APPENDIX C

Paper III

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Modelling and analyses of heat exchangers in a biomass boiler plant

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SUMMARY

A boiler plant is presented, in which the fuel is dried before combustion in a silo with air. The drying air is heated in a recuperative heat exchanger by the heat of flue gases. Hot air is then blown through the bed of fuel in the drying silo, while the fuel dries and the air cools down and becomes humidified. Heat of the moist exhaust air of the silo is recovered for the drying air and combustion air by a recuperative heat exchanger. Modelling of the thermal behaviour of the plant helps in understanding complex interdependencies of the two heat exchangers, the boiler and the dryer. The models of the heat exchangers and applications in analysing the boiler system are described in this paper. Calculating the combinations of extreme operational conditions gives the input data needed in comparing different types of heat exchangers, dimensioning the heat transfer area, choosing the control strategy and selecting the operating parameters and set-values of the control system. Results of verification measurements and practical operation at a $40\,kW_{th}$ pilot plant and a $500\,kW_{th}$ demonstration plant are also discussed. Using engineering correlation formulas for heat and mass transfer, an adequate accuracy between the model and the measurements was achieved.

Fouling was detected to be a major problem with the flue gas heat exchanger. However, in absence of condensation, the increase of a fouling layer with respect to time was observed to be low. Fouling was also a problem with the drying exhaust gas heat exchanger, but after the installation of a simple dust collector, a reasonable cleaning period was achieved. A mixed-flow configuration was found to be the most appropriate for the flue gas heat exchanger. In order to avoid condensation of the flue gas the drying exhaust gas heat exchanger is indispensable in Finnish climate in the considered system. In addition to this, it decreases the need of fuel. A parallel-flow type was found the most appropriate as the drying exhaust gas heat exchanger. Copyright © 2004 John Wiley & Sons, Ltd.

KEY WORDS: bio-fuel; chip; control; deposit; fouling; heat recovery; heat transfer; wood chip

1. INTRODUCTION

Efforts to restrict the emissions of greenhouse gases are going to increase the use of bio fuels. This makes these low-quality fuels an interesting potential resource also in the small-scale heat production. Moisture of fuel is one of the most significant difficulties in the combustion of forest residues, wood chips, bark or saw dust in grate boiler plants. Moisture content may be lowered

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before delivering fuel to the boiler by storing it outside or with special drying systems. Nakkila Works Ltd. and Satakunta Polytechnic have co-operated in developing a boiler plant, in which wooden fuel is dried in a silo by hot air before combustion, see Figure 1. The drying air is heated in a recuperative heat exchanger by the heat of flue gases, see Figure 2. The hot air is then blown through the bed of fuel particles in the drying silo, where the particles dry and the air cools down and becomes humidified. Heat of the moist exhaust air from the silo is recovered to the drying air and combustion air by a recuperative heat exchanger, see Figure 7. The boiler is an ordinary warm water grate-type boiler for dry bio-fuels. This kind of system enables the use of both wet and dry bio-fuel in producing heat for single buildings, industrial plants or district heating systems. It reduces the need of storing the fuel and makes it possible to use cheap fuels with low heating value. The additional heat exchangers and the dryer increase the requirements for the automation system, but the control of combustion process remains similar to ordinary dry bio-fuel boilers.

The operating conditions of biomass boiler plants vary greatly in the Scandinavian area. The variation scale of moisture content is about 20–60% (wb) when using forest residues and by-products of the timber and carpentry industry as a fuel. A typical guarantee value for the load variation of the plant is 20–100% in unmanned operation. The outdoor air temperature varies from about -30–20°C. These three dimensioning parameters influence both the rate and the state of primary and secondary mass flows of heat exchangers. Every main component of the plant has a direct or indirect influence to the others. To simulate the thermal behaviour of the whole system, the models for heat exchangers, boiler and dryer were carried out. By combining the models, it is possible to predict the efficiency of the plant and clarify the performance and potential operational risks when changing the operation parameters or component sizes.

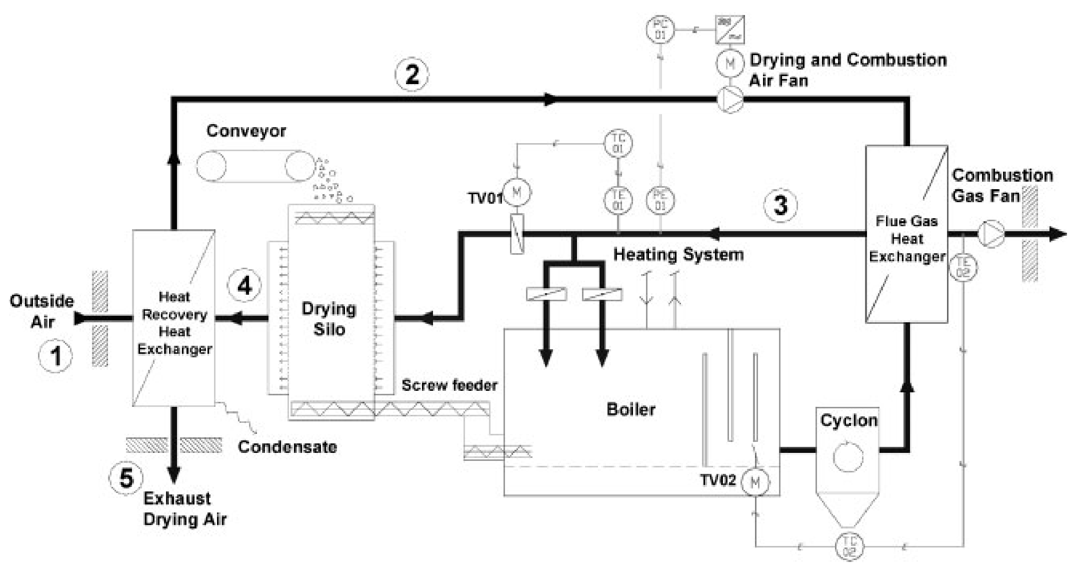

Figure 1. A schematic diagram of the boiler plant and the control strategy for heat exchangers.

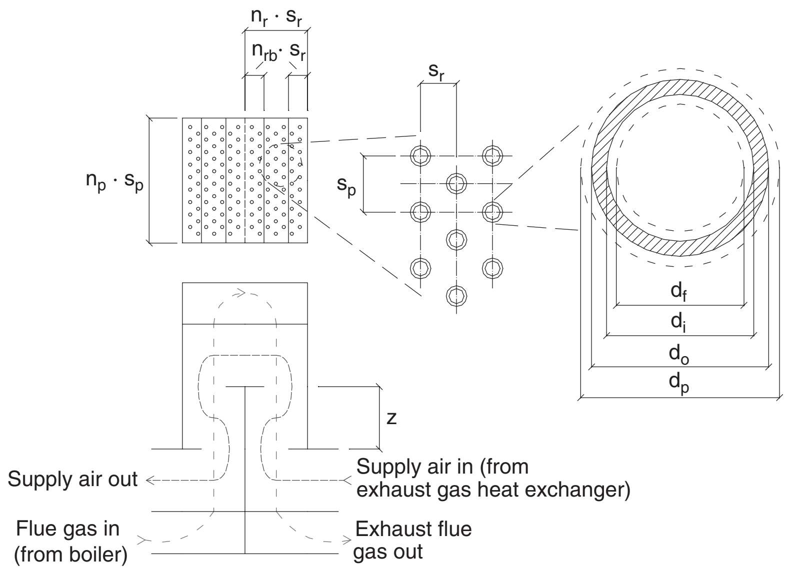

Figure 2. The principle construction of the FGHEXs in the pilot and demonstration plant.

Two boiler plants were built during the study: one 40 kW_{th} pilot plant purely for experimental purposes and one 500 kW_{th} demonstration plant both for testing and for producing heat for the users. Three different heat exchanger types were tested.

The models of heat exchangers and their applications in analysing the boiler system are described in this paper. Results of verifying measurements, emissions of drying and practical operation of the system are also discussed.

2. MODELLING AND SYSTEM ANALYSES

2.1. Flue gas heat exchanger (FGHEX)

The primary flow of the flue gas heat exchanger (FGHEX) is hot flue gas and the secondary flow is outside air, which is preheated by the exhaust gas heat exchanger, see Figure 1. The heat exchanger is of the pipe-bundle type, where the flue gas flows in pipes and outside air in the shell side. The cross-flow elements are arranged in a counter-flow configuration, see Figure 2.

The heat exchanger is divided into computational cells where the surface temperatures of tubes and outlet temperatures of primary and secondary mass flows are numerically solved by the energy balance Equation (1), see Figure 3.

$$\pi d_f L \alpha_i (t_{fg} - t_f) = \frac{\pi L}{\dfrac{1}{2\lambda_f}\ln\dfrac{d_i}{d_f} + \dfrac{1}{2\lambda_p}\ln\dfrac{d_o}{d_i}}(t_f - t_o) = \pi d_o L \alpha_o (t_o - t_a) = \dot{m}_g \Delta h_{fg} = \dot{m}_a \Delta h_a \quad (1)$$

where the heat transfer coefficients are calculated by

$$\alpha_i = \mathrm{Nu}_{i,lam/turb}\lambda_{fg}/d_f \quad \alpha_o = \mathrm{Nu}_o \lambda_a/(d_o/2) \quad (2)$$

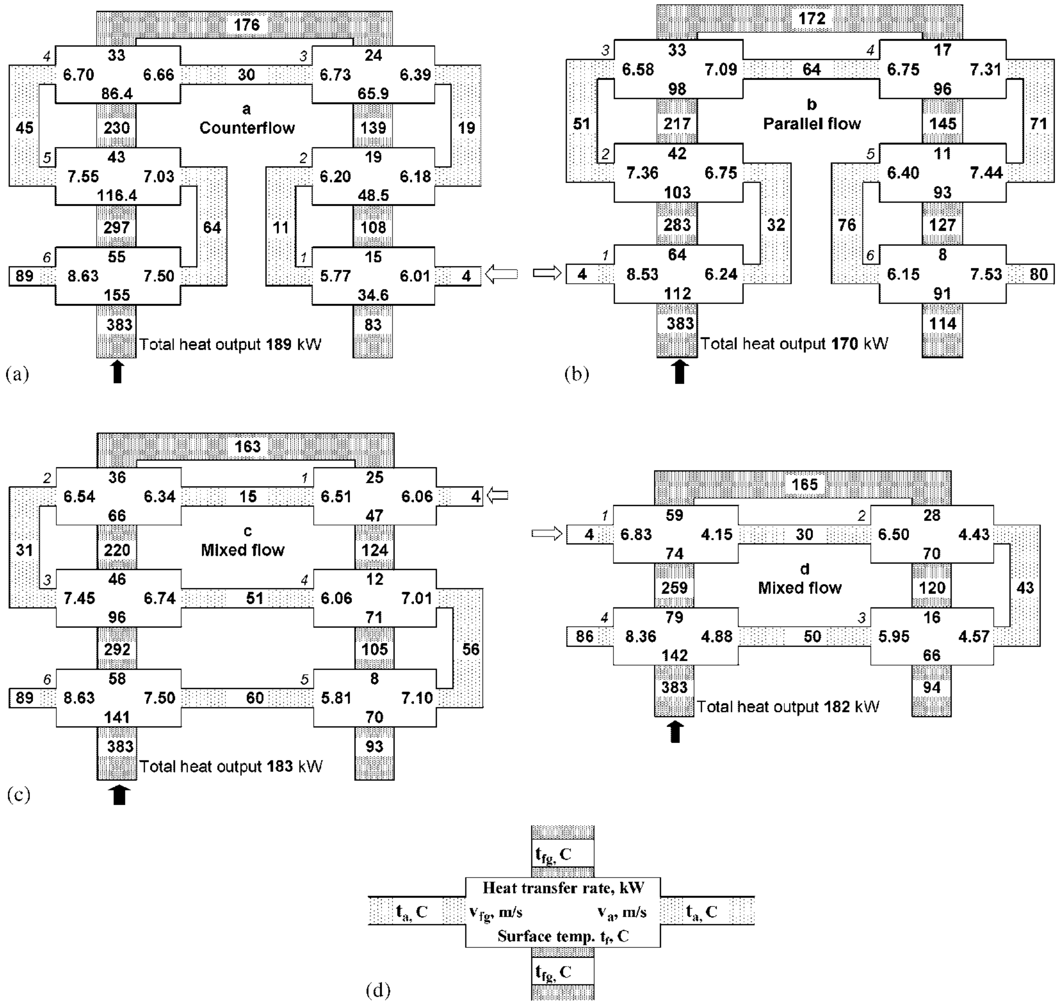

Figure 3. Comparison of four types of heat exchangers: (a) counter-flow, (b) parallel-flow, (c) mixed-flow with two baffles and (d) mixed-flow with one baffle.

where the Nusselt numbers for the flue gas in the tube side are (Wagner, 1988)

$$\mathrm{Nu}_{\mathrm{i,lam}} = 3.66 + 0.0677(\mathrm{Re}\,\mathrm{Pr} d_{\mathrm{f}}/z)^{1.33}/[1 + 0.1\mathrm{Pr}(\mathrm{Re} d_{\mathrm{f}}/z)^{0.83}] \tag{3}$$

$$\mathrm{Nu}_{\mathrm{i,turb}} = 0.0235\mathrm{Re}^{0.8}\,\mathrm{Pr}/[1 + 1.947(\mathrm{Pr}^{2/3} - 1)/\mathrm{Re}^{0.1}] \tag{4}$$

The Nusselt numbers in the shell side are obtained by means of Nu numbers of flows perpendicular (Nu_1) and parallel (Nu_2) to tubes (Wagner, 1988; VDI-GVC, 1991)

$$\mathrm{Nu}_{\mathrm{o}} = (n_{\mathrm{rb}}\mathrm{Nu}_1 f_1 f_2 + (n_{\mathrm{r}} - n_{\mathrm{rb}})\mathrm{Nu}_2) f_3/n_r \tag{5}$$

$$\mathrm{Nu}_1 = 0.3 + \sqrt{\mathrm{Nu}_{1,\mathrm{lam}}^2 + \mathrm{Nu}_{1,\mathrm{turb}}^2} \tag{6}$$

$$\mathrm{Nu}_{1,\mathrm{lam}} = 0.664\sqrt{\mathrm{Re}}\sqrt[3]{\mathrm{Pr}} \tag{7}$$

$$\mathrm{Nu}_{1,\mathrm{turb}} = 0.037\mathrm{Re}^{0.8}\,\mathrm{Pr}/[1 + 2.443\mathrm{Re}^{-0.1}(\mathrm{Pr}^{2/3} - 1)] \tag{8}$$

$$\mathrm{Nu}_2 = 0.037\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.48} \tag{9}$$

The enthalpy and the specific heat capacity of dry gas are

$$h = ct + (l + c_\mathrm{v}t)Y/(1 - Y) \tag{10}$$

$$c = \sum Y_\mathrm{i}c_\mathrm{i}(t) \tag{11}$$

The calculation procedure of α_i is described in more detail in Wagner, 1988; and α_i in (VDI-GVC (1991). The heat and mass transfer properties $\lambda(t)$, $v(t)$, $\rho(t)$ and $\mathrm{Pr}(t)$ are calculated according to molar composition of gases (on pages Da1...Dc2 in VDI-GVC (1991)).

Some simplifying assumptions are made. Firstly, the heat resistance of the fouling inside the tube is included, but outside the tubes excluded. No sub-model for predicting deposit layer is implemented, but the assumed thickness and thermal conductivity of the fouling layer are input data as shown in Equation (1). Secondly, the isolated heat exchanger is assumed to be fully tight and adiabatic, thus the leakage flow and heat transfer from/to the environment are excluded. Also the leakage between primary and secondary flows is excluded, but in the shell the leakage stream through the gap around the tubes and by-pass stream through the gap between the baffle and the shell are taken into account. These are taken into account with the correction factor f_2 in calculating the Nusselt number of the shell side of the heat exchanger (VDI-GVC, 1991). Thirdly, the condensation of water vapour of flue gas is excluded in practical calculations; see the argumentation in Sections 4 and 5. Also the effect of radiation is excluded, because the diameter of tubes and the partial pressure of carbon dioxide and water vapour are small and the temperature of the flue gas is relatively low. Finally, in calculating the convective heat transfer coefficient the effects of the entrance region of turbulent flow and mixed convection are omitted, because the tubes are relatively long compared to their diameter and the natural convection is assumed to be low.

Analysis 1: *Comparison of the heat exchanger types*: Changing the arrangement of the flows influences the characteristics of the heat exchanger. The comparison of four different flow arrangements is shown in Figure 3. Input data of calculations is shown in Table I. The boiler plant is operating at peak load 500 kW. The molar composition of inlet flue gas is $CO_2(H_2O)_{1.81}(N_2)_{7.81}(O_2)_{1.05}Ar_{0.08}$. The inlet outdoor air ($t = -20°C$, RH = 80%) is preheated in the exhaust drying gas heat exchanger to 4°C.

The heat exchangers have the same heat transfer area and tube geometry. The only exception is heat exchanger type **d**, which has only one baffle resulting in a larger cross-sectional area for the secondary flow and thus a smaller velocity of air. The symbols describing the geometry of the FGHEX are shown in Figure 2.

The counter-flow type of heat exchanger gives the highest outlet temperature of air. That is beneficial, because after the heat exchanger, the air is led to combustion and drying. If combustion air temperature increases, the propagation velocity of the ignition front in the fuel bed and the heat effect of the boiler increase. If the drying air temperature increases, the needed

Table I. Input data of the FGHEX calculations.

Data of flows		
Flue gas mass flow rate ($kg\,s^{-1}$)	0.576	
Air mass flow rate ($kg\,s^{-1}$)	2.227	
Geometry and material properties	1. Section	2. Section
Tube outer diameter, d_o (mm)	42.4	42.4
Tube wall thickness, $(d_o-d_i)/2$ (mm)	2.6	1.5
Fouling layer thickness, $(d_i-d_f)/2$ (mm)	0.5	0.5
Distance between. adjacent pipes in the row, s_p (mm)	60	60
Distance between adjacent rows, s_r (mm)	80	80
Number of pipe rows, n_r	10	8
Number of pipe rows in a baffled zone, n_{rh}	5	4
Number of pipes in one row, n_p	11	11
Distance between baffles, z^* (m)	1	1
Diameter of perforation in the baffle, d_p (mm)	44	44
Gap between the baffle and shell (mm)	2	2
$^*z$ in heat exchanger type **d** (m)	1.5	1.5
Heat conductivity of the tube material ($W\,K^{-1}\,m^{-1}$)	52	25
Heat conductivity of the fouling ($W\,K^{-1}\,m^{-1}$)	0.12	0.12

mass flow rate decreases (Yrjölä, 2001). On the other hand, it is not desirable that the surface temperature of the inside wall of tubes falls below the dew point of water vapour in the flue gas, see Sections 4 and 5. The dew point in the considered flue gas is 56°C, which is broken in the type **a** in cells 1 (34.6°C) and 2 (48.5°C) and in the type **c** in cell 1 (47°C).

The heat output of the FGHEX varies depending on the heat load of the boiler plant, moisture content of the fuel, and the outdoor temperature, in addition to the operating parameters of the plant. The results of heat and mass balance calculations for the system are shown in Figure 4. The subscripts of symbols refer to Figure 1.

The load of the boiler plant has the strongest influence on the operation conditions of the FGHEX. The partial load at frost is not a relevant dimensioning criterion and therefore it is excluded in Figure 4. The heat transfer rate varies from 5 to 100%, the mass flow rate of air from 7 to 100% and that of flue gas from 13 to 100%. The air mass flow rate is clearly higher than the flue gas mass flow rate, and their enthalpy flow rate ratio varies about 2–3.5. Point number 4 (full load, dry fuel, warm weather) illustrates the effect of operating parameters; both mass flow rates are higher than at point number 3 (full load, dry fuel, cold weather). This is because the flue gas is cooled to the set value at TE02 by first closing the damper TV02 in the convection section of the boiler, see Figure 1. When TV02 is fully closed and the flue gas temperature (t_6) after the boiler (point number 6 in Figure 1) is at minimum, the damper TV01 begins to open and the frequency converter controlled supply air fan starts to increase the air flow rate to the FGHEX. The fuel is dried and warmed up effectively. This is not the most efficient way to operate the plant, but it is chosen because of safety reasons; a high air mass flow rate prevents its temperature to rise above the risk level of causing a danger of fire.

The information in Figure 4 allows the conclusion that it is not possible to achieve the temperature 90°C for the drying air with a parallel-flow heat exchanger, type **b**. At partial load even the weighted average temperature of air and flue gas is below 80°C. Further, in Figure 3 can be seen that with full load and wet fuel already at $t_1 = -20$°C the drying air temperature

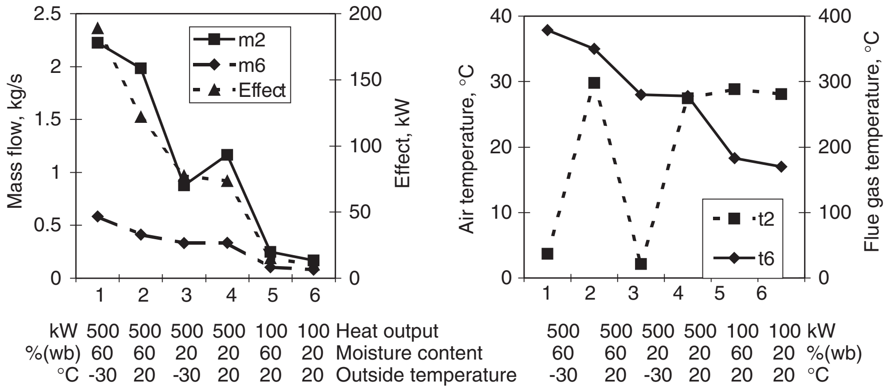

Figure 4. The mass flow rates and temperatures of preheated air ($\dot{m}_2$, t_2) and flue gas ($\dot{m}_6$, t_6) and the heat transfer rate at six different combinations of extreme boiler load, moisture content of fuel and the outside temperature.

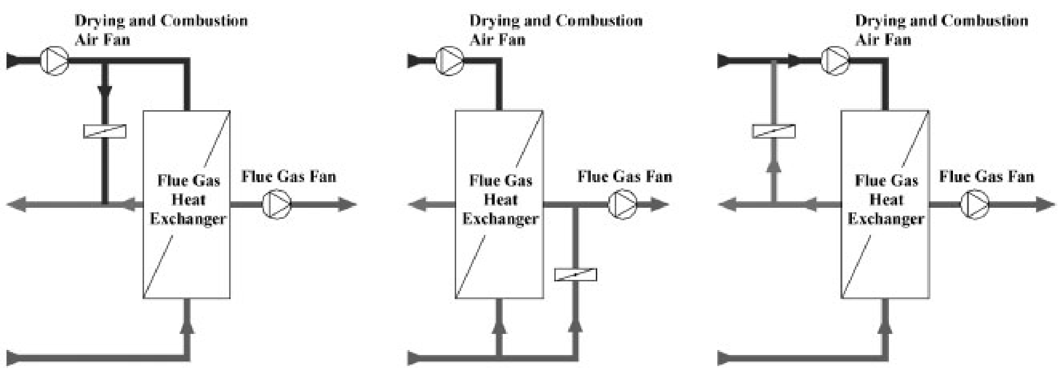

Figure 5. Control alternatives for the FGHEX. By-pass of air on the left, by-pass of flue gas in the middle and recirculation of the air on the right.

reaches only 80°C. Thus, the rate of drying is not high enough and thereby it is not possible to get the needed heat output from the boiler. In addition, the total heat output of type **d** is only 4% less than that of the most effective type **a**. Therefore, type **d** should be seriously considered when choosing a heat exchanger for the next boiler plants.

Analysis 2: *Comparison of by-pass alternatives of the FGHEX*: At partial load of the boiler (points 5 and 6 in Figure 4) the heat transfer rate of the FGHEX is low with respect to heat transfer area (15.1 and 10.5 kW, respectively). Therefore, the flue gas cools down below the dew point of the flue gas water vapour. In order to avoid condensation a control system is necessary for the FGHEX. The heat transfer area is technically difficult to regulate; therefore three alternative control strategies of the heat exchanger are discussed, see Figure 5.

The by-pass of flue gas is overlooked, because it reduces the flue gas mass flow rate through the heat exchanger resulting the flue gas water vapour to condense. The other two methods are compared in Figure 6.

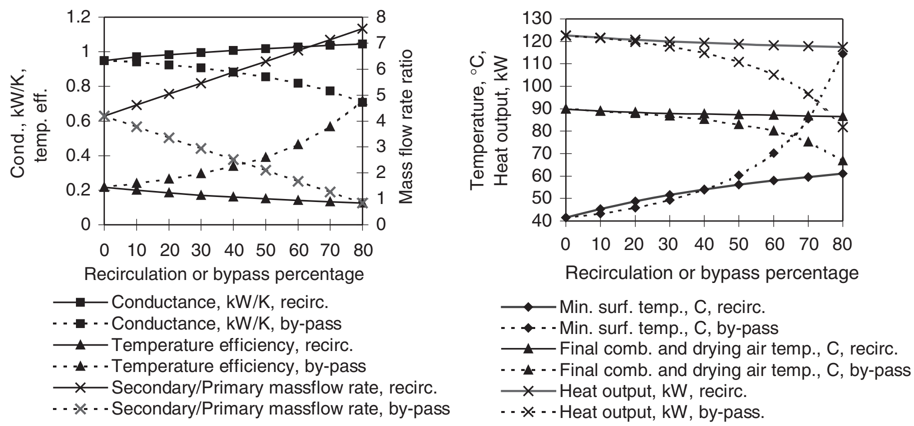

Figure 6. Comparison of two control strategies of the FGHEX: the by-pass and recirculation of the air. The heat exchangers are identical; the inlet temperatures of flue gas and air are 340 and 20°C, respectively.

The flue gas mass flow rate is constant and the recirculation or by-pass percentage of the air mass flow rate is changed from 0 to 80%. On the left it is seen that the mass flow rate ratio ($\dot{m}_a/\dot{m}_{fg}$) increases by 80% (from 4.20 to 7.55) with the recirculation control but decreases by 80% (from 4.20 to 0.84) with the by-pass control. As a result of this the temperature efficiency increases with by-pass control (from 0.22 to 0.72) and decreases with recirculation control (from 0.22 to 0.13). On the other hand, the conductance of the heat exchanger increases with the recirculation (from 0.95 to 1.05 $\mathrm{kW\,K^{-1}}$), because the effective velocity of air increases and the convective heat transfer from the outer surface of the pipes improves. With the by-pass control it decreases (from 0.95 to 0.71 $\mathrm{kW\,K^{-1}}$). On the right it is seen that the heat transfer rate decreases 5 kW with the recirculation and 41 kW with the by-pass. Consequently, the final temperature of combustion and drying air decreases 3°C with the recirculation and 23°C with the by-pass. The dew point is 59°C. The heat exchanger is of type **a** in Figure 3 and the minimum surface temperature of the inner tube wall is situated in its computational cell 1. It exceeds the dew point with the by-pass percentage of 51% and with the recirculation percentage of 80%. The heat transfer rates are 110 and 117 kW and the final temperature of the combustion and drying air 83 and 86.5°C, respectively. Considering the dew point as a comparison point, the recirculation system gives higher temperature and heat output, but needs more electrical power for the fan, than the by-pass system. The air volume rate of the recirculation system is 260% higher and as a result the need of fan power approximately 1200% higher than of the by-pass system. In most cases this disadvantage may be decisive and with similar constructions the by-pass control strategy is more economical. By choosing a more spacious geometry for the secondary side, the influence of the pressure drop decreases and the conclusion may change.

2.2. Drying exhaust gas heat exchanger (DEGHEX)

The primary flow of the drying exhaust gas heat exchanger (DEGHEX) is moist exhaust gas from the drying silo. Its temperature range is about 20–40°C and it is nearly saturated. The secondary flow is outside air, see Figure 1. A parallel-flow type heat exchanger was chosen due to its smallest risk for freezing, see Figure 7.

The condensing mass flux of water vapour in drying exhaust gas is calculated with Equation (12), (Lampinen, 1997; Seppänen, 1988)

$$\dot{m}''_{\mathrm{c}} = \alpha_{\mathrm{i}} \frac{\mathrm{Le}^{1-n}}{C} M_{\mathrm{v}} \ln \frac{p - p_{\mathrm{v,s}}}{p - p_{\mathrm{v}}} \tag{12}$$

where

$$\mathrm{Le} = D\rho c/\lambda \tag{13}$$

$$p_{\mathrm{v,s}} = \exp(77.345 + 0.0057 T_{\mathrm{fg}} - 7235/T_{\mathrm{fg}})/T_{\mathrm{fg}}^{8.2} \tag{14}$$

$$p_{\mathrm{v}} = Y M_{\mathrm{g}}(p - p_{\mathrm{v,s}})/[M_{\mathrm{v}}(1 - Y)] \tag{15}$$

The heat exchanger is divided into computational cells where the outlet temperature and surface temperature of primary and secondary air are solved with mass and energy balance Equations (16) and (17)

$$\dot{m}_{\mathrm{eg}} \Delta Y = \dot{m}''_{\mathrm{c}} A \tag{16}$$

$$\alpha_{\mathrm{i}} A(t_{\mathrm{eg}} - t_{\mathrm{i}}) + \dot{m}_{\mathrm{eg}} l = \lambda A(t_{\mathrm{i}} - t_{\mathrm{o}})/s_{\mathrm{pl}} = \alpha_{\mathrm{o}} A(t_{\mathrm{o}} - t_{\mathrm{a}}) = \dot{m}_{\mathrm{eg}} \Delta h_{\mathrm{eg}} = \dot{m}_{\mathrm{a}} \Delta h_{\mathrm{a}} \tag{17}$$

where the Nusselt number for calculating α_{i} and α_{o} is (Wagner, 1988)

$$\mathrm{Nu} = 0.0235 \mathrm{Re}^{0.8}\, \mathrm{Pr}^{0.48} \tag{18}$$

and the heat of evaporation (Lampinen, 1997)

$$l = 2501 - (c_{\mathrm{w}} - c_{\mathrm{v}})t \tag{19}$$

Some simplifying assumptions are made. Firstly, the heat resistance of the fouling is excluded. Secondly, the isolated heat exchanger is assumed to be fully tight and adiabatic, and the leakage flow and heat transfer from/to the ambient are excluded. Also the leakage between primary and secondary flows is excluded. Thirdly, the heat resistance of the condensate layer is excluded and the outlet condensate is assumed to be equal in temperature with the outlet gas.

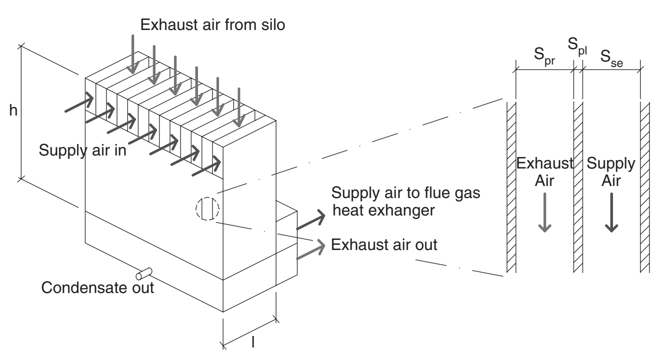

Figure 7. The principle construction of the DEGHEX in the demonstration plant.

Analysis 3: *The effect of drying exhaust gas heat exchanger in the system*: The DEGHEX reduces the FGHEXs heat output demand, which is needed to raise the drying air temperature to a sufficient level. Furthermore, it improves the efficiency of the boiler plant by recovering some heat of the exhaust gas to drying and combustion air. In that sense, this boiler plant operates to some degree similarly to the boiler plants with a flue gas condensing system; the exhaust gas heat exchanger condenses a part of the water evaporated from the fuel particles. The input data of a case, where the moisture content of fuel is 45% (wb) as received is shown in Table II. The calculation results are shown in Figure 8.

In Figure 8 the mass flow rates (on the left) and inlet temperature (in the middle) of the exhaust drying gas are taken from the heat and mass balance calculations of the whole system. The mass flow rates decrease slightly, when the outdoor temperature is below 0°C, because with the decreasing outdoor temperature, the damper TV01 starts to shut slowly. The temperatures of both the flue gas (t_7) and the combustion/drying air (t_3) are kept by the control system constant at 110 and 90°C, respectively. While the outdoor temperature decreases, the heat output demand of the FGHEX increases. The mass flow rate of the combustion air is

Table II. Input data of calculations. The symbols describing the geometry of the heat exchanger are shown in Figure 7.

	Primary		Secondary			
Distance between plates, s_{pr}, s_{sc} (mm)	15		10			
Number of passes, n_{pr}, n_{sc}	15		16			
Thickness of the plates, s_{pl} (mm)		2				
Conductivity of the plate material ($W\,K^{-1}\,m^{-1}$)		52				
Depth of the plates, l (m)		1				
Height of the plates, h (m)		4				
		Outside temperature t_1,°C				
	−30	−20	−10	0	10	20
Heat output of the plant (kW)	500	500	500	500	293	125
Heat output of the flue gas heat exch. (kW)	101	99	96	90	49	7
RH%, outside air	80	80	80	80	80	70
RH%, drying exhaust gas	86	87	88	89	93	32

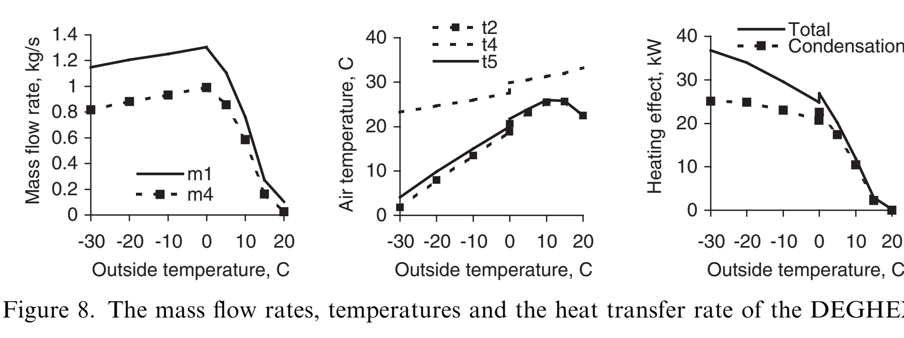

Figure 8. The mass flow rates, temperatures and the heat transfer rate of the DEGHEX.

independent of the outdoor temperature. Thus, the only adjustable variable is the mass flow rate of the drying air. The damper TV01 reduces both the primary and secondary mass flow rates of the DEGHEX. The heat load of the boiler plant reduces, when the outdoors temperature exceeds 0°C, see Table II, therefore also the mass flow rates decrease distinctively. The smaller the mass flow rate of the fuel is, the smaller are the mass flow rates of all the gases.

There is practically no more heat available by enlarging the plate area of the parallel-flow heat exchanger. In the middle of Figure 8 is shown, that the difference of the outlet temperatures varies from 0.01 ($t_1 = 20$°C) to 2.3°C ($t_1 = -30$°C). On the other hand, the minimum outlet temperature of the primary flow is 4°C ($t_1 = -30$°C) and the risk of freezing exists in changing the flow arrangement into cross-flow or counter-flow. Furthermore, it has been shown in Yrjölä (2001) that no significant increase in heat output is achieved with a cross-flow or a counter-flow type heat exchanger. That is because of a big share of condensation in the total heat output of the exhaust gas heat exchanger (68–88% in Figure 8, on the right).

When the outside temperature exceeds 0°C, there is a jump in the primary inlet temperature (t_4). In this stationary modelling, it is assumed that at frost the fuel is icy and at 0°C it immediately melts. Because the drying air temperature is kept constant and no heat is needed for melting any more, the temperature of the exhaust drying air increases by 2.3°C in this example. Naturally there appears to be a temperature step of the outlet mass flows (t_2, t_5), too.

Moving parts for freezing protection have not been applied for reliability reasons. The heat output of the exhaust gas heat exchanger increases with increasing the heat load of the plant. In that sense, the exhaust gas heat exchanger has a kind of self-regulation characteristics. This is a benefit for the system; the possible mass flow rate of the drying air heated only by the FGHEX would be too low to dry enough fuel in the drying silo. When the boiler at this kind of stage begins to receive wetter fuel, the combustion temperature and the propagation of ignition on the grate decrease. Accordingly, the decreasing heat transfer rate decreases further the flue gas temperature. Finally the consequence of this circle may be the extinction of combustion. It can be concluded on the basis of Figure 4 that, without the exhaust gas heat exchanger, the outlet temperature of the flue gas will decrease below the dew point. On the other hand, when designing the system so that the outlet temperature of the flue gas always remains above the dew point, the outlet combustion gas temperature from the boiler should exceed 500°C at the dimensioning condition. This is considered to be difficult to realise without special solutions in the boiler designing. Consequently, the practical alternative for the exhaust gas heat exchanger may be the storage of dry fuel for the duration of extreme cold weather.

The exhaust gas heat exchanger decreases the yearly fuel consumption. This has been evaluated by Equation (20).

$$e = \int_{i_y}^{0} \phi_{\text{eghe}}(i)\,\mathrm{d}i \Big/ \int_{i_y}^{0} \phi_{\text{fuel}}(i)\,\mathrm{d}i \tag{20}$$

With the initial values shown in Table II and Figure 8, the calculation gives an estimation of 4.5% for the yearly fuel consumption.

2.3. *Combined heat exchanger (CHEX)*

A schematic diagram of the system, where the previously separate heat exchangers are combined, is shown in Figure 9. The idea is to mix flue gas and drying air exhaust gas and cool this mixture below the dew point. The heat exchanger consists of three parts and the mixing of

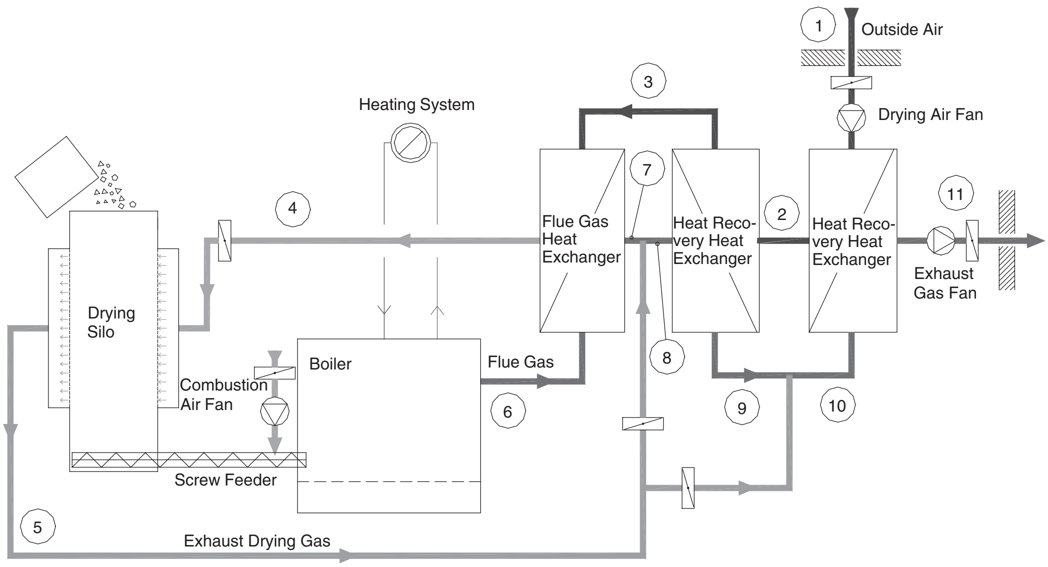

Figure 9. The scheme of the pilot boiler plant with the CHEX.

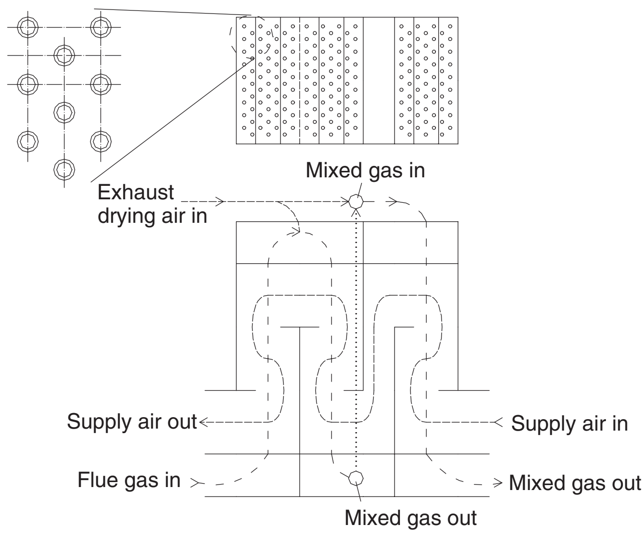

Figure 10. The construction scheme of the CHEX.

gases is carried out in two stages. The principle construction of the combined heat exchanger (CHEX) is shown in Figure 10. It is of the pipe-bundle type, where the flue gas and the mixed gas flow in pipes and the outside air in the shell side.

The modelling of the CHEX is a combination of models for the separate heat exchangers. The heat transfer coefficients are calculated analogously as in the case of the FGHEX and the condensing mass flux analogously as in the case of the DEGHEX. The state of the mixed gas is calculated by the mass and energy balance, Equations (21) and (22).

$$\dot{m}_{\mathrm{eg}} + \dot{m}_{\mathrm{fg}} = \dot{m}_{\mathrm{mg}} \tag{21}$$

$$\dot{m}_{\mathrm{eg}}h_{\mathrm{eg}} + \dot{m}_{\mathrm{fg}}h_{\mathrm{fg}} = \dot{m}_{\mathrm{mg}}h_{\mathrm{mg}} \tag{22}$$

Analysis 4: *Comparison of the CHEX to separate heat exchangers*: Some benefits are achieved by combining the heat exchangers compared to the system in Figure 1. The CHEX is easier to make, while the manufacturing process is similar to the whole of the construction and easier to service, because they are integrated. On the other hand, the leaving exhaust gas is nearly saturated, which may cause at least visual problems. The volume flow rate is higher, and the consequence is a larger diameter of the chimneystack.

A boiler plant with a CHEX operates with a higher efficiency, because the average leaving mixed gas temperature is lower than with separate heat exchangers, see Sections 4 and 5. The schemes of the processes are shown on the Mollier-graph of moist air, see Figure 11. The bordered numbers refer to Figures 1 and 9.

This kind of graph visualises the nature of differences between processes of the CHEX and separate heat exchangers (even if it is, to be precise, incorrect to draw the changes of states of the flue gas and the air in the same diagram). The efficiency of the boiler plant increases mainly, because the condensation heat is higher in the CHEX. Also the recovered sensible heat of the flue gas and drying exhaust gas is slightly higher. The outlet temperature of the mixed gas (t_{11}, on the right) is higher than the outlet temperature of the drying exhaust gas (t_5, on the left), but clearly lower than the temperature of the flue gas from separate heat exchangers (t_7, on the left), see Figure 11. The purpose in dividing the heat exchanger in three parts is to exploit the heat transfer area effectively. The temperature difference between hot and cold stream is higher compared to the construction where the flue gas and the drying exhaust gas would be mixed at one stage.

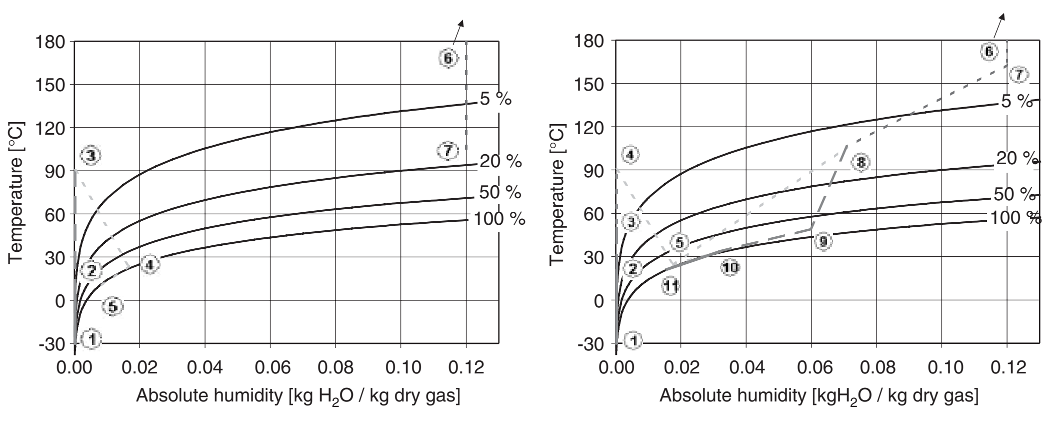

Figure 11. Principle comparison of processes with the FGHEX and the DEGHEX (on the left) and with a CHEX (on the right).

3. EXPERIMENTAL

The aim of experiments was to clarify the thermal performance of the heat exchangers, verify the calculations and to find out whether the emissions of the drying are disadvantageous for the DEGHEX or for the environment. In situ measurements of four heat exchangers, emissions from the drying silo and the contents of condensate were done. Air flow rates of the heat exchangers were measured with an Electrothermal Anemometer ALNOR GGA-65P (relative inaccuracy 4%), an Air Flow Measurement Device Halton for Round Ducts (relative inaccuracy 5%) and a Pitot pipe with an ALNOR MP6-KSR Micromanometer (relative inaccuracy of the micromanometer 1%). Temperatures were measured with K-type thermocouples and a Grant Squirrel 1200-Series datalogger (resolution 0.2°C). The relative humidity of air was measured with an ALNOR Therm 2286-2 Psychrometer (relative inaccuracy 1%). The calculated values are compared to the measured values in Table III. The t_i symbols refer to Figures 9 and 1.

The measurement point of the condensing mass flow rate ($kg\,h^{-1}$) is presented in Figure 1. The presented results are mean values of the 1–2 h data logging periods. The accordance between the measured and calculated values with the DGHEX and the FGHEX is fairly good, but for the CHEX poorer. One reason for this is probably the inaccuracy in measurements. The fuel was not appropriate for the stoker burner in the pilot-plant, generating slag on the burner during the measurements. Therefore, it was difficult to operate the plant at a stationary state. Further, the installation is compact and the straight-line distances are short on both sides of the volume flow rate measurement device. The temperature measurements show reasonable accuracy, but the condensate mass flow rate measurements show poorest accuracy. However, the modelling of the condensing mass flux is similar to the case of the DEGHEX, where the accuracy was fairly good.

In addition to water, also some part of volatile organic compounds vaporizes when drying chips by hot air. Hot air is blown through the moving chip bed via the perforated plate walls of the drying silo. The diameter of the holes is 3 mm, whereby some amount of small particles will be carried out from the silo by the drying air. Gaseous and dust emissions of the silo were measured at the demonstration plant (Koivisto, 2000), see Table IV. The t_i symbols refer to Figure 1. The gaseous components were measured with a GASMET FTIR and the dust by weighing the filter according to the SFS5624.

The concentrations of CO and R-limonene are not presented because they were <1 ppm in all cases. The starting moisture content and the sieved mass fraction with $>24/>16/>5/<5$ mm sieves of wood chips was 41–42.5% (wb) and 0.15/0.38/0.35/0.12 and of forest residue chips 68% and 0.12/0.23/0.26/0.39. The fresh forest residue was very wet and the mass fraction of smallest sieved particles was 225% higher than with the wood chips. Therefore also the mass flow rate of the evaporated water was higher. The temperature of the particles stays rather constant as long as their surface is wet (Yrjölä and Saastamoinen, 2002). The small particles dry faster than the big ones and so their temperature is higher. This may explain, why the concentrations of turpentine components β-pinene, 3-carene and terpinol and formic acid are clearly higher with forest residue than with wood chips. The difference in the wood species and the notable amount of needles of the forest residue chips may also affect the concentration. The results of the condensate analysis of the heat exchanger are shown in Table V.

The condensate is acidulous and the conductivity detects that there are dissolved inorganic salts in it. The sample was filtered for separating the solid particles, because the purpose was to

Table III. Results of verifying measurements of the FGHEX, DEGHEX and CHEX.

Fuel gas heat exchanger										
Meas. *t*6 (°C)	Meas. t2 (°C)	Meas. t7 (°C)	Calc. t7 (°C)	Relat. diff.	Meas. t3 (°C)	Calc. t3 (°C)	Relat. diff.			
330	18	54	60	−0.11	60	62	−0.03			
340	17	60	58	0.03	65	60	0.08			
350	18	70	69	0.01	70	80	−0.14			
		122	127	−0.04	68	64	0.06			
Average value of magnitudes				0.05			0.08			
Drying exhause gas heat exchanger										
Meas. *t*1 (°C)	Meas. *t*4 (°C)	Meas. *t*2 (°C)	Calc. *t*2 (°C)	Relat. diff.	Meas. *t*5 (°C)	Calc. *t*5 (°C)	Relat. diff.	Meas. m_c ($kg\,h^{-1}$)	Calc.m_c ($kg\,h^{-1}$)	Relat. diff.
−5.8	30.6	7.9	6.3	0.21	25.5	27.6	−0.08	16.7	13.8	0.18
−5.5	31.2	8.7	6.9	0.21	26.2	28.1	−0.07	13.3	14.3	−0.08
4.7	31.0	10.2	7.8	0.23	26.9	28.2	−0.05	14.1	14.6	−0.04
3.7	32.6	13.8	13.3	0.04	26.6	30.2	−0.14		13.1	
3.8	40.9	16.1	16.7	−0.04	26.9	30.9	−0.15		9.9	
4.0	37.0	15.3	15.6	−0.02	29.8	34.3	−0.15		12	
6.2	39.0	24.8	23.6	0.05	28.5	28.4	−0.00	10	14.3	−0.43
10.4	36.5	20.1	19.1	0.05	33.8	34.1	−0.01	9.8	8.5	0.13
10.6	38.4	20.7	19.9	0.04	34.7	35.5	−0.02	8.1	7.6	0.06
11.0	35.3	27.0	26.0	0.04	29.9	28.8	0.04			
Average value of magnitudes				0.09			0.07			0.15
Combined heat exchanger										
Meas. *t*1 (°C)	Meas. *t*6 (°C)	Meas. *t*4 (°C)	Calc. *t*4 (°C)	Relat. diff.	Meas. *t*11 (°C)	Calc. *t*11 (°C)	Relat. diff.	Meas. m_c ($kg\,h^{-1}$)	Calc.m_c ($kg\,h^{-1}$)	Relat. diff.
5.4	348.0	92.8	88.5	0.05	32.7	35.2	−0.08	2.2	1.1	0.48
7.2	387.6	109.3	112.9	−0.03	36.1	40.3	−0.12	1.9	0.8	0.56
6.8	332.0	92.8	96.5	−0.04	33.7	36.9	−0.09	2.5	1.8	0.31
13.3	399.0	94.6	115.3	−0.22	36.1	44.2	−0.22	2.2	0.8	0.63
8.6	348.3	75.8	91.2	−0.20	31.7	36.8	−0.16	3.1	1.7	0.44
17.0	345.5	76.7	99.3	−0.30	33.5	42.8	−0.28	0.7	0.0	1.00
Average value of magnitudes				0.14			0.16			0.57

Table IV. The average concentrations of gaseous emissions at different inlet/outlet temperatures of drying air (Koivisto, 2000).

t_3/t_4 (°C)	CO_2 (%)	CH_4 (ppm)	NO (ppm)	SO_2 (ppm)	α-pin (ppm)	β-pin (ppm)	3-car (ppm)	HCOOH (ppm)	Terpinol (ppm)	Dust (mg m^{-3})
Wood chips										
90/32°C	0.041	1	16	2	<0.5	0.5	<0.5	0.5	1	0.6
110/37°C	0.042	1	10	1	<0.5	0.8	<0.5	<0.5	1	0.6
130/36°C	0.042	<0.5	7	1	0.7	1.2	0.9	<0.5	1	
Forest residue chips										
90/30.28°C	0.041	<0.5	12	3	<0.5	3.2	2.5	1.3	6	109*, 5.7
110/36.45°C	0.041	<0.5	14	3	<0.5	4.9	5.2	2.2	11	1.3, 6.6
Extended uncert.	0.005	± 2	± 4	± 3	± 3	± 3	± 3	± 4	± 3	± 35%

The expression < is used, when the measured value is under the limit of identification in question.
*There was one extra large particle in the sample.

Table V. The measured properties of condensate of the DEGHEX, measurement devices or standards and as a reference value respective properties of lake water (Koivisto, 2000).

	pH	Conductivity (μS cm^{-1})	CODMn (mg dm^{-3})	Solids cont. (mg dm^{-3})	Phosphor cont. (μg l^{-1})	TOC (mg l^{-1})
Device/standard	Metrohm 632	E518 Metrohm Herisau	SFS3036	SFS 3037	SFS 3026	
Wood chips	5.87–6.15	33–41	42–80	23–27	326–360	36–43
Forest residue chips	5.90–6.04	55–92	55–126	14–217	396–1538	72–139
Finnish lake water	6.6	5–6	15		13	7.7

find the dissolved amount of carbon. The Total Organic Carbon (TOC) is defined by subtracting the inorganic carbon from the total carbon of the condensate.

4. PRACTICAL OPERATION RESULTS

During the winters 1997 and 2001 qualitative operation characteristics were collected at the $40\,kW_{th}$ pilot plant, and during the heating seasons 1998...1999 and 1999...2001 at the $500\,kW_{th}$ demonstration boiler plant at Pori Forest Institute. The demonstration plant produces heat for the buildings and the sawmill of the Institute. Some operational problems and requirements, which came up during the long-term testing period are discussed here.

One FGHEX was tested at the pilot plant and one at the demonstration plant. Both of them are of the counter-flow type, which was estimated as the best type at the beginning of the study. This choice caused condensation of flue gas at low outside temperatures and always during the start-up of the plant. The partial load situations also lead to condensation, because no control (by-pass or recirculation) of the flows is installed in those plants. It was observed that the fly ash particles stuck on the inner wet surface of the tubes causing fouling, which finally resulted in a plug preventing the flow in the condensing area. The condensate and particles formed a sticky solution in the cells 1 and 2, see Figure 3. After drying, the deposit was difficult to remove. The thin easily removable layer of grey pulverized fly ash was discovered in the tubes in cells 4–6.

Two DEGHEXs were tested at the demonstration plant. The first was a small-scale device mainly for verifying the modelling and observing the operational characteristics and the second was a full-scale device for minimizing the condensation problem of the FGHEX.

The dust leaving the drying silo causes fouling on the surfaces of the heat exchanger. To minimize dust emissions a dust collector was installed between the dryer and the heat exchanger, see Figure 12. However, the heat exchanger must then be cleansed approximately once a month.

Because of the pressure loss of the primary flow, the drying air must be maintained over-pressurised before the drying silo. The drying silo is open to the ambient air at the top and to the screw feeder at the bottom. The over-pressure caused leakage of the moist air to the conveyor channel and the fuel storage. This led to freezing problems in the fuel storage at frost. An extra exhaust gas fan was installed after the heat exchanger.

The CHEX represents a different approach to prevent long-term fouling problems. A hypothesis was made: If the surfaces of the primary flow are wet all the time, the high mass flow rate of the condensate will wash the fouling down. This might also clean the flue gas. The testing period was, however, too short to make conclusions on whether any self-cleaning effect is really achieved. In the centre part no condensation was detected, but due to the low temperature of the primary gas it will probably condense occasionally. This increases the risk of harmful deposit in this part, even if the latter part would remain clean enough. The exhaust gas fan will maintain the drying silo in under-pressure. This decreases the air leakage problem, which was found with the exhaust drying air heat exchanger. On the other hand, if the dust emissions from drying silo form fouling on the surfaces of the centre and latter parts, it will by time decrease the

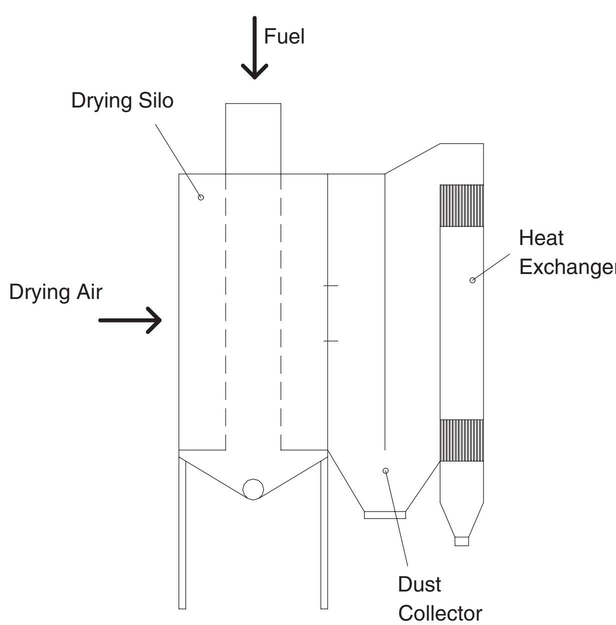

Figure 12. The dust collector between the drying silo and the exhaust gas heat exchanger.

under-pressure in the boiler, which will disturb the combustion. This makes the fouling more critical for the operation of the plant than in the case of separate heat exchangers. The dimensioning of the CHEX's heat transfer area was successful; the desired performance was achieved.

5. DISCUSSION

The counter-flow type FGHEX without by-pass or recirculation control appeared to suffer from fouling problems mainly in the parts where the tube surface was wet. When the water vapour of the flue gas is condensing onto the inner wall of tubes, it draws along the particles. In this kind of deposition mechanism, called diffusiophoresis, the mass flux of particles onto the walls is directly proportional to the water vapour condensation rate (Waldmann and Schmitt, 1966). Another appearing deposition mechanism, thermophoresis, is the result of thermal gradient. Gas molecules on the hotter side of the particle collide with higher energy than on the colder side, thus causing a net transport in the direction of colder temperature (Lehtinen *et al.*, 2002). Further, the impact of particles onto the tube wall due to the turbulent flow increases at a growing rate, the thicker and rougher the deposit layer is. The flow is clearly turbulent, as the Reynolds number of flue gas in the clean tube is in the scale 7000...10 000 (the typical velocity of the flue gas is shown in Figure 3). Finally, the chemical reactions form deposit. The flue gas contains many solvable uncombustibles, a part of them commonly named tars, which may form a sticky liquid with the condensate. This adds the accumulation of unwanted particles on the surfaces. Similar fouling appears on the superheaters of steam boilers using biomass. The wooden fuels contain Cl, especially if there is a notable amount of green needles within. The ash contains oxides of both alkali metals, like CaO and MgO and earth alkali metals like Na_2O and K_2O, which may be involved in reactions leading to ash fouling (Werther *et al.*, 2000; Jenkins *et al.*, 1998; Obernberger, 1998). They volatilise and react in a complex way in the combustion chamber, and mainly K and Na chlorides and hydroxides form deposit on the surfaces while condensing at the temperature 600–900°C (Miles *et al.*). At the temperature of 50–300°C in the FGHEX, those alkali compounds are in solid phase, but they may be involved in particulate fouling. The heat transfer surface may react with the flowing fluid. The condensation of water vapour may cause corrosion problems in the presence of acidic CO_2 (Hewitt *et al.*, 1998). The dissolving of gaseous HCl into condensate may assist the corrosion. The condensate may also assist chemical reactions, leading to sintering and thus difficulties in removing the particles.

In the exhaust drying air heat exchanger, the diffusiophoresis, thermophoresis and particle impact mechanisms are proposed to cause fouling. The drying air inlet temperature to the silo is 80–100°C. Air cools down and becomes humidified in the silo, resulting in the outlet temperature of 25–35°C and RH near 100%. Even if the volatile matter represents about 80% of dry wood mass (Werther *et al.*, 2000), a very low amount of it evaporates under the temperature of 100°C (Fagernäs and Sipilä, 1996). The results of measurements at the demonstration plant agree with this; the traced contents of the exhaust drying gas were very low, near or below the limit of identification. Thus, the partial pressure of VOC appears to be too low to react significantly with the surface. For environmental reasons, the condensate was found problematic. The COD_{Mn} and TOC values indicate that the condensate contains organic matter, which have adverse effect when falling into waters. The decomposing of organic matter consumes the oxygen from waters, which is especially harmful at wintertime. The phosphorous has overfertilizing effect on waters. Even if the condensing mass flow rate is relatively low, a treatment for the phosphorus content and COD_{Mn} should be considered. Furthermore, the condensate contains inorganic salts, which may react with phosphorus causing crystallisation of sparingly soluble compounds on the heat transfer surfaces.

The operational aspects of the whole system limit the possibilities to optimise the heat effect and dimensions of the heat exchangers. The maximum inlet temperature to the heat exchanger from the type of boiler considered is approximately 400°C. If it increases, more expensive materials will be needed in the combustion chamber and the probability of slag forming increases. The softening temperature of wood ash is 1100–1200°C (Wilen *et al*., 1996), but slagging may form at a temperature clearly below this. Thus, the risk of local slagging on the grate increases, if the masonry of the walls of the combustion chamber is added. The minimum exhaust flue gas temperature from the FGHEX derives from the dew point of flue gas, being in the scale of 70–100°C depending among other things on the moisture content of fuel, and on the construction of the heat exchanger. The temperature of the drying air must be clearly below 250°C, which is the ignition temperature of wood. The control strategy must ensure that the temperature does not even accidentally exceed the chosen safety level. On the other hand, increasing the drying air temperature improves the efficiency of the boiler plant. The maximum inlet air temperature depends on the dimensioning of the exhaust drying gas heat exchanger. The dimensioning outdoor temperature in Finland is $-26\ldots-38$°C depending on the region of the country. The inlet temperature to the heat exchanger is in the scale of 20–35°C and it is almost saturated. Consequently, the freezing risk limits the minimum exhaust air temperature.

6. CONCLUSIONS

The main conclusions of this study are:

1. Mathematical modelling proved to be a useful means to develop a heating plant for moist and dry bio fuel. Modelling of thermal behaviour of the plant helps in understanding interdependencies of two heat exchangers, boiler and dryer. Calculating combinations of extreme operating conditions (heat load, moisture content of fuel, outside temperature) gives the input data needed in comparing different types of heat exchangers, dimensioning the heat transfer area, choosing the control strategy and selecting operating parameters and set-values of control system.
2. One $40\,kW_{th}$ pilot plant was built purely for experimental purposes and one $500\,kW_{th}$ demonstration plant, both for testing and producing heat for the users during the development work. Using well-known engineering correlation formulas for heat and mass transfer, an adequate accuracy between model and verifying measurements was achieved.
3. Fouling was detected as a major problem with the FGHEX. However, in the absence of condensation, the increase of the fouling layer with respect to time was observed low. Fouling was a problem also with the DEGHEX, but the condensation of moisture in the exhaust drying air was not found to have an adverse effect on cleaning. After the installation of a simple dust collector a reasonable cleaning period was achieved.
4. Contrary to expectations, a counter-flow type heat exchanger was discovered inapplicable in the considered boiler plant. A mixed-flow type was found most appropriate as the FGHEX. Condensation is unavoidable with a counter-flow type if the fuel is wet and the

heat output is insufficient with a parallel-flow type. A parallel-flow type was found most appropriate as the DEGHEX because of the smallest risk of freezing and nearly equivalent heat output with other considered types. The latter is due to the big share of condensation heat.

5. It is necessary to control the heat output of the FGHEX at partial load in order to avoid condensation. The by-pass control accomplished lower consumption of fan power and the recirculation control appeared to reach higher heat output without condensing.
6. The DEGHEX decreases the required heat transfer rate from the flue gas to the drying air by preheating the drying air. This characteristic is indispensable in order to minimize condensation in the FGHEX in Finnish climate. It also saves fuel by recovering heat from the exhaust drying air back to the system. Its yearly fuel savings in typical district heating use was estimated to be 4.5%.
7. The VOC emissions of drying are low and do not cause corrosion or fouling problems on the heat transfer surfaces of DEGHEX. For environmental reasons a treatment of COD_{Mn} and phosphorous of the condensate should be considered.
8. A CHEX improves the efficiency of the plant because the average exhaust gas temperature is lower and the recovered phase change heat is higher. Fouling problems were not discovered during the short testing period, but they are anticipated in the long term.

NOMENCLATURE

A	= area (m^2)
D	= diffusivity ($m^2\,s^{-1}$)
d	= diameter (m)
C	= molar heat capacity ($J\,mol^{-1}\,K^{-1}$)
c	= specific heat capacity ($J\,kg^{-1}\,K^{-1}$)
e	= relative energy saving
f_1	= factor function (s_r) considering the staggering of tubes
f_2	= factor function (d_o, d_p, n_p, n_{rb}, s_p, s_r, z) considering the leakage and by-pass streams
f_3	= factor function ($T_{a,bl}$, T_w) considering the direction of the heat flux
h	= specific enthalpy ($J\,kg^{-1}$)
i	= time (h)
L	= length (m)
l	= heat of phase change ($J\,kg^{-1}$)
Le	= Lewis number
M	= molar mass ($g\,mol^{-1}$)
$\dot{m}$	= mass flow rate ($kg\,s^{-1}$)
$\dot{m}''$	= mass flux ($kg\,m^{-2}\,s^{-1}$)
n	= number
Nu	= Nusselt number
p	= pressure (Pa)
Pr	= Prandtl number
Re	= Reynolds number

RH	=	relative humidity (%)
s	=	distance of plates (mm)
t	=	temperature (°C)
T	=	temperature (K)
v	=	velocity ($\mathrm{m\,s^{-1}}$)
x	=	depth (m)
Y	=	mass fraction
y	=	height (m)
z	=	distance between baffles (m)
α	=	heat transfer coefficient ($\mathrm{W\,m^{2}\,K^{-1}}$)
ϕ	=	effect (W)
λ	=	heat conductivity ($\mathrm{W\,m^{-1}\,K^{-1}}$)
ν	=	kinematic viscosity ($\mathrm{m^{2}\,s^{-1}}$)
ρ	=	density ($\mathrm{kg\,m^{-3}}$)

Subscripts

a	=	air
b	=	baffled
bl	=	boundary layer
c	=	condensate
eg	=	exhaust gas
f	=	fouling
fg	=	flue gas
g	=	gas
i	=	inner, inside
lam	=	laminar flow
mg	=	mixed gas
o	=	outer, outside
p	=	pipe, perforation
pl	=	plate
pr	=	primary
r	=	row
rb	=	rows in a baffled zone
s	=	saturated
se	=	secondary
v	=	water vapour
w	=	water
y	=	year
turb	=	turbulent flow

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