UP-DATE ON CYCLONIC COMBUSTION AND CYCLONIC BOILERS

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Abstract. *The boiler concept has been around for more than 70 years, and there are many types available. Boilers provide steam or hot water for industrial and commercial use. The Federal University of Pará (UFPA) through the research group EBMA (Energy,Biomass and Environment) has been developing cyclonic furnace with a water wall, a boiler, aiming to use regional timbers (sawdust) and agro-industries residues as fuel to produce steam to be used in industrial processess as well as in power generation,. The use of cyclonic combustion for burning waste instead of burning in a fixed bed is mainly due to two factors efficiency improvement causing a more compact boiler and less risk of explosion, since their process does not generate an accumulation of volatile. Present state-of-art for commercial cyclone boilers has as set up a cyclone combustor with two combustion chambers, in fluid communication, where there ducts for supplying air and fuel directly into the first chamber and for forming a cyclonic flow pattern and a heat exchanger surrounding the second chamber for keeping low combustion temperature in both chambers. This paper shows the results of a literature review about design, construction and operation of cyclonic boilers using solid, liquid or gaseous fuel. This information has been used for the design of a cyclone boiler to be constructed at UFPA for research pruposes and its basic concept is presented at the end of this article.*

Keywords*: cyclonic combustion, boilers, bioenergy*

1. INTRODUCTION

The cyclonic combustor is technically feasible and good performance solution on replacement of conventional combustion furnace and has fuel flexibility advantage what is to operate with different types of fuels such as wood chips, bark, waste, petroleum coke (Ushima, 1998). Cyclonic combustor has the capability to convert, more efficiently and with more compact equipment, such solid fuel into usable energy to generate electricity or steam. The cyclonic combustor is basically a cylindrical chamber in which the solid fuel particles burn in a cyclonic motion because the air is introduced tangentially into combustor (Ragland, 1998).

In cyclonic combustion, the burning occurs at a positive pressure, rather than the slight negative pressure associated with conventional systems. The vigor which cyclonic action occurs facilitates a rapid combustion and solid products removal. The flow inside in this type of combustor is strongly rotational or swirling. Centrifugal force moves the particles towards the cylindrical wall of the burner facilitating the rapid combustion what occurs at boundary layer near the wall (Tillman, 1991). Cyclonic burners remove 70% of the solid products of combustion as slag and only 30% of solid combustion products leaving the burner enter into the boiler convective area to be lately removed as fly ash. This facilitates the use of small boilers with high rates of heat release and tighter tube spacing than conventional systems (Tillman, 1991).

According Gazel (2009) the high working temperatures of cyclonic combustor combined with the complex flow inside the combustor complicates the experimental measurements, and causing the atmospheric nitrogen dissociation that by combining with oxygen results in high nitrogen oxide emissions, which restricts the use of cyclonic combustors. This is why most of the studies to turn to the minimization of such pollutants and are performed through a numerical study

According Turns (2000), the swirling flow is used for two reasons: first, swirl flow can stabilize a flame creating a recirculation zone if the swirl is strong enough. Second, because the flame length can be controlled by the intensity of swirl. In accordance with Syred (2006) the mechanisms and benefits of stabilizing the swirl combustion depend on the formation of a central toroidal recirculation zone, which recirculates the heat and reactive chemical species to the base of the flame. This recirculation that allows the flame stabilization occurs in regions of low relative velocity, where the low flow and turbulent flame speed can be compensated and assisted by the recirculation of heat and reactive chemical species.

Chenand and Driscoll (1989) conducted a study about the role of the recirculation vortex in improving fuel-air mixture in swirling flames. Their conclusion was that swirl flames are unique to promote fuel-air mixture at high rate due two facts what control the swirl intensity. The first, recirculation zone acts as a large swirl with large dimensions with a characteristic velocity and length scale that are much larger than those associated with swirls in a simple jet, therefore, the air is drawn into the vortex mainly in the downstream region. The second, because there is no incidence of opposite jets ahead, there is no stagnation point. Thus, pressure gradients, which are not present in simple jet, increase the mixing rates.

Syred et al (1974) discusses the combustion rotational flow that occurs in swirl and cyclonic combustors seeking to understand and use the rotational flow. It explains that the rotational flow increase considerably the stability limits of most of the flames, and provide very long residence times for the air-fuel mixture. According to this research, the entry of fuel or air in staged or stratified way can be used to minimize emissions of $\mathrm{NO_X}$ and hydrocarbons in the rotational combustors.

This work present a summary of literature review done with the objective to collect information to be used during the design of a cyclone boiler with water wall by the research group EBMA (Energy, Biomass and Environment) at the Federal University of Pará. The relevant information is shown here and also the final cyclonic boiler set up designed to be constructed aiming to understand cyclone furnaces.

2. CYCLONIC FLOW

The turbulent flow has been studied using equations obtained through an average procedure from the instantaneous equations Navier-Stokes. For the average process, initially the solution variables of the instantaneous equations Navier-Stokes are decomposed into average and fluctuating components. For the components of velocity $\mathrm{u_i}$ are demonstrated in Eq. 1.

$$\mathrm{u_i} = \mathrm{\overline{u}_i} + \mathrm{u'_i} \tag{1}$$

Where $\mathrm{\overline{u}_i}$ and $\mathrm{u'_i}$ are average and fluctuating components, respectively.

Replacing u_i in the instantaneous equations of continuity and momentum conservation and using the temporal average a set of average equations is obtained, which according to Silveira Neto (2001), for case of an incompressible flow of a Newtonian fluid can be written in tensor notation, as in Eq. 2 and Eq. 3.

$$\frac{\partial}{\partial x_i}(\overline{u}_i) = 0 \tag{2}$$

$$\frac{\partial}{\partial t}(\overline{u}_i) + \frac{\partial}{\partial x_j}(\overline{u}_i\overline{u}_j) = -\frac{1}{\rho}\frac{\partial\overline{p}}{\partial x_i} + \frac{\partial}{\partial x_i}\left[v\left(\frac{\partial\overline{u'_i}}{\partial x} + \frac{\partial\overline{u'_j}}{\partial x}\right) - \overline{u'_iu'_j}\right] \tag{3}$$

The above equations are known as the Reynolds equations. These equations are in the same form of equations from which they were derived, but there has additional terms that represent the effects of turbulence named Reynolds stress, $\overline{\mathrm{u'_iu'_j}}$. These additional terms must be modeled so that Reynolds equations can be solved.

The Reynolds equations need additional equations to be solved. The closure of theses equation is accomplished through the turbulence models that provide additional terms. Cunha (2005) used two models of turbulence to determine the additional terms, RNG k-ε model and Reynolds Stress Model, RSM.

With respect to the calculation of the flow in the boundary layer near wall, in many cases, for high Reynolds numbers, wall law can be used to model the viscous effects. For the case of the flow, an universally term used for the calculation of viscous effects in the boundary layer close to the wall is known as the wall law and is presented by Prandtl, as in Eq. 4 (Eaton et al, 1999).

$$u^+ = \frac{1}{k}\ln(Ey^+) \tag{4}$$

Where:

$$u^+ \equiv \frac{u}{u_t}\ ,\ y^+ \equiv u_t\frac{y}{V}\ e\ u_t \equiv \sqrt{\frac{\tau_w}{\rho}} \tag{5}$$

$\mathrm{u_t}$ is the friction velocity, ν the kinematic viscosity, τ_w is the local strain in the wall, E is an empirical constant and k is Von Kármán's constant.

The wall law version proposed by Launder and Spalding is widely used in industrial flows. Fluent (2001) refers the Launder and Spalding wall law as Standard Wall Function. The standard wall function for average velocity is show in Eq. 6.

$$u^* = \frac{1}{k}\ln(Ey^*) \tag{6}$$

Where:

$$u^* \equiv \frac{u_p C_\mu^{1/4} k_p^{1/2}}{\tau_w/\rho} \tag{7}$$

$$y^* \equiv \frac{\rho C_\mu^{1/4} k_p^{1/2} y_p}{\mu} \tag{8}$$

k is the Von Kármán constant (= 0.42), E is an empirical constant (= 9.81), u_p is the average velocity of the fluid at any point, k_p is the turbulent kinetic energy at this point, y_p is the distance this point to the wall, μ is the dynamic viscosity of the fluid and. According Cunha (2005), C_μ is a constant that assumes the default value of 0.09 if using the RSM model or 0.0845 if using the RNG k-ε model.

The description of particle motion is accomplished through a Lagrangian approach. The prediction of the trajectory of a particle is made by integrating the force balance equation on a particle. The force balance of compares inertia with surface forces acting on a particle, as in Eq. 9.

$$\frac{du_p}{dt} = F_D(u - u_p) + \frac{g_x(\rho_p - \rho)}{\rho_p} + F_x \tag{9}$$

Where $\mathrm{F_D}(\mathrm{u} - \mathrm{u_p})$ is the drag force per unit mass of the particle, as in Eq. 10.

$$F_D = \frac{18\mu}{\rho_P d_p^2}\frac{C_D Re}{24} \tag{10}$$

Where u is the velocity of the fluid phase, u_p the particle velocity, μ the molecular viscosity of the fluid, ρ the fluid density, ρ_p is the particle density and d_p is the particle diameter. Re is the relative Reynolds number on, as in Eq. 11:

$$Re \equiv \frac{\rho d_p |u_p - u|}{\mu} \tag{11}$$

Cunha (2005) showed that the drag coefficient ($\mathrm{C_D}$), in Fluent (2001), is determined by the Eq. 12.

$$C_D = \frac{24}{Re} \times (1 + b_1 Re^{b_2}) + \frac{b_3 Re}{b_4 + Re} \tag{12}$$

Which in,

$$b_1 = \exp(2.3288 - 6.4581\phi + 2.4486\phi^2) \tag{13}$$

$$b_2 = 0.0964 + 0.5565\phi \tag{14}$$

$$b_3 = \exp(4.905 - 13.8944\phi + 18.4222\phi^2 - 10.2599\phi^3) \tag{15}$$

$$b_4 = \exp(1.4681 + 12.2584\phi - 20.7322\phi^2 + 15.8855\phi^3) \tag{16}$$

The form factor ϕ is demonstrated in Eq. 17.

$$\phi = \frac{s}{S} \tag{17}$$

Where, s is the surface area of sphere having same volume as the particle and S is the surface area of the real particle.

3. CYCLONIC BOILERS AND SWIRL BURNERS

Boilers can be classified into two basic types, flametube and watertube. The flametube boiler has combustion gases crossing inside the boiler tubes which are surrounded by water, transferring heat to it. The watertube boiler has tubes located outside of the boiler combustion chamber. In this, the water circulating inside the tubes and the hot gases are found in contact with its outer surface.

Except for bubbling and circulating fluidized bed boilers, all other boilers use burners as the primary source of energy. The burner plays an important role, because the ignition of the fuel, combustion conditions and aerodynamic are all governed primarily by the construction and arrangement of this burners. The burner performance determines whether the combustion equipment will operate reliably and economically. Swirl burners are of three major types: volute burners, axial vane burners and tangential vane burners. (Basu, Kefa, Jestin, 2002)

Taupin (2003) studied the burning cyclonic applied in biomass boiler with following dimensions: combustion chamber with 1.26 m in diameter and 1.4 m long with maximum operating pressure of 5 bars and staged combustion to reduce pollutants finding the boiler efficiency of 89% and emissions in accordance with European legislation.

Madhiyanon, Lapirattanakun et al. (2006) studied an inverted cyclonic combustion chamber of internal diameter 0.40 m and length of 0.85 m used for burning rice husk. The project purpose was to integrate the cyclonic and fluidized combustion to increase efficiency of the process, while maintaining compatibility. They analyzed the thermal efficiency, volumetric combustion intensity and emissions of pollutants, to reach the performance of the incinerator, which presented a combustion efficiency of 88% and maximum volume intensity of 0.95 MW.

Vasconcelos (2008) studied a cyclonic combustor that had a cylindrical geometry with internal diameter 0.84 m, external diameter 1.66 mm and combustor length of 4.41 m with lean combustion. He assessed the inlet air/fuel ratio, the inner wall temperature and gas concentration profile and also at the chimney.

Kang, Culick and Ratner (2007) used a low-swirl axial burner to propose a methodology to assess the combustion dynamics through the Rayleigh index, which quantifies the thermo-acoustics coupling and flame response functions, measuring the system response to outside disturbances. The burner has an internal diameter nozzle of 2.54 cm with a swirler located 3.81 cm below the outlet burner. The angle of the vanes related to the horizontal plane was approximately 65° and the ratio of axial internal and rotational external flow was $1/2$. The pre-mixer diameter was 7.62 cm and 46 cm length. It had a grate inserted between the upper and lower parts to improve fuel/air mixture. This study found that thermo-acoustic coupling was evident, especially in the shear mixing zone, producing a distribution pattern of Rayleigh toroidal index. Also showed, that the phase shift of the flame fluctuation from the imposed acoustic wave appears to be closely linked to the vortices generated at the border flame due to mixed shear, thus inducing the alternating toroidal structures. They demonstrated that the peak value of the flame response function coincides with the peak absolute value of the Rayleigh index.

Cheng et al. (2000) also studied the low-swirl burners. They studied two burners, with internal diameter of 5.28 and 7.68 cm, configured to accept a novel vane-swirler design were evaluated up to a firing rate of 73 kW and 280 kW, respectively. These burners were tested in a boiler. The results show that constant velocity criterion is valid for scaling the burner diameter to accept higher thermal inputs. However, the turbulence number required for stable operation should be scaled independently through a criterion of constant residence time. Between 210 and 280 kW and 0.8 <equivalence ratio <0.9, the emissions of NO_x and CO are below 15 ppm and 10 ppm, respectively.

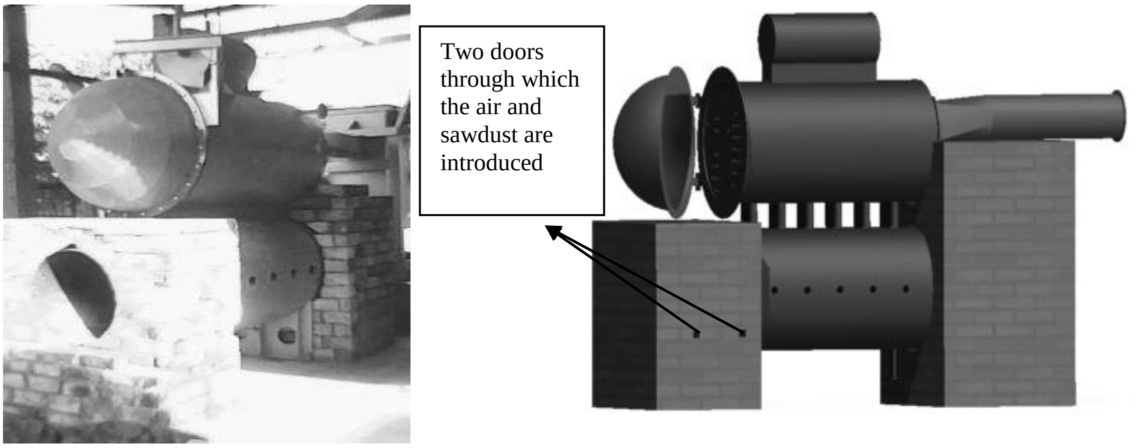
United States Department of Energy (2000) evaluated the efficiency and emission of NO_x of a boiler that combines firetube and watertube technology using the burning of natural gas and oil. The boiler was housed in a cylindrical body encasing a combustion chamber with watertubes extending along the waterwall. The watertube furnace section was connected to the firetube convective section in the steam drum by a turning box with the waterwall side and end walls. The waterwall tubes have an outlet leading to a top header that is connected to the steam drum and two water inlets from the bottom water header. The turning box end wall contains points to access the watertube furnace section and the steam drum and firetube sections. A single-pass convective section was located in the steam drum. The heat transfer was enhanced between the flue gas and the water in the steam drum by the addition of metal gas flow restrictors. This study showed that for natural gas, the boiler had, for the most emissions, a value of 20 ppm NO_x and 30 ppm CO, while those in conventional boilers showed emissions that were higher than 40 ppm CO and 110 ppm NO_x. Using oil, the boiler had, for the most emissions, which is close to 30 ppm CO and 60 ppm of NO_x, while those in conventional boilers showed results emissions similar to those obtained with natural gas. The boiler presented efficiencies of 87% and 85%, using oil and natural gas respectively, while those in conventional systems the efficiency is close to 80%.

Hoffert, Milligan and Morrison (1989) patented a pressurized cyclonic combustion method and a cylindrical burner apparatus for pressurized combustion of particulate solid fuels to produce a pressurized clean effluent gas. The particulate solids are fed tangentially into a primary combustion chamber at its inlet and flow at high tangential velocity in a helical path through the burner. The combustion gas is supplied tangentially at high velocity through multiple ports spaced along the burner length to maintain and increase the high tangential velocity and produce high burner volumetric heat release rates exceeding about 4140 kW/m^3. A secondary combustion chamber is provide downstream of the choke opening, this centrally located at the combustion chamber outlet end for promoting prolonged combustion of solid fuel particles upstream of a quench zone. In the choked zone, a quench gas stream was introduced for mix itself with the hot effluent gas, with the aim of reducing the temperature of these gases, usually to about 760 °C to 1093 °C. They showed that their invention had a much higher centrifugal force than the conventional cyclonic boilers. For example, using tangential velocity of 30.48 m/s, for both boilers, the conventional boilers generated a centrifugal force of approximately 1.47 N, while the innovation showed approximately 2.94 N. This prototype showed an internal pressure of 5 atm and a volumetric heat release 20700 kW/m³, while those conventional boilers showed an internal pressure of 1 atm and 3105 kW/m³ of volumetric heat release.

Jacob Korenberg and Mark Khinks (1990) patented a boiler with cyclonic combustor with two combustion chambers in fluid communication. Ducts supply air and fuel directly into the first chamber forming a cyclonic flow pattern of hot gases for combustion within the first and second chambers. An exit throat at the end of the second chamber and a heat exchanger surrounding the second chamber was used for keeping low combustion temperature in both chambers. They showed, for the maximum capacity, the boiler would require a swirl and Reynolds number at least of 0.7 and 18000 respectively. In an experiment using a swirl number of 0.6 and Reynolds number of 18000, they had obtained values of heat release exceeding 4070 kW/m³ and NO_x concentrations of 60-120 ppm and 120-180 ppm for the combustion of natural gas and light fuel oil, respectively. According to the authors, stable combustion, even at low boiler capacity, is achieved by not cooling the walls of first chamber where the air and fuel are injected, but cooling only the walls of the second chamber. This stable combustion enables high turndown ratios to be accomplished. In this prototype the turndown ratio can be increased from 4:1 up to and higher than 10:1. Excess air can be decreased from 25-30% to 5%, and kept constant at 5% over the high turndown ratio of 10:1. The flame temperature can be decreased to 1093 °C, as opposed to about 1648 °C for conventional fire tube boilers.

4. CYCLONIC BOILER DESIGNED AT UFPA

The boiler developed combines the characteristics of watertube and flametube boilers, built on the concept of cyclonic burning and staged combustion system. In cyclonic burning the particles fuel burns in cyclonic motion due to the air introduced tangentially into the combustion chamber. In staged combustion, the air needed for combustion is introduced into the burner refractory cylindrical through two doors. The cyclonic boiler is shown in Figure 01 and its technical specifications are presented in Table 01.



(a) (b)

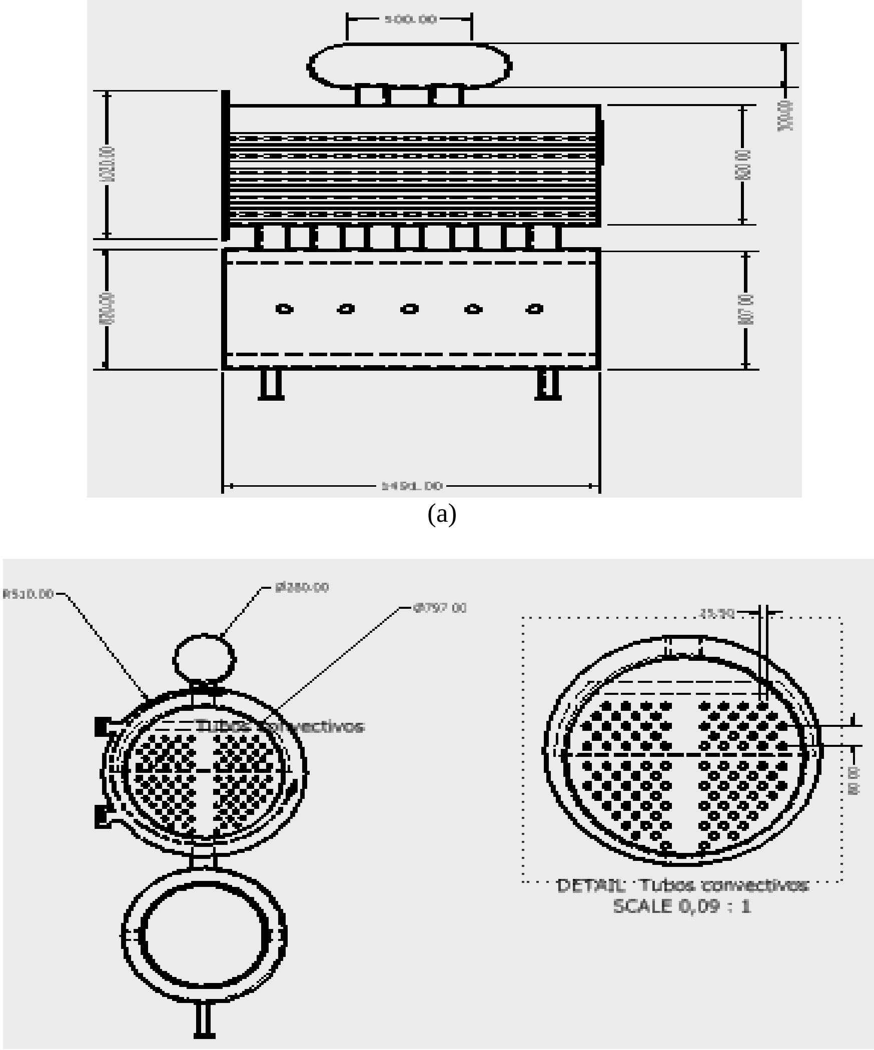
Figure 01 – (a) Shows the boiler in final phase of construction and
(b) shows the design used to boiler construction, where the two doors are posted Source: Gazel, 2009

Trough the first door, (left one at Figure 01-b) would sawdust and air will be injected tangentially into the combustion chamber at high velocity with the aid of a centrifugal fan (It is the primary port). In this primary stage, the cyclonic staged combustion occurs with combustion rich in fuel and the combustion process is characterized by a strongly rotational flow (swirling) and high internal recirculation. To complete the combustion, only air will be introduced through the second door (right one at the Figure 01-b), also with the aid of centrifugal fan (It is the secondary port). The secondary staged air helps to reduce NO_x emissions. Combustion with rich fuel prevents the formation of NO_x in the primary zone since any oxygen present in air is used to oxidize the fuel. The combustion chamber was constructed with steel plate ASTM A-285 - lined with insulation glass wool appropriate for relevant temperatures - with an internal diameter of 617 mm and length 1491 mm, with an approximate volume of 0.44 m³. The combustor volume has a second steel wall creating a volume 0.34 m^3 where a water wall will be locates. Table 1 shows the boiler technical specification.

Table 1 - Technical specifications of the cyclonic boiler design

Specifications	**Values**
Steam production	100 kg/h
Steam pressure	10 atm
Total intensity energy	115 kW
Theoretical thermal efficiency of the boiler	85%
Fuel mass flow rate	34 kg/h
Air mass flow heat	190 kg/h
Average expected temperature in combustor	1223 K

The combustor chamber is divided in two sections. The first, lined with refractory bricks, is where the combustion starts and responsible for flame stabilization. The second section, with a cylindrical shape having a steel wall with water wall outside, is name radiation section where the cyclonic combustion progress transferring heat to the water wall. At the end of the radiation section, the cyclonic move is destroid and the flow is transferred to the convection section, where bundles of horizontal tubes force the hot gases to make two (2) passes hot gases transfer heat to the water. On the top of the convective section, there is a reservoir where the interface liquid steam will be. It means that both, radiation and convection water wall will be at liquid phase and the vapor phase will be collect at the steam tank (Figure 2). The convective section presents an internal diameter of 794 mm, length 1491 mm and its convective tube has a diameter of 25.4 mm.

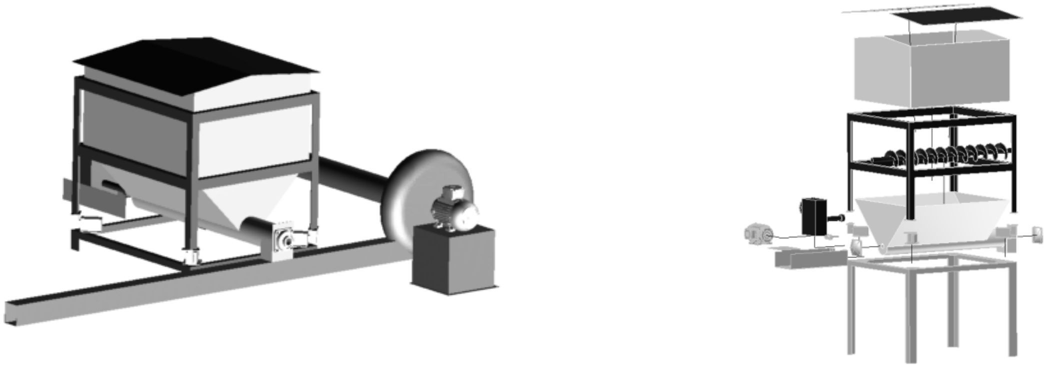


(a)

(b)

Figure 2 – (a) Shows the design used in the construction of the drum convective and (b) shows the steam tank detailing the convective tubes. Source: Gazel, 2009.

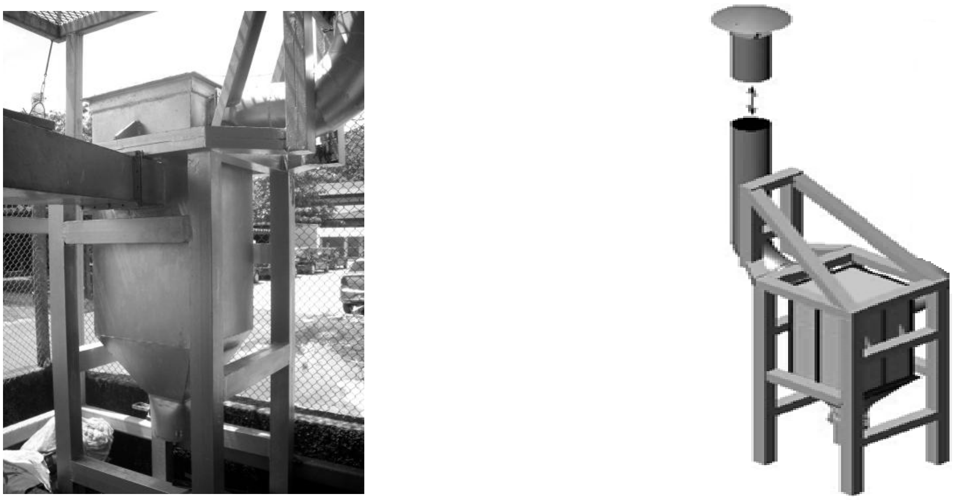
The supply system of air and sawdust (Figure 3) is mainly composed of a silo, screw conveyor, two centrifugal fans and cells for weighing the biomass. The rotation of the fans can range 0 - 1740 rpm, and screw conveyor, driven by a gear, with ratio 1:58 of 0-1740 rpm.



(a) (b)

Figure 3 – (a) Shows supply system with the biomass tank coupled with the centrifugal fan and
(b) Shows the expanded view of the design used in construction of the supply system. Source: Gazel, 2009

The flue gas system will pass through a cyclone filters to remove the particulate residues (Figure 4). This equipment was previously designed, constructed and operated successfully during previous investigation work on cyclone combustion furnaces. It was designed and constructed by (Cunha, 2005) and Operated by (Vasconcelos, 2008). The system was adapted to capture fly particulate in the waste combustion gases with dimensions greater than 5 microns, through cyclonic filters containment.



(a) (b)

Figure 4 - (a) shows the collection system and filtering the flue gas and
(b) Shows the design used for their construction of it. Source: Akel, 2008

The design of this cyclone boiler used information obtained through previous studies related to burn in a cyclonic combustor, which has generated the dissertation by Cunha (2005) and Vasconcelos (2008). This experience has enabled a new phase which is the construction and deployment of this for future studies where there will measure the efficiency of the boiler, and the behavior of the cyclonic combustion process inside.

5. ACKNOWLEDGMENTS

The authors express here the gratitude to the CNPq for financing the project “Combustor Ciclônico com Parede de Água para Geração de Vapor Aplicado para Geração de Energia Elétrica”, Processo: 555133/2006-0, that make possible this investigation and boiler construction.

Felipe Alfaia do Carmo has a PIBIC Grant by CNPq.

The authors thank Pedro Andrey Cavalacanti Sampaio for his cooperation in this project and also for useful discussion and suggestions.

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