In situ study on the condensate latent heat recovery and its economic impact in the case of a 60 kW condensing boilers system

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Abstract. Implementation of the Energy Related Products directive in September 2015 had focused attention on the condensing boilers more than ever before since they are the unique boiler technology complying with the requirements of this new regulation. The main benefit of condensing boilers comes from the latent heat recovery by condensing the water vapors from flue gas; the higher the condensation fraction, the higher the energy saving is. The experimental study refers on the heat gained and fuel cost saving by condensing water vapors from flue gas in the case of a heating system with two condensing boilers having nominal outputs of 25 kW and 35 kW, respectively. The maximum condensation fraction of the heating system, determined in real operating conditions, was 0.52, which corresponds to an energy saving rate of roughly 2 MJ/m³ of fuel. The cost of the energy saved, converted into fuel saving cost, is 22.28 Euro/1000 m³ for the natural gas price of 426 Euro/1000 m³ (39.9 Euro/MWh or 0.01108 Euro/MJ) in Iași, Romania, February 2020.

1. Introduction
Condensing boilers capitalize attention for more than twenty years due to the significantly higher performance and lower emissions compared to the traditional (non-condensing) boilers. Thus, annual fuel savings of 17.5 % can be usually achieved by replacing traditional boilers with condensing boilers of equal power [1] while the environmental impact is 23 % lower [2].

The implementation of the Energy Related Products directive in 2015 has considerably increased the interest for these boilers and for condensing technology in general. Most of the current commercial condensing boilers are operating with gaseous fuel – natural gas or LPG (liquid petroleum gas) in some cases. The economic and environmental benefits of latent heat harnessing by condensing the water vapors of the flue gas motivated the interest for condensing technology extrapolation of to a wider category of fuels. Thus, a theoretical analysis of the condensing technology potential for several gaseous, liquid and solid fuels was conducted in [3] and shown that significant benefits could be obtained in the case of biofuels, from both economic and environmental perspectives. An experimental study on performance of a condensing boiler fired by wood pellets was performed in [4]. This study indicates a maximum efficiency in terms of lower heating value of 99.4 % corresponding to the maximum condensation fraction of 33.7 %. An ingenious solution for conversion of small traditional wood fired boiler into condensing boilers was analyzed in [5]. Solution consists of an additional heat exchanger
with electrical charging, where not only the heat recovery by condensation of water vapor from flue gas takes place but also the capture and removal of more than 80% of the fine solid particles from flue gas could be performed.

A main concern regarding gas fired condensing boilers is the maximizing of the combustion efficiency and minimizing the emissions produced by combustion. The study in [6] reveals that thermal efficiency of a multi-hole burner (the mostly used in domestic and industrial boilers) is higher when inline configuration of holes is assumed but the surface heat flux is more uniform when staggered configuration is used. Formation of pollutants and emissions of such a burner was analyzed in [7], both theoretically and experimentally. It was shown that NO reduction by lowering the temperature of burned gas is a limited method since other three pathways for NO formation rely less on temperature and more on the concentration of the radicals in the flame front. The chemiluminescence analysis of a multi-hole cylindrical burner, performed in [8], shows that maximum surface temperature of this burner is achieved when equivalence ratio is 0.82. Experimental study in [9] indicates an optimum equivalence ratio in the range 0.7–0.75, which corresponds to NOx and CO concentrations in burned gases of 40 ppm and 30 ppm, respectively. These results are confirmed by the image processing method in [10], which revealed that optimum operation of the burner, defined by stable blue flame, was achieved for equivalence ratio of 0.7–0.73.

It is obvious that heat gained by condensing the water vapors from the flue gas is proportional to quantity of the condensate collected. Consequently, the gain can be improved by increasing the condensing rate. A possible solution, proposed in [11], is heat transfer enhancement using coatings for the heat exchange surfaces. An ingenious method was analyzed in [12] and consists in rising the dew-point temperature of the water vapors in flue gas by increasing the relative humidity of the combustion air with recirculated condensate; a maximum thermodynamic efficiency of 93.91% is claimed in this case. The study in [13] refers to a similar technical solution but, in this case, the condensate is introduced in the combustion air stream by spraying; it was shown that thermal efficiencies up to 95% could be achieved by this method. The condensation efficiency was analyzed even in the case of a coal fired power plant, in [14]. More specifically, it is a theoretical study referring to a rotary regenerative condensing flue gas-air heat exchanger, which indicates a condensation efficiency of up to 43% when matrix material is ceramic and air mass flow is three times greater than flue gas mass flow.

The current study also refers to heat gained due to the condensation of water vapors from the flue gas. The analysis was conducted on the heat generation system of the Thermal Machines Laboratory – “Gheorghe Asachi” Technical University of Iasi, which has an installed heating capacity of 60 kW. All the tests were performed in situ, in real weather conditions, which are characteristic to the period of the year when the study was developed. Thus, the results reflect the real behavior of the heat generation system and not the performance achieved by implementing a laboratory testing program.

2. Experimental installation and method
The heat generation system of the Thermal Machines Laboratory consists of two condensing boilers with nominal outputs of 25 kW and 35 kW, respectively. The heat generation system and the heating network are connected by means of a low loss header, as can be seen in figure 1. A picture of the system is presented in figure 2, where notations have the same meaning as in figure 1.

The two boilers have premix combustion system with multi-hole burner and spiral stainless steel primary heat exchanger. Flow and return water temperatures of boilers were measured with probes Twf1, Twf2 and Twr1, Twr2 while flue gas temperatures were measured with probes Tfg1 and Tfg2, according to figure 1; all the temperature probes are NTC type, Brahma ST07 model. Fuel gas consumption was measured with the gas consumption meter GCM, Aerotech BK-G6 MT model. Quantity of condensate was accounted during each test. In this regard, the condensate drain ducts were redirected from sewage system to a collector vessel and quantity of the collected condensate was measured by using glass graduated measuring cylinders MV.

The tests were performed from 13th to 21st of February 2020. There were analyzed eight operating regimes, each described by setting a certain value of the flow temperatures of the two boilers
\( t_{wf1} = t_{wf2} = t_{wf} \), from 80 °C to 45 °C, step 5 K. The measurements were performed in all cases at least one hour after the return temperatures of the boilers \( t_{wr1} \) and \( t_{wr2} \) as well as flue gas temperatures \( t_{fg1} \) and \( t_{fg2} \) were stabilized, which meant variations of less than 0.5 K during the entire previous hour.

**Figure 1.** Schematic of the analysed heat generation system.

**Figure 2.** Heat generation system of Thermal Machines laboratory.
Based on the quantity of the collected condensate, $V_c$, in cm$^3$, the specific condensate production was expressed as

$\chi = 10^3 \cdot V_c \cdot \tau_c^{-1} \cdot \dot{Q}_{hi}^{-1} \left[ \text{cm}^3 / \text{MJ} \right], \quad (1)$

where $\tau_c$ is time interval in which condensate was collected, in s, and $\dot{Q}_{hi}$ is overall gross heat input rate to the burners of the two boilers, given by

$\dot{Q}_{hi} = 10^3 \cdot V_g \cdot \tau_g^{-1} \cdot H_{hg} \left[ \text{kW} \right]. \quad (2)$

The following notations were used in equation (2):

$V_g$ - fuel gas consumption, in m$^3$;
$\tau_g$ - duration of measurement for gas consumption, in s;
$H_{hg}$ - higher heating value of the fuel gas, in MJ/m$^3$; it were assumed the public data from the official website of the fuel gas supplier, Delgaz Grid S.A.

Condensation fraction was calculated as

$fr = 3.6 \cdot 10^{-6} \cdot \chi \cdot \rho_c \cdot m_{H_2O}^{-1}, \quad (3)$

where $\rho_c = 1000$ kg/m$^3$ is the density of condensate while $m_{H_2O}$ is specific mass flow rate of water content in flue gas; it was assumed $m_{H_2O} = 0.159$ (kg/h)/kW for natural gas, according to [3].

The energy saving rate was defined by

$e_r = fr \cdot e_{sp} \left[ \text{MJ} / \text{m}^3 \right], \quad (4)$

where $e_{sp}$ is the energy saving potential of the fuel (natural gas); it was assumed $e_{sp} = 0.1 \cdot H_{hg}$, according to [1]. Based on the energy saving rate, the cost of the energy saved was expressed as

$CES = 10^3 \cdot P_{NG} \cdot e_r \left[ \text{Euro} / 1000 \text{ m}^3 \right], \quad (5)$

where $P_{NG}$ is the price of natural gas, in Euro/MJ. It was considered that $P_{NG} = 39.9$ Euro/MWh = 0.01108 Euro/MJ, which leads to 426 Euro/1000 m$^3$ on February 21, 2020, in Iași, Romania.

3. Results and interpretation

The measured quantity of the collected condensate at each of the eight testing regimes of the boilers, the fuel gas consumptions and the corresponding time intervals are given in table 1. Besides, the higher heating values of the fuel gas, made available for each day by the fuel gas supplier, are also indicated.

| Date       | $H_{hg}$ [kWh/m$^3$] | $H_{hg}$ [MJ/m$^3$] | $t_{wf}$ [°C] | $V_c$ [ml] | $\tau_c$ [s] | $V_g$ [m$^3$] | $\tau_g$ [s] |
|------------|----------------------|----------------------|--------------|------------|-------------|-------------|-------------|
| 13.02.2020 | 10.647               | 38.329               | 80           | 133        | 3699        | 3.830       | 3757        |
| 14.02.2020 | 10.636               | 38.290               | 75           | 258        | 4064        | 3.635       | 3832        |
| 14.02.2020 | 10.636               | 38.290               | 70           | 430        | 6768        | 5.283       | 6442        |
| 15.02.2020 | 10.693               | 38.495               | 65           | 564        | 7695        | 6.023       | 8867        |
| 17.02.2020 | 10.689               | 38.480               | 60           | 417        | 3308        | 2.680       | 4165        |
| 18.02.2020 | 10.689               | 38.480               | 55           | 1216       | 5635        | 2.887       | 5506        |
| 19.02.2020 | 10.693               | 38.495               | 50           | 2052.5     | 8006        | 3.247       | 7950        |
| 21.02.2020 | 10.679               | 38.444               | 45           | 2155.5     | 6844        | 2.422       | 6832        |
The influence of the flow water temperature over the return water temperatures and flue gas temperatures are shown in figure 3. As effect of the low loss header, \( t_{\text{wr1}} \) and \( t_{\text{wr2}} \) are quasi-equal at each operating regime. They increase with \( t_{\text{fw}} \) from 40.2 °C to 66.7 °C. In all cases, \( t_{\text{fg1}} > t_{\text{fg2}} \), which indicates that flue gas loss is higher in the case of boiler 1. Thus, \( t_{\text{fg1}} \) varies from 47.4 °C to 76 °C while \( t_{\text{fg2}} \) varies from 45.3 °C to 72.1 °C. It should be mentioned that condensate was collected from both boilers in each operating regime, which means that both \( t_{\text{fg1}} \) and \( t_{\text{fg2}} \) were lower than dew point temperature in all cases.

![Figure 3](image-url)

**Figure 3.** Variation of the return water temperatures and flue gas temperatures with flow water temperature.

![Figure 4](image-url)

**Figure 4.** Variation of the specific condensate production, \( \chi \), and condensation fraction, \( fr \), with flow water temperature.
Figure 5. Variation of the energy saving potential of the fuel, energy saving rate and cost of the energy saved with flow water temperature.

The specific condensate production varied from 0.92 cm$^3$/MJ (for $t_{wf} = 80 \, ^\circ$C) to 23.11 cm$^3$/MJ (for $t_{wf} = 45 \, ^\circ$C). Accordingly, the condensation fraction varies from 0.02 to 0.52 (see figure 4). As shown in figure 5, the energy saving potential is quasi-constant while the energy saving rate by condensing water vapors from flue gas varies from 0.080 MJ/m$^3$ (for $t_{wf} = 80 \, ^\circ$C) to 2.011 MJ/m$^3$ (for $t_{wf} = 45 \, ^\circ$C). In terms of fuel cost, this means an economy of 0.88 Euro/1000 m$^3$ to 22.28 Euro/1000 m$^3$. Taking into account that average annual fuel consumption of the condensing heating system is 5500 m$^3$, it means that annual fuel cost economy is up to 122 Euro.

It should be noted that even in the most convenient case, described by $t_{wf} = 45 \, ^\circ$C, almost half of the energy saving potential of the fuel, namely 20.31 Euro/1000 m$^3$, is lost (the average value of $e_{sp}$ is 3.841 MJ/m$^3$). The maximum condensation fraction of 0.52 – achieved in this case, as mentioned above – highlights this issue. In order to increase the condensate production and, consequently, to get a higher condensation fraction, the flue gas temperature should be reduced even more, which means to reduce the return water temperatures. This is technical possible by extending the heating installation (the heat demand increases) or/and by reducing the water flow rate. The second solution is not convenient in this case since implies the reduction of the heat supplied by boilers; beside the risk of incompliance with the heat demand, reduction of the heat supply can lead to on/off operating cycles of boilers, which should be avoided for economic and environmental considerations.

4. Conclusions

In all eight real operating regimes of the 60 kW condensing heating system, experimented in situ in the study, described by water flow temperatures in the range 45…80 °C, both boilers were operated in the condensing regime, as confirmed by condensate draining.

The highest condensation fraction and specific condensate production were 0.52 and 23.11 cm$^3$/MJ, respectively. As expected, they were achieved at minimum water flow temperature (45 °C), when the water return temperatures as well as flue gas temperatures of both boilers were also minimum. Accordingly, the energy saving rate by condensing water vapors from flue gas and the cost of energy saved were maximum at this operating regime. Their values are 2.011 MJ/m$^3$ and 22.28 Euro/1000 m$^3$, respectively. In these conditions, the estimated annual fuel cost economy is up to 122 Euro.
The technical solution to improve the condensation fraction and, consequently, to get additional economic benefits is the extending of the heating installation.

5. References

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