THERMAL PERFORMANCE AND COMBUSTION HYGIENE OF A BRIQUETTE BURNING DOMESTIC BOILER

DESEMPENHO TÉRMICO E HIGIENE DE COMBUSTÃO DE UMA CALDEIRA DOMÉSTICA A BRIQUETES

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RESUMO

O presente trabalho incide no estudo do desempenho térmico e na higiene da combustão de uma caldeira doméstica a lenha, destinada ao aquecimento de água. Nos ensaios de queima utilizaram-se briquetes comerciais, briquetes produzidos a partir de resíduos de biomassa, como serrim de pinho e aparas de poda de videira, e ainda lenha de eucalipto. O rendimento térmico foi calculado pelo método direto, concluindo-se que a caldeira tem um funcionamento muito semelhante para os diversos tipos de combustível e independentemente do caudal mássico de alimentação. Os valores do rendimento térmico situaram-se na gama entre 60,12 e 77,27%. Foi também monitorizada a composição molar, na base seca, dos gases de combustão (O_2, CO_2, CO e NO_x), verificando-se maior emissão de CO no ensaio com briquetes de aparas de poda de videira, sendo o tipo de briquetes com maior teor de cinzas, e maior emissão de NO_x nos ensaios usando briquetes comerciais, sendo este tipo de briquetes o que apresenta maior teor de azoto na sua composição. O aumento do caudal mássico de combustível provocou a diminuição do excesso de ar que se refletiu no aumento da temperatura dos gases de exaustão e na diminuição da emissão de CO).

ABSTRACT

This work presents the study of the thermal performance and combustion hygiene of a domestic briquette burning hot water boiler. The fuels tested were commercial briquettes, briquettes made from pine sawdust and vine pruning waste and eucalyptus logs. The thermal efficiency was determined by the direct method and it was found that the boiler thermal performance was quite similar for the different types of fuel tested and did not depend on the fuel feeding rate. The boiler efficiency values were in the 60.12 to 77.27% range. The dry basis molar composition of the boiler exhaust gases (O_2, CO_2, CO and NO_x) was continuously monitored. In terms of the CO emissions, the worst situation was for the combustion of vine pruning waste briquettes, the fuel with the highest ash content, whereas as far as the NO_x emissions were concerned the worst situation was for the combustion of commercial briquettes, which were those with the highest nitrogen content. Increasing the fuel mass flow rate lead to a decrease of the furnace excess air, which reflected in an increase of the exhaust gases temperature and a reduction of CO emission.

1. INTRODUCTION

In 2017 the world biomass primary energy consumption was around 10%, being the biomass the fourth mostly use energy source, behind the petroleum, the coal and the natural gas (IEA, 2017). In the last few years, there has been a growing interest on the biomass use as an energy source, motivated by political, social and environmental factors. In countries like Portugal, that have no abundant fossil energy resources, biomass use might reduce the external energy dependence of such countries, besides creating new job opportunities through the exploitation of this energy source and simultaneously reducing the greenhouse gas emissions (Ferreira *et al*., 2013a; Ferreira *et al*., 2014).

The biomass was one of the first energy resources used by humankind and can be considered as a practical manifestation of the solar energy because its origin comes from the photosynthesis (Demirbas *et al*., 2009; Toklu, 2017). In fact during this process the solar energy is stored as chemical energy through the chlorophyll function and subsequently released during the thermochemical or biochemical biomass conversion processes (McKendry, 2002; Abbasi and Abbasi, 2010). The main properties affecting the biomass performance as a fuel and consequently affecting any choice of the adequate energetic conversion process are: the heating value, the moisture content, the density, the amount of volatile matter, the amount of fixed carbon, the ash content, the alkaline content and the cellulose ratio (Ferreira *et al*., 2013b).

The biomass in its natural state has many disadvantages as fuel, mainly for high handling, transportation and storage costs as well as due to some limitations during the burning process. There are however some biomass densification processes that improve the physical properties of the solid biomass, like the pelletization and briquetting, creating a fuel with more uniform size and composition, reducing storage and transportation costs (Ferreira *et al*., 2013b; Almeida *et al*., 2014). These processes also increase the volumetric heating value of the biomass (its *LHV* will be equal to five times the non densified biomass) leading to a more uniform combustion with lower particulate emission rates (Werther *et al.*, 2000; Silva *et al.*, 2015).

As far as the biomass conversion into useful energy is concerned, this can be carried out by means of physical-chemical processes, thermochemical processes or biochemical processes, being combustion the most common and direct process used. The biomass combustion is considered neutral in terms of CO_2 emissions as the amount released during it is the same amount being captured and stored during the plant growing period (Zhang *et al*., 2010). So the biomass combustion when managed on a sustainability basis through a careful forestation will not change the CO_2 atmospheric with time, in strong contrast with the fossil fuel combustion.

The main objective of the present study is to evaluate the environmental and energy efficiency of the combustion of a set of biofuels as specified ahead.

2. MATERIALS AND METHODS

2.1. Fuel characterization

The combustion experiments were carried out with commercial briquettes made by a Portuguese company, Briquetes Raro – Sociedade de Aproveitamento de Resíduos Lda., with briquettes made from pine sawdust and vine pruning waste at the Instituto Politécnico de Viseu (IPV) and also with eucalyptus wood logs, Figure 1. The pine sawdust came from the Viseu region, the vine pruning waste from the Viana do Castelo region and eucalyptus logs from the Beira Litoral region.

The three types of briquettes had similar and almost constant diameters and lengths, whereas the eucalyptus logs had to be cut to get the correct size. The commercial briquettes were rather heterogeneous, while those manufactured at IPV from pine sawdust and vine pruning waste were quite homogeneous.

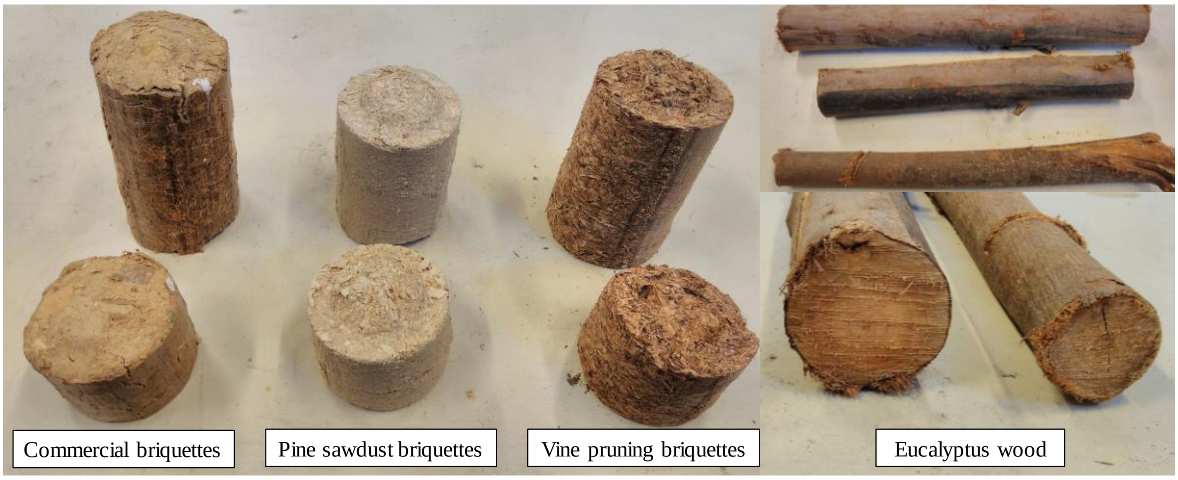

Fig. 1 - Used fuels.The biomass moisture content was obtained through samples from each biofuel and its mass loss was evaluated according to the FprEN 14774-2:2009. Previously weighted samples were fully dried inside a Binder laboratory oven, model FD-115, at 105 ºC.

The fuels proximate and immediate analysis as well as the corresponding lower heating values are in Table 1. For the commercial and vine pruning waste briquettes, the proximate and ultimate analysis were determined by Centro para a Valorização de Resíduos (CVR); for the pine sawdust briquettes, the proximate analysis was also determined by the CVR, whereas the ultimate analysis was from the literature (Núez Regueira *et al*., 2001); for the eucalyptus wood, the moisture content was determined in the laboratory oven according to FprEN 14774-2:2009 and the proximate and ultimate analysis from the literature (Núez Regueira *et al*., 2001).

2.2. Experimental setup

In Figure 2 a schematics of the experimental setup shows all the components used along the experiments. Two balances were used, one model EW6000-1M from Kern to weigh the briquettes to be tested and the other a Gram scale model Tortuga GL1500 underneath the boiler to evaluate the fuel mass flow rate. The water inlet and outlet temperature, T_1 and T_2, the exhaust gases and flame temperatures, T_3 and T_4 respectively, and also the boiler external surface temperatures T_5 to T_8, were all measured through K type thermocouples. The water mass flow rate was measured by turbine flow meter from Parker, model DFC.9000.100, and the mass flow rate of combustion gases was measured by means of a standard Pitot pipe. During the experiments the ambient pressure, temperature and absolute humidity were measured by means of a Vaisala meteorological plant, model PTU301.

For the analysis of the composition of the exhaust combustion gases, the following analyzers were used: a paramagnetic oxygen analyzer from Signal, model 8000M; an infra-red carbon dioxide analyzer from Signal, model 7000MF; a carbon monoxide infra-red analyzer from ADC, model MGA3000 and finally a chemiluminescence NO_x analyzer from Environnement, model MIR 9000CLD. Temperature, water mass flow rate and gaseous composition readings were automatically registered in the computer through the DASYLab software.

2.3. The boiler

The boiler used in the experiments was a hot water wood burning boiler from Solzaima, model SMZ IW, with a nominal power output of 24 kW. The combustion chamber is divided in two regions, an upper and a lower chamber. Figure 3 presents some pictures of the boiler and its combustion chambers. The boiler was periodically fed through a manual door during the combustion process. The extraction of the combustion gases is carried out in a way such that these gases will flow downward from the upper combustion chamber into the lower one, i. e. the boiler operates under the inverted flame principle. Thus, the combustion takes place in a fixed bed with the comburent flowing downwards (red arrows), Figure 3.

Table 1 – Characteristics of the tested fuels

	Commercial	Pine sawdust	Vine pruning	Eucalyptus
Moisture [% wb]	10.6	6.9	10.3	13.4
Volatile 900 °C [%]	72.8	85.4	69.5	-
Ash 550 °C [%]	1.2	0.7	3.2	1.0
Fixed carbon [%]	15.4	7.0	17.0	-
C [% db]	55.6	54.9	47.0	53.0
N [% db]	3.3	0.2	0.6	0.2
H [% db]	6.4	5.2	6.2	7.2
S [% db]	0.09	0.2	0.05	0.3
O [% db]	34.6	39.5	46.1	39.2
HHV [MJ/kg]	19.2	19.6	20.0	18.7
LHV [MJ/kg]	17.8	18.5	18.6	17.1

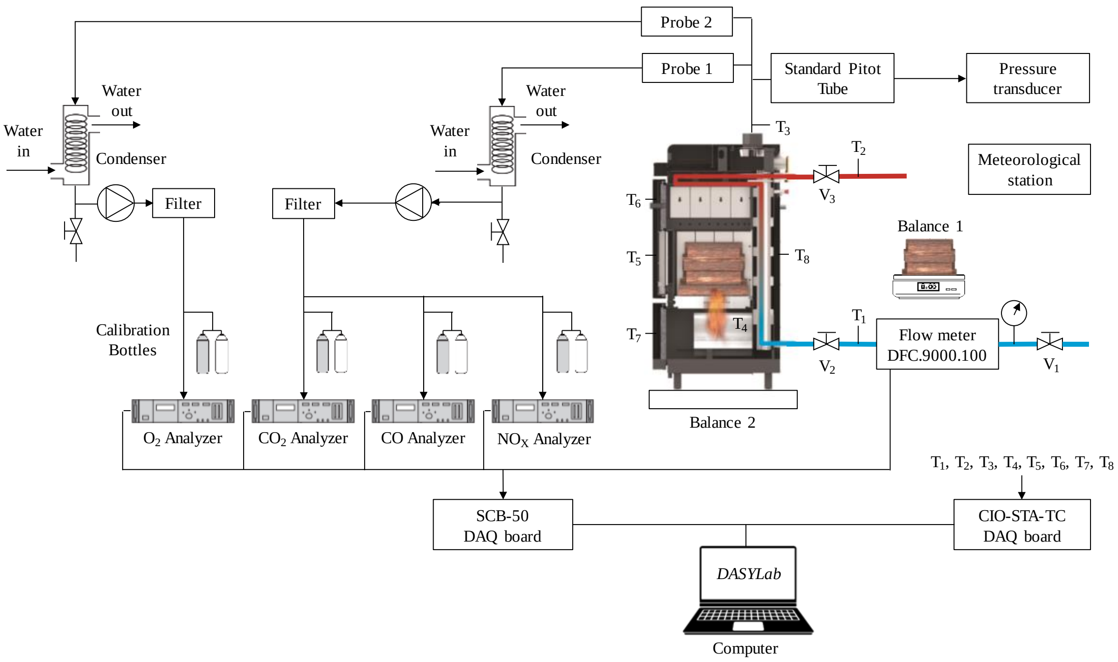

Fig. 2 – Schematics of the experimental setup.

Fig. 3 – Pictures of the boiler showing the upper and the lower combustion chamber.

2.4. Experimental procedure

During the combustion experiment, the first step is the boiler startup. The fuel was ignited and after reaching steady state operating conditions, the normal testing conditions are attained. The objective is to evaluate the influence of the different types of fuel, and of different fuel feeding rates, upon the boiler thermal performance and pollutants emissions. For the first set of experiments, where the influence of fuel characteristics were evaluated, similar operating conditions for the water mass flow rate and identical fuel batch sizes were supplied every 15 minutes. For the second set of experiments, where the fuel-feeding rate was changed, the water mass flow rate was also kept constant and identical to the first set of tests. In both situations, the boiler was kept under extreme operating conditions and so the water inlet was always at the water mains ambient temperature. Such conditions lead to a more stressing operating mode, thus penalizing the boiler thermal performance as well as the combustion hygiene.

3. CALCULATION PROCEDURE

For each experiment, the thermal efficiency of the boiler was calculated through the direct method and subsequently the corresponding energy losses were quantified. The average values of several parameters were measured, namely water inlet and outlet temperatures, the temperature of the boiler exhaust gas flow, the water mass flow rate, the dry basis molar fractions of oxygen, carbon monoxide, carbon dioxide, nitrogen oxides, ambient temperature, pressure and absolute humidity, static pressure in the chimney, and the dynamic pressure in several points along the boiler exhaust pipe.

In the overall chemical equation for the real combustion presented below, the nitrogen oxide emissions are not considered as they are not relevant in energy terms, only from the point of view of combustion hygiene. In this equation, subscript *b* refers to the briquette composition, subscript *a* concerns the atmospheric air components and finally subscript *c* refers to the components formed during combustion,

$$\begin{aligned} &n_{R1}(\mathrm{C})_b + n_{R2}(\mathrm{H_2})_b + n_{R3}(\mathrm{O_2})_b + \\ &n_{R4}(\mathrm{N_2})_b + n_{R5}(\mathrm{S_2})_b + n_{R6}(\mathrm{H_2O})_b + \\ &n_{R7}(\mathrm{H_2O})_a + n_{R0}(1+e)(\mathrm{O_2} + 3{,}76\mathrm{N_2})_a \rightarrow \\ &n_{P1}(\mathrm{CO_2})_c + n_{P2}(\mathrm{H_2O})_b + n_{P3}(\mathrm{H_2O})_c + \\ &n_{P4}(\mathrm{H_2O})_a + n_{P5}(\mathrm{N_2})_b + n_{P6}(\mathrm{N_2})_{ar} + \\ &n_{P7}(\mathrm{SO_2})_c + n_{P8}(\mathrm{O_2})_a + n_{P9}(\mathrm{CO})_c \end{aligned} \quad (1)$$

being e the excess air fraction, n_{Ri} the coefficients of the reactants and n_{Pi} the coefficients of the products.

The coefficients for the briquettes components n_{R1} to n_{R6} were calculated by means of the wet base chemical composition of the briquettes, whereas n_{R7} depends upon the atmospheric air composition. The coefficients n_{P1} to n_{P7} were calculated through the mass conservation of the chemical species, namely carbon, hydrogen, oxygen, nitrogen and sulfur, and finally the coefficients n_{P8} and n_{P9} were calculated through the combustion gas composition measured by the gas analyzers.

The fuel mass flow rate was calculated with the fuel mass feeding weight determined in balance 2 and the corresponding time interval. The combustion gases mass flow rate was determined by two different methods, through the Standard Pitot Tube placed at the boiler exhaust pipe, and also through the coefficients of the real combustion equation, Eq. (2),

$$\dot{m}_{geq} = \left(\sum n_{Pi} M_i\right)\dot{m}_{fuel} \quad (2)$$

where $\dot{m}_{geq}$ is the mass flow rate of the combustion gases calculated from the coefficients of the real combustion equation, M_i is the molecular mass of the component i and $\dot{m}_{fuel}$ is the fuel mass flow rate.

The boiler thermal efficiency η_t, determined by the direct method was calculated by means of,

$$\eta_t = \frac{\dot{Q}_{output}}{\dot{Q}_{input}} = \frac{\dot{m}_{\mathrm{H_2O}}\tilde{c}_{\mathrm{H_2O}}(T_{out} - T_{in})}{\dot{m}_{fuel} LHV_{fuel}} \quad (3)$$

where $\dot{Q}_{input}$ is the thermal energy released by the briquettes combustion, $\dot{Q}_{output}$ is the useful thermal power transferred to the water flowing through the boiler, LHV_{fuel} is the lower heating fuel of the briquettes, $\dot{m}_{\mathrm{H_2O}}$ is the water mass flor rate, $\tilde{c}_{\mathrm{H_2O}}$ is the average specific heat of the water, T_{out} is the water outlet temperature and T_{in} is the water inlet temperature.

The boiler thermal losses were divided, according their origin, into: sensible and latent thermal losses in the exhaust gas flow, unburned losses either due to incomplete combustion products in the exhaust gas flow or due to unburned fuel in the ashes, and finally radiation, convection and conduction losses through the boiler walls.

The thermal losses in the combustion exhausts gases $\dot{Q}_g$, were calculated through,

$$\begin{aligned}\dot{Q}_g &= \dot{Q}_{g,s} + \dot{Q}_{g,l} = \\ &= \left\{\tilde{c}_{P_g}\dot{m}_g(T_g - T_{amb})\right\}_s + \left\{\alpha\dot{m}_{fuel}h_{lv}\right\}_l = \\ &= \{[(n_P M\tilde{c}_P)_{\mathrm{H_2O}} + (n_P M\tilde{c}_P)_{\mathrm{CO_2}} + (n_P M\tilde{c}_P)_{\mathrm{O_2}} + (n_P M\tilde{c}_P)_{\mathrm{CO}} + (n_P M\tilde{c}_P)_{\mathrm{N_2}} + (n_P M\tilde{c}_P)_{\mathrm{SO_2}}]\dot{m}_{fuel}(T_g - T_{amb})\}_s + \\ &\quad \left\{\alpha\dot{m}_{fuel}h_{lv}\right\}_l\end{aligned} \tag{4}$$

where $\dot{Q}_{g,s}$ and $\dot{Q}_{g,l}$ are the sensible and latent losses in the exhaust flow, respectively, $\tilde{c}_{P_g}$ is the average constant pressure specific heat of the exhaust gases for the T_g to T_{amb} temperature range, $n_P(i)$ is the number of kmol of the exhaust combustion gases per kg of fuel, $\tilde{c}_P(i)$ is the average constant pressure specific heat for the product component i in the T_g to T_{amb} temperature range, T_g is the exhaust temperature of the combustion products, T_{amb} is the ambient temperature, α is the amount of water in the fuel in kg/kg$_{\mathrm{fuel}}$ and h_{lv} is the latent heat of vaporization of water at 25 °C, 2442 kJ/kg. The latent heat losses came from the moisture in the fuel, while the ambient moisture is already in gaseous phase.

The thermal losses connected to the carbon monoxide incomplete combustion product $\dot{Q}_{\mathrm{CO}}$, were given by,

$$\dot{Q}_{\mathrm{CO}} = \dot{m}_{\mathrm{CO}}HV_{\mathrm{CO}} = (n_P M)_{\mathrm{CO}}\dot{m}_{fuel}HV_{\mathrm{CO}} \tag{5}$$

where $\dot{m}_{\mathrm{CO}}$ is the mass flow rate of CO in the combustion products obtained through the real combustion chemical equation and HV_{CO} is the CO heating value, 10102 kJ/kg (Van Loo and Koppejan, 2008).

The unburned fuel losses in the bottom ashes and the convection and radiation losses from the boiler towards the environment were not calculated in the present study, their total amount was determined by the closure of the boiler energy balance. So the thermal power transferred to the water flowing through the boiler plus all the thermal losses must be equal to the fuel power supplied to the boiler,

$$\dot{Q}_{input} = \dot{Q}_{output} + \dot{Q}_{gs} + \dot{Q}_{gl} + \dot{Q}_{CO} + \dot{Q}_{others} \tag{6}$$

where $\dot{Q}_{others}$ concerns the unaccounted losses like the unburned fuel in the ashes, and the thermal losses by conduction, convection and radiation.

4. RESULTS AND DISCUSSION

In Table 2 are indicated the boiler operating conditions, the combustion gases mass flow rate either obtained through the real combustion chemical equation $\dot{m}_{geq}$ or through the Standard Pitot Tube $\dot{m}_{gPitot}$. The experiments were carried out with commercial briquettes (tests C1 to C6), with briquettes made with pine dust (test P1), briquettes made with vine pruning's (test V1) and eucalyptus wood (tests E1 and E2). In the same table is the relative error obtained when the combustion gases mass flow rate is determined by means of the Standard Pitot Tube, which reached around 10 % for some experiments. These relatively high errors can be explained by the bad location of the Standard Pitot Tube, in a region of the exhaust pipe where the gas flow was not completely developed. In fact, a straight pipe development should be free of any obstacles for 20 diameters upstream and 5 diameters downstream of the Standard Pitot Tube (Coelho, 2014). However, such rule could not be followed, due to physical space constraints in the experimental setup.

Table 2 – Important mass flow rates

Test	$\dot{m}_{fuel}$ [kg/h]	$\dot{m}_{H_2O}$ [kg/h]	$\dot{m}_{geq}$ [kg/h]	$\dot{m}_{gPitot}$ [kg/h]	$error_{Pitot}$ [%]
C1	3.65	9.30	116.43	108.17	7.10
C2	4.24	9.30	117.21	106.62	9.04
C3	4.32	9.04	109.77	105.53	3.86
C4	4.60	9.41	99.82	104.65	4.84
C5	4.69	9.32	113.38	110.09	2.91
C6	4.84	9.15	108.68	106.63	1.89
P1	4.90	9.08	101.39	101.53	0.14
V1	4.68	9.16	93.82	103.42	10.23
E1	4.38	9.13	112.74	109.57	2.81
E2	4.44	9.25	114.34	103.66	9.34

Figures 4 and 5 present respectively the evolution of the exhaust gases temperature and the boiler efficiency, during a 60 minutes period. The boiler was manually fed and consequently its operating regime was affected every time the furnace door was opened for fuel admission.

The oscillation of the exhaust gases temperature is very clear, Fig. 4, the same happening with the exhaust gas composition and consequently with the boiler efficiency, Fig. 5. It must be stressed that, the definition of steady state boiler operating conditions has, in the present study, a rather broad definition. In other words, it can be assumed that along a given experiment, that the boiler operates in roughly a constant operating condition, with some oscillations due to the fuel feeding process.

Figure 6 presents the average thermal efficiency of the boiler, its maximum and minimum deviation, in terms of the feeding fuel mass flow rate. There is no clear connection between the fuel mass flow rate

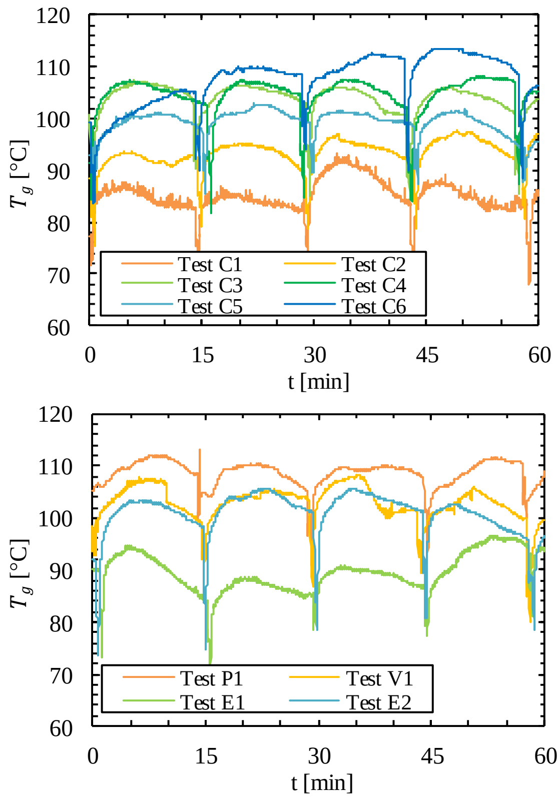

Fig. 4 - Time evolution of the exhaust gas temperature.

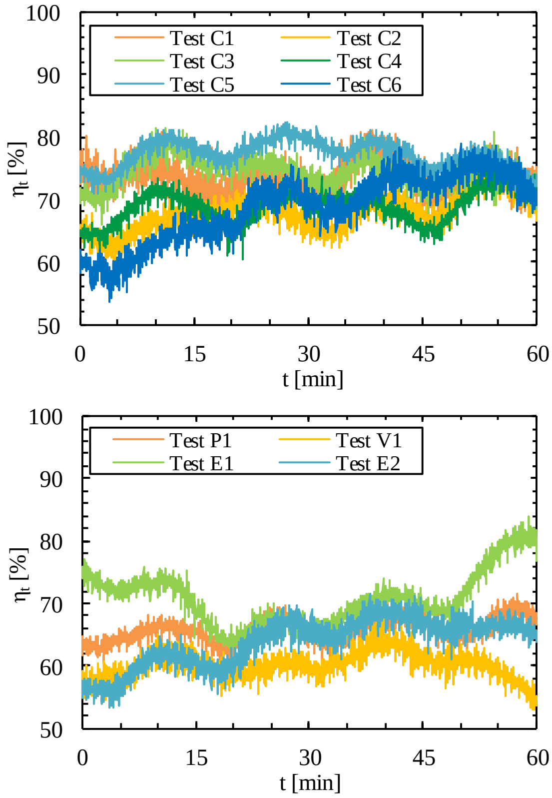

Fig. 5 – Time evolution of the boiler efficiency.

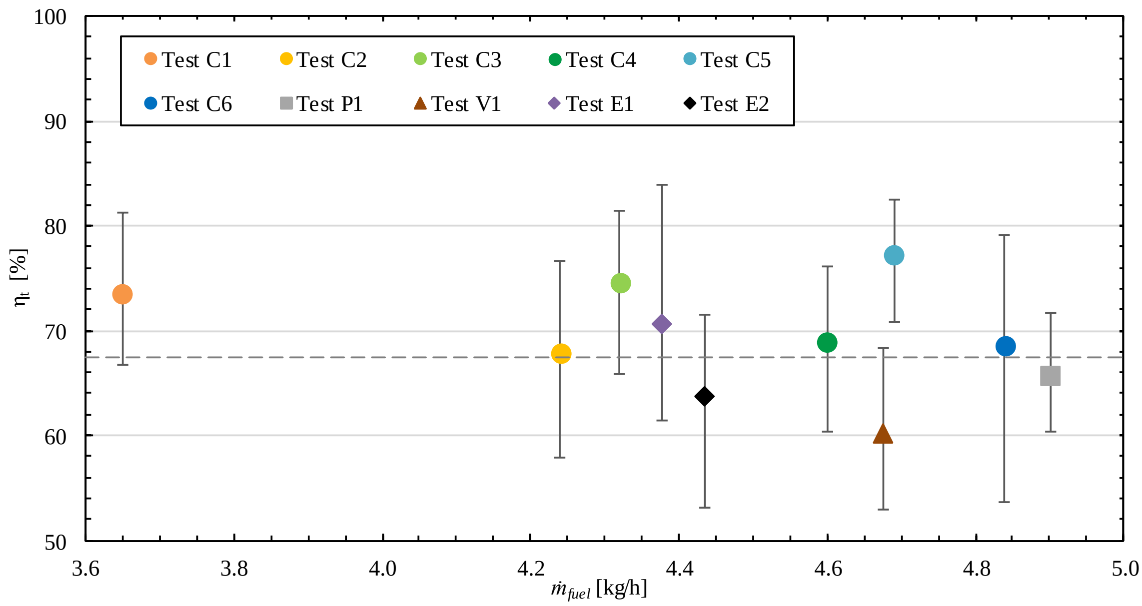

Fig. 6 – Evolution of the boiler thermal efficiency with the feeding mass flow rate.

and the boiler efficiency and the boiler operates at an approximately steady state condition independently of the fuel-feeding rate.

The European Standard EN 12809:2015 for solid fuel domestic independent boilers with nominal thermal output below 50 kW imposes limits on the boiler's efficiency. For a 24 kW nominal thermal output, the minimum established efficiency is around 67.5% and this limit is shown as a dashed line in Figure 6. So according to this limit, the experiments P1, V1 and E2 do not fulfill its requirements. However, it must be stressed once more, that in the present set of experiments the imposed operating conditions, with a low water inlet temperature, were very demanding, thus imposing tougher operating conditions for the boiler.

In Fig. 7 the boiler thermal efficiency and the allocation of losses are shown for all tested situations. The thermal losses in the boiler exhaust gases $\dot{Q}_g$, are in the range of 11 to 15% of the fuel input. These losses do not change with the fuel feeding rate. On the other hand, the unburned losses, quantified through the CO in the exhaust combustion gases $\dot{Q}_{\mathrm{CO}}$, have a maximum of 2.72% for the C1 test, obtained with the lowest fuel flow rate considered.

Figure 8 presents results for the average temperature of the exhaust gases and the amount of excess air supplied to the boiler, for the experiments carried out with commercial

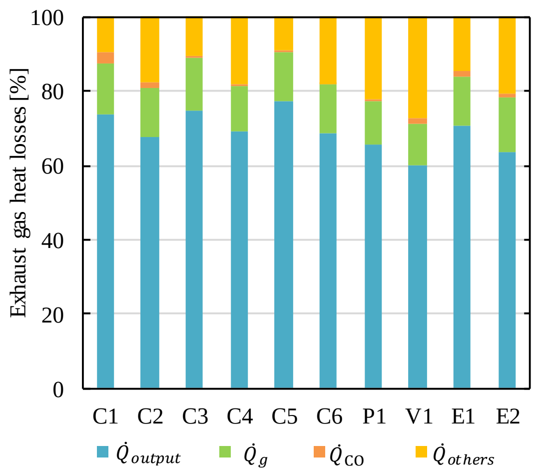

Fig. 7 – Allocation of useful and thermal losses.

briquettes. To have a better view of the importance of the fuel feeding rate on the time evolution of these two parameters, the tests were divided into three groups: a low feeding rate experiment (C1), medium feeding rate experiments (C2 and C3) and high feeding rate experiments (C4, C5 and C6). Full data points refer to experimental results while the empty points refer to average values.

As seen in Fig. 8, the increase of the fuel feeding rate leads to a reduction of the excess air, diminishing the oxygen concentration in the exhaust flow and increasing its temperature. The increase of the fuel feed rate was not followed by a correspondent increase in the airflow, as the boiler has no automatic adjustment procedure for acting upon the inlet oxidant flow.

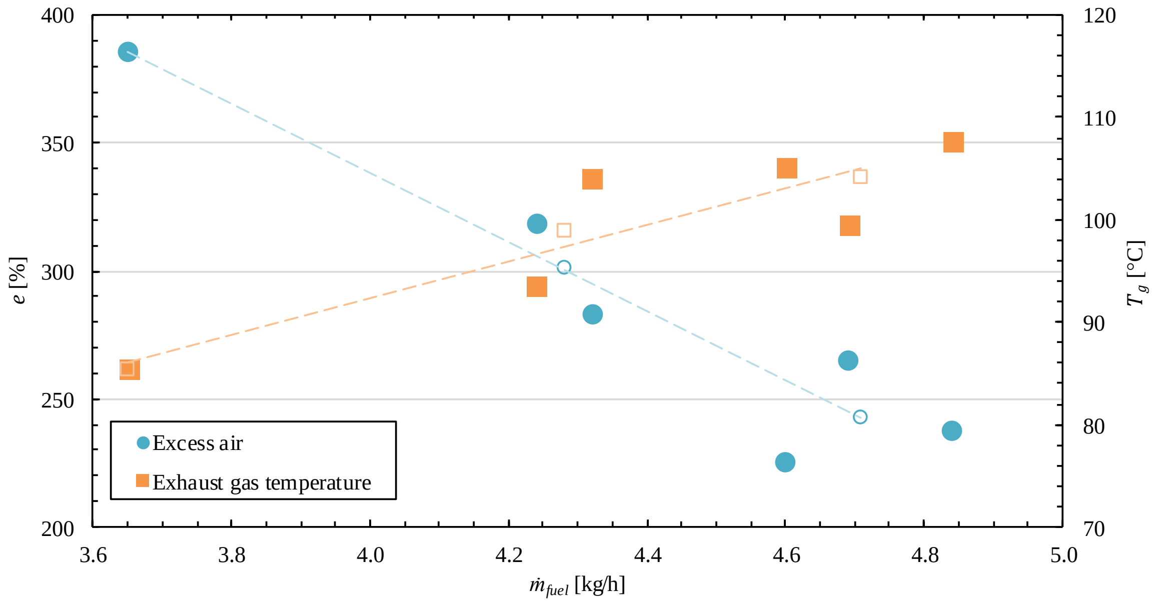

Fig. 8 – Evolution of the excess air and the exhaust gases temperature with the fuel feed rate.

To compare the emission values according to the EN 14785:2008 European Standard, for wood burning domestic ambient heaters, the reference oxygen molar fraction in the exhaust gases is 13%. All the emission data were then accordingly corrected. As far as the CO_2 emissions are concerned, they are similar in all tested situations.

Figure 9 presents the influence of the type of fuel upon the CO emissions. In the same figure, the fuel ash mass fractions are also plotted.

According to the EN 14785:2008 European Standard, the CO limit for the emissions is 600 ppm (13% O_2) and this limit is shown in Fig. 9 by the dashed grey line. For high load operation the commercial and pine sawdust briquettes have emissions below this limit, 397 and 546 ppm, respectively. The other two fuels, vine pruning and eucalyptus do not fulfil the standard requirements, and the CO emissions for the vine pruning can reach up to four times the limit. Khan *et al.* (2009) say that higher CO emissions can be connected with large particles sizes and high ash contents of the solid fuel. Looking at Table 1 it is clear that the fuel with highest ash content is precisely the vine pruning briquettes, thus justifying the obtained high CO emissions.

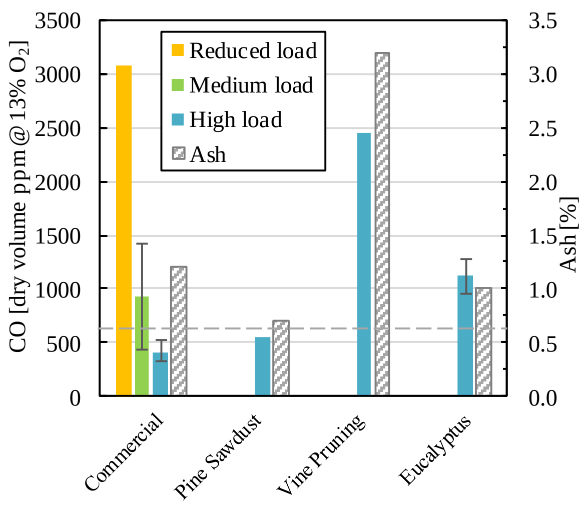

Fig. 9 - Corrected CO emissions and fuel ash mass fractions.

On the other end, the CO formation depends on several other factors like, limited residence time inside the furnace, bad air fuel mixture, inappropriate excess air leading to a lower combustion temperature (Johansson *et al.*, 2004; Roy *et al.*, 2013). For the experiments with commercial briquettes, there is a reduction of the CO emissions with the increase of fuel feed rate. This lowered the excess of combustion air leading to a higher combustion temperature, Figure 10.

Another point to take into account is the low inlet water temperature promoting lower temperatures at the external boundary layer of the heat transfer membrane walls of the boiler.

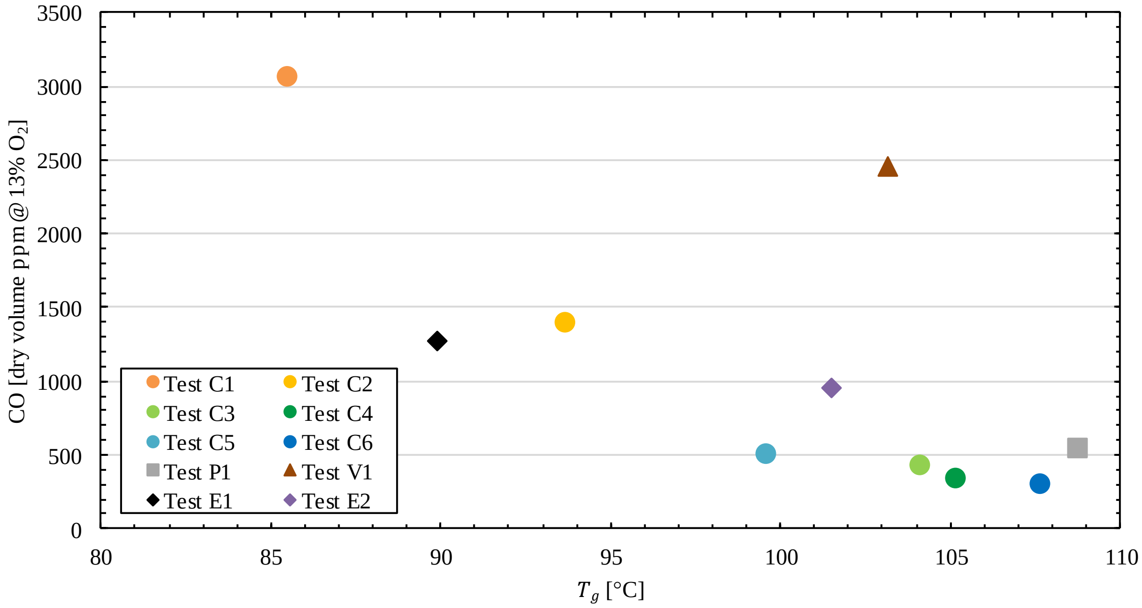

Fig. 10 - CO emissions as a function of the exhaust gas temperature.

Such will have a stronger quenching effect on the gas phase reactions taking place close to these walls. In fact, for solid fuel combustion of large particles (well above 1 mm) the oxygen reacts at the surface of the particles to oxidize carbon to CO while CO burns away from the solid particles (Pinho, 2011). As the CO oxidation to CO_2 can be subjected to quenching effects near cooler heat transfer walls of the boiler furnace (Warnatz *et al.*, 2001), this effect is strongly affected by the inlet water temperature.

Figure 11 shows the influence of fuel type on the NO_x emissions and also the nitrogen content of each fuel in the hatched bars.

The NO formation depends on three mechanisms, namely, the thermal-NO, the fuel-NO and the prompt-NO. The thermal-NO mechanism requires furnace temperatures above 1300 ºC, value not found in domestic boilers and so this mechanism does not contribute for the NO emissions in the present situation (Pinho, 2011; Roy *et al.*, 2013). From the experimental data it is clear that the commercial briquettes tests are those with the highest NO emissions, while the eucalyptus tests gave the lowest emissions. In fact the commercial briquettes are the fuel with the highest nitrogen content (3.30%, w/w) while the eucalyptus is the fuel with the lowest nitrogen content (0.2%, w/w) and consequently it can be concluded that in the present situation the NO formation mechanism is the fuel-NO. The prompt-NO is typical of rich air fuel mixtures, found for example in the spark ignition internal combustion engines, a very different universe from the present one and thus can be easily discarded.

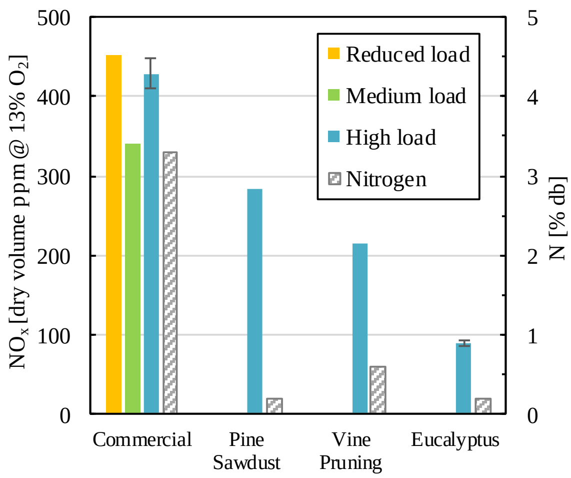

Fig. 11 – NOx emissions.

5. CONCLUSIONS

For the steady state operation, the boiler presented an approximately constant activity independent upon the fuel mass flow rate. The highest thermal efficiency, 77.27% was for commercial briquettes (C5 experiment) and the lowest, 60.12%, was for briquettes made from vine pruning waste (V1 experiment).

An increase on the briquettes mass flow

rate has led to a reduction on the amount of free oxygen in the exhaust combustion gases, in other words, a reduction of the excess of the combustion air, as the boiler has no automatic control device, acting upon the inlet combustion air.

As far as the CO emissions are concerned, a reduction of the fuel mass flow rate has led to an increase of the combustion excess air, thus lowering the combustion temperature and consequently promoting the increase of CO emissions. The type of fuel has also a strong influence on the CO emissions. For high load boiler operation, the commercial briquettes had lower CO emissions, while the vine pruning briquettes presented the highest emissions.

From the analysis of the NO emissions the conclusion is that, the main responsible is the fuel-NO. The commercial briquettes, the fuel with the highest nitrogen content, presented the highest NO emissions, while the eucalyptus log, the fuel with the lowest nitrogen content, presented the lowest NO emissions.

ACKNOWLEDGMENTS

The present work was supported by the INTERREG Project N. PR321705 - BIOMASA_AP and carried out in the Combustion Laboratory of INEGI.

One of the authors, Mariana Vale da Silva, also acknowledges the award “Prémio Professor J F Silva Gomes/FREZITE-2017” for the best experimental work master dissertation, presented at the Department of Mechanical Engineering, Faculty of Engineering of the University of Porto.

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