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TRANSIENT BEHAVIOR FOR THERMAL EFFICIENCY OF A COMBUSTION HEATER SUBJECTED TO ASH FOULING

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Abstract. *Constant demands for the reduction of energy waste and increase of the thermal efficiency, reducing operating costs, have been an important parameter in processes that make use of thermal machines, like boilers. Within the draft boilers, air preheaters play a key role in determining the absolute efficiency of heat recovery of low level, such as the flue gases before sending it to the atmosphere. In Overall, for a common boiler every 22°C recovered of the flue gases by an air preheater, the absolute efficiency of the equipment may rise by about 1%. The proposed work experimentally aims to show the loss of thermal efficiency of an air preheater of a superheated steam boiler during its operation. This analysis covers the relationship of loss of heat exchange by fouling and clogging of the heat exchanger tube bank during its operation process. To analyze the thermal losses were used measurements of maximum velocities and average velocities by pitot tube and venturi tube, together with Bernoulli equation, by calculating the mass flow rate of the input current of air and flue gas, connecting them with the inlet and outlet temperature for each flow.*

Keywords*: Boiler, Air preheater, Pitot tube, Venturi tube, Bernoulli equation.*

1. INTRODUCTION

The steam generators have been widely used to produce energy in more than 200 (two hundred) years, being one of the pioneers of the application of engineering principles of heat exchangers (Fraas, 1989). The product of this equipment can be used for different applications like residential, industrial processes, or more commonly used for the production of electricity. The different types of application also suggest different types of fuels that may vary depending on the availability of raw material per region or construction concepts. Within the application of heat exchange in steam generators, it is extremely important to the combustion efficiency the use of an air preheater. In general, the air preheater recover the heat from flue gases that are released to the atmosphere, or different forms of hot stream. Studies have shown that the overall efficiency of a steam generator can rise from 5 to 10% when using heated air for combustion (Babcock and Wilcox, 2005). Strategically, the preheater is located at the rear of the boiler, where uses the hot gas from the combustion (which exchanged heat with the other systems inside the steam generator), and cold air, which are inflated with the use of fans. The air preheater is separated by regenerative or recuperative. In regenerative preheaters, heat transfers occur indirectly through convection. A medium heat store periodically transfers heat to the hot or cold stream, being the rotary preheater Ljungström the most commonly used and its efficiency and losses usually studied. (Drobnic, Oman and Tuma 2006) (Wang et al. 2009). To the recuperative preheater, the heat transfer is transmitted directly and continuously by a stationary system which transfers heat through solid surface, which the cold stream and the hot stream flows separated (Babcock and Wilcox, 2005).

This study aims to establish the operation efficiency of a recuperative air preheater (tubular and multipass) experimentally, thus finding the loss of efficiency by fouling and clogging of the tube bank. Through this equipment flows two air streams separated, known as primary air and secondary air. The two streams are used in the steam generator to increase the combustion efficiency, being the primary air used to keep the bagasse biomass in suspension and the secondary air stream is used to carry out the vortex within the combustion chamber, standardizing the bagasse burning and temperature throughout the chamber. Although this type of preheater hardly present leakages mixing air with gas, problems can occur due to oxidation by condensation of gases in the tube wall. This problem was studied by other authors comparing this preheater to other one like a heat pipe preheater for example (Shi et al. 2011).

2. METHODOLOGY

The experiment was designed to establish the mass flows inlet and outlet of the heat exchanger according to Fig. 1. To define the streams was used for both air inlets (primary and secondary) Pitot tube in order to establish the maximum velocity inside the air duct, as reported by other studies (Pesarini et al. 2002). Each fan is provided with an inverter frequency. With the use of a differential pressure transmitter and PLC (programmable logic controller) was held the historical data of the Pitot tube pressure along with the rotation of the fan motor. Through a comparison chart of these values within the Microsoft Excel, the equation of this correlation data was found. The same procedure was replicated to the other air stream, being found thus the pressure in mmH2O only with the fan rotation.

To determination of the required gas flow a venturi tube was installed inside the duct of the exhaust flue gas to the air preheater. The use of the Venturi for determining gas flow has already been studied (Jitschin, 2004) (Xu et al. 2003). To determine the time of efficiency loss, a differential pressure transmitter was used an integral manner, with a PLC (programmable logic controller) collecting all usable information.

The temperature data were measured by transmitters, and their data were collected together with the information that determines the air and gas flow in every 10 seconds to reach a more accurate conclusion.

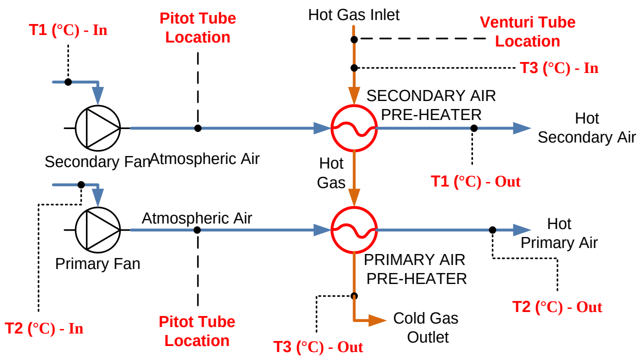

Figure 1. Instrumentation and Process Flow Diagram

2.1 ANALYSIS CONDITION

To get a better phenomenon analysis and get reliability data from the equipment efficiency, the following situations were assumed.

- The air relative humidity scorned;
- The air was established as incompressible fluid due to low Mach number <0.3 (Fox et al. 2011);
- The furnace pressure was established as permanent behavior;
- The friction inside the Pitot tubes and Venturi were considered negligible;
- The heat loss to the atmosphere was considered negligible;
- The air (secondary and primary) and combustion gases flow were considered laminar;
- The transient operation hours (beginning and stopping) were not considered in the calculus;
- No air leakage in ASME method.

2.2 EQUIPAMENT DESCRIPTION

The studied air preheater Fig. 2 is part of a monodrum boiler capable of producing 150 tons of steam per hour with 67kgf/cm² pressure and 520°C temperature. The equipment is fully automated and has temperature information at each stage of the process.

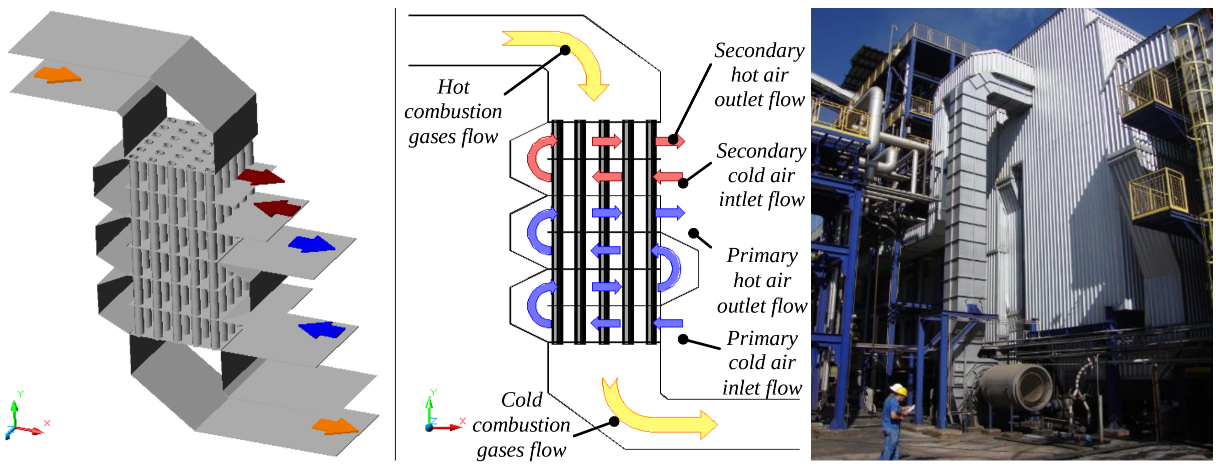

Figure 2. Preheater Detail – Monodrum Boiler

Through Tab. 1 the preheater specification of surface exchange are shown.

Table 1. Heat Exchanger Surface and Pipe Specification

Heat Surface (m²)		Pipe Specification				
Primary air	Secondary air	Diameter (mm)	Thickness (mm)	Number	Length (mm)	Material
6649	1385	63.5	2.25	3472	11730	SAE 1008

2.3 TYPICAL INSTRUMENTATION INSTALLATION

To get the results of velocity and flow, a Pitot tube was installed in the secondary air and primary air, Fig. 3 and Fig. 4. In this case, a differential pressure transmitter was used to get the data to correlate with fan rotation in RPM.

Figure 3. Pitot Tube Installation – Secondary Air Duct

Figure 4. Pitot tube installation – Primary Air Duct

Another typical installation was the Venturi tube, developed to this application, taking measurements of the hot combustion gases flow before entering in the air preheater in Fig. 5.

Figure 5. Venturi Tube – Calibration and Installation

3. RESULTS

To begin the experiment analysis was necessary to calculate the mass flow of each stream through the medium velocity for a rectangular duct (even secondary and primary duct, as the combustion gases duct). For each stream, the laminar flow was considered.

To get start the flow analysis (trough the medium velocity for each mass flow), several flow rates of primary and secondary air were taken according to the graphs equation that are related the fan rotation, Eq. (1) and Eq. (2), like mentioned in item 2. The graphs, presented in Fig. 6 and Fig. 7 have been raised to do not need use a pressure transmitter for each air inlet, and therefore only the venture tube was monitored by differential pressure transmitter.

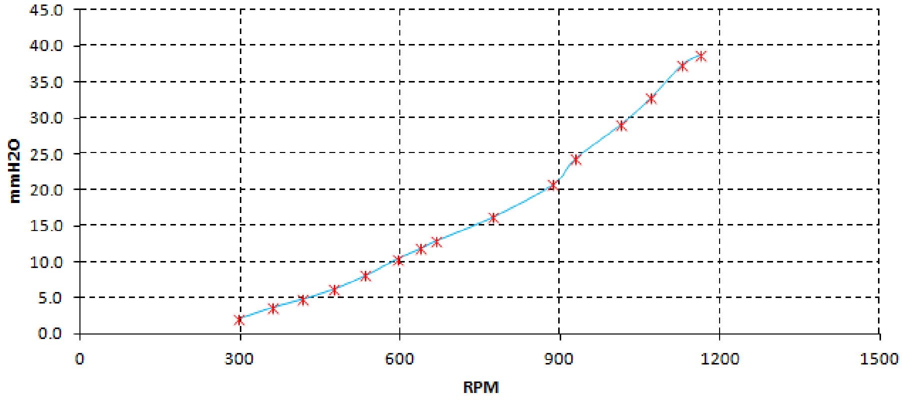

Figure 6. Pitot Tube – Primary air

$$Y = -3.66E-13 * x^5 + 1.32E-09 * x^4 - 1.8E-06 * x^3 + 1.17E-03 * x^2 - 0.339 * x + 37,27 \quad (1)$$

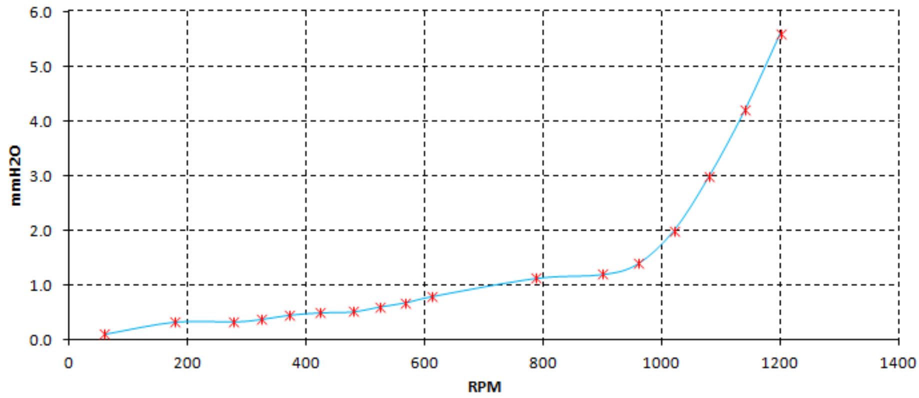

Figure 7. Pitot Tube – Secondary air

$$Y = -3.67E-16 * x^6 + 1.61E-12 * x^5 - 2.77E-09 * x^4 + 2.41E-06 * x^3 - 0.0011 * x^2 + 0.26 * x - 23.63 \quad (2)$$

Where Y is the differential pressure in mmH20 obtained by the fan rotation axis x.

The determination of the air velocity in the primary and secondary ducts is established through the Bernoulli equation Eq. (3):

$$V_{max} = \sqrt{\frac{2 * (p_0 - p)}{\rho_{air}}} \quad (3)$$

Where p_0 is the stagnation pressure and p is the static pressure taken in the duct wall, and ρ is the air density.

The venturi calibration curve in Fig. 8 shows the data obtained to find the Eq. (4) that connects the pressure with the volumetric flow of the combustion gases.

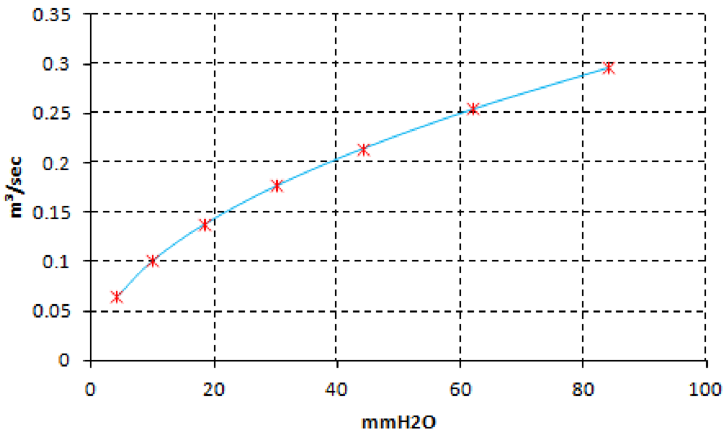

Figure 8. Venturi Tube – Calibration Curve

$$Y = 0.0324 * x^{0.5} \tag{4}$$

Where Y is the volumetric flow of the flue gases and x the differential pressure of the venturi.

To determine the medium velocity inside the duct (such primary and secondary flow as combustion gases flow) the Eq. (5) similar to Hagen-Poiseuille equation (Bird et. al, 1960) was used.

$$v_{med} = \frac{2}{3} * V_{\max} \tag{5}$$

Where v_{med} is the medium velocity and $V_{\max}$ is the maximum velocity.

The energy balance was definite taking as reference the control volume from Fig. 9. The Eq. (6) describes the energy balance used to find the heat loss to the atmosphere. The evolution graph of the heat loss to atmosphere (energy balance resume) is shown in Fig. 10.

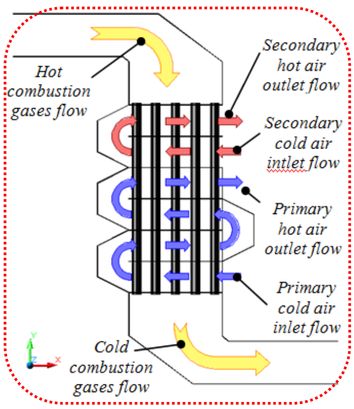

Figure 9. Energy Balance – Control Volume

$$\text{Heat Loss to Atmosphere} = \dot{Q}_{gas} - (\dot{Q}_{primary\ air} + \dot{Q}_{secundary\ air}) \tag{6}$$

Where $\dot{Q}$ is the heat flow.

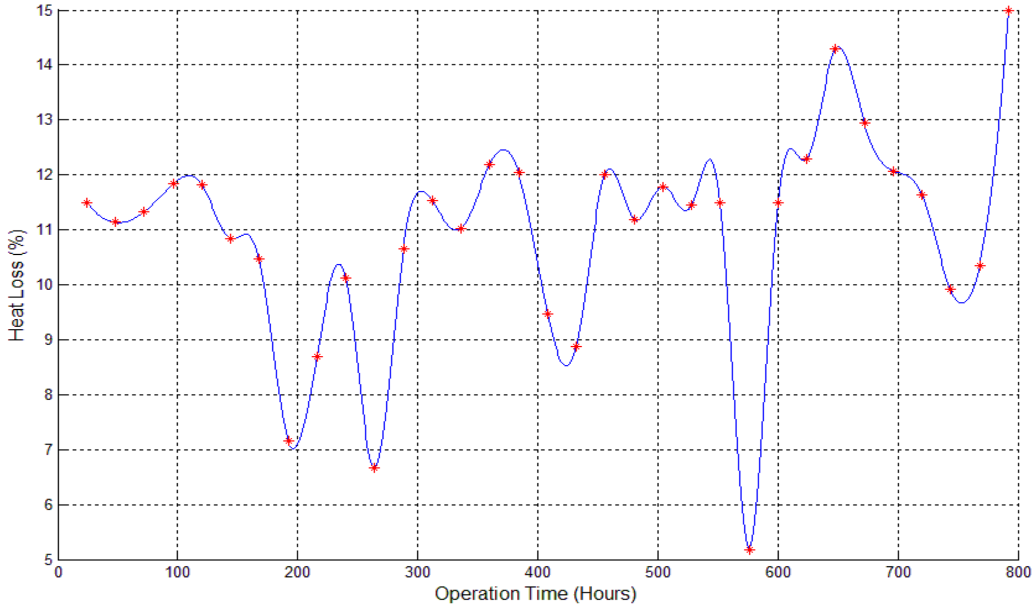

Figure 10. Energy Balance Resume – Heat Loss to Atmosphere

The graph of transient behavior of thermal efficiency Fig. 11 is comparing the ASME PTC 4.1 method (ASME, 1968) with the method used in this work. The data were extracted from the Eq. (7), Eq. (8), and the ASME method Eq. (9). All of these equation give a sample of efficiency during the hours of operation of the equipment, including too the operation time with very low production. The differential of gas temperature is calculated through the total availability of energy (until the ambient temperature).

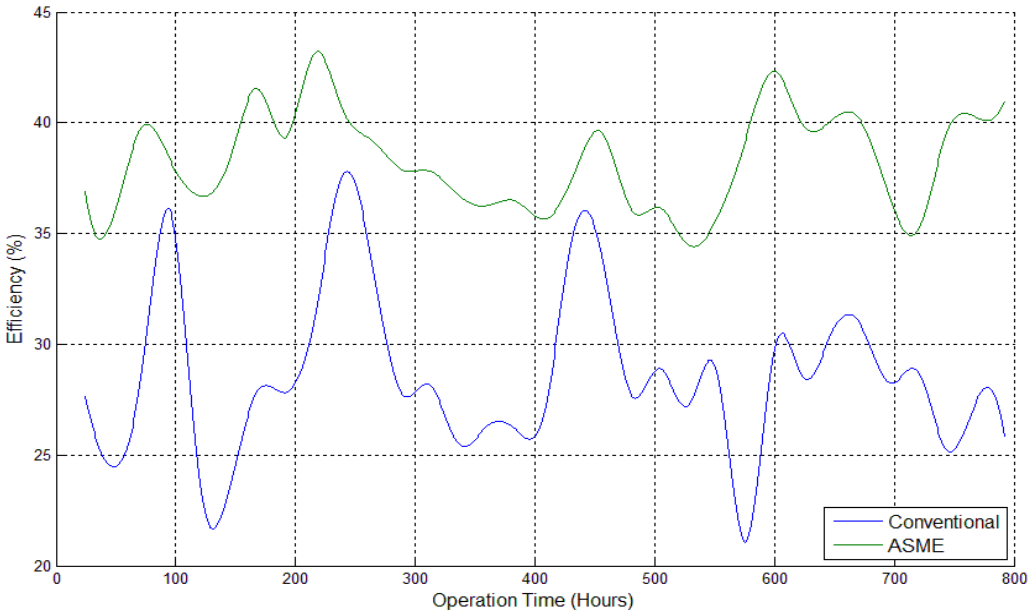

Figure 11. Transient Behavior for Thermal Efficiency

$$\dot{Q} = \dot{m}.C_p.\Delta T \tag{7}$$

$$Efficiency = \frac{\dot{Q}_{primary\ air} + \dot{Q}_{secundary\ air}}{\dot{Q}_{gas}} \tag{8}$$

Where $\dot{Q}$ is the heat flow and $\dot{m}$ is the mass flow, C_p is the heat coefficient and ΔT is the temperature difference between the inlet and outlet stream in the heat exchanger.

Another way to calculate the preheater efficiency is the ASME procedure, (ASME, 1968). The Eq. (9) exemplifies the data showed in Fig. 11.

$$Efficiency = \frac{t_{in\ gas} - \left(\frac{A_L * C_{pA} * \left(t_{out\ gas} - t_{in\ air}\right)}{100 * C_{pG}} + t_{out\ gas}\right)}{t_{in\ gas} - t_{in\ air}} * 100 \tag{9}$$

Where $t_{in\ gas}$ is the inlet gas temperature, $t_{out\ gas}$ is the outlet gas temperature, A_L is the air leakage percentage, C_{pA} is the air heat coefficient, C_{pG} is the gas heat coefficient, $t_{in\ air}$ is the inlet air temperature (ambient temperature).

The effects of the calculated efficiency can be physically evaluated in Fig. 12 that show the beginning of the ash fouling in the internal wall of the tube bank (gas side) after about 792 operation hours.

Figure 12. Preheater Tube Bank after 792 Operation Hours.

4. CONCLUSIONS

The very low efficiency is related to the gas out temperature, that is about 158°C. This high temperature from gas out is easily explained through the security parameters that protect the physical parts (materials) of the gas out flow duct against the oxidation. The combustion gases when submitted to temperatures under about 150°C can condense forming acids that deteriorates the internal duct wall. This is the main reason to the efficiency under 50%, because the difference between 158°C to ambient temperature is not recovered.

Although the data trend in Fig. 11 through the method used in this work (conventional line) varies all the time, is possible analyze the drop trend of the efficiency. The beginning of this drop trend is related to the ash fouling. The ash can contain sand and an others materials from field during the sugar cane harvest. For this particular case studied, the bagass consumption was about 35 ton per hour (in full operation is 66 ton per hour), and it can contain about 5% of ash (depends of the bagass quality), totalizing about 1.75 ton per hour of ash. Considering the full operation, it can further increasing incidence of ash fouling.

This experiment shows the important data that a recuperative air preheater (tubular and multipass) of a steam boiler initiate a heat transfer efficiency degradation. Through these evolution parameters it is possible to determine an ideal time to stop the equipment for cleaning and maintenance. Through these data is possible to analyze the bagass quality impact in the global boiler efficiency or maybe study the feasibility of corrosion resistant materials to get a better heat recover.

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