered uniform in the direction parallel to the plate.
- The heat transfer coefficient from the flue gas to the surface was indicated clearly to be the most influential parameter on the heat transfer rate of the convection section in the range of biomass boilers.
- According to the verifying measurements, the simulation seemed to describe the effects of changes in the operating conditions with reasonable accuracy. However, additional measurements are needed for convection sections of different shapes or sizes to ensure an accurate result for a certain operating stage.

Using the model, it is possible to study the performance of different convection sections equipped with rectangular longitudinal fins of constant thickness, and it can be used as a tool for dimensioning and development work. It also helps the operation staff to estimate the appropriate cleaning period.

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