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MODELLING OF A DOMESTIC GAS-FIRED CONDENSING BOILER

Sub - Task 2.1H

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CONTENTS

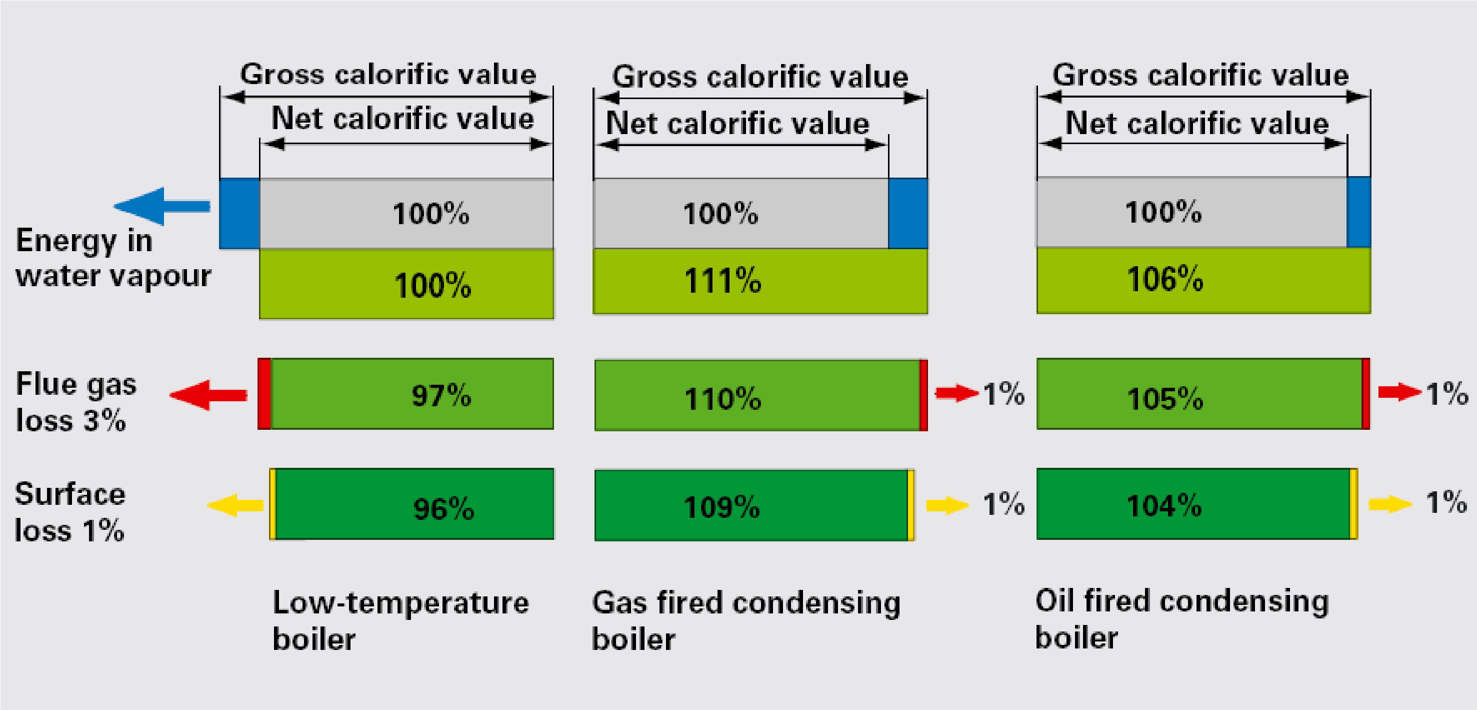
- Introduction (why?, investment and operating costs)
- Model description
- Test bench used to validate the model
- Model results
- Conclusions
- Acknowledgments

Condensing boilers (1)

- Energy efficiency improvements are needed to attain Kyoto targets (CO_2 emissions reduction)
- Exit flue gas temperature of conventional boilers is too high $\Rightarrow$ great amount of energy is lost to the environment
- Condensing boiler : both sensible and latent heats are recovered $\Rightarrow \eta \uparrow$

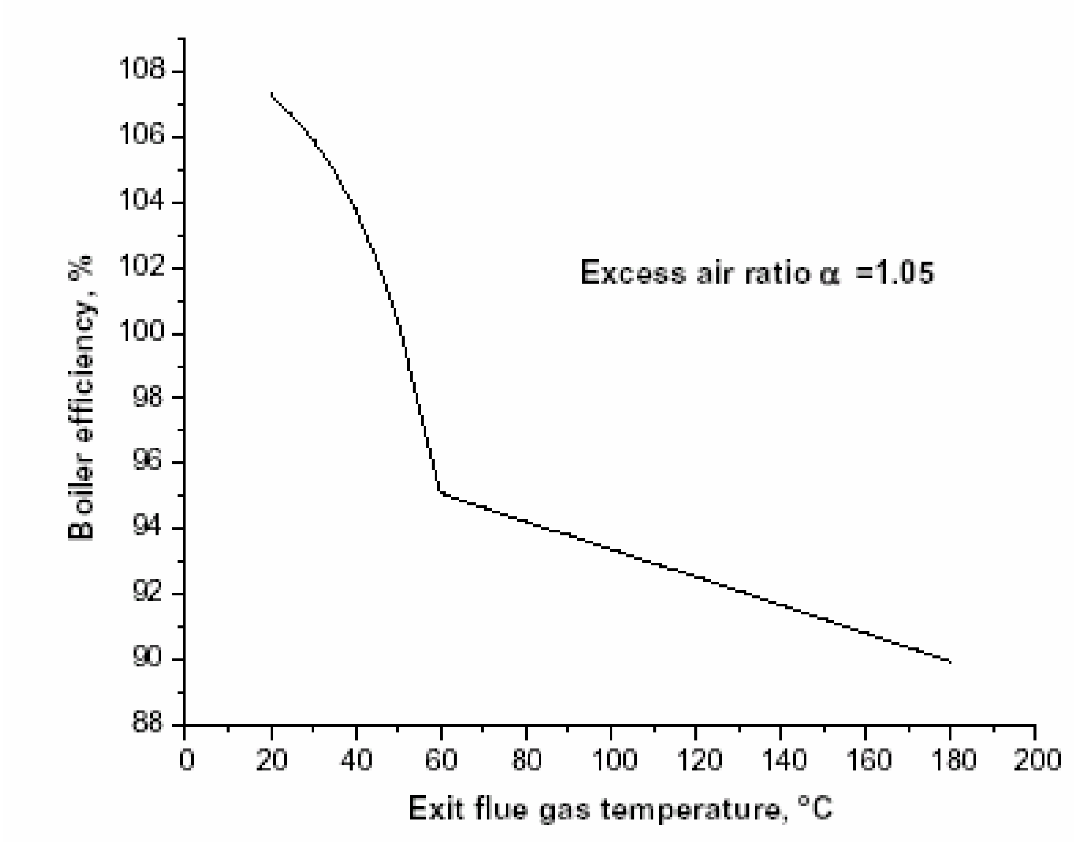
Condensing boilers (2)

Condensing boilers versus low-temperature boilers



Ref: Viessmann

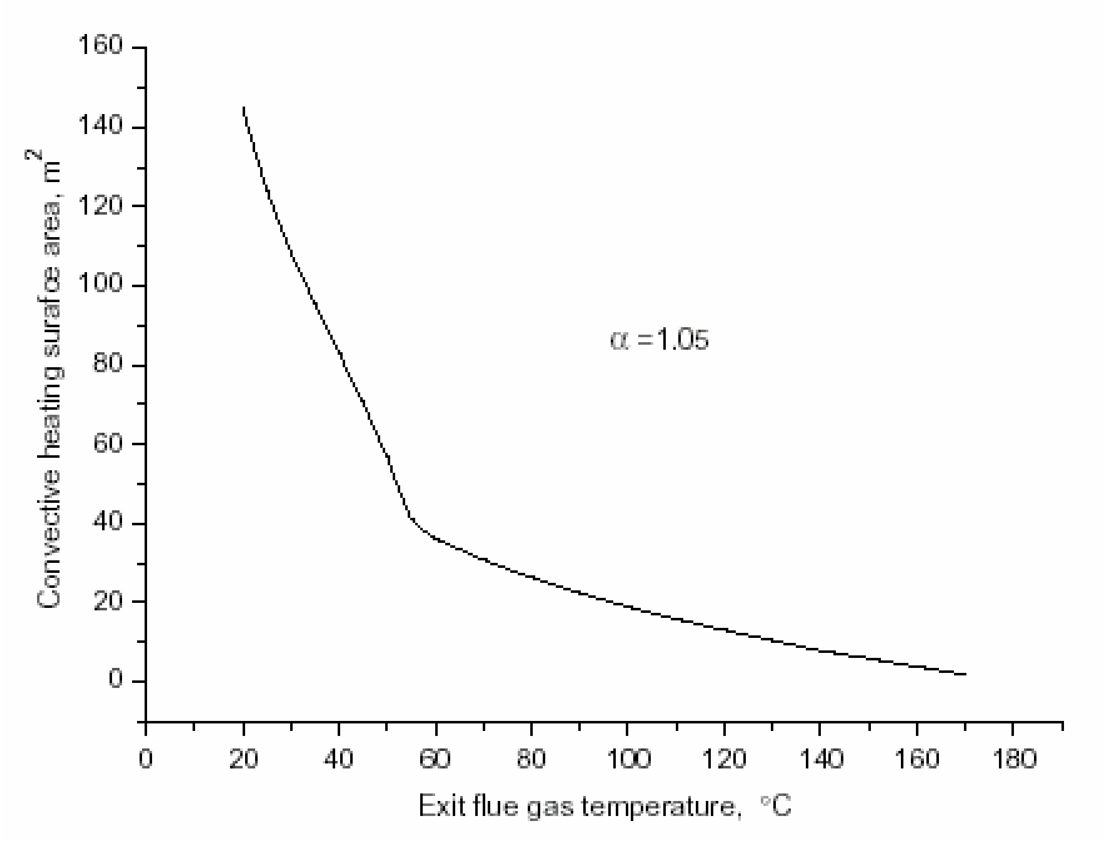
Condensing boilers (3)



Boiler efficiency based on LHV !

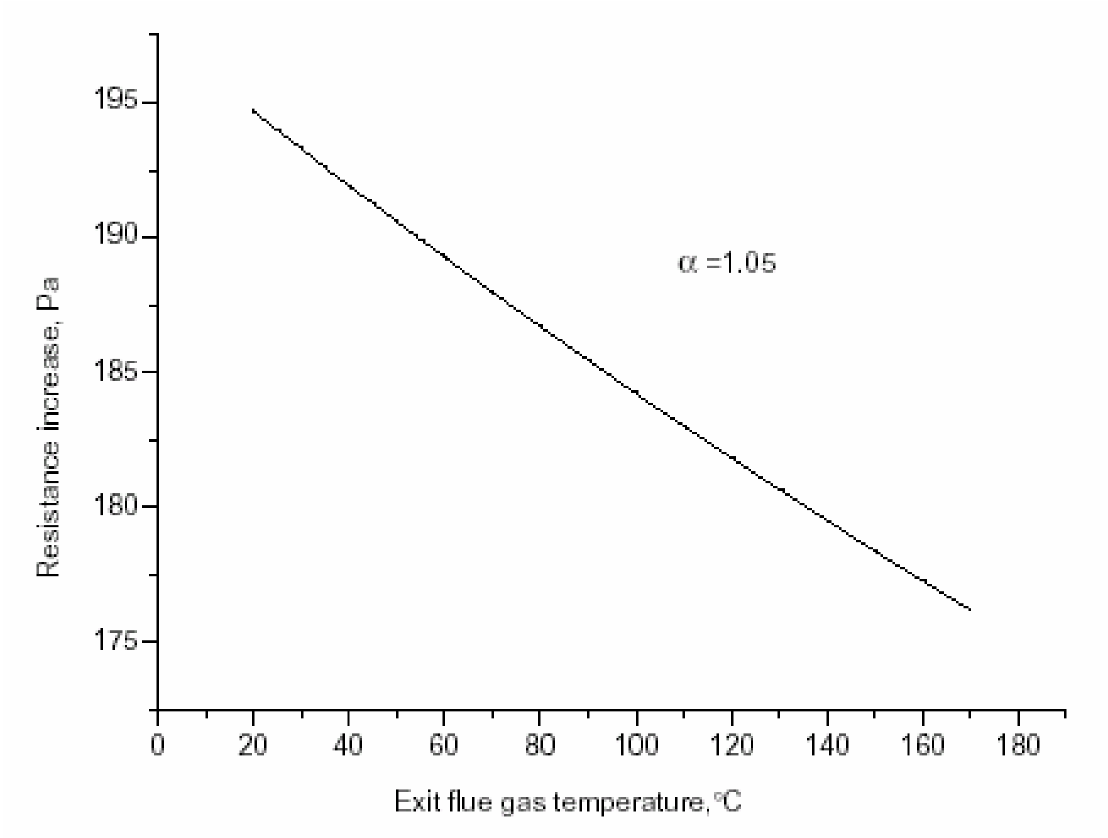
Ref : Che D.F., Y.H. Liu, and C.Y. Gao, *Evaluation of retrofitting a conventional natural gas fired boiler into a condensing boiler.* Energy Conversion and Management, 2004. **45**(20): p. 3251-3266.

Condensing boilers (4)



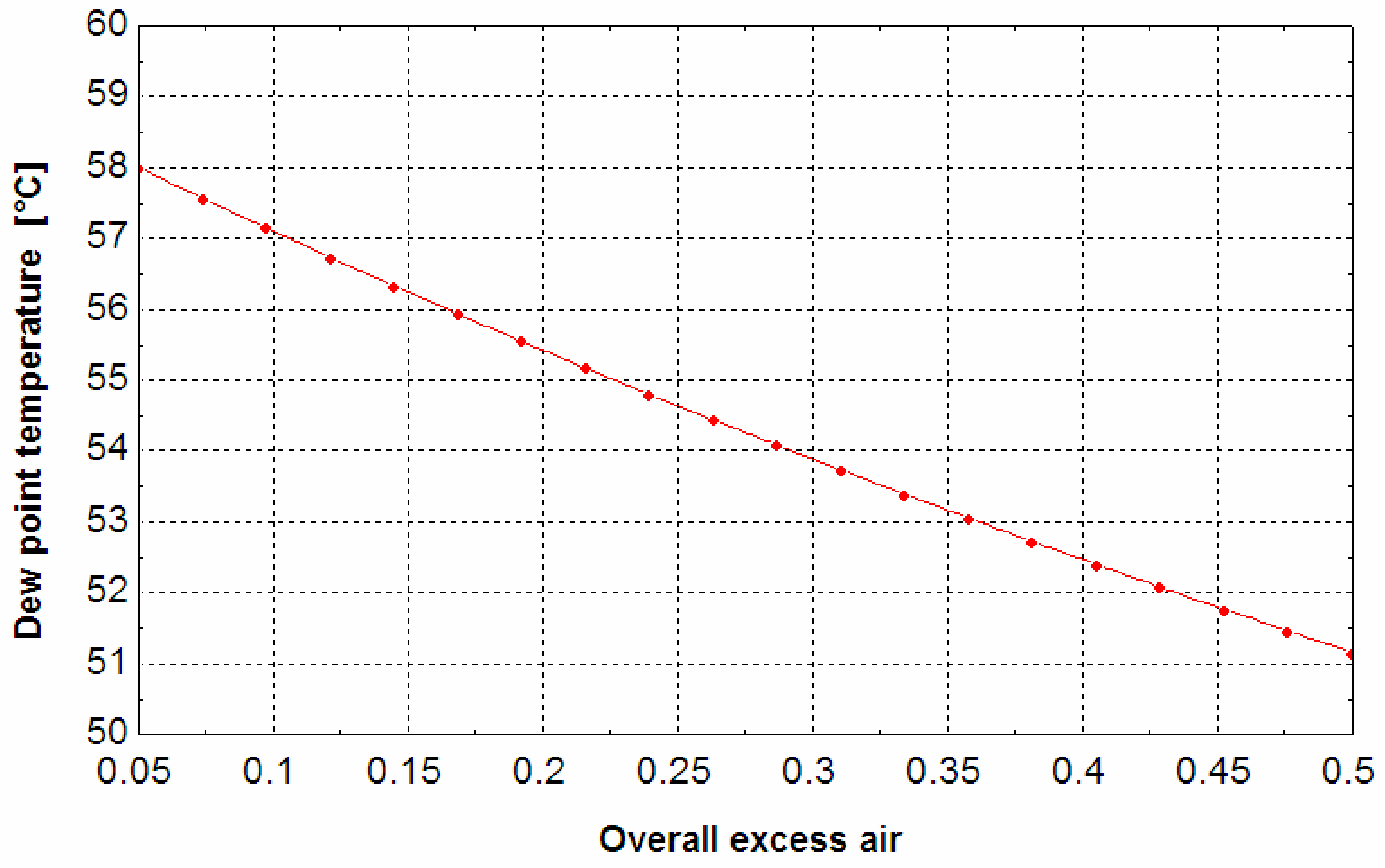
Ref : Che D.F., Y.H. Liu, and C.Y. Gao, *Evaluation of retrofitting a conventional natural gas fired boiler into a condensing boiler.* Energy Conversion and Management, 2004. **45**(20): p. 3251-3266.

Condensing boilers (5)

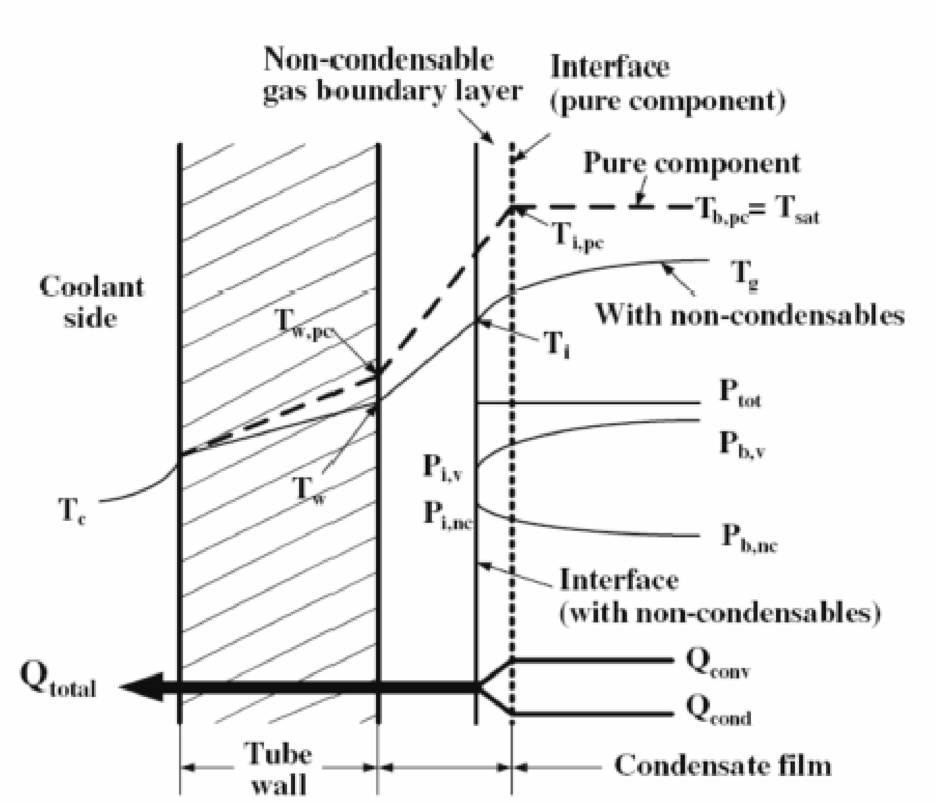


Ref : Che D.F., Y.H. Liu, and C.Y. Gao, *Evaluation of retrofitting a conventional natural gas fired boiler into a condensing boiler.* Energy Conversion and Management, 2004. **45**(20): p. 3251-3266.

Dew point temperature

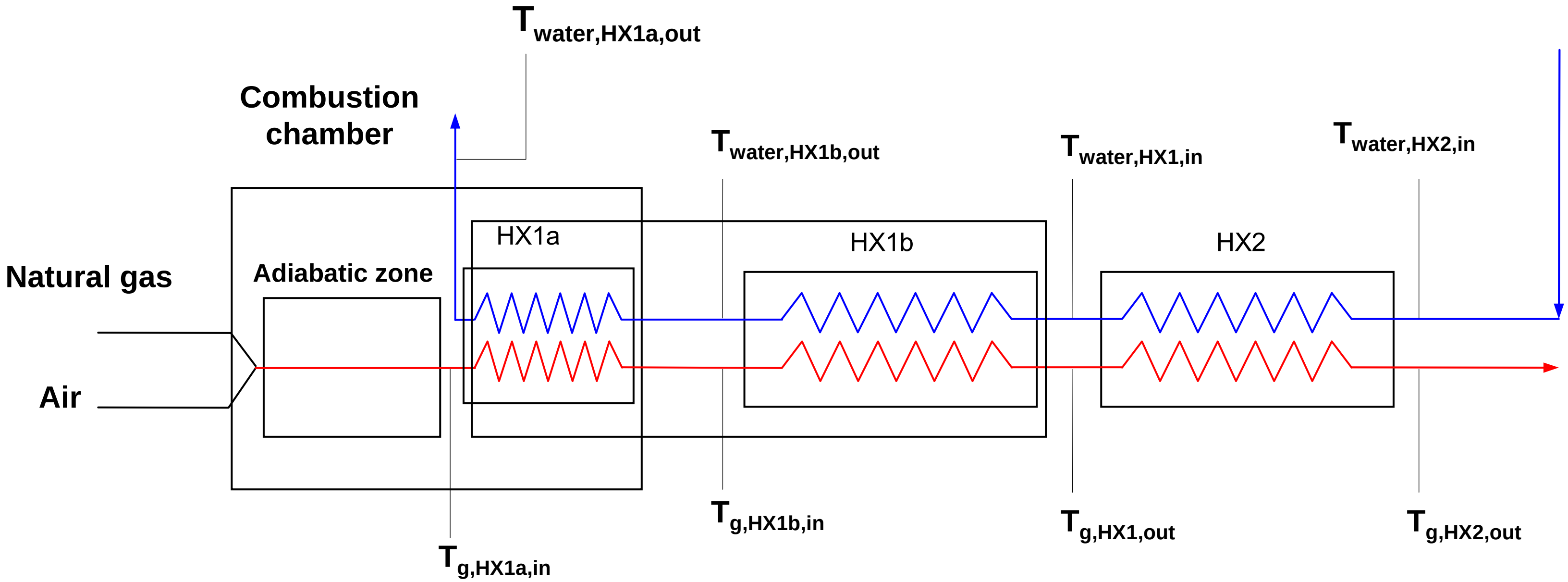


Model description (0)



Ref : Liang, Y., Che D., and Kang Y., *Effect of vapor condensation on forced convection heat transfer of moistened gas.* Heat and mass transfer, 2007. **43**(7): p. 677-686.

Model description (1)



Three main zones: the adiabatic zone, the main heat exchanger (dry) and the wet heat exchanger (condensing)

Model description (2)

1. **<u>Adiabatic zone</u>**

Combustion part of the model.
Adiabatic temperature is calculated taking into account dissociation at high temperature.

2. **<u>Main Heat Exchanger (HX1)</u>**

Exact geometry of the real heat exchanger is unkown

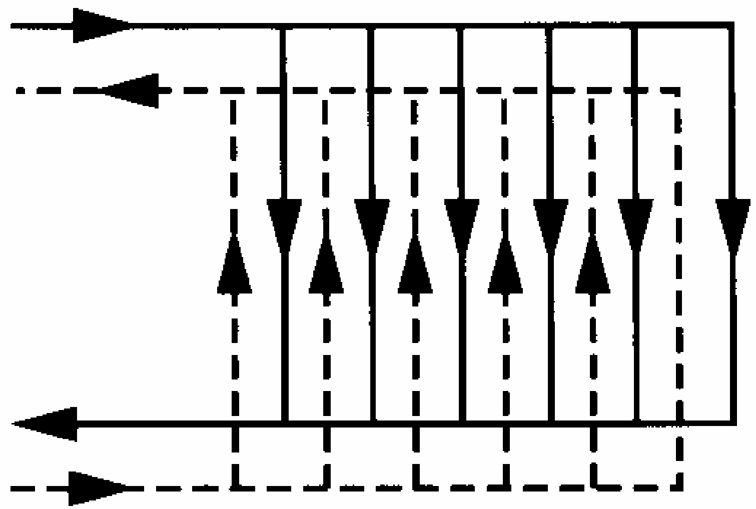
→ Simplified by a counterflow heat exchanger
→ Overall transfer coefficient calculated by:

$$UA_{HX1} = UA_{HX1,N}\left(\frac{\dot{M}_g}{\dot{M}_{g,N}}\right)^{0,8}$$

Model description (3)

3. Wet Heat Exchanger (HX2)

Simulated by a Plate Heat Exchanger with chevrons



Single pass counterflow U-arrangement

Model description (4)

Heat Exchanger (HX2) divided into N elements. For each element :

Dry heat transfer coefficient h_{dry} calculation

↓

Flue gas – Wall – Water Temperatures

↓

When wall temperature at the hot side < Dew point → Condensation

↓

Wet heat transfert h_{wet} when condensation

Model description (5)

Dry heat transfer coefficient h_{dry} is calculated by Martin's Model for Plate Heat Exchangers.

Two main correlations:

$$Nu = 0.122 * Pr^{1/3} * \left(\mu / \mu_w\right)^{1/6} * \left[\xi * Re^2 * \sin(2\varphi)\right]^{0.374}$$

$$\frac{1}{\sqrt{\xi}} = \frac{\cos\varphi}{\sqrt{0.045 * \tan\varphi + 0.09 * \sin\varphi + \xi_0(Re)/\cos\varphi}} + \frac{1-\cos\varphi}{\sqrt{\xi_1(Re)}}$$

Model description (6)

For laminar flows (Re<2000):

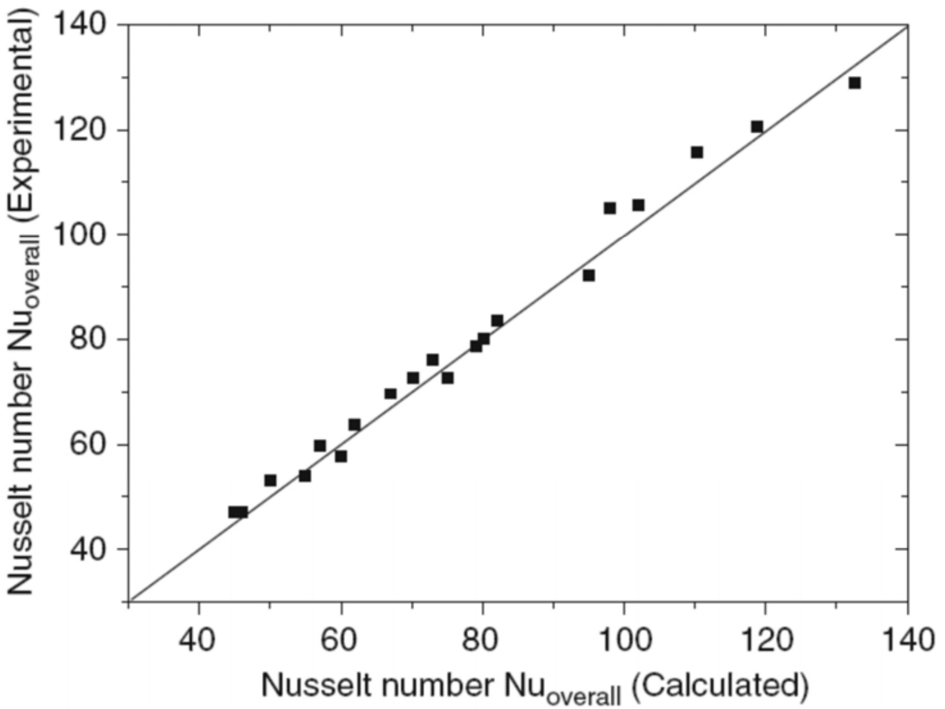
$$\xi_0 = \frac{16}{\mathrm{Re}} \qquad \xi_1 = 3.8 * \left(\frac{149.25}{\mathrm{Re}} + 0.9625 \right)$$

For turbulent flows (Re≥2000):

$$\xi_0 = \left(1.56 * \ln(\mathrm{Re}) - 3\right)^{-2} \qquad \xi_1 = 3.8 * \left(\frac{9.75}{\mathrm{Re}^{0.289}} \right)$$

Ref : Martin H., *A theoretical approach to predict the performance of chevron-type plate heat exchangers.* Chemical engineering and processing, 1996. **35**(4): p. 301-310.

Model description (7)



$$h_{wet} = \alpha * h_{dry} \qquad 1 < \alpha < 3.5$$

Ref : Jia, L., *Effects of water vapor condensation on the convection heat transfer of wet flue gas in a vertical tube.* International journal of heat and mass transfer, 2001. **44**(22): p. 4257-4265.

Model description (8)

Convective heat transfer coefficient in the elements where condensation occurs

$$\alpha_j = \frac{h^*_j}{h_{dry}}$$

$$\dot{Q}_j = h^*_j * S_j * (T_{g,m,j} - T_{w,m,j})$$

$$\dot{Q}_j = h_{dry} * S_j * (T_{g,m,j} - T_{w,m,j}) + h_{fg} * \dot{M}_{cond,j}$$

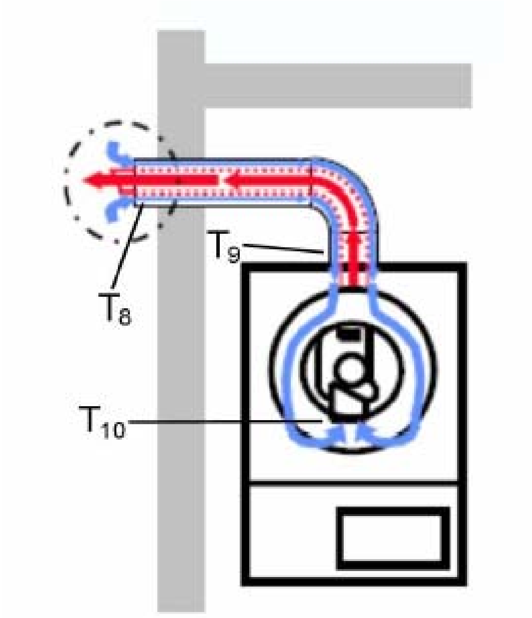
$$\dot{M}_{cond,j} = \beta * \frac{\dot{M}_{cond}}{N_{cond}}$$

Test Bench (1)

Viessmann Vitodens 300
24 kW
Inox Radial Heat
Exchanger

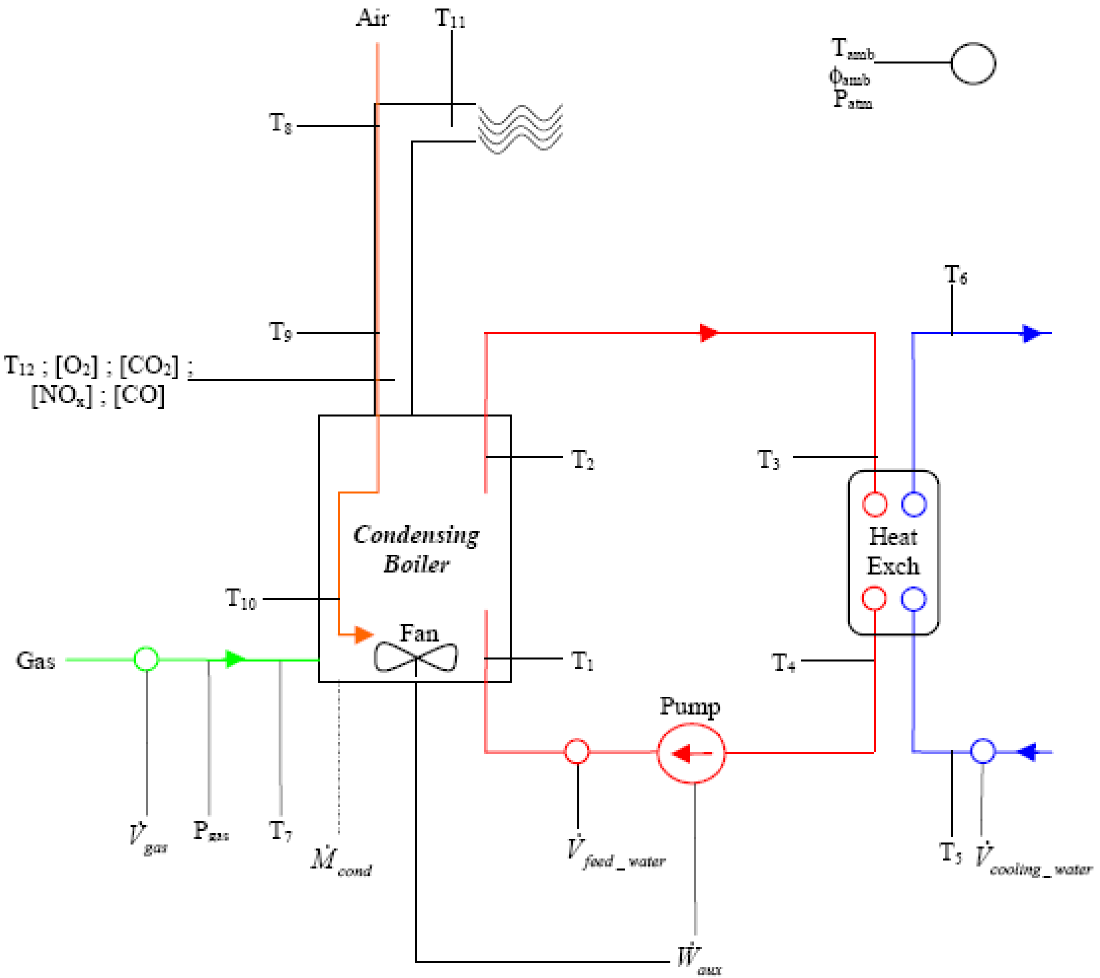
Ref: Viessmann

Test Bench (2)

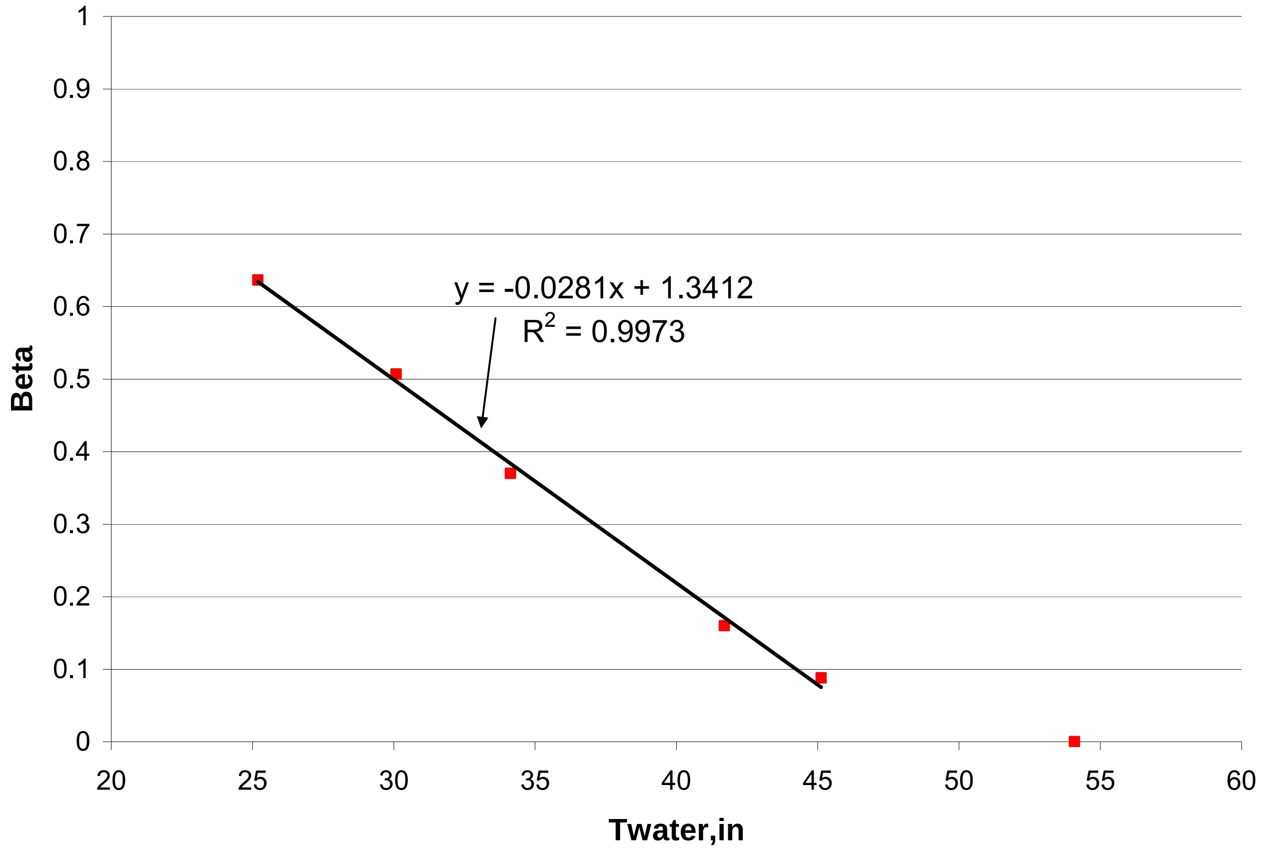


Condensing gas-fired boiler with preheated air combustion

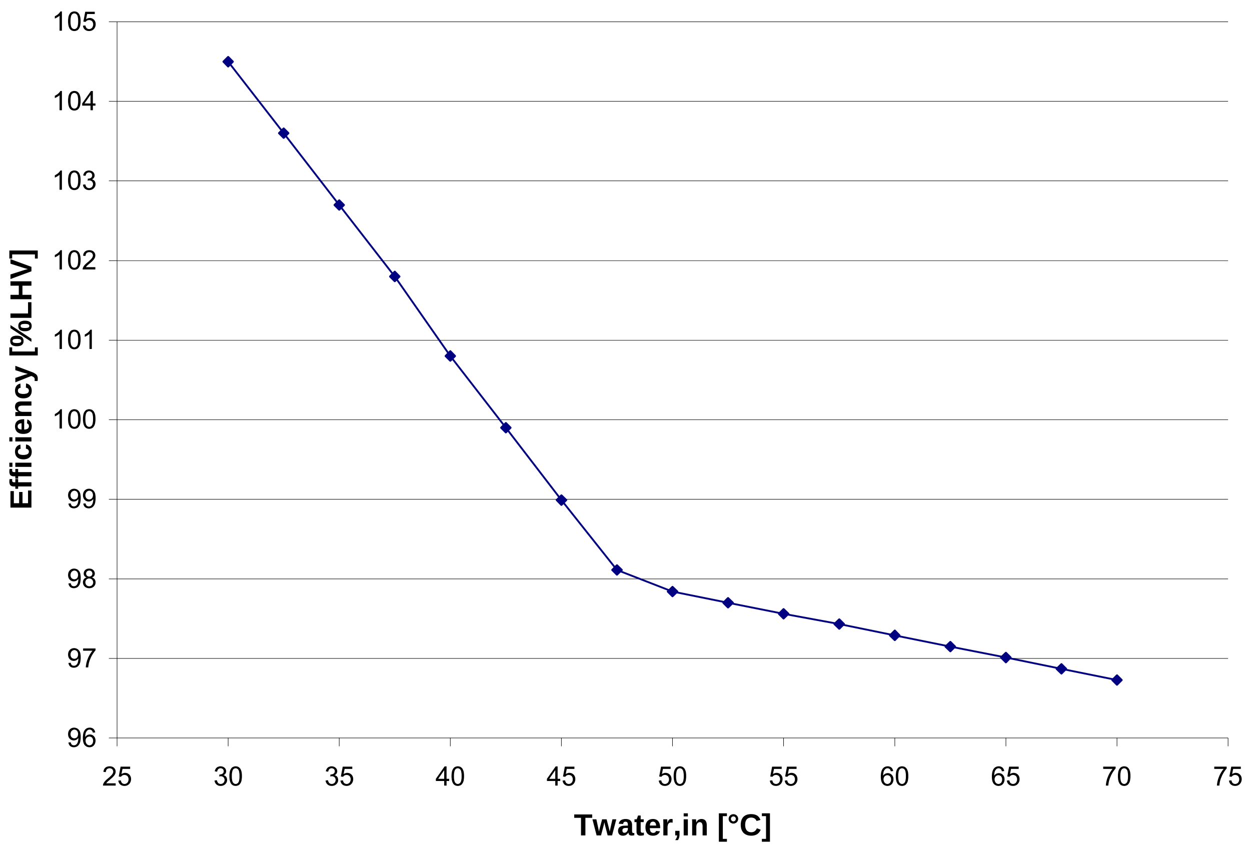
Test Bench (3)


Air
T_{11}
T_{amb}
ϕ_{amb}
P_{atm}
T_8
T_6
T_9
T_{12} ; $[O_2]$; $[CO_2]$;
$[NO_x]$; [CO]
T_2
T_3
Condensing
Boiler
Heat
Exch
T_{10}
Fan
Gas
T_1
T_4
Pump
$\dot{V}_{gas}$
P_{gas}
T_7
$\dot{M}_{cond}$
$\dot{V}_{feed\_water}$
T_5
$\dot{V}_{cooling\_water}$
$\dot{W}_{aux}$

Results (1)

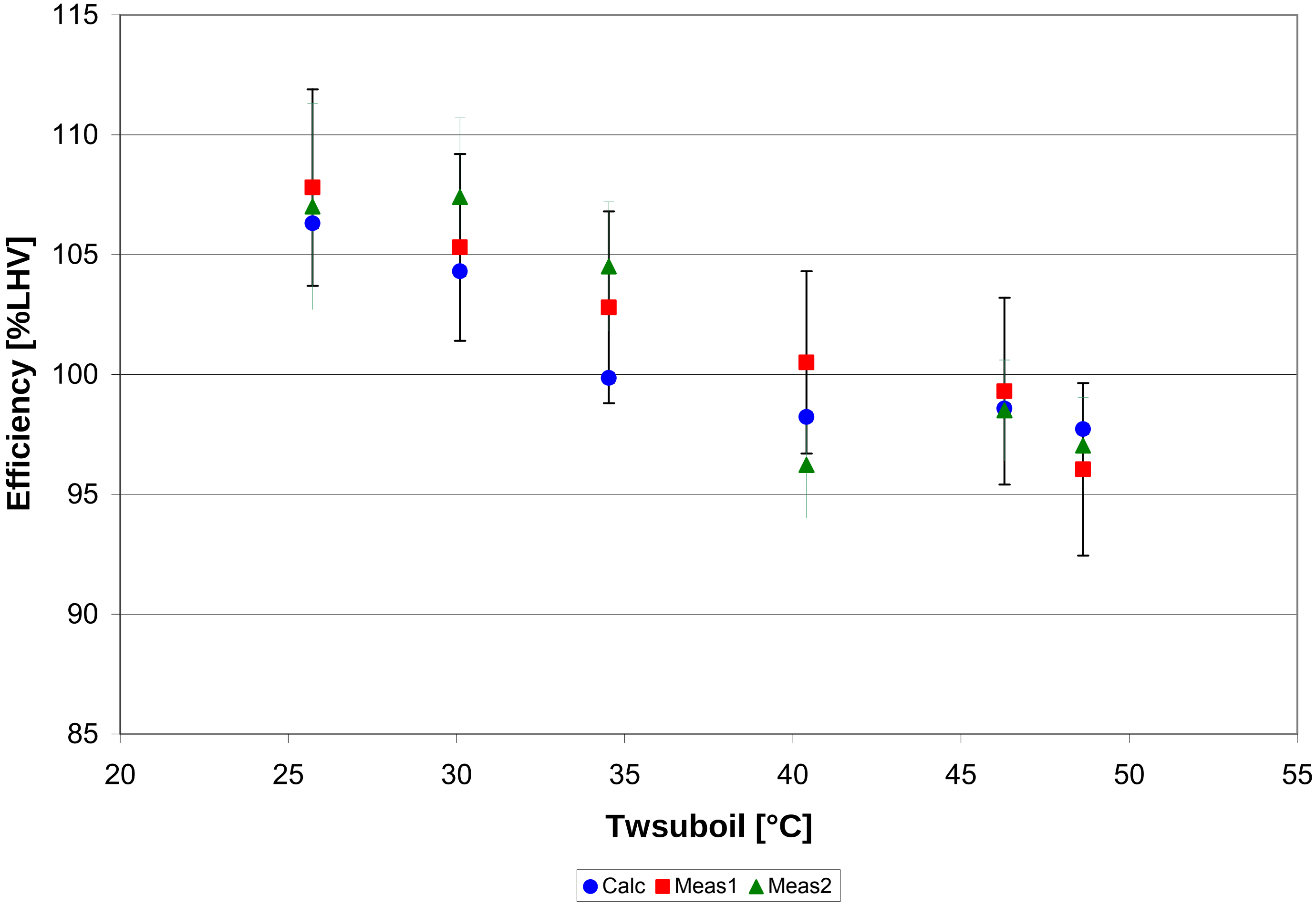

y = -0.0281x + 1.3412
R2 = 0.9973
Beta
Twater,in
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
20
25
30
35
40
45
50
55
60

Results (2)



105
104
103
102
101
100
99
98
97
96
Efficiency [%LHV]
25
30
35
40
45
50
55
60
65
70
75
Twater,in [°C]

Results (3)



Efficiency [%LHV]
115
110
105
100
95
90
85
20
25
30
35
40
45
50
55
Twsuboil [°C]
Calc
Meas1
Meas2

Conclusions

- Condensing boilers are more efficient than non-condensing boilers:
  - - 11% of retrievable latent heat
  - - Sensible heat recovery
- Boilers efficiency should be based on Gross Calorific Value (HHV)
- The model developped gives the correct trend for the boiler thermal efficiency
- Validation of the model to be continued

Acknowledgements

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Thank you for your attention

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Abstract

Nowadays, condensing technology is widely used in domestic heating. This paper presents a model that simulates the performance in steady-state regime of a domestic gas-fired condensing boiler. The model is similar to a conventional boiler model with a dry heat exchanger at which a wet heat exchanger is added to simulate the heat recovered by condensation. Experimental parameters are used to simulate the different components. The model has been applied to a 24 kW gas-fired condensing boiler test bench available in the laboratory. According to the experimental results (at full load), the model predicts correctly the thermal efficiency of the boiler. Indeed, calculated values are in good agreement with measured values.

List of symbols

α_j	Enhanced convection heat transfer coefficient factor
β	Condensing rate parameter
c_p	Specific heat at constant pressure
d_h	Hydraulic diameter
φ	Corrugation inclination angle
ξ	Fanning friction factor
h	Convection heat transfer coefficient
h_{fg}	Enthalpy of condensation
k	Thermal conductivity
N	Number of intervals used for the discretization of the condensing heat exchanger
N_{pl}	Number of plates (parameter of the condensing heat exchanger)
NTU	Number of transfer units
Nu	Nusselt number = hd_h/k
Re	Reynolds number = $\rho u d_h/\mu$
ρ	Density
Pr	Prandtl number = $\mu c_p/k$
S	Surface
T	Temperature
u	Flow velocity
UA	Overall heat transfer coefficient
μ	Dynamic viscosity

Subscripts

b	bulk
dry	in dry condition
g	flue gas
i	interface
in	inlet
j	for one interval
N	Nominal values
m	mean
out	outlet
v	vapor
w	at wall temperature

1. Introduction

Within the actual context of energy savings and pollutant emissions reduction, installing condensing boilers becomes indispensable because of their high potential efficiency compared to conventional (non condensing) boilers.

Conventional boilers are designed in order to avoid flue gas condensation and thus low temperature corrosion. Their exit flue gas temperature is typically around 180°C. This high exhaust temperature leads to up to 10% of sensible heat loss. As the latent heat of natural gas exhaust is around 11% of the calorific value of the gas, the total exhaust heat loss is about 16-22% of the gross calorific value of the gas.

Condensing technology consists of recovering part of this heat loss. For a gas-fired condensing boiler, the exhaust must be cooled below the dew point temperature(around 57°C for an overall air excess of 10% - Figure 1) so that the latent heat can be partly recovered as well as the sensible heat.

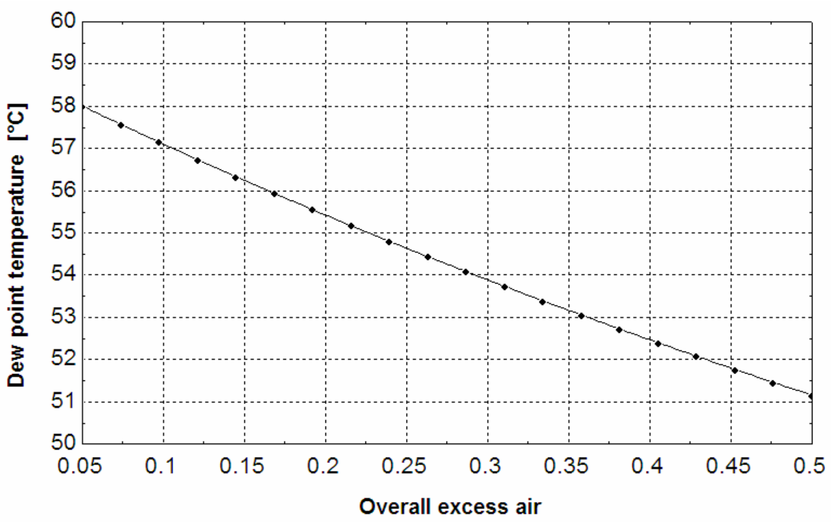

Figure 1 : Dew point temperature versus overall excess air for natural gas (~ CH_4)

The thermal efficiency of the boiler is thus significantly increased. Figure 2 shows for an overall excess air ratio of 1.05 the theoretical relationship of the boiler thermal efficiency based on lower heating value versus exit flue gas temperature [1]. It can be seen on that figure that after a relatively gradual increase in the temperature range of 60-180°C, the boiler efficiency rises sharply with condensation to attain up to 108% for a flue gas temperature equal to 20°C.

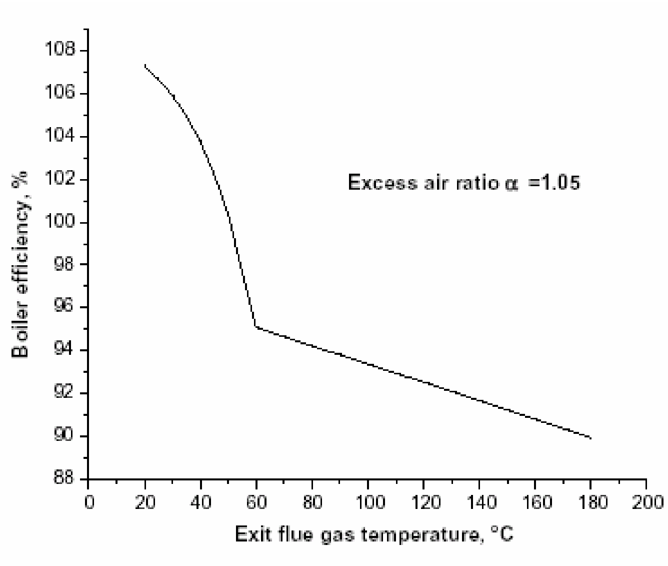

Figure 2 : Boiler efficiency versus exit flue gas temperature [1]

To ensure continuous condensing operation, heat exchange surfaces must be high enough and, therefore, return water must be below the dew point temperature. That means that the heating system must be designed so that its return water temperature is low enough (underfloor heating systems for example). A simulation model able to predict boiler performance and condensing rate in the boiler heat exchanger is very valuable for building simulation programs. There has been few published work in this area [2] contrary to the condensation of pure vapor and the data is mostly available only from manufacturers testing. Moreover, existing models are based on the sensible heat ratio method usually used in cooling coil performance calculation [2].

The model developed within this work is a one-dimension model of a domestic gas-fired condensing boiler. The aim of the model is to know whether or not the boiler works in good condensing conditions.

2. Model description

The structure of the model is similar to a conventional boiler model [3] with a combustion chamber and a dry heat exchanger (HX1), at which a wet heat exchanger (HX2) is added downstream (Figure 3).

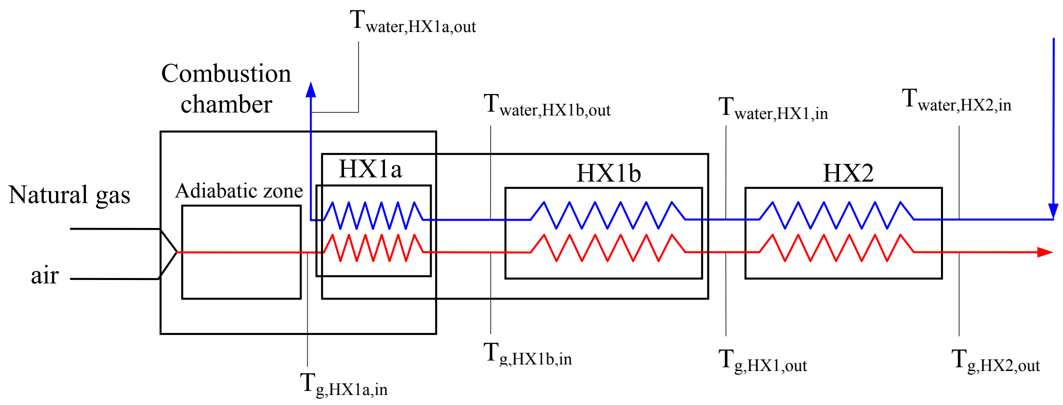

Figure 3: Schematic diagram of the condensing boiler model

- **The adiabatic zone**

In this zone, the oxygen content in the flue gas at the chimney being known, the adiabatic temperature ($T_{fg,HX1a,in}$) as well as the combustion products composition are calculated.

- **The main heat exchanger (HX1)**

Heat transfer from combustion products cannot be calculated using conventional radiation and convection relationships because the exact heat exchanger geometry is generally unknown.

The gas-water heat exchanger is simplified in this work by a counterflow heat exchanger. This exchanger considers the part of the heat exchanged around the combustion chamber and the sensible heat before the condensing heat exchanger. As the outlet temperatures of the fluids are not initially known, the epsilon-NTU method is used to get converged values. From Dittus-Boelter correlation for turbulent flows, the global transfer coefficient can be determined by :

$$UA_{HX1} = UA_{HX1,N}\left(\frac{\dot{M}_g}{\dot{M}_{g,N}}\right)^{0,8} \quad (1)$$

where nominal values ($UA_{HX1,N}$ et $\dot{M}_{g,N}$) are fitted with experimental data.

- **The condensing heat exchanger (HX2)**

The wet heat exchanger is simulated by one of the most commonly used heat exchanger design: plate heat exchangers with chevrons. A single-pass counterflow U-arrangement was chosen (Figure 4). The number of plates (N_{pl}) is used to fit the plate heat exchanger model to the real condensing heat exchanger. The basic plate geometry is taken from an existing one: a plate surface of 0.0125m² , thickness of 1mm and corrugation inclination angle (φ) of 60°. The distance between two plates is equal to 2.5mm.

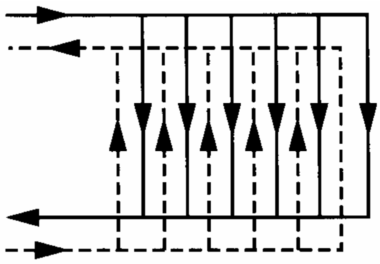

Figure 4: Single-pass counterflow plate heat exchanger with U-arrangement

Martin [4] provides comprehensive correlations to calculate friction factors and heat transfer coefficients as a function of plate heat exchangers geometry and Reynolds number. The equations are the following:

$$\mathrm{Nu} = 0.122 * \mathrm{Pr}^{1/3} * \left(\mu / \mu_w \right)^{1/6} * \left[\xi * \mathrm{Re}^2 * \sin(2\varphi) \right]^{0.374} \quad (2)$$

$$\frac{1}{\sqrt{\xi}} = \frac{\cos\varphi}{\sqrt{0.045 * \tan\varphi + 0.09 * \sin\varphi + \xi_0(\mathrm{Re}) / \cos\varphi}} + \frac{1 - \cos\varphi}{\sqrt{\xi_1(\mathrm{Re})}} \quad (3)$$

For laminar flows:

$$\xi_0 = \frac{16}{\mathrm{Re}} \quad \text{and} \quad \xi_1 = 3.8 * \left(\frac{149.25}{\mathrm{Re}} + 0.9625 \right) \text{ if } \mathrm{Re} < 2000 \,. \quad (4)$$

For turbulent flows:

$$\xi_0 = \left(1.56 * \ln(\mathrm{Re}) - 3\right)^{-2} \quad \text{and} \quad \xi_1 = 3.8 * \left(\frac{9.75}{\mathrm{Re}^{0.289}} \right) \text{ if } \mathrm{Re} \geq 2000 \quad (5)$$

Martin's model is used to calculate heat transfer coefficients at the water and gas sides considering that the flue gas does not condense. According to previous studies [1, 5-7], the heat transfer coefficient for vapor-gas mixture with a large amount of non-condensable gas is higher than that of forced convection without condensation and much lower than that of condensation of pure steam. Jia et al [5] found that vapor condensation in the wet flue gas enhances convection heat transfer and excessive difference between bulk and wall temperature has a strong influence on convection heat transfer, especially for cases of low vapor fraction. Che et al [1, 6] recommends to take the convective heat transfer coefficient with condensation 1.5-2 times the corresponding convective heat transfer coefficient without condensation. Liang et al [7] found that convective heat transfer coefficient increases as the gas-vapor mixture flow rate and the mass fraction of water vapor increase. In their experimental range, this coefficient was 1-3.5 times that of forced convection without condensation.

The heat transfer coefficient for the vapor-gas mixture cannot be held constant if the heat exchanger wall temperature falls below the dew point temperature. In the developed model, the condensing heat exchanger has been divided into N intervals. First, calculation is performed considering that the vapor does not condense. Gas, water and wall temperatures are calculated for each part. The flue gas starts to condense at the heat exchanger interval where the wall temperature at the gas side is below the dew point temperature. For this interval and the followings, calculation is performed a second time with an enhanced convection heat transfer coefficient factor (α_j) calculated by equations (6,7,8).

$$\alpha_j = \frac{h^*_j}{h_{dry}} \quad (6)$$

$$\dot{Q}_j = h^*_j * S_j * (T_{g,m,j} - T_{w,m,j}) \quad (7)$$

$$\dot{Q}_j = h_{dry} * S_j * (T_{g,m,j} - T_{w,m,j}) + h_{fg} * \dot{M}_{cond,j} \quad (8)$$

The mass fraction of condensed vapor on each element is unknown. However, the maximum condensing rate ($\dot{M}_{cond}$) can be calculated from the vapor content in the flue gas. In a first approximation, it is considered that the vapor condenses linearly on each element and the condensing rate on one element ($\dot{M}_{cond,j}$) is calculated by equation (9) where N_{cond} is the number of "wet" elements. In most cases, there will be remaining vapor in the flue gas so that the maximum condensing rate on one element is multiplied by a factor $\beta<1$. That factor is the ratio of vapor which is really condensed in the boiler to the maximum vapor content in the flue gas. It is fitted to experimental results and is a function of inlet water temperature in the heat exchanger.

$$\dot{M}_{cond,j} = \beta * \frac{\dot{M}_{cond}}{N_{cond}} \quad (9)$$

3. Model results and discussion

The model has been applied to a Viessmann Vitodens 300 gas-fired boiler whose nominal output is 24 kW. Figure 5 is a schematic representation of the test bench. A secondary heat exchanger has been installed to simulate the heat emitters and get the desired boiler water inlet temperature. As all the system is well insulated, the useful power transferred into the water can be measured by two ways: from the boiler closed loop or from the open loop at the secondary heat exchanger simulating the emitters.

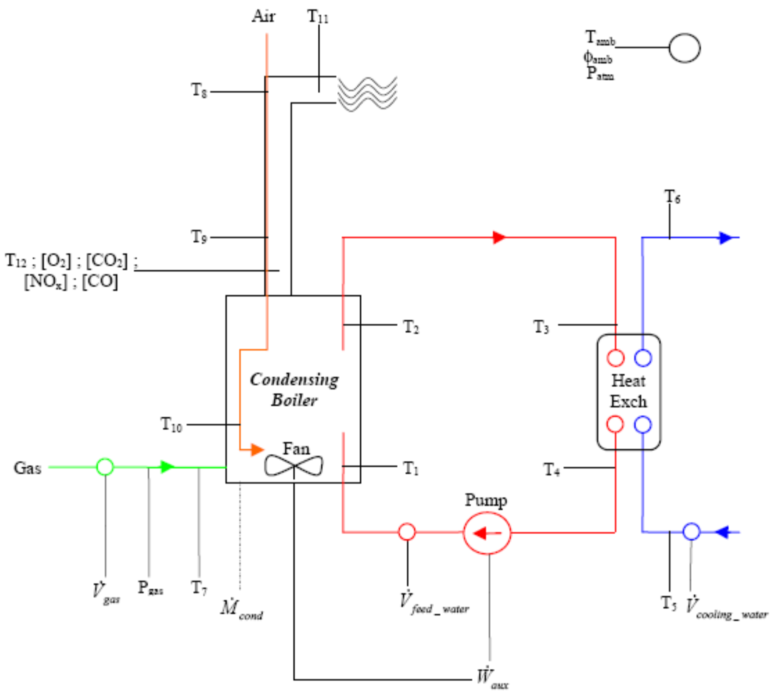

Figure 5: Test bench of the condensing boiler

Tests have been performed in steady-state regime in order to determine the model parameters (Table 2). Natural gas composition for all tests is given in Table 1. The nominal gas flow rate and the heat transfer coefficient of the dry heat exchanger have been fitted when the boiler was working in complete dry conditions at 6% of oxygen at the chimney. The useful power transferred to the water was taken as the reference. For the β-factor, an experimental relationship (Figure 6) was determined as the condensed water was collected and weighted continuously during the tests for boiler water inlet temperatures ranging from 25°C to 55°C. On figure 6, one observes that β is equal to zero when the boiler water inlet temperature is higher than 48°C.

Table 1: Natural gas composition (molar fractions)

x_{N2}	0.028832
x_{CO2}	0.018899
x_{CH4}	0.87425
x_{C2H6}	0.06126
x_{C3H8}	0.01229
x_{C4H10}	0.00313
x_{C5H12}	0.00036
x_{C6H14}	0.00083
x_{He}	0.0002

Table 2: Nominal parameters @ 6% O_2

$UA_{HX1,N}[K/W]$	$\dot{M}_{g,N}[kg/s]$	N_{pl}
12.32	0.0125	31

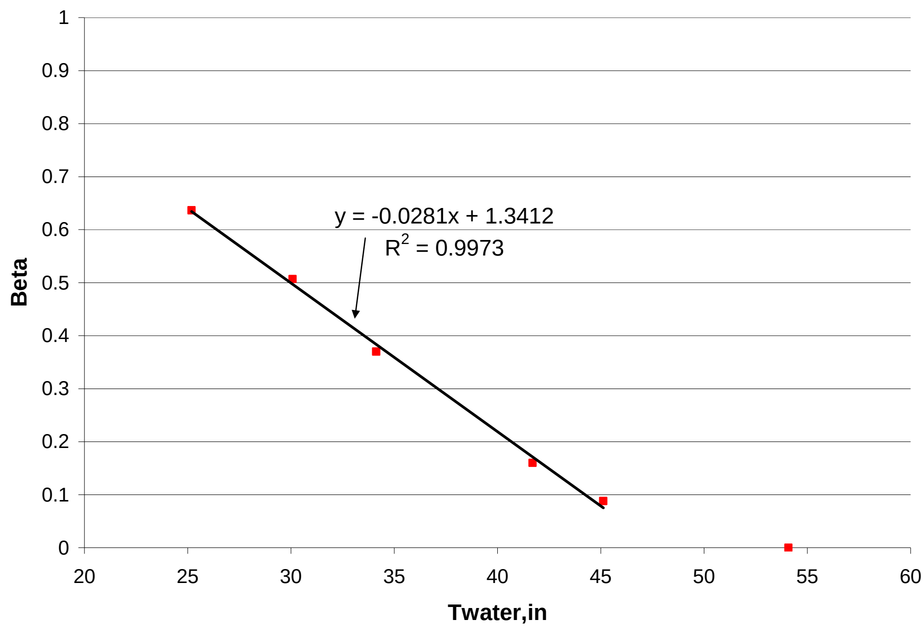

Figure 6 : β factor as a function of the boiler water inlet temperature [°C]

In order to check the model behavior, the water inlet temperature was varied from 30°C to 70°C keeping all other inputs constant (see Table 3). The output of the model is shown on Figure 7 where the boiler efficiency is plotted as a function of the boiler water inlet temperature. That figure shows that the model gives the correct trend compared to Figure 2. Moreover, the convective heat transfer coefficient with condensation (α_j) was found to be 1.1 to 2 times the heat transfer coefficient without condensation, which is in good agreement with Che et al [1, 6] and Liang et al [7].

To validate the model, complementary tests have been performed at nominal power (full load) and 6% of oxygen at the chimney. Calculated thermal efficiency has been compared to the measured one (Figure 8). In fact, uncertainty on temperature measurement is 0.5°C while it is 1% on water flow rates measurement. Total uncertainty on efficiency is around 4% in the case of useful power measured on the boiler closed loop ("meas1") while it is only 2.5% in the case of useful power measured at the secondary heat exchanger ("meas2"). That discrepancy in the measured boiler thermal efficiencies is explained by a larger temperature difference at the secondary of the heat exchanger compared to the temperature difference on the boiler closed loop. Considering measurement uncertainty, the calculated efficiencies are acceptable.

The boiler thermal efficiency is underestimated for high condensing rates whereas it seems to be overestimated for low condensing rates. This trend could be explained by an underestimation of the β-factor for high condensing rates. Indeed, when condensation occurs, heat transfer coefficient is underestimated because combustion gas exhaust temperatures are higher than the measured ones (up to 30°C higher). When there is no condensation (water inlet temperature above 48°C), the model seems to correctly evaluate the thermal efficiency.

Table 3: Model inputs

$T_{water,in}$	From 30°C to 70°C
Q_{NG}	2.239 m³/h
$Q_{water,boiler}$	958.2 l/h
T_{air}	23.7 °C
T_{NG}	48.1 °C
P_{atm}	990 mbar
P_{NG}	19.27 mbar
O2	6.1 %
LHV	45.10 MJ/kg

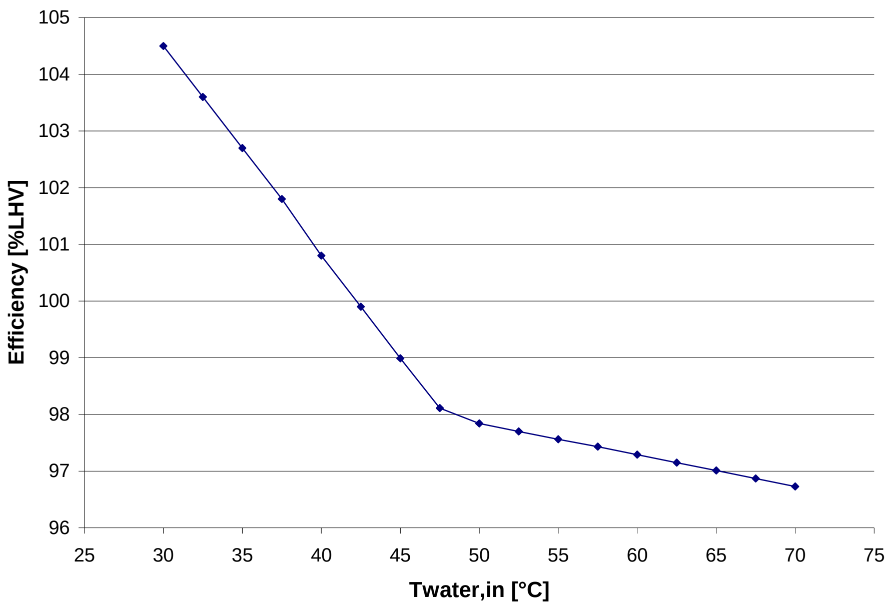

Figure 7: Boiler efficiency (on LHV basis) versus water inlet temperature

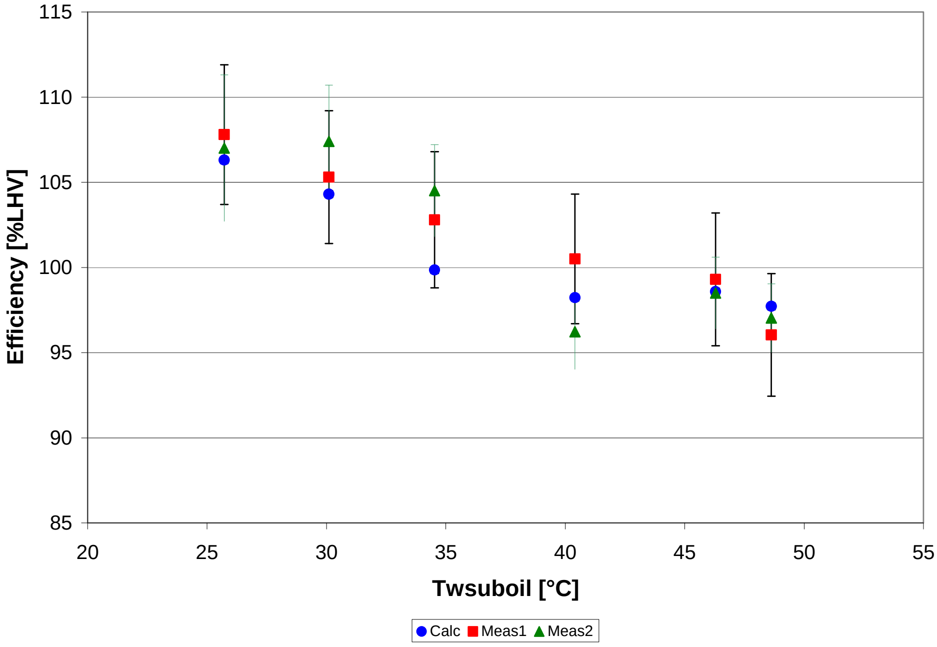

Figure 8: Boiler efficiency (on LHV basis) versus water inlet temperature – comparison of calculated values with measured values ("meas1" for the boiler closed loop and "meas2" for the secondary heat exchanger)

4. Conclusion

A model that simulates the performance in steady-state regime of a domestic gas-fired condensing boiler has been developed.

The model was applied to a 24 kW gas-fired condensing boiler installed on a test bench. According to experimental results, the model gives the correct trend for the thermal efficiency. However, it seems that thermal efficiency is underestimated for high condensing rates while it is overestimated for low condensing rates. The model gives the good results when there is no condensation.

The model has only been validated at nominal output (full load) and 6% of oxygen at the chimney. Further tests should be carried out to determine model parameters for other overall excess air and to validate the model at part loads.

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