APPENDIX F

Paper VI

Reprinted from Yrjölä J, Paavilainen J, Sillanpää M. Modelling and experimental studies on heat transfer in the convection section of a biomass boiler. Int. J. Energy Research, 2006;30 (12):939 – 953, with permission from John Wiley & Sons.

© 2006 John Wiley & Sons

INTERNATIONAL JOURNAL OF ENERGY RESEARCH
Int. J. Energy Res. (in press)
Published online in Wiley InterScience (www.interscience.wiley.com). DOI: 10.1002/er.1196

Modelling and experimental studies on heat transfer in the convection section of a biomass boiler

Jukka Yrjölä[1,*,†], Janne Paavilainen[1] and Matti Sillanpää[2]

[1] *Satakunta Polytechnic, Tekniikantie 2, FIN-28600 Pori, Finland*
[2] *Thermia Ltd., P.O. Box 59, FIN-43101 Saarijärvi, Finland*

SUMMARY

This paper describes a model of heat transfer for the convection section of a biomass boiler. The predictions obtained with the model are compared to the measurement results from two boilers, a 50 kW_{th} pellet boiler and a 4000 kW_{th} wood chips boiler. An adequate accuracy was achieved on the wood chips boiler. As for the pellet boiler, the calculated and measured heat transfer rates differed more than expected on the basis of the inaccuracies in correlation reported in the literature. The most uncertain aspect of the model was assumed to be the correlation equation of the entrance region. Hence, the model was adjusted to improve the correlation. As a result of this, a high degree of accuracy was also obtained with the pellet boiler. The next step was to analyse the effect of design and the operating parameters on the pellet boiler. Firstly, the portion of radiation was established at 3–13 per cent, and the portion of entrance region at 39–52 per cent of the entire heat transfer rate under typical operating conditions. The effect of natural convection was small. Secondly, the heat transfer rate seemed to increase when dividing the convection section into more passes, even when the heat transfer surface area remained constant. This is because the effect of the entrance region is recurrent. Thirdly, when using smaller tube diameters the heat transfer area is more energy-efficient, even when the bulk velocity of the flow remains constant. Copyright © 2006 John Wiley & Sons, Ltd.

KEY WORDS: entrance region; heat transfer; mixed convection; natural convection

1. INTRODUCTION

The aim of this study is to present a model of heat transfer for the convection section of a biomass boiler. The model helps a design or development engineer to optimize the geometry of the convection section. A need to improve the boiler design arises from the currently expanding markets for biomass boilers in an effort to restrict the emissions of greenhouse gases. Burning

*Correspondence to: Jukka Yrjölä, Satakunta Polytechnic, Tekniikantie 2, FIN-28600 Pori, Finland.
†E-mail: jukka.yrjola@samk.fi

Contract/grant sponsor: National Technology Agency of Finland; contract/grant number: 40241/01
Contract/grant sponsor: Nakkila Works
Contract/grant sponsor: Thermia Ltd.
Contract/grant sponsor: Timarcon Ltd.

Received 24 June 2004
Revised 6 December 2005
Accepted 15 December 2005

Copyright © 2006 John Wiley & Sons, Ltd.

woody biomass has almost zero net greenhouse effect. With biomass boilers both heat demand and fuel quality (moisture content, particle size) may vary to a large extent during the heating season. Modelling helps the designer to understand and predict the thermal performance of a boiler and to improve its efficiency.

Traditionally, the convection section of a small-scale biomass boiler is constructed of circular tubes. Inside the tubes, flue gas flows in a single phase, while there is hot boiler water outside the tubes. A heat transfer problem such as this is a classic. It is generally known that in heat exchange between liquid and gas, the greatest resistance to heat transfer is present between gas and the heat transfer surface while the resistances of the separating wall and that on the liquid side of the wall are small. Accordingly, the rate of heat transfer of the flue gas side controls the total rate. The ranges of the flue gas flow rate and that of temperature are large. Therefore, the model must incorporate a wide range of Reynolds numbers from turbulent to laminar flow, and also a nonlinear temperature dependency relative to the heat transfer properties of flue gas and boiler water. During the past few decades numerous scientists have presented correlation equations for heat transfer of the round tubes. For example, Prandtl, Karman, Nikuradse, Gnielinski, and also Petukhov, Kirillov and Popov have devised correlations for fully developed turbulent flow. The Nusselt number correlations use Reynolds and Prandtl numbers and the relative roughness of the tube surface as input data, and they have fully been described, e.g. in the reference book by Aung *et al*. (1987). The simple correlations of Dittus-Boelter and Colburn for smooth and circular tubes are well-known (DeWitt, 1996). Ceylan and Kelbaliyev have summarized the results of the recent research concerning the effect of roughness on heat transfer (Ceylan and Kelbaliyev, 2003). For laminar flow conditions, correlations have been presented among others in sources VDI-Heat Atlas (1993), Wagner (1988) and Aung *et al*. (1987).

Despite the many available correlations, designing a model for total heat transfer in a full-scale convection section is challenging. Firstly, the correlations typically describe one aspect of heat transfer, such as radiation, forced convection, natural convection, or effect of the entrance region. Therefore, when designing a model, several equations need to be combined, such as the correlations of the laminar and fully-developed turbulent flow at the transition flow region. Aicher and Martin have used a weighed average method (Aicher and Martin, 1997), but a different strategy is chosen in this paper. Secondly, full-scale equipment typically differs from an experimental apparatus used in the laboratory. For instance, the configuration of the convection section entrance differs from that used in laboratory tests. Grass (Grass, 1956), Mills (Mills, 1995) and Sukomel (Isachenko *et al*., 1977) have presented tests results of the constructions which resemble the entrance region of a convection section.

To give credible results, the model needs to be adjusted since there are often practical limitations on conducting a full-scale verification measurement program. Therefore, a strategy was needed showing how to adjust the model to suit a program which only uses few measuring data. This paper presents the measurement results of two boilers, a 50 kW_{th} wood pellet boiler and a 4000 kW_{th} wood chips boiler. The calculated rate of heat transfer of the pellet boiler was 40% smaller than the rate measured. This is definitely more than the error expected on the basis of the correlations reported in the literature.

The scheme of the wood pellet boiler and the wood chips boiler in this study are presented in Figure 1. The pellet boiler is used for residential heating and the wood chips boiler for a small district-heating network. Both of them produce hot water.

Copyright © 2006 John Wiley & Sons, Ltd.

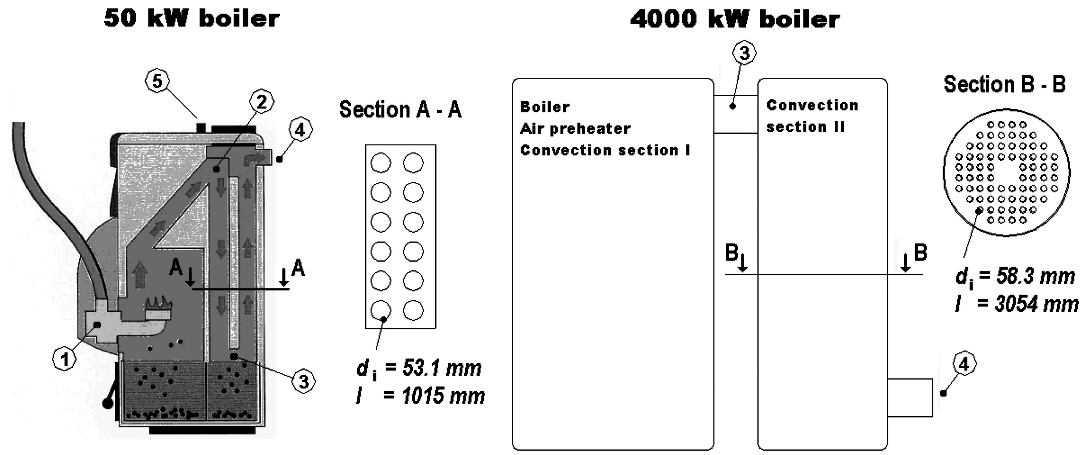

Figure 1. Principal schema of wood pellet boiler 50 kW_{th} and Wood Chips Boiler 4000 kW_{th} and their convection sections. Numbers refer to the experimental information in Table I.

Possible reasons for the error are discussed, and the most likely cause is attributed to the effect of the entrance region. Neshumayev *et al.* have discovered a similar feature, also attributing the most probable cause for error to the effect of the entrance region (Neshumayev *et al.*, 2003). After correcting the correlation of the entrance region the model gives accurate results. Finally, the thermal performance of the convection section is analysed using the corrected model.

2. MODELLING

Heat transfer on the gas side consists of radiation and convection, $h_g = h_r + h_c$. The heat transfer coefficient of radiation is calculated in accordance with VDI-Heat Atlas (1993).

$$h_r = \varepsilon_s \sigma(\varepsilon_g T_g^4 - \alpha_{g,s} T_s^4)/\{[1 - (1 - \varepsilon_s)(1 - \alpha_g)](T_g - T_s)\} \tag{1}$$

The radiation heat transfer is less significant in the convection section. This is due to the small diameter of the tubes, a small amount of fly ash as well as small concentrations of the emitting gas components of CO_2 and H_2O, and a relatively low temperature (VDI-Heat Atlas, 1993). Initial and final temperatures are typically in the range of 250–600°C and 100–250°C, respectively. The convection heat transfer coefficient is calculated using

$$h_c = \mathrm{Nu} k_g / d \tag{2}$$

For a *fully developed turbulent flow* Petukhov, Kirillov and Popov have proposed Equation (3), valid for $4000 \leqslant \mathrm{Re} \leqslant 5{*}10^6$ and $0.5 \leqslant \mathrm{Pr} \leqslant 10^6$ with an accuracy of $\pm$ 5 per cent as compared to the experimental data (Aung *et al.*, 1987)

$$\mathrm{Nu_t} = (f/8)(\mathrm{Re} - B)\mathrm{Pr}/[A + 12.7\sqrt{f/8}(\mathrm{Pr}^{2/3} - 1)] \tag{3}$$

where $B = 0$ and

$$A = 1.07 + 900/\mathrm{Re} - 0.63/(1 + 10\mathrm{Pr}) \tag{4}$$

The friction factor f can be calculated by using the explicit Equation (5) as proposed by Zigrang and Sylvester (1982)

$$f = \{-2.0\,\log\{k_s/(7.4R) - 5.02\,\log[k_s/(7.4R) + 13/\mathrm{Re}]/\mathrm{Re}\}\}^{-2} \tag{5}$$

Equation (3) is valid for relatively smooth pipes. If the inner surface is roughened with a view to increase the heat transfer rate, a correlation other than Equation (3) should be considered.

At *the entrance region*, the heat transfer is more efficient in comparison with the fully developed turbulent flow because of the undeveloped velocity and temperature profiles. A general Equation (6) for the relation of the mean Nusselt number in the entrance region, $z(= l/d_i) < 60$, and the Nusselt number for fully developed turbulent flow have been presented in (Aung *et al.*, 1987; DeWitt, 1996). In this paper, this relation is called the Nu correction factor.

$$\overline{\mathrm{Nu}}/\mathrm{Nu_t} = 1 + C/z^m \tag{6}$$

The values of constants C and m depend on the configuration of the entrance. By choosing the most appropriate configuration as compared with the case of convection section in Figure 1 of the pellet boiler, Hausen proposes $C = 1$, $m = 2/3$ (Hausen, 1943), Grass $C = 2.3$, $m = 1$ (Grass, 1956), Mills $C = 2.4$, $m = 0.68$ (Mills, 1995) and Sukomel $C = 0.83$, $m = 0.67$ ($z < 15$), $C = 0$ ($z \geqslant 15$) (Isachenko *et al.*, 1977). To illustrate the effect of the entrance region, these four are compared in Figure 2.

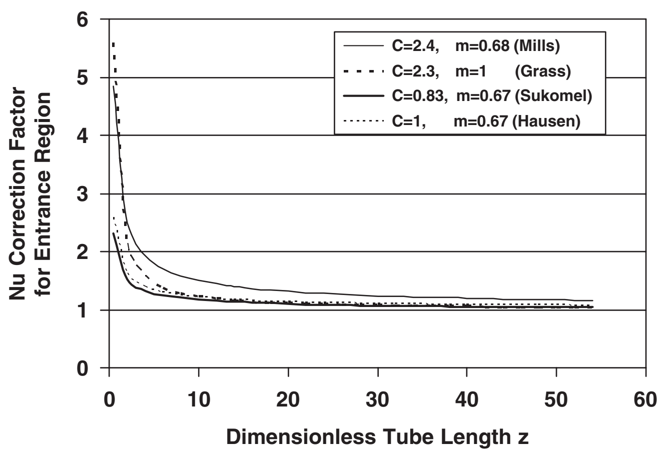

Figure 2. Mean Nu correction factor of the entrance region for turbulent flow depending on the dimensionless tube length.

Copyright © 2006 John Wiley & Sons, Ltd.

The flue gas cools down in the convection section, and its velocity and fluid properties change all along the tube. To improve the accuracy of the model, the tube is divided into an optional number of computational cells in the direction of flow, if needed. Therefore, the local Nu correction factor is solved from Equation (6). The mean and local Nusselt numbers are shown in

$$\overline{\mathrm{Nu}} = \mathrm{Nu}_\mathrm{t}(1 + C/z^m) = z^{-1} \int_0^z \mathrm{Nu}\, \mathrm{d}z \tag{7}$$

This yields for the local Nu

$$\mathrm{Nu} = [1 + (1 - m)Cz^{-m}]\mathrm{Nu}_\mathrm{t} \tag{8}$$

Integrating Equation (8) from z_i to z_{i+1}, a mean Nusselt number for each part of the tube can be calculated

$$\mathrm{Nu}_{z_{i+0.5}} = \mathrm{Nu}_\mathrm{t}[z_{i+1} - z_i + C(z_{i+1}^{1-m} - z_i^{1-m})]/(z_{i+1} - z_i) \tag{9}$$

Laminar flow conditions of the flue gas may occur at low heating loads. The operating range of a boiler for heating residential buildings is large, with a minimum heat output of approximately 10–20 per cent of the peak load. The mean Nusselt number of the laminar flow at a constant surface temperature of the tube is calculated using Equation (10), valid for $0 < \mathrm{Gz} < \infty$ (VDI-Heat Atlas, 1993)

$$\mathrm{Nu}_\mathrm{l} = \{49.731 + (1.615\mathrm{Gz}^{1/3} - 0.7)^3 + \{[2/(1 + 22\mathrm{Pr})]^{1/6}\sqrt{\mathrm{Gz}}\}^3\}^{1/3} \tag{10}$$

The equation is based on the assumption of constant surface temperature because the temperature gradient of water is small in comparison with that of flue gas. Equation (10) takes into account the entrance region. The Nusselt numbers of the laminar and turbulent flow are compared and the higher of them is used in the calculation.

Mixed natural and forced turbulent convection may have significant effect on the Nusselt number. As for the vertical tubes, Aicher has summarized the previous experimental work and proposed empirical correlations for the aided and opposed mixed convection, Equations (11)–(14) (Aicher and Martin, 1997). They are valid within an operating range typical of the biomass boilers. When the pressure gradient and the density gradient act as driving forces in opposite directions, the correlation is

$$\mathrm{Nu}_{\mathrm{m,o}} = \sqrt{\mathrm{Nu}_\mathrm{t}^2 + \mathrm{Nu}_\mathrm{n}^2} \tag{11}$$

where Nu_n is the Nusselt number for natural convection

$$\mathrm{Nu}_\mathrm{n} = 0.122\mathrm{Ra}^{0.333}[1 + (0.492/\mathrm{Pr})]^{-0.03} \tag{12}$$

This is the situation in the second pass of the convection section of the wood pellet boiler where the fluid is cooled, flowing upward (see Figure 1). The Nusselt number of the opposing mixed convection is always higher than that of the pure forced convection as can be seen in Equation (11).

Copyright © 2006 John Wiley & Sons, Ltd.

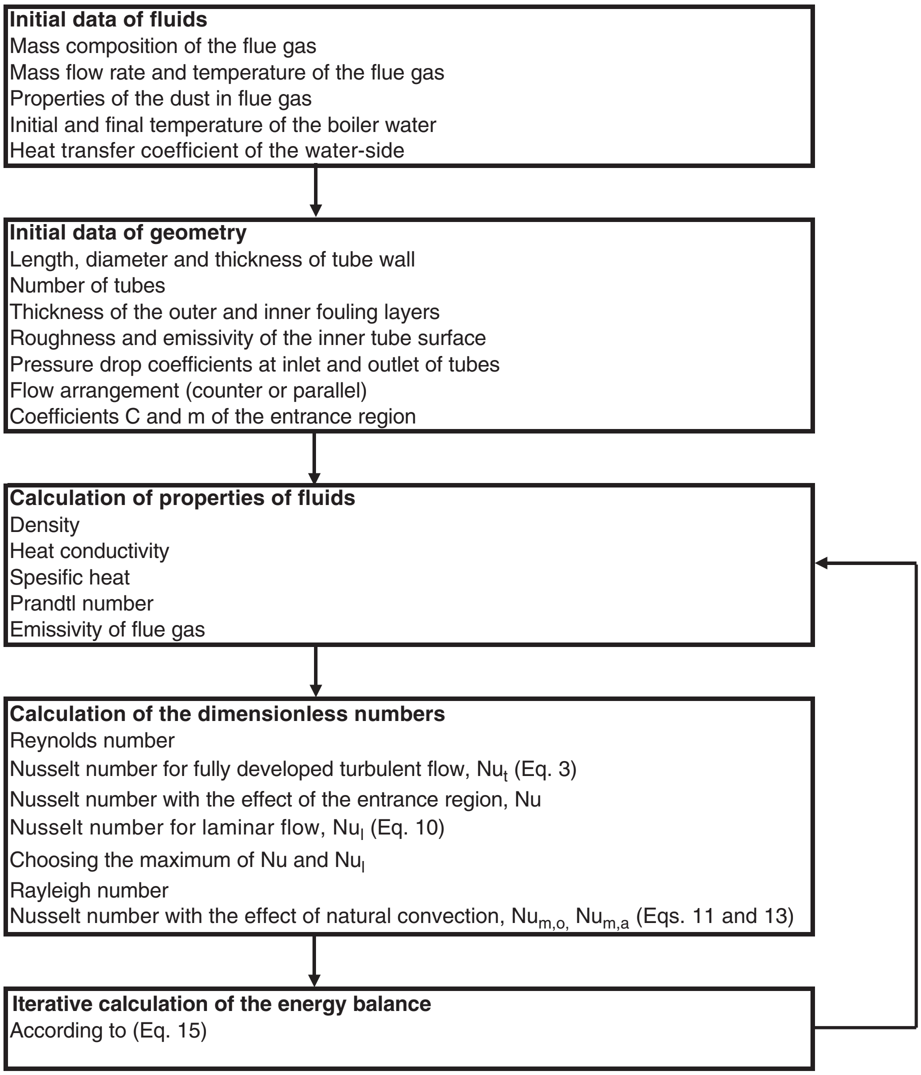

Figure 3. Progress of the heat transfer calculations.

When the pressure and density gradients act as driving forces in the same direction, the correlation for the aiding mixed convection is

$$\mathrm{Nu}_{m,a} = \mathrm{Nu}_{m,o}\{1 - (1 - 2\mathrm{Nu}_l/\mathrm{Nu}_{m,o}) \exp\{-1.3[P/(1 - |P|) + 0.5]^2\}\} \tag{13}$$

$$P = (\mathrm{Nu}_n - \mathrm{Nu}_t)/\mathrm{Nu}_{m,o} \tag{14}$$

This is the situation in the first pass of the convection section where the fluid is cooled, flowing downwards. The Nusselt number of the aided mixed convection can be lower than the Nu of the pure forced convection as can be seen in Equation (13). This is because the diffusive energy

Copyright © 2006 John Wiley & Sons, Ltd.

transport controls the heat transfer from the core of the turbulent flow to the border of the viscous layer. At a certain velocity the buoyancy force causes laminarization of the turbulent flow and thus declines the heat transfer. The calculation results of the mixed convection are stated in Chapter ANALYSES.

The *energy balance* Equation (15) is solved numerically for each computational cell

$$\begin{aligned} -\dot{m}_{\mathrm{g}} c_p (T_{\mathrm{g},z_{i+1}} - T_{\mathrm{g},z_i}) &= h_{\mathrm{g}} A_{\mathrm{i}} (\bar{T}_{\mathrm{g}} - \bar{T}_{\mathrm{si}}) \\ &= \frac{n\pi (z_{i+1} - z_i) d_{\mathrm{i}}}{R_{\mathrm{i}} + \dfrac{\ln(d_{\mathrm{o}}/d_{\mathrm{i}})}{2k_{\mathrm{p}}} + R_{\mathrm{o}}} (\bar{T}_{\mathrm{si}} - \bar{T}_{\mathrm{so}}) = h_{\mathrm{w}} A_{\mathrm{o}} (\bar{T}_{\mathrm{so}} - \bar{T}_{\mathrm{w}}) \end{aligned} \tag{15}$$

The heat transfer coefficient of the waterside, h_{w}, is assumed constant, because it is typically one order of magnitude greater than h_{g}. The heat transfer resistances of the fouling layer and the heat conductivity of the tube wall material are used as input data, so are the mass flow rate, initial temperature and composition of the flue gas. When calculating the Reynolds number, the kinematic viscosity is calculated at the mean temperature of the bulk flow, while the other heat transfer properties are calculated at the mean temperature of the boundary layer. The properties are calculated according to the molar composition of the flue gas with the method described in VDI-Heat Atlas (1993).

3. EXPERIMENT

The performance of the convection section of the wood pellet boiler was measured at the laboratory of Thermia Ltd in Saarijärvi and that of the wood chips boiler on the Suomussalmi site in Finland.

Pellets made by compressing powdered wood chips, carpentry residue or sawdust were the fuel source for the 50 $\mathrm{kW_{th}}$ boiler. The fuel source of the 4000 $\mathrm{kW_{th}}$ boiler (Experiments 3–7) was a mixture of 60–80% wood chips, 0–30% bark and 10–20% REF.

The measurement and calculation results are shown in Table I. The subscript identifiers refer to the measured quantities and measuring points shown in Figure 1. The mass flow rate of the

Table I. Measurement and calculation results of the pellet boiler (Experiments 1 and 2) and wood chips boiler (Experiments 3–7).

Experiment	$\dot{m}_{\mathrm{f1}}$ (kg/s)	t_{wo5} (°C)	t_{wi5} (°C)	q_{w5} ($\mathrm{m^3/h}$)	t_{fg2} (°C)	t_{fg3} (°C)	t_{fg4} (°C)	$O_{2,\mathrm{fg4}}$ (%)†	CO_{fg4} (ppm)†	$\dot{m}_{\mathrm{fg4}}$ (kg/s)	Re*	Nu*	h_i^* ($\mathrm{W/m^2K}$)
1	3.24×10^{-3}	77.5	59.0	2.08	546	286.0	163.0	3.7	800	24.0×10^{-3}	3460	17	11
2	2.63×10^{-3}	82.5	62.0	1.44	450	235.0	140.0	7.3	178	24.3×10^{-3}	2460	17	11
3	0.309*	105.3	52.2	58.38		318.7	180.6	8.44	3450	2.27	36660	115	64
4	0.343*	101.5	54.5	76.95		331.8	171.9	6.74	1114	2.51	37600	118	66
5	0.292*	103.7	53.0	58.86		318.9	164.3	7.54	826	1.94	38400	130	73
6	0.470*	91.7	56.9	112.09		330.3	174.2	7.45	394	2.67	40390	126	71
7	0.362*	102.1	56.7	61.86		290.4	154.6	8.03	1103	2.56	40350	126	69

*Calculated.
†Dry gas.

Copyright © 2006 John Wiley & Sons, Ltd.

fuel was measured directly by weighing in Experiments 1 and 2, and it was calculated by means of heat output, fuel characteristics, temperature and flue gas concentration of $O_{2,fg4}$ and CO_{fg} in Experiments 3–7. The concentrations of oxygen and carbon monoxide were measured using an analyzer (KM 9106 Quintox). Temperatures were measured by K-type thermoelements, and data was collected by a logger (Grant Squirrel). The dynamic pressure was measured by a Pitot tube and an electronic micromanometer (Alnor Micromanometer MP6KSR).

The presented Reynolds and Nusselt numbers and the heat transfer coefficient of the flue gas side are mean values of the whole convection section.

The measured results of the final temperature and the heat transfer rate of the convection section are compared to the predictions produced by the model in Figure 4. The boiler was run at a constant heating load. Each of the presented temperatures is the mean value of a measuring period of 30–70 min.

The heat transfer resistance caused by fouling is omitted from calculations because the pellet boiler was new and the wood chips boiler had recently been swept. Establishing the final temperatures and the heat transfer rates of the convection section constitute the main results of the calculations. The entrance region is taken into account in accordance with Equation (10)

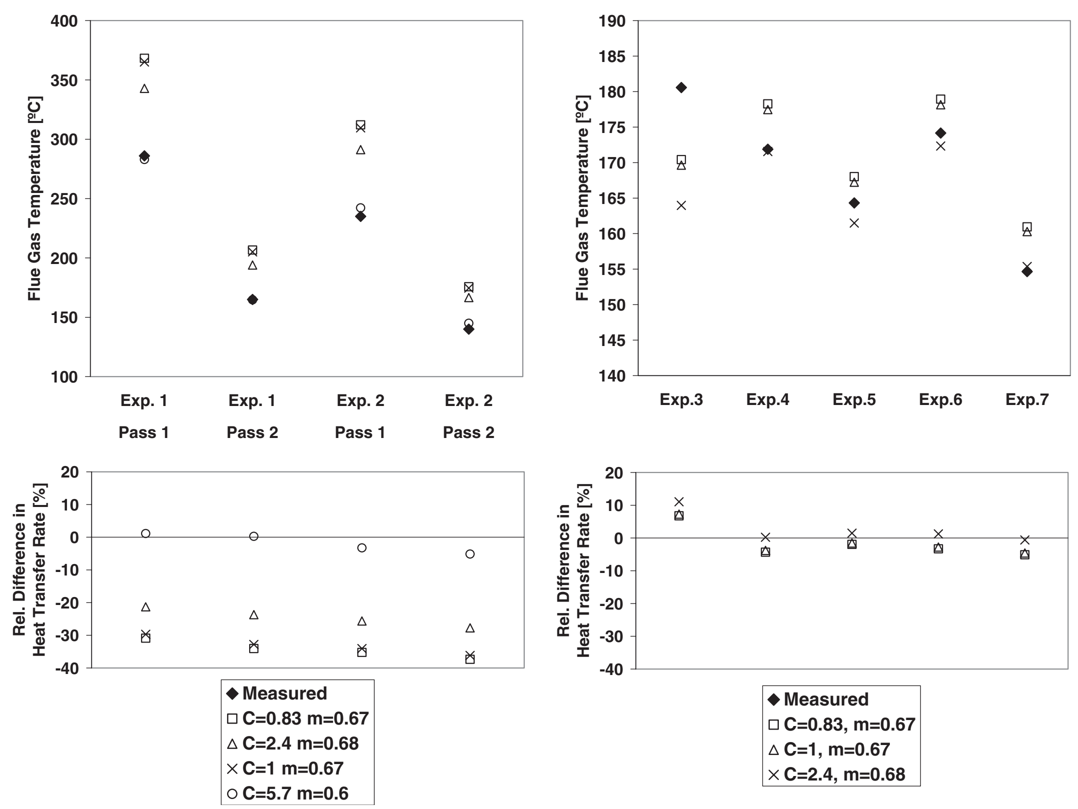

Figure 4. Comparison of measured and calculated temperatures, and relative difference between calculated and measured heat transfer rate of the convection section. Values of the pellet boiler are on the left and those of the wood chips boiler on the right.

Copyright © 2006 John Wiley & Sons, Ltd.

where the constants C and m are defined as referred to by Hausen (1943), Mills (1995) and Isachenko *et al*. (1977) (see Figure 2). The dimensionless length z of a tube is 19.1 on the pellet boiler and 60.1 on the wood chips boiler. On the pellet boiler the effect of the entrance region is strong while the calculated temperature after the first pass is 84–109°C higher than the measured one, and after the second pass 37–53°C, respectively. The mean heat transfer rate is 37–44% smaller than the measured rate, when using correlations from Mills (1995) and Isachenko *et al*. (1977), respectively. To correct the values of the constants in Equation (10) they were chosen so that the mean heat transfer rate would be accurate enough in comparison with the measured values. There are several options for values C and m of which $C = 5.7$ and $m = 0.6$ were chosen. Given the values the relative difference between measured and calculated heat transfer rates varied between -1.1 and 2.5% in all cases.

Contrary to the above, the accuracy on the wood chips boiler is significantly better. The calculated final temperatures of the wood chips boiler differ by -6 to 2°C from the measured ones in Experiments 4–7, and by 10–17°C in Experiment 3. The calculated heat transfer rates differ by -5.1 to -1.5% from the measured ones in Experiments 4–7, and in Experiment 3 by 6.8–11.1%. Conducted on the first day of measurement Experiment 3 differs from the others. The experiments were performed on three subsequent days, and thermocouples were removed at the end of each day and duly reinstalled the following day. The installation may have been defective the first day. Nevertheless, the mean value of the magnitude of relative difference in the heat transfer rate is 3.7%. Therefore, no correction to the values of C and m is needed.

4. ANALYSES

All analyses are calculated for a wood pellet boiler, having a full load condition as a reference point. When increasing the mass flow rate of flue gas from 33 to 200% of the full load value (Experiment 1, pass 2) of the pellet boiler, the effect of radiation decreases from 12.9 to 3.0% and the effect of entrance region increases from 39 to 50%. This is because with increasing mass flow rates the convective heat transfer is enhanced. Under typical operating conditions the effect of natural convection is small, and at a mass flow rate of 80–180% of the full load value the effect is negative. This is caused by the laminarization effect of buoyancy forces on turbulent flow which decreases heat transfer. However, at a mass flow rate of below 80% the effect is positive, reaching 14.6% at a rate 33% of the full load.

The next four analyses depict the designer's possibilities to exploit the simulation method for optimizing the construction.

4.1. Analyses 1 and 2

When using straight and smooth tubes the designer is able to change three geometrical parameters of a tube: number, n, length, l, and diameter, d. In Figures 5 and 6 the parameters vary but the initial data is the same as in Table I, Experiment 1.

The convection section of the pellet boiler has two passes. The effect of dividing it into one or three passes is shown in Figure 5 on the left (n, d and total l are constant). The tube diameter and the heat transfer area are constant. It means that with one pass the tube length is double and with three passes it is 2/3 compared to the reference case. With three passes the heat transfer rate increases by 6%, and with one pass it decreases by 11%. The final flue gas temperature decreases

Copyright © 2006 John Wiley & Sons, Ltd.

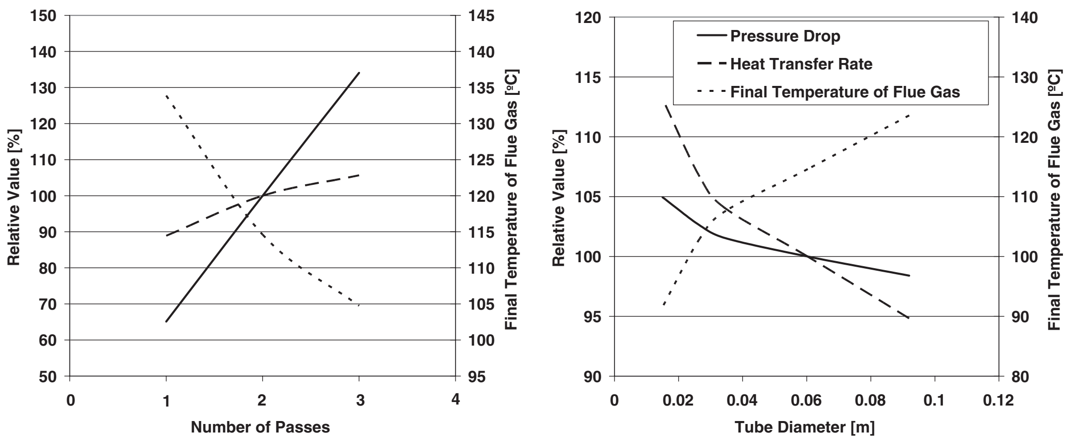

Figure 5. Effect of dividing the convection section into 1, 2 or 3 passes (on the left) and effect of using different tube diameters (on the right). To the right, total cross-section area and inner heat transfer surface area of tubes remain constant.

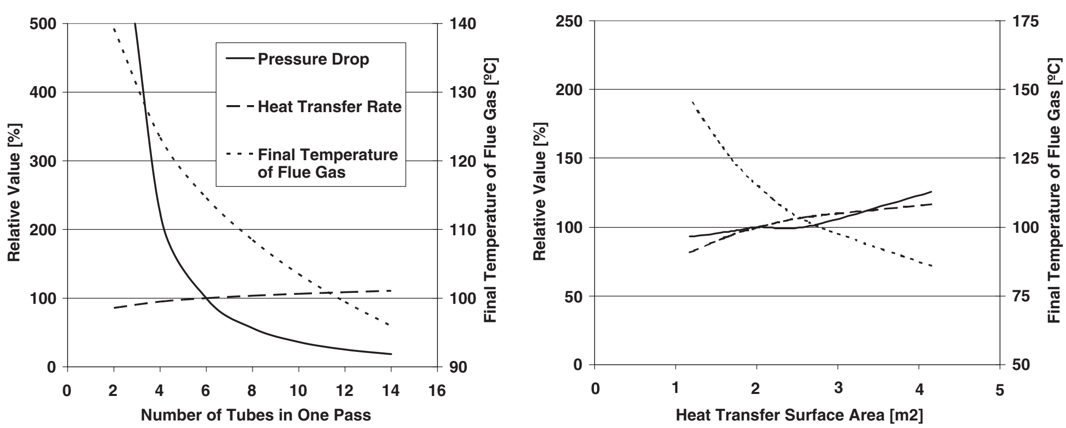

Figure 6. The effect of decreasing or increasing the number of convection tubes from the original value $n = 6$ (on the left) and decreasing or increasing the heat transfer area from the original value (on the right).

by 10°C and increases 19°C, respectively. This is caused by the effect of the entrance region. However, the gradient of the heat transfer rate decreases when the number of passes increases whilst Figure 3 would suggest an increase in gradient. The reason for this is that concurrently with the decreasing difference in temperature between flue gas and water the heat transfer rate also decreases.

The pressure drop becomes double when adding the number of passes from 1 to 3. Hence, the inlet and the outlet of the tubes dominate the pressure drop. Due to friction the pressure drop for the entire tube length is equal to the pressure drop of the inlet and outlet of one pass.

Copyright © 2006 John Wiley & Sons, Ltd.

The outer diameter of the pellet boiler tube is 60.3 mm. Tube diameters are changed using one larger and three smaller (see Figure 5 on the right). The inner heat transfer area and the cross sectional flow area are constant (i.e. also n and l of the tubes change). When decreasing the diameter the initial bulk flow velocity stays constant, but the Reynolds number decreases and the friction factor slightly increases. It also follows that the value of Nu decreases but not so fast as the diameter. Therefore, the relative heat transfer rate increases more than the pressure drop. It means that, using smaller diameters, the heat transfer area can be exploited in a more efficient way. Modern bio-fuel boilers have an impeller for combustion air or flue gas so that natural draft is not the only force driving the flow. In principle the selection of tube diameter is a question of economic optimization.

4.2. *Analyses 3 and 4*

In Figure 6 the heat transfer area is changed using two different methods. On the left, the number of tubes n varies (d and l are constant), and on the right, d and n vary keeping the total cross-flow section area of the tubes constant (i.e. l and initial v are constant).

The heat transfer rate decreases by 15% when the number of tubes decreases from the original 6 to 2, and the rate increases by 9% when the number of tubes increases from 6 to 12 (see Figure 6 on the left). The gain with 8 additional tubes is smaller than the loss with 4 tubes less. Due to the decrease in flue gas velocity the inner heat transfer coefficient decreases when the number of tubes is increased. Secondly, the mean temperature difference between flue gas and boiler water decreases which decreases the heat transfer rate. On the other hand, the pressure drop increases by 800% and decreases by 75%, respectively.

When the heat transfer area is increased keeping the total cross-flow section area of tubes constant the heat transfer rate increases more than in the previous case (see Figure 6, on the right). The increase is 16% when the heat transfer is doubled from the original value $A = 2.03\,\mathrm{m}^2$. This is because the changes of the inner heat transfer coefficient are smaller due to the constant initial velocity of the flue gas. Unlike the previous case, the pressure drop changes in the opposite direction because the friction factor increases with decreasing tube diameters. However, the range of the pressure drop is smaller, from 92 to 125% when the heat transfer increases from 56 to 200%.

5. DISCUSSION

A significant difference was detected between the measured and calculated heat transfer rates of the wood pellet boiler. Possible reasons are proposed:

Configuration of the entrance region: The flue gas enters the first pass with an incidence angle of approximately 135° which changes to 180° when entering the second pass (see Figure 1). There is a calming chamber before the entrance of the passes. There was no reference to this kind of configuration in the literature, and the closest equivalents, termed an *open-end 90°-edge* (Mills, 1995) and a *square entrance* (Aung *et al*., 1987), were used. The effect of the entrance region is difficult to model. A modelling error may become significant on the boiler under study because the tubes are rather short ($z = 19.1$).

Inaccuracy of the correlation equations: There is a gap of $2300 < \mathrm{Re} < 4000$ in the validity range of the laminar and turbulent flow correlations Equations (3) and (10). Near the validity

Copyright © 2006 John Wiley & Sons, Ltd.

range limit the accuracy of Equation (3) may be poor. In the transition region the Reynolds number was $2300 < Re < 10\,000$ under the measured conditions. A weighed average value of Nu_l and Nu_t is used within that range (VDI-Heat Atlas, 1993; Aicher and Martin) but a maximum of between Nu_l and Nu_t is used for this model. This method gives higher Nusselt numbers than the weighed average value method does, and therefore the inaccuracy of the correlation equations is not assumed to be the reason for the significant underestimation in calculation.

Fluctuation in the flue gas flow: When burning fuel pellets or other biomass the oscillating nature of the flame becomes apparent, and therefore the flue gas flow may also fluctuate. In non-stationary flow the convective heat transfer is more efficient than in stationary flow. In specific pulsating combustors, a 100% increase is reported in the convective heat transfer coefficient (Hanby, 1969). Furthermore, the combustion is cyclic because the heating effect of the boiler is controlled by changing duration of the fuel feeds and the rest periods. However, one test run was performed at a peak load. Then, fuel was fed continuously which is why the cyclic combustion characteristic was not present and cannot be the reason for the difference.

The full-scale equipment differs from the experimental apparatus used in the laboratory setting. When compared to those in the literature, the clearest difference is in the configuration of the entrance region. Therefore, the effect of such configuration is estimated to be the most probable cause for the difference between the measured and calculated values. Secondly, the accuracy was good in the wood chips boiler in which the effect of entrance region is insignificant ($z = 60.1$). An adequate accuracy has also been reported on air–flue gas heat exchangers with long tubes ($z = 81$) (Yrjölä and Paavilainen, 2004). Further still, Neshumayev *et al.* have found a similar feature during the measurement of one $200\,kW_{th}$ oil-fired boiler and a $20\,kW_{th}$ wood pellet boiler. The experimental convective heat transfer was higher than that obtained using the calculation data. Neshumayev *et al.* explain the difference by the presence of strong entrance effect (Neshumayev *et al.*, 2003). Thus, correcting the calculation addresses the constants C and m in Equation (10). Unfortunately, the experimental program was not extensive enough to come to a definite conclusion.

The model offers optimization possibilities for the designer. There are, however, practical limitations on the construction:

- The demand for thermal efficiency and the risk of local condensation of water vapour set an upper and lower limit on the flue gas temperature, respectively.
- The convection section must have a self-cleanable feature which sets a minimum flue gas velocity and an upper limit on the diameter. Sweeping is difficult on small diameter tubes which sets a lower limit on the diameter.
- Manufacturing technology, transportation, and installation delimit the number of alternative shapes and sizes.

After selecting valid alternatives, it is possible to calculate the thermal performance and pressure drop. In addition to this, decision-making requires taking manufacturing and material costs into account.

The model is also suitable for water boilers using different fuels with no limitation on the size of the boiler. It also offers a sizing strategy, but additional verification measurements are needed if the geometry of the convection section is different. Furthermore, simulation promotes cognitive approach to the thermal behaviour of the convection section.

Copyright © 2006 John Wiley & Sons, Ltd.

6. CONCLUSIONS

A model for dimensioning and optimizing the convection section of a biomass boiler is described. The model takes into account the radiation and convection of the laminar and turbulent flow, tube roughness, fouling resistance, effect of the entrance region, and the effect of free convection. Verification measurements were performed on a 50 kW_{th} pellet boiler and a 4000 kW_{th} wood chips boiler. The calculated heat transfer rate of the pellet boiler was 37–44 per cent smaller than the measured rate. This is more than the inaccuracies in heat transfer correlation reported in the literature had suggested. The accuracy established on the wood chips boiler was adequate, the mean difference being 3.7 per cent. It is noteworthy that the wood chips boiler has long tubes whereas the wood pellet boiler has short tubes. For that reason, the effect of the entrance region seemed to constitute the most uncertain element in the model and consequently necessary modifications were made to correlation in the entrance region. After modifications, the model predicts the varying operating conditions of the pellet boiler with a good degree of accuracy.

The possibilities provided by the model are described by calculations carried out for the pellet boiler. The main conclusions of these analyses are:

1. The portion of radiation is 3–13% of the whole heat transfer rate under typical operating conditions of a biomass boiler. As for the pellet boiler, the effect of the entrance region is 39–52% whereas the effect of natural convection is small, mostly reducing heat transfer but at a minimum load increasing it reached up to +14%.
2. Dividing the convection section into more passes causes the rate of heat transfer to increase, even if the heat transfer surface area is constant. This is because of the recurring effect of the entrance region.
3. The heat transfer area is exploited more efficiently by using smaller tube diameters, even if the bulk velocity of the flow remains constant.
4. An increase in the heat transfer area obtained by more tubes with constant diameter only increases the heat transfer rate slightly. By reducing the diameter while increasing the number of tubes, it is possible to keep the cross-flow area constant and achieve an improved heat transfer rate.

NOMENCLATURE

A	= area (m^2)
c_p	= specific heat capacity ($kJ\,kg^{-1}\,K^{-1}$)
d	= diameter (m)
f	= friction factor
Gz	= Graetz number = $\mathrm{RePr}dl^{-1}$
h	= heat transfer coefficient ($W\,m^{-2}\,K^{-1}$)
k	= heat conductivity ($W\,K^{-1}\,m^{-1}$)
k_s	= roughness (mm)
l	= length (m)
m	= coefficient
$\dot{m}$	= mass flow rate ($kg\,s^{-1}$)

Copyright © 2006 John Wiley & Sons, Ltd.

n	= number of pipes
Nu	= Nusselt number
Pr	= Prantdl number = $\eta c_p/\lambda$
R	= radius (m), heat transfer resistance of fouling ($\mathrm{K\,m\,W^{-1}}$)
Ra	= Rayleigh number = GrPr
Re	= Reynolds number = vd/v
T	= temperature (K)
α	= absorptance
v	= velocity ($\mathrm{m\,s^{-1}}$)
z	= dimensionless length $= l/d_i$
ε	= emissivity
σ	= Stefan–Boltzmann coefficient = 5.67×10^{-8} ($\mathrm{W\,m^{-2}\,K^{-4}}$)
η	= dynamic viscosity (Pa s)
v	= kinematic viscosity ($\mathrm{m^2\,s^{-1}}$)

Subscripts

c	= convection
cs	= cross-section
d	= dust, dynamic
f	= fuel
fg	= flue gas
g	= gas, gas side
g,s	= gas at wall inner surface temperature
i	= inner
l	= laminar
m,a	= mixed, aided
m,o	= mixed, opposed
n	= natural convection, number
o	= outer
p	= pipe
r	= radiation
s	= surface
si	= inner surface
so	= outer surface
t	= turbulent, fully developed
w	= water
wi	= water in
wo	= water out

ACKNOWLEDGEMENTS

This study was funded by grants from the National Technology Agency of Finland (TEKES, Contract 40241/01), Nakkila Works, Thermia Ltd. and Timarcon Ltd.

Copyright © 2006 John Wiley & Sons, Ltd.

REFERENCES

Aicher T, Martin H. 1997. New correlations for mixed turbulent natural and forced convection heat transfer in vertical tubes. *International Journal of Heat and Mass Transfer* **40**(15):3617–3626.

Aung W, Kakac S, Shah RK (eds). 1987. *Handbook of Single-Phase Convective Heat Transfer*. Wiley: New York.

Ceylan K, Kelbaliyev G. 2003. The roughness effects on friction and heat transfer in the fully developed turbulent flow in pipes. *Applied Thermal Engineering* **23**:557–570.

DeWitt DP. 1996. *Fundamentals of Heat and Mass Transfer* (4th edn). Wiley: New York.

Grass G. 1956. Wärmeübertragung an turbulent strömende Gase im Rohreinlauf. *Allgemeine Wärmetechnik* **7**(3):58–64.

Hanby VI. 1969. Convective heat transfer in a gas fired pulsating combustor. *Journal of Engineering for Power* (ASME) **91**:48–52.

Hausen H. 1943. Darstellung des Wärmeüberganges in Rohren durch verallgemeinerte Potenzbeziehungen. *Zeitschrift des Vereines Deutscher Ingenieure, Beiheft Verfahrenstech* **4**:91–98.

Isachenko VP, Osipova VA, Sukomel AS. 1977. *Heat Transfer*. Mir Publishers: Moscow.

Mills AF. 1995. *Heat and Mass Transfer*. Richard D Irwin Inc.: U.S.A.

Neshumayev D, Ots A, Laid J, Tiikma T. 2003. Heat transfer augmentation and pressure drop of turbulator inserts in gas-heated channels. In *Advances in Heat Transfer Engineering*, *4th Baltic Heat Transfer Conference*, Kaunas, Lithuania, Sundén B, Vilemas J (eds). Begell House/Lithuanian Energy Institute: Kaunas, Lithuania; 565–572.

Wagner W. 1988. Wärmeübertragung, 2. überarbeitete Aufl., Vogel, Würtzburg.

VDI-Heat Atlas, 1. 1993. Association of German Engineers VDI—Society for Chemical and Process Engineering. VDI-Verlag GmbH: Düsseldorf (English Edition).

Yrjölä J, Paavilainen J. 2004. Modelling and analyses of heat exchangers in a biomass boiler plant. *International Journal of Energy Research* **28**:473–494.

Zigrang DJ, Sylvester ND. 1982. Explicit approximations to the solution of Colebrook's friction factor equation. *AIChE Journal* **28**:514–515.

Copyright © 2006 John Wiley & Sons, Ltd.