

Thermal Behavior of a Pressurized Water Heater

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Abstract. *Some experimental and numeric studies around a prototype of a gas burning domestic water heater using a premixed combustion in a perforated ceramic burner are presented. The numerical approach was adopted as a simple methodology to define adequate steps to take during the prototype improvement in order to better its thermal and environmental performance. In the mathematical model it was assumed that the combustion reactions could be represented by a pseudo-single step overall reaction mechanism whereas, as far the heat transfer was concerned, the furnace was considered a well stirred reactor with participating grey combustion gases. Results of the thermal performance and pollutants emissions of the prototype are compared those obtained for some heaters found in the market.*

Keywords. *water heaters, pollutants emission, thermal behavior*

1. Introduction

Gas burning water heaters are used for the production of domestic hot water. The great majority of commercially available models use a combination of atmospheric partially premixed and diffusion burners, Torregiani (1992). The gaseous combustible injected through a calibrated orifice into a tuyere drags primary air and the resulting gaseous mixture is sent to the burner mouth where secondary air is added. Working range of such heaters is adequate to the majority of applications but studies have been carried out to look for new appliances capable to follow future and more restricted standards.

Traditional gas burning water heaters present high levels of NO_x emissions (Marques and Pinho, 1998) which puts these appliances under a low grade classification in the European standards for such type of equipments (EN26, 1997; EN 297:1994/prA3:1996/prA5:1997, 1997). Water heater manufacturers were thus faced with the urgency to develop new products to respect future tighter emission limits. To achieve such target the simplest approach is to go towards pressurized premixed gas burners using either perforated or porous ceramic burner plates (Barnes et al., 1988; Williams et al., 1992; Wood, 1994). Porous ceramic burners present the best performance, wider thermal range and lower pollutants emissions, but are very expensive and in some situations their construction can involve high pollution. A cheaper approach is to use perforated ceramic plates instead, and although such burner plates have lower thermal and environmental performances, when compared with the porous ones, they can still fulfill the objectives.

With premixed and pressurized gas burners poor reactant mixtures are burned and combustion regimes are away from those adequate for the prompt or Fennimore NO mechanism. On the other end, to avoid NO formation through the thermal or Zeldovich mechanism, the excess air was adjusted so that the prototype could only work in the 20 to 30 % range, (Borman and Ragland, 1998; Huppa and Kilpinen, 1995; Miller and Bowman, 1989; Schuler et al., 1997; Turns, 1996; Wood, 1994). The formation of thermal NO and CO have contradictory requirements, but in the above mentioned range of excess air, a compromise is reached provided a convenient furnace height is used.

In the design of the prototype several basic ideas were followed:

- Minimization of the gaseous pollutants emissions;
- A wide thermal output range in spite of low elasticity of premixed burners;
- Reduction of combustion noise;
- Maximum heater compactness.

The solution that was obtained for the burner plates, furnace and heat exchanger dimensions, was the result of the compromise of sometimes contradictory requirements to achieve low CO and NO_x emissions, wide output range and compactness.

A smaller burner plate will lead to higher burning densities, higher combustion gas temperatures inside the furnace and heat transfer region, optimal conditions for thermal NO formation. With a shallower furnace flames may reach the exchanger fins and tubes, quenching the combustion and leading to CO and unburned hydrocarbons formation. A higher furnace results into cleaner combustion, but at the expense of compactness.

Although the present legislation imposes a CO maximum of 1000 ppm (for 3 % of O₂ in the products), it was adopted as reference, a value of 200 ppm of CO for all the fuels. As far as NO_x was concerned, the maximum accepted value was of 91 mg/kWh for butane, 89 mg/kWh for propane (and 70 mg/kWh for natural gas), according to the future european legislation for these equipments, (EN26, 1997; EN 297:1994/prA3:1996/prA5:1997, 1997).

With the adopted solution for the ceramic burner plates, furnace and heat exchanger, a solution was achieved as far as pollutants emissions were concerned. In practical terms a water heater has to work with all the gases from the same family, this means that a LPG family gas appliance must be able to handle with commercial butane and propane without further adjustments and so the settings are not the most convenient for each gas, but are again a compromise. Besides, there are composition changes on LPG between summer and winter periods, imposing restrictions on the emission performance.

In conclusion, a fully premixed and pressurized burner can handle with two counteracting actions, reduction on pollutants emissions, particularly NO_x, and compactness.

In this paper a rough description of the steps taken for the development of the prototype are presented. The work was a combination of measurements carried out according to the experimental procedures clearly outlined in the european legislation (EN26, 1997; EN 297:1994/prA3:1996/prA5:1997, 1997) and the information gathered through a simple numerical simulation of the main components of the prototype. A close description of the steps taken in the development of the prototype is impossible, due to deontological constraints imposed by the project financier.

2. Numerical simulation of the prototype

To reach a deep understanding of important parameters relevant to the prototype performance, there were two approaches: the use of extensive experimental measurements imposed by the dimensions of the prototype, or the using a reduced experimental lay-out supported by a simple numerical model. It was this second strategy that was chosen.

In the next sections, a short resume of the thermal behavior study that was elaborated is presented. The intention of using a simple mathematical model was not only the comprehension of what was occurring inside the reaction volume but also the development of a simple tool that could be used in the design and simulation of larger prototypes. Having always in mind the simplicity of approach, the mathematical model was divided into two sub-models for combustion and heat transfer.

A schematic representation of the water heater is shown on Fig. (1).

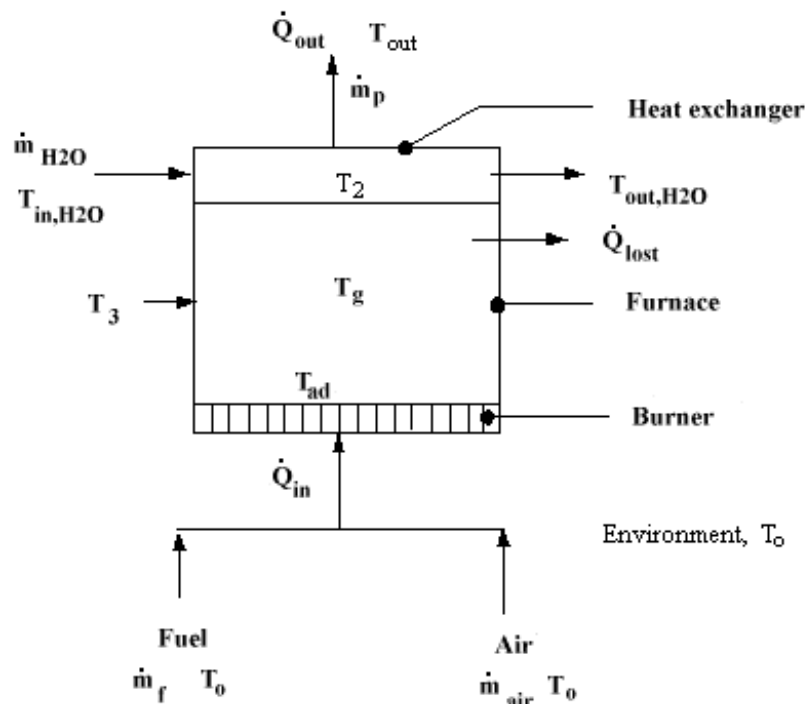


Figure 1. Schematic representation of the water heater.

2. Combustion sub-model and results

The combustion is a complex phenomena and difficult to simulate unless a simple approach is adopted, specially in a situation where the general behavior of a pre-industrial prototype had to be analyzed in short notice. For the development of the model four important points were accounted for:

- Fluid mechanics of the reaction zone;
- Heat transfer mechanisms;

- Reaction rates;
- Boundary conditions.

The last item was the least and so it will not be subject of any attention in the subsequent explanation.

The prototype was analysed under steady state conditions, according to the european standards (EN26, 1997; EN 297:1994/prA3:1996/prA5:1997, 1997). So, the ideal reactor type is the plug-flow reactor model for the combustion chamber. For the sake of simplicity combustion was assumed adiabatic and the heat transfer was assumed to take place immediately afterwards, as products rise in the interior of the furnace towards the heat exchanger. The next consideration was about reaction rates, the simplified single step kinetic mechanism proposed by Westbrook and Dryer (1981a, 1981b, 1984) was adopted. These researchers carried out several studies about simplified reaction mechanisms that could be easily coupled with complex thermal and fluid dynamic models, avoiding the need to account for dozens of intermediate reaction steps and allowing a simple and multidisciplinary analysis of such complex phenomena. The use of a simplified single step combustion mechanisms imposes some limitations on the model effectiveness and conclusions (Hautman et al., 1981), but this simplified approach was more than enough for engineering purposes, as suggested by Turns (1996).

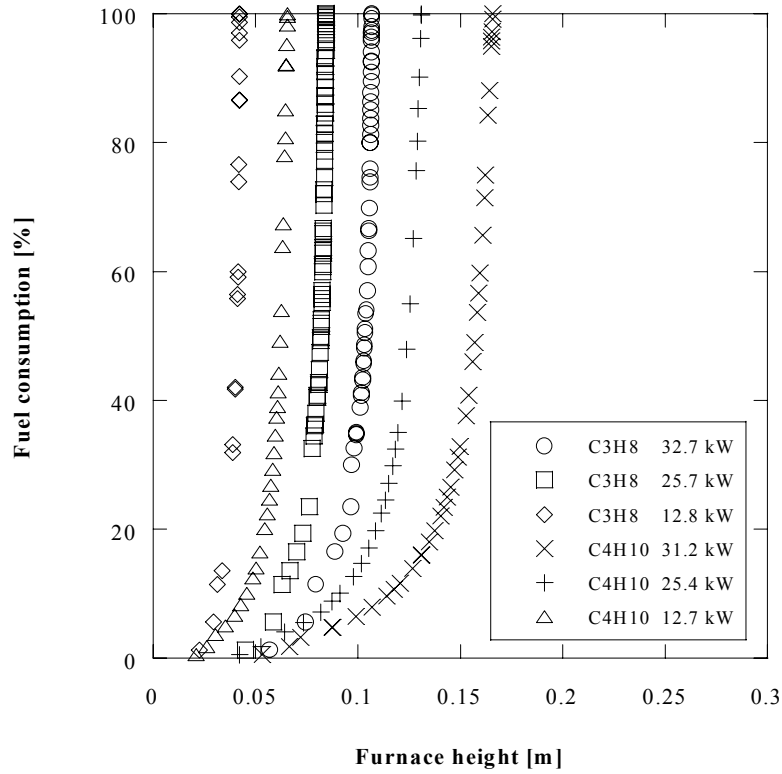


Figure 2. Importance of furnace height on combustion completion.

The corresponding reaction rate \dot{w}_F was given by an equation like,

$$\dot{w}_F = -\frac{d[C_xH_y]}{dt} = -A \exp\left(-\frac{E}{\bar{R}T}\right) [C_xH_y]^a [O_2]^b \quad (1)$$

where A is the pre-exponential factor and a and b are constants given by Westbrook and Dryer (1981a, 1981b, 1984), $[C_xH_y]$ and $[O_2]$ are the fuel and oxygen molar concentrations. \bar{R} is the universal gas constant, E is the activation energy and T is the temperature. This equation was coupled with thermal balance equations for the plug flow combustion region following the development proposed by Turns (1996). The mixture density ρ and temperature evolution in the reaction zone were given by,

$$\frac{d\rho}{dx} = \frac{\frac{\rho \bar{R}}{v_x c_{pmix} M_{mix}} \sum_{i=1}^N M_{mix} \dot{w}_i \left(h_i - \frac{M_{mix}}{M_i} c_{pmix} T \right)}{p \left(1 + \frac{v_x^2}{c_{pmix} T} \right) - \rho v_x^2} \quad (2)$$

$$\frac{dT}{dx} = \frac{v_x^2}{\rho c_{pmix}} \frac{d\rho}{dx} - \frac{1}{v_x \rho c_{pmix}} \sum_{i=1}^N h_i \dot{w}_i M_i \quad (3)$$

In the above equations M_{mix} is the molecular mass of the mixture, M_i is the molecular mass of component i , v_x is the gas phase velocity, c_{pmix} is the mixture specific heat at constant pressure and h_i is the enthalpy of gaseous component i .

The subsequent results refer only to propane and butane combustion where CO formation occurs caused by flame impingement on the heat transfer surfaces due to furnace height limitations. Of these two hydrocarbons, it is the butane which has tighter constraints.

The influence of combustion heat release rate on the fuel consumption height is seen in the numerical results shown on Fig. (2). These results refer to 30 % of excess air, as it was found experimentally and numerically that above this value the reduction of furnace height was meaningless. The experimental data for the CO formation were not obtained from experimental measurements inside the furnace, but by testing several furnace heights and measuring the CO in the exhaust gases. Furnace heights of 70 mm, 110 mm and 130 mm were tested. Above 110 mm of furnace height, performance increase in terms of reduction of CO emissions was not worth the extra furnace size. This can also be concluded from Fig. (2), for the thermal output range referred in the legend.

3. Heat transfer sub-model and results

After the composition and temperature of combustion products were known as a result of the combustion modeling, the output data were sent to the heat transfer modeling. The final target was the determination of the heat exchanger wall temperature T_2 , the average temperature of the gaseous products of combustion inside the furnace T_g , the furnace wall temperatures and the exhaust gases temperature T_{out} , Fig. (1).

The developed model was based on the following main assumptions:

- The furnace is a well stirred reactor (Hottel, 1960; Hottel and Sarofin, 1967; Siegel and Howell, 1982; Pilão and Pinho, 1998) and inside the furnace gases are at an uniform temperature T_g and are considered grey;
- Metallic surface of the heat exchanger, is grey and at an average equivalent temperature T_2 ;
- Metallic lateral surface of the furnace is also grey and at an equivalent average temperature T_3 . Such assumptions were also considered by Hottel and Sarofin (1976) and Siegel and Howell (1982);
- Convective heat transfer losses through lateral walls are neglected as radiation is the dominating mechanism (Hottel and Sarofin, 1976; Lobo and Evans, 1939).

The furnace confining walls were called respectively surface 1, 2 and 3 for the ceramic burner, the heat exchanger and the lateral walls. Combustion gases will reach an adiabatic flame temperature T_{ad} but due to radiative heat transfer among the burner surface (surface 1), the gas inside the furnace, the heat exchanger (surface 2) and the lateral walls (surface 3), the burner superficial equilibrium temperature T_1 is different from T_{ad} . The total irradiation and the total radiosity for a generic surface i are respectively (Incropera and de Witt, 1990)

$$E_i = \sigma \varepsilon_g T_g^4 + \sum_k \tau_{gki} J_k F_{ik} \quad (4)$$

$$J_i = \alpha \varepsilon_i T_i^4 + \rho_i E_i \quad (5)$$

where, τ_{gki} is the gas transmission coefficient for the radiative energy being exchanged between surfaces k and i , ρ_i is the reflection coefficient for the surface i , ε_i is the surface i emissivity and ε_g is the gas emissivity. For the gas phase only the CO_2 and the H_2O were considered main emitters and absorbers (Taylor and Foster, 1974; Siegel and Howard, 1982; Incropera and de Witt, 1990). Geometric considerations concerning the gaseous phase emissions followed Hottel and Sarofin (1976) analysis.

The energy balance of the heat exchanger gives the next equation,

$$-\frac{\varepsilon_2}{1-\varepsilon_2} A_2 \sigma T_2^4 + \frac{\varepsilon_2}{1-\varepsilon_2} A_2 J_2 + \frac{A_i U_i}{2} T_g + \frac{A_i U_i}{2} T_{out} = A_i U_i \bar{T}_{H_2O} + \dot{m}_{H_2O} c_w \sigma T \quad (6)$$

while the energy balance of the lateral walls of the furnace results in

$$\left(\frac{\varepsilon_3}{1-\varepsilon_3} A_3 \sigma + A_3 \varepsilon_3 \sigma \right) T_3^4 - \frac{\varepsilon_3}{1-\varepsilon_3} A_3 J_3 + h_3 A_3 T_3 = h_3 A_3 T_o + A_3 \sigma T_o^4 \quad (7)$$

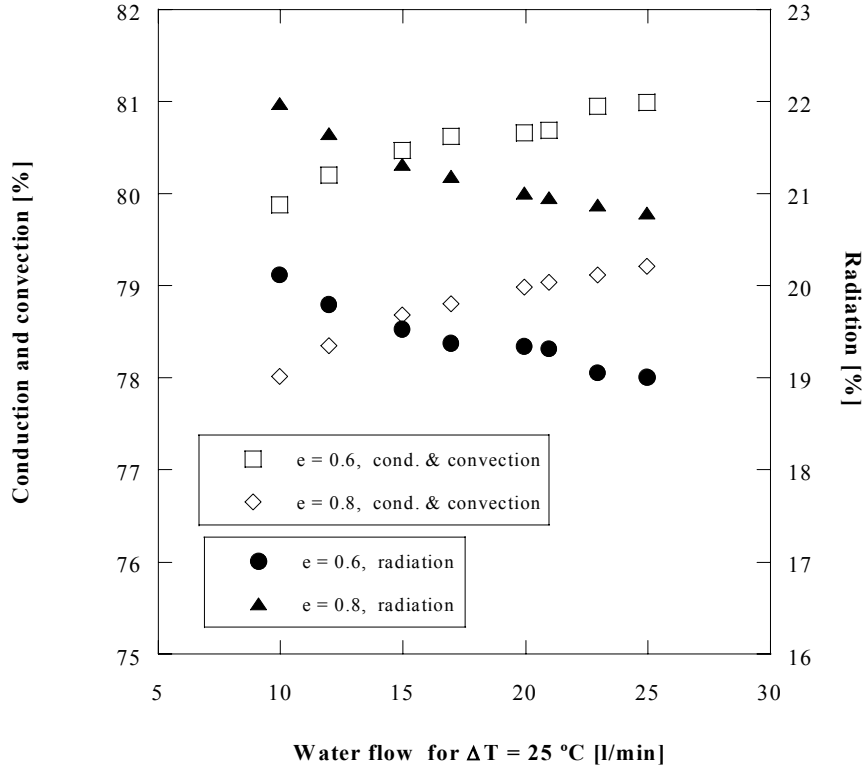


Figure 3. Relative importance of heat transfer mechanisms. Fuel, propane.

In the above equations A_i is the area of the generic surface i , σ is the Stefan-Boltzmann constant, U_i is the overall heat transfer coefficient in the heat exchanger, \bar{T}_{H_2O} is an average mean logarithmic temperature for the water and c_w is the water specific heat

The energy exchanged by the combustion products inside the furnace is given by two equations,

$$\sigma T_g^4 (\overline{GS}_1 + \overline{GS}_2 + \overline{GS}_3) - \overline{GS}_1 J_1 - \overline{GS}_2 J_2 - \overline{GS}_3 J_3 = 0 \quad (8)$$

$$h_3 A_3 (T_3 - T_o) + A_3 \sigma (\varepsilon_3 T_3^4 - T_o^4) + \dot{m}_{H_2O} c_w \Delta T = \dot{m}_p c_{pmist} (T_{ad} - T_{out}) \quad (9)$$

In Eq. (8) $\overline{GS}_{1,2,3}$ are the global area exchange factors between the gas and the surfaces while $J_{1,2,3}$ are the corresponding radiosities. In Eq.(9) ΔT is water temperature augmentation while flowing through the heat exchanger.

The energy radiated by the combustion products mass flow \dot{m}_p while its temperature falls from the adiabatic flame temperature T_{ad} towards the burner steady state superficial temperature T_1 is given by,

$$\frac{\varepsilon_1}{1 - \varepsilon_1} A_1 (\sigma T_1^4 - J_1) - \dot{m}_p c_{pmist} (T_{ad} - T_1) = 0 \quad (10)$$

Solving equations (4) to (10) it is possible to calculate temperatures and energy balances for all components of the prototype. Figure (3) is an example of the obtained results and shows the weight of the different mechanisms on the heat being transferred towards the water.

4. Comparison of the prototype with commercially available heaters

To assess the validity of design and construction options, a comparison was carried out among the 110 mm height furnace prototype and commercially available water heaters, either with atmospheric or pressurized burners. Reference to commercially available devices cannot be done, the tested equipments were then referred as Heater-A to Heater-D. Tests were carried out according to the already mentioned european standards.

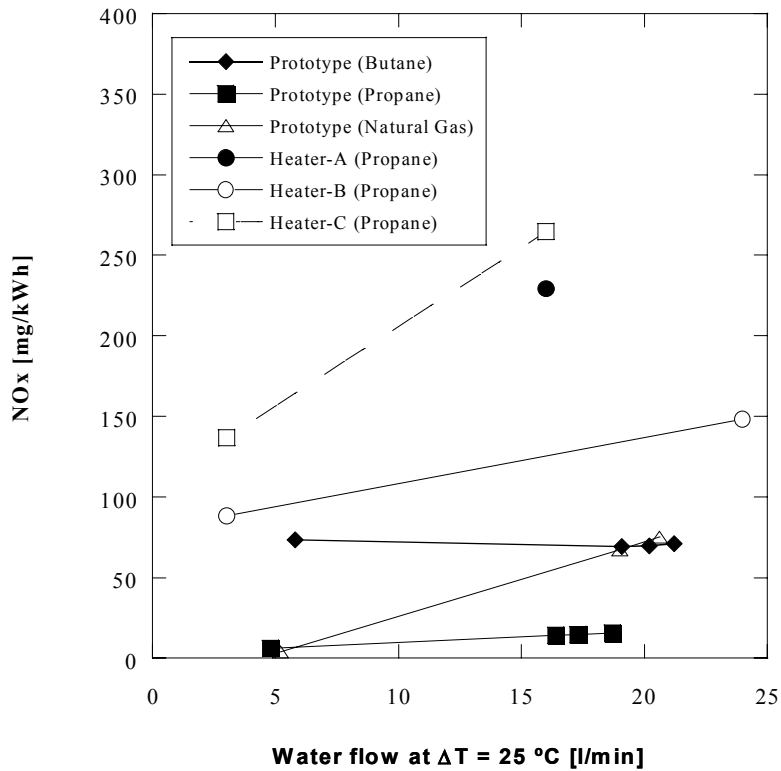


Figure 4. Comparative performance of the prototype and commercial appliances. NO_x emissions.

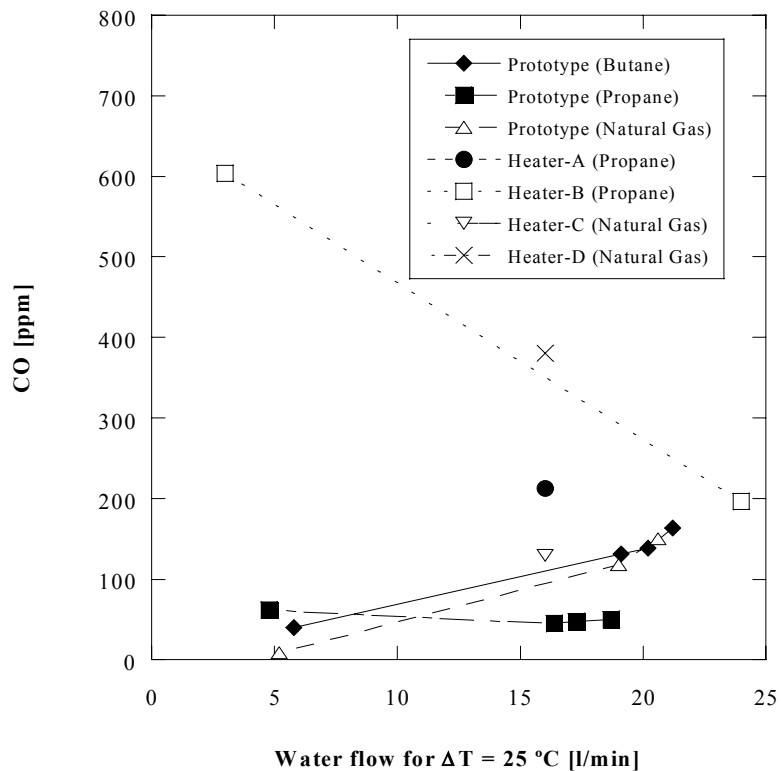


Figure 5. Comparative performance of the prototype and commercial appliances. CO emissions.

The two previous figures present a resume of the most important conclusions in terms of gaseous pollutants emissions. It must be stressed that data are from a comparison among a development prototype and commercially available appliances and so must be viewed with caution.

The next figure, Fig. (6), presents results obtained for the prototype burning propane with two different furnace heights. The emissions reduction, associated to the higher furnace, is a proof of the delicate balance required by the

compromise between compactness and low pollution. The search of adequate experimental results was carried out through adjustments in the configuration of main components of the prototype, like the burner and the furnace for example, and the choice of such adjustments was aided by the above referred numerical simulations.

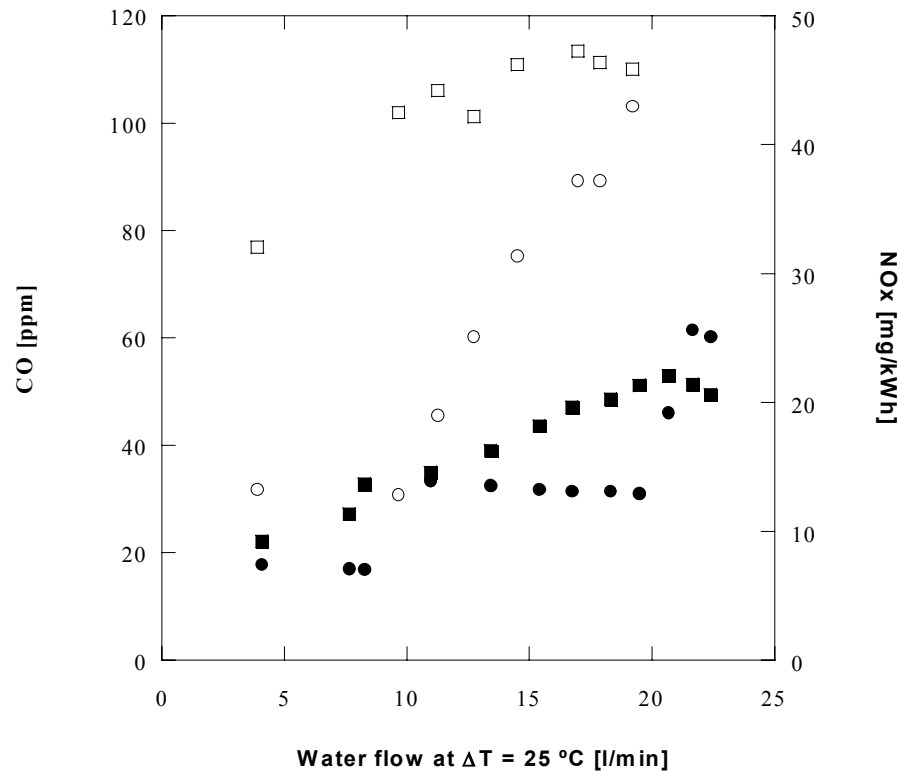


Figure 6. Evolution of CO and NO_x emissions with furnace height. CO, ○ in 70 mm furnace and ● in 110 mm furnace. NO_x, □ in 70 mm furnace and ■ in 110 mm furnace. Fuel, propane.

5. Conclusions

A brief description of the steps taken on the research and development of a prototype for pressurized burner water heater was presented.

The development of a commercial product has time constraints that impose simplified approaches to achieve reasonable answers with reduced cost and in short notice. A good way to reach such targets is through the combination of a trial and error experimental procedure combined with a simple mathematical modeling.

In a given moment of the prototype development process, the combination of compactness and low pollutant emissions requirements, lead to some incompatibilities which were solved by the above mentioned strategy. Comparison of the prototype emissions with commercially available appliances showed the correctness of the methodology.

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