ENERGETIC ANALYSIS OF A HEATING KILN TO FORGING PROCESS

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Abstract *– This work deals with a thermal analysis of a kiln used to heat steel plates up to the forging temperature (around 1,250°C). The kiln works with natural gas and the analysis intends to determine its heat efficiency and the heat losses through an energetic balance, creating proposals to save energy. The heating kiln analyzed has two combustion chambers: one of preheating and one of soaking. Based on the energetic balance it was developed three proposals to improve the good use of the heat internally available. The first proposal suggests a change on the internal geometry of the kiln through which it would be possible to reduce the consumption of natural gas in approximately 17 Nm^3/h. The second proposal examines the feasibility to install a heat exchanger to preheat the combustion air, which could result in a reduction of 5 Nm^3/h in the consumption of natural gas. Based on the value of the necessary investment and also on the costs involved it was done a study of the Net Present Value (NPV), which determines that both proposals are practicable. The third proposal consists on the change of configuration (first proposal) and also the change of the materials that are applied to the thermal insulation of the soaking chamber. On the basis of the radiation and convection heat transfer analysis by the external surfaces, a better configuration for the thermal insulation was determined*

Keywords: heat transfer, energetic balance, natural gas, heat exchanger, economical availability, heat efficiency.

1. INTRODUCTION

The serial production of steel plates that requires uniformity and quality with high productivity is served by the process of forging, because this reduces the waste of material and increases the mechanical strength considerably. Forging is the process of hot shaping, where the deformation is caused by the application of static or dynamic force which plastically modify the form of a metal block (Schaeffer, 2001). According to Chiaverini (1986), the forging operations are performed over metal recrystallization temperature, although some metals may be cold forged. Forging maximum temperature is limited by the occurrence of incipient fusion or acceleration of oxidation, while the minimum temperature is that below which starts hardening. For carbon steels the range of normal temperature is around 1,000ºC. To highly alloyed steels, the temperatures employed are higher, around 1,150 to 1,200ºC, due to the complexity of the structural material.

Forging steel industries use furnace of two categories: intermittent and continuous furnaces, according to the technical possibilities of implementation in the plant, but also the geometry and quantity of steel to be heated (Chiaverini, 1997).

The primary characteristic of a furnace, whatever is its purpose is to transfer to the material the heat generated by a source, with maximum efficiency, uniformity and safety. To get these characteristics, the construction of a furnace requires a careful study on heat transfer, i.e., on its movement and means to minimize energy loss (Gilchrist, 1977).

The heat source for heating the furnace, both continuous and intermittent, can have several origins, however, in furnaces currently in use, can be of two types: combustion and electricity.

In the combustion furnace, the choice of fuel depends on a number of factors such as cost of acquisition, storage and pre-conditions for burning (such as temperature, pumping, pre-heating), fuel use and environmental impact (rate between the fuel burned, the waste and the quantity of heat generated), (Correa, 2002).

For the combustion analysis of the process, some fundamental fuel characteristics are needed. The energy contained in chemical bonds of the fuel substances is manifested through its lower and higher calorific values. Each fuel has a calorific value which depends on the chemical elements contained in its composition (Wylen, Sonntag and Borgnakke, 2003).

According to Souza (2003), to reduce fuel consumption and get a great thermal efficiency in the operation of a furnace, is necessary the following procedures: use the right amount of excess air to have a complete burning; use burners technologically advanced; make the most of the sensible heat of exhaust gas; minimize the amount of fuel and air for combustion; not tolerate unburnt fuel; keep the equipment with great insulation and maintain a positive internal pressure to prevent infiltration of cold air in the furnace. The pre-heating of combustion air contributes to fuel economy and also to increase combustion efficiency, resulting in an increase in temperature of the gases produced in the combustion (Janh, 2007).

For the preheating of combustion air is usually used heat recuperators, that use part of the residual energy, available in the combustion gases leaving the furnace. Second Incropera and Dewitt (2003) and Kreith and Bohn (2003), when only the temperature of the input of cold and hot fluid are known is preferable to use the method of the effectiveness (ε) – number of transfer units (*NTU*) for analysis of heat exchanger.

2. MATERIAL AND METHODS

The heating furnace here studied has two chambers, a pre-heating chamber that heats the steel plates from environment temperature to around 550°C and works with four burners of 90 kW and a second soaking chamber that heats the plates up to 1,230°C. This chamber operates with a burner of 1,000 kW and has a length of 4,855mm and 1,470 mm in width. The pre-heating chamber has a length of 3,400mm and 1,320mm in width. Figure 1 shows an internal view of the current design of the furnace.

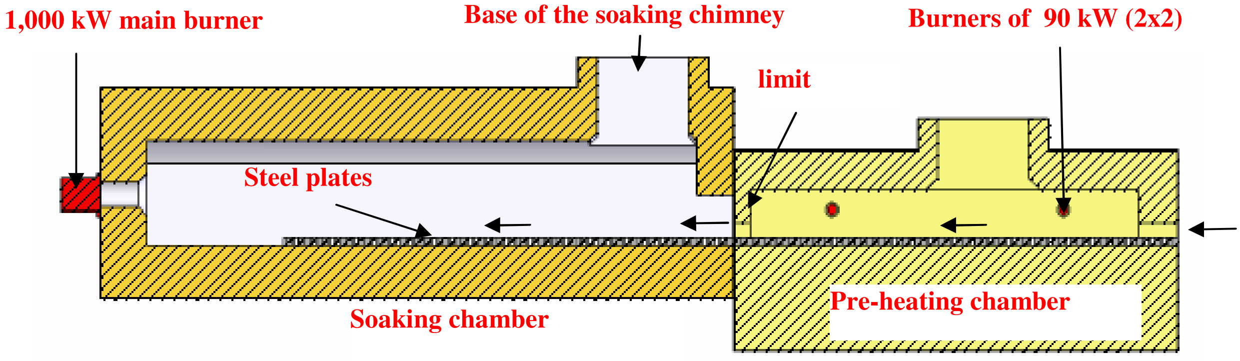

Figure 1. Current design of the furnace

The work in this article refers to the energy analysis of the furnace shown in "Figure 1" to heat steel plates that will be forged. The heating furnace burners operate on natural gas (NG). Although NG is less polluting than other fossil fuels due to their chemical composition, being predominantly methane (CH_4), which is a light hydrocarbon, also presents the same problems like other fuels when it isn't properly mixed with air on burners. Table 1, reports the main technical information related to the NG used.

Table 1. Characteristics of NG [Source: www.sulgas.rs.gov.br]

Composite	Concentration (%V)	Parameter	Value
Methane (CH_4)	91.8	Density Molar	0.0409kmol/m^3
Ethane (C_2H_6)	5.58	F. compressibility	0.99705
Propane (C_3H_8)	0.97	Critical temperature	198.78 K
N-Butane (C_4H_{10})	0.02	Critical pressure	45.76 atm
Carbon dioxide (CO_2)	0.8	Mass spec.@ 20°C	0.71031 kg/m^3
Nitrogen	0.7	Factor K (Cp/Cv)	1.2952
Iso-Butane (C_4H_{10})	0.030	Ratio gas / air	(1/9.96) m^3/m^3
Pentane (C_5H_{12})	0.1	LHV volumetric	37.59 MJ/Nm^3
		Mol. weight average	17.367g/mol
		HHV volumetric	41.624 MJ/Nm^3

For the NG supplied by Sulgás (2008) is necessary about 10 Nm^3 of theoretical air for each 1Nm^3 of fuel gas.

The plates used to forging are of alloyed steel GG15B28 of *Siderúrgica Gerdau Riograndense* (similar to SAE 1,030 steel but with the addition of boron - B): B=0.003%; Si=0.30%; Mn=1.25% and C=0.28%. Due to the characteristics of the material of the plates, the usual range of temperature for forging is from 1,100 to 1,160°C. The specific heat at different temperature ranges and the enthalpy of formation of the scale for this material are obtained from the *FactSage* software at the Federal University of Rio Grande do Sul - *UFRGS*.

An energy balance, according to the First Law of Thermodynamics (Wylen, Sonntag and Borgnakke, 2003), allows a thorough analysis of equipment's energy performance and helps to prioritize efforts to save energy in a financially feasible way.

The energy input of the furnace is considered as: the heat generated by burning NG, and the sensible heat contained in the NG, in the combustion air, and in the steel plates, each evaluated at the inlet temperature of the furnace. Moreover, is accounted the heat of formation of scale.

The energy output is computed as: the heat absorbed by the plates, the heat lost by gas exhaust; and the heat lost by convection and radiation to external walls.

The heat flow rate by free convection (natural) of the external walls of the furnace is obtained by Newton's law of cooling, according to Kreith and Bohn, (2003):

$$q_{conv} = \bar{h} \times A_{\sup} \times (T_{\sup} - T_{\infty}) \tag{1}$$

where: $\bar{h}$ is obtained by using the method of solution for free convection on surfaces that makes use of empirical correlations developed for engineering calculations in Incropera and Dewitt (2003) and Kreith and Bohn (2003):

$$\bar{h} = \frac{\overline{Nu}_L \times k}{L} \tag{2}$$

where: Nu_L is the mean Nusselt number for the surface, $\bar{h}$ is the mean convection coefficient [W.m^{-2}.K^{-1}], L is a characteristic length [m], k is the thermal conductivity [W.m^{-1}.K^{-1}].

The Nusselt number to vertical plate is given by the following equations:

$$\overline{Nul} = \left\{ 0.825 + \frac{0.387 \times {Ra_L}^{1/6}}{\left[1 + \left(0.492 / P_r\right)^{9/16}\right]^{8/27}} \right\}^2 \tag{3}$$

$$\overline{Nu}_L = 0.15 \times {Ra_L}^{1/3} \text{, to upper surface of a heated plate} \tag{4}$$

$$\overline{Nu}_L = 0.27 \times {Ra_L}^{1/4} \text{, to lower surface of a heated plate} \tag{5}$$

Where the Rayleigh number is given by:

$$Ra_L = \frac{g \times \beta \times (T_{\sup} - T_{\infty}) \times L^3}{\nu \times \alpha} \tag{6}$$

where: Ra_L is the Rayleigh number, g is the gravity acceleration [m.s^{-2}], β is the thermal expansion coefficient (air is considered as an ideal gas) [K^{-1}], T_{sup} is the surface temperature [K], ν is the kinematic viscosity [m^2.s^{-1}] and α is the thermal diffusivity [m^2.s^{-1}].

The heat flow rate by radiation is given by:

$$q_{rad} = \varepsilon \times \sigma \times F_{1-2} \times A_{\sup} \times ({T_{\sup}}^4 - {T_{viz}}^4) \tag{7}$$

where: A is the surface area [m^2], $F_{1\text{-}2}$ is the shape factor between surfaces 1 and 2, ε is the emissivity factor and σ is the Stephan-Boltzmann constant, equal to 5.67 x 10^{-8}, [W.m^{-2}.K^{-4}].

For calculations of the heat loss in the external walls of the furnace was measured the surface temperatures using thermography camera Flir T360 model Western. An emissivity ε of 0.9 in the camera was adopted and a form factor for external radiation of 1 was used.

The calculation of the loss of heat by the combustion gases leaving the furnace is obtained from *Acomb v5* - Industrial Combustion software, developed by IPT, (2004) in which the composition of the NG supplied by Sulgás as Table 1, the temperatures and flow of inlet in the burners of the NG and air for combustion, the coefficient of excess air and temperature of gases leaving the chimney, are inserted as input data. With these input information, the software calculates all the powers involved in the gas side and the values are for a complete burning of fuel.

To perform the mass and energy balances, the flow and temperatures of the several mass flows at the limits of the furnace is needed, as well as the temperatures of the external surfaces and the temperature of the plates passing from the pre-heating chamber to the soaking chamber. These measurements were performed for the plates of higher production per month, which present dimensions of 23 x 63 x 266 mm, with a mass of 3.026 kg and a standard production of 395 pieces/h. During the tests the measurements were performed by the methods and equipment described below:

- total consumption of NG obtained from the station supply of the Sulgás;
- total consumption of NG for 4 burner of the pre-heating chamber obtained by a turbine type flow meter, model-MTL's MTG of Nykon Dwyler;
- outlet temperature of the plates of the soaking chamber obtained by an optical pyrometer, brand Minolta Land, model Cyclops-52;
- inlet temperature of the plates in the soaking chamber obtained by a laser pyrometer Ecil, model PM Plus, through an inspection door located between the pre-heating chamber and soaking chamber;
- inlet temperature of the plates input the pre-heating chamber obtained by the laser pyrometer Ecil;
- outlet temperature of the gases leaving the furnace in the chimney of soaking and pre-heating chambers, measured by a thermocouple type "K" Consistec and a gas analyzer Eurothron, model GreenLine 8,000, which are inserted at the base of the chimney;
- inside temperature of the soaking chamber obtained by a thermocouple type "S" Consistec, installed on the roof of the chamber;
- inside temperature of the chamber pre-heating obtained by the thermocouple type "S" Consistec, installed in the rear side of the chamber;
- amount of the scale obtained by difference in weight of the plate before input the furnace and when it leaves the interior of the same and the scale is removed;
- consumption of combustion air in burner of pre-heating chamber obtained by an orifice plate installed in the air inlet tube coupled a column of manometer;
- total consumption of combustion air for the furnace obtained by measuring the velocity of the air with a fan anemometer Kestrel 4,000 positioned in a square duct of 270 x 270 mm installed in the suction fan. The total air flow consumed by the furnace was obtained multiplying the average speed of two measurement points by the transversal area of the duct;
- admission temperature of combustion air obtained by a digital thermometer inserted in the air inlet tube of the main burner;
- external surfaces temperatures of the furnace obtained by a laser pyrometer and compared with the temperatures obtained by a thermography camera;
- environment temperature obtained by a digital thermometer; e
- analysis of the gases that leaving the combustion chamber using a gas analyzer mentioned before, which the probe is inserted into the base of the chimney.

3. MASS AND ENERGY BALANCE RESULTS

Tables 2 and 3 show the flow of mass and energy that cross the control volume around the furnace analyzed, being computed the values of energy balance, with indications on the percentage of total energy that enter in each chamber.

Table 2. Mass and power balance of the pre-heating chamber

N°	Input	Flow (kg/h)	Temp. (°C)	Power (kW)	(%)
1	Consumption of NG	12.29	27	0,19 (1)	0.10
2	Combustion air	257.25	38	2,74 (1)	1.48
3	Steel plates	996.00	21	2,64 (1)	1.42
4	Combustion			180,65 (1)	97.00
	Total	1,265.54		186,22	100.00
	Output	Flow (kg/h)	Temp. (°C)	Power (kW)	(%)
5	Flue gas	269.54	840.9	93.46 (2)	50.19
6	Steel plates heated	996.00	527	82.55 (1)	44.33
7	Heat loss by walls			6.93 (3)	3.72
8	Other losses			3.28	1.76
	Total	1,265.54		186.22	100.00

(1): Power calculated on the reference temperature (0ºC)
(2): Power calculated by *Acomb* (wet gases enthalpy = 1,248.20 kJ/kg)
(3): Power calculated in relation to environment temperature (23°C), considering convection and radiation.

The values reported in Table 2 show that there is a high amount of energy transported by the combustion gases (5) out of the furnace. The other losses (8) correspond to energy losses by the bottom of the pre-heating chamber (not calculated) and also by the small portion of the gases leaving the inlet of the plates in the furnace. The thermal efficiency of the pre-heating chamber calculated based on the LHV is 44.24%, obtained by: $\eta = \{[(82.55 - 2.64) / 180.65] \times 100\}$, which represents a loss to the external environment of 55.76% of the energy released by burning fuel.

Table 3. Mass and power balance of the soaking chamber

N°	Input	Flow (kg/h)	Temp. (°C)	Power (kW)	(%)
9	Consumption of NG	41.20	27	0.64 (1)	0.10
10	Combustion air	890.00	38	9.47 (1)	1.34
11	Steel plates	996.00	527	82.55 (1)	11.70
12	Combustion			605.63 (1)	85.78
13	Heat of formation of scale			7.75 (1)	1.08
	Total	1,927.20		706.00	100.00
	Output	Flow (kg/h)	Temp. (°C)	Power (kW)	(%)
14	Flue gas	931.20	1,093	406.83 (2)	57.63
15	Steel plates heated	996.00	1,215	223.54 (1)	31.66
16	Heat loss by walls			30.10 (3)	4.26
17	Other losses			45.53	6.45
	Total	1,927.20		706.00	100.00

(1): Power calculated on the reference temperature (0ºC)
(2): Power calculated by *Acomb* (wet gases enthalpy = 1,572.80 kJ/kg)
(3): Power calculated in relation to environment temperature (23°C), considering convection and radiation.

The other losses (17) to the soaking chamber correspond to losses of energy by the bottom of the chamber (not calculated), by radiation loss in the mouth of the discharge and the energy that goes into the chamber for pre-heating through the wall limit.

The thermal efficiency of soaking chamber calculated on the basis of LHV is 23.28%, obtained by: $\eta = \{[(223.54 - 82.55) / 605.63] \times 100\}$, representing a loss to the external environment of 76.72% of energy supplied by the main burner, since the energy contained in the gases leaving the soaking chamber (14) is greater than the energy required to elevate the temperature of the plates from 0°C to 1.215°C (N°15). The overall thermal efficiency of the furnace is around 28.10% $\{[(223.54 - 2.64) / (605.63 + 180.65)] \times 100\}$ i.e., the total waste of energy is 71.90%.

4. PROPOSALS TO IMPROVE THE THERMAL EFFICIENCY OF THE FURNACE

In the energy balance developed in section 3, is identified the low heat efficiency of the furnace and the high percentage of energy lost to the environment. According to these results, three proposals were prepared to improve the use of heat generated in the furnace.

4.1 First proposal

The first proposal is for design improvements of the current furnace, to improve the use of the heat generated in the soaking chamber. The energy which leaves the soaking chamber with the combustion gases is approximately 406.83 kW with a flow rate of 931.2 kg/h at 1,093°C, while the available energy in the pre-heating chamber is only 186.22 kW (smaller than half the energy that leaves the soaking chamber) with a flow of combustion gases of 269.54 kg/h at 840.9°C. So, the energy available at the base of the chimney of the soaking chamber is greater than the quantity necessary to heat the plates from 0°C to 1,215°C which are 223.54 kW (identified in table 3). According to this analysis, it is proposed:

- remove the burners of pre-heating area, to reduce the 17.3 Nm^3/h of NG that are consumed currently by 4 burners;
- unify the chimneys of the soaking and pre-heating chambers, moving it to the inlet region of the material, causing the combustion gases to go the opposite way of the plates, i.e., in counter flow;
- install a regulation damper in the transition between the soaking and pre-heating chambers to control the speed of the combustion gases of the main burner over the plates in displacement through the pre-heating chamber, to increase heat transfer capacity.

These improvements can be seen in Figure 2, which shows the new design proposal for the furnace.

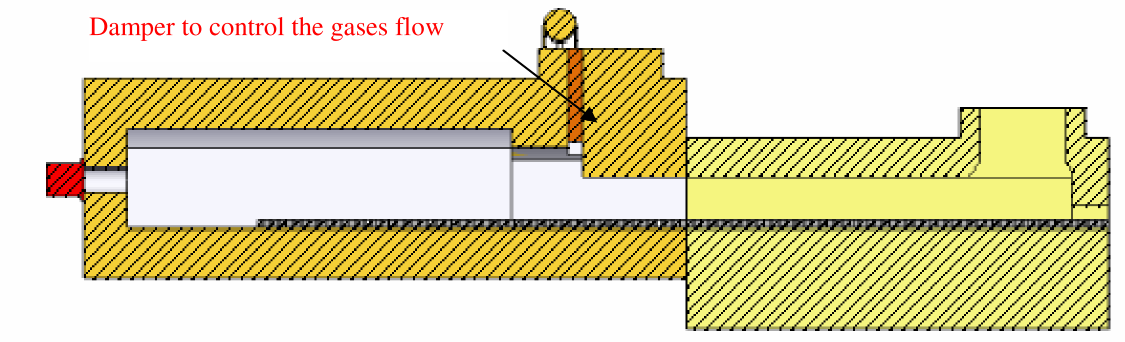

Figure 2. New design for the furnace with control damper of combustion gases and only one chimney

To this new configuration is adopted that the plates will have a gain of 10% more energy compared with the energy absorbed in the current pre-heating chamber, whose value is 82.55 kW. This gain is a function of increased flow of combustion gases at a higher temperature in counter flow with the plates that currently leave the soaking chamber chimney, and with the new design will leave the single chimney (on the right of Figure 2). With the increase of 10%, the energy with which the plates will enter the soaking chamber will be approximately 91 kW. This value together the 10.21 kW relative to losses "Nº 7" and "Nº 8" indicate in Table 2 of the currently configuration that is supposed still exist in the new design, will totalize 101.21 kW. Therefore, discounting the 101.21 kW of the 406.83 kW available at the chimney of the current soaking chamber, the energy that will leave the furnace with wet gases "q_{gases}" in the new design will be of 305.62 kW.

To estimate the temperature "T_g" of the gases entering the single chimney of the new furnace, the same inlet conditions of the main burner to the current soaking chamber are adopted: temperature of the combustion air of 38°C, input temperature of the NG of 27°C, air excess of 30% and the same currently NG consumption of 58 Nm^3/h to obtain more conservative results of investment analysis even adopting the increase of the heat flow rate in the pre-heating chamber. These information were inputted into the *Acomb* software to determine the output temperature of the gases whose enthalpy was 1,180 kJ/kg, obtained from the equation:

$$h_{gases} = \frac{q_{gases}}{\dot{m}_{gases}} \tag{8}$$

where: "$\dot{m}_{gases}$" is the mass flow rate of combustion gases of 0.259 kg/s found in the mass balance in Table 3. Therefore, inserting the value of "h_{gases}" into the *Acomb* software the temperature "T_g" results in approximately 797°C.

With this new design of the heating furnace is estimated the following:

- a reduction of 17 Nm^3/h on NG consumption, that represents a saving of more than R$ 54,286.00 per year. This value is obtained considering the operation time of 13.2 h per day, working 20 days per month, with NG at a cost of R$ 1.008/ Nm^3;
- an improvement on the heat transfer between the combustion gases and the plates, making better use of energy generated, consequently reducing the power of the main burner, but that will not be considered of investment analysis;
- an increase in the furnace thermal efficiency from 28.1% to 36.9% [(223.54 / 605.63) x 100], increasing the share of the energy generated by NG burning that is really used for heating the plates.

The total cost for implementation of the first proposal to improve the heating kiln is R$ 19,931.00.

The feasibility of the proposal is examined by the method of the Net Present Value (NPV). This is considered a sophisticated technique for the analysis of budget capital, just considering the value of money over time. The rate used to discount the input cash, bringing the value to the present time, is the Minimum Rate of Attractiveness (MRA). The MRA is the rate at which the investor is obtaining financial income (Correa, 2002). The MRA of the company where was made this study is 15% per year. To calculate the NPV of each month, the equation used is (Casarotto, 1992):

$$NPV = P + A\frac{(1+i)^n - 1}{i \times (i+1)^n} \tag{9}$$

where: P is the initial investment in R$, A is the contributions of cash in R$ in the period n.

The value of the investment for first proposal is R$19,931.00. As is estimated reduce 4,488 Nm^3/month of NG, it is possible to say that the input for cash to the proposal is R$ 4,523.90 per month. Figure 3 shows the economic analysis with the NPV of first proposal, calculated for 24 months after payment of the investment.

From Figure 3 is observed that the economy with NG, the investment on thee new conception of the furnace it will be paid in the period between 4 and 5 months, i.e., the pay back will be of about 5 months. The NPV at the end of 24 months is R$ 74,238.00 which proves the feasibility of the project.

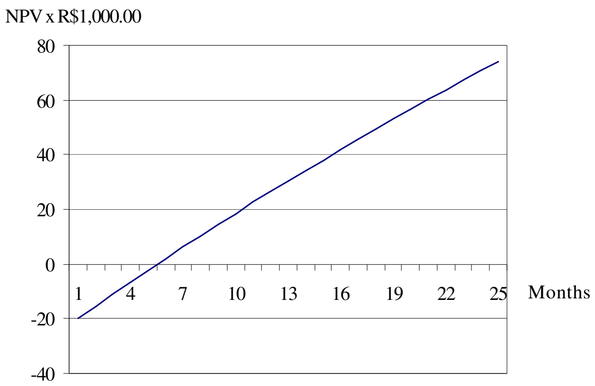

Figure 3. Net Present Value of first proposal for 24 months

4.2. Second proposal

The second proposal aims to analyze the feasibility of installing a recovery heat exchanger to pre-heat the combustion air due to the high temperature and flow of the gases leaving the furnace by the chimney. The heat recuperator includes the installation of a tubular heat exchanger in the exit of the gases, which will be positioned at the end of the furnace according to the new configuration. The combustion gases will pass inside the tubes and the combustion air will pass outside the tubes. The distribution tubes study determined a cross area of 96,762 mm^2. Currently, the soaking chamber chimney cross area is 90,792 mm^2. Figure 4 shows a 3D figure of the recovery heat exchanger designed.

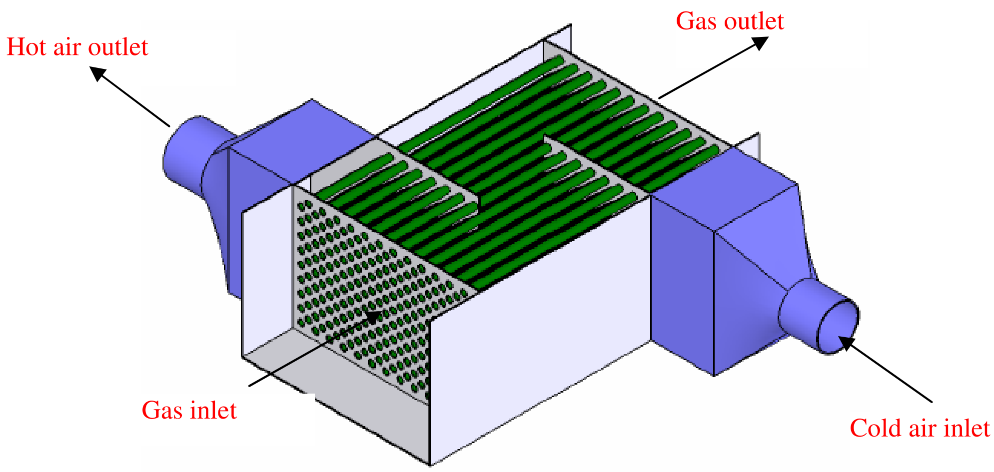

Figure 4. Configuration of the recovery heat exchanger proposed

The heat exchanger is composed of 225 steel tubes AISI 309, placed in triangular arrangement, with 25,4 mm of external diameter, 1 mm of wall thickness, 1m of length, and the total area used in the sizing is 17.245 m^2 , refered to mean diameter of the tubes. It is estimated that the heated air exits the heat exchanger at a temperature of 275°C, and assuming a heat loss of 10% between the heat exchanger tubing and the main burner, the arrival temperature of the air will be 250°C. The maximum working temperature indicated by valve and electronic components suppliers is 300°C, and so a safety temperature at the exit of heat exchanger of 275°C was admitted.

The head loss in the passage of the gases inside the pipes was calculated in 2.076 mmH_2O, and as the combustion system of the furnace was sized for an internal pressure of the combustion chamber of 5 mmH_2O, it appears that the exchange will not harm the circulation of the furnace nor will increase its internal pressure. The head loss in the air stream outside the tubes was estimated in approximately 23.85mmH_2O, which can be considered low compared to the 900 mmH_2O of total pressure at a flow of 972 Nm^3/h of the centrifugal fan used in the plant. The estimated value of the overall heat transfer coefficient "U" to the configuration of Figure 4 is 8.52 $W/m^2.K$, this value is about 14.8% smaller than the "U" minimum find in the literature and hence, the percentage of difference is acceptable, because some correlations are used. Incropera and Dewitt (2003) and Kreith and Bohn (2003) reported values of the coefficient "U" for several different conditions of fluid flow and interacting in a heat exchanger. Particularly to gas-gas heat exchanger with both fluids in flow, (Kreith and Bohn, 2003) mention the value of "U" between 10 and 30 W/m^2 K.

By the method of ε – *NTU* an effectiveness value of 32.56% was obtained, which is the ratio between the actual rate of heat transfer (63.39 kW) and the maximum rate of heat transfer (194.68 kW) possible in the heat exchanger. The temperature of the gases leaving the heat exchanger, "th, $_o$," according to Kreith and Bohn (2003) is:

$$T_{h,o} = T_{h,i} - \varepsilon \Delta T_{max} = 545\ °C \tag{10}$$

The ratio between the rates of heat capacity of the hot and cold fluids "Cr" is 0.8596. Using the relationship for *NTU* of heat exchangers with cross flow obtained from Incropera and Dewitt (2003):

$$NUT = -\left(\frac{1}{C_r}\right) \times \ln\left[\left(C_r \times \ln(1-\varepsilon)+1\right]\right. = 0.4809 \tag{11}$$

With the values of *NTU*, ε and C_{min} (251.52 W/K), the heat exchanger "U" value can be obtained which is 7.015 $W/m^2.K$. The percentage difference between the "U" found by the calculations of 8.52 $W/m^2.K$ and the "U" of 7.015 $W/m^2.K$, necessary for the heat exchanger heating the air from 23 to 275°C is 21.45%. This difference assures that in practice the heat exchanger heats the air up to the desired temperature.

With the installation of the recovery heat exchanger, associated to the heating furnace configuration change is estimated the following:

- an energy gain that entries the furnace with the heated combustion air at 250°C in relation to the current configuration at 38°C, i.e.:

$$q_{actual} = \dot{m}_{air} \times c_{air} \times (T_{actual} - T_{old}) = 0.2473 \text{ x } 1.01718 \text{ x } (250 - 38) = 53.33 \text{ kW} \tag{12}$$

- a reduction of 5.1 Nm^3/h in the NG consumption, representing a saving per year of R$17,056.00;
- an increase of the thermal efficiency of the new configuration of the furnace to 40.48%, i.e.,

$$\eta_T = \frac{q_{ap}}{q_{gas} - q_{actual}} \times 100 = \frac{223.54}{605.63 - 53.33} \times 100 = 40.48\% \tag{13}$$

The value of investment for the 2nd proposal is R$21,405.00. As it is estimated a reduction of 1,410 Nm^3/month of NG, the cash input to the proposal is R$1,421.00 per month. Figure 5 shows the NPV of the second proposal, calculated for 24 months after payment of the investment. It appears from Figure 5 that due to the economy obtained on the NG consumption, the pay back in the investment of the heat recovery exchanger will be of 17 months. The NPV at the end of 24 months is R$ 8,174.00 which proves the feasibility of the project, but with less significant gains relatively to the first proposal.

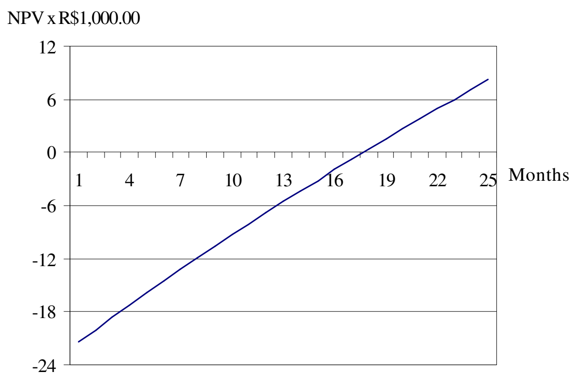

Figure 5. Net Present Value of second proposal to 24 months

4.3. Third proposal

This third proposal assesses the thermal insulation of the soaking chamber and compare with different configurations and refractory materials. The temperatures and the heat lost by convection and radiation to the environment through the walls of the current soaking chamber are shown in Table 4. In the current configuration of the furnace thermal insulation is made with insulating firebrick with approximately 65% of Al_2O_3 and density 1.50 g/cm^3 and a plate of calcium silicate. This type of brick is fragile to the system of intermittent work of the furnace (turn- off) and has little resistance to erosion caused by gases at high temperature.

To evaluate the different configurations of the thermal insulation of the soaking chamber was used the software available on Togni S/A (2008). From this software, the externals wall temperature of all situations is obtained.

Table 4 - Information from each surface of the current soaking chamber

Surface of the furnace	Area (m^2)	Mean Temperature (°C)	Power Loss (kW)	Environment Temperature (°C)
Roof	7.13	146	15.35	23
Rear wall	2.264	101	2.33	23
Lateral front	7.47	97	7.18	23
Lateral back	7.47	81	5.25	23

In table 5, the external wall temperature, the total thickness (sum of thicknesses of all components that are part of the wall) and also the cost of acquisition per square meter of the material are presented for each composition of the wall. That is, the compositions layers of the walls are as follows, corresponding the N°1 for the roof and the other to the laterals walls:

- N°1, composed of a refractory brick (72% of Al_2O_3 and thickness, e=152mm), an aluminous cement with low cement content (e=25mm), a 0.8 g/cm^3 insulating firebrick (e=114mm) and a ceramic fiber blanket (128 kg/m^3 and 50mm);
- N°2, composed of a refractory brick (72% of Al_2O_3 and e=229mm), a 0,8 g/cm^3 insulating firebrick (e=114mm) and a ceramic fiber blanket (128 kg/m^3 e=50mm);
- N°3, composed of a refractory brick (72% of Al_2O_3 and e=114mm), a 0,8 g/cm^3 insulating firebrick (e=229mm) and a ceramic fiber blanket (128 kg/m^3 and e=50mm);
- N°4, composed of a refractory brick (72% of Al_2O_3 and e=229mm), a 0,8 g/cm^3 insulating firebrick (e=76mm) and a ceramic fiber blanket (128 kg/m^3 and e=50mm); and
- N°5, consisting of a refractory brick (72% of Al_2O_3 and=229mm), a 0,8 g/cm^3 insulating firebrick (e=63mm) and a ceramic fiber blanket (128 kg/m^3 and e=50mm).

Table 5 - analysis of different configurations of thermal insulation to the soaking chamber

Configuration	Total Thickness of The Wall (mm)	Cost (R$/m^2)	Wall temp. (°C)
N°1	**341**	**2,139.00**	**81**
N°2	393	2,210.00	90
N°3	**393**	**1,548.50**	**79**
N°4	355	2,122.00	96
N°5	342	2,124.50	101

Regarding the values reported in Table 5, it is observed that for the walls of the soaking chamber the configuration "Nº 3" is more indicated, because even presenting the lowest temperature of cold face and having the greatest total wall thickness it presents the lower cost of deployment. The energy lost by convection ($\bar{h}$ = 4.98 W/m^2.K) and radiation (ε = 0.9) through the external walls of the furnace for this configuration is 4.8 kW and 6.74 kW respectively totalizing 11.54 kW. The energy loss was obtained by the methods and equations presented in section 2. This configuration shows a reduction of approximately 22% of the current loss of energy which is of 14.76 kW.

The configuration N°1, proposed to the roof of the soaking chamber, presents a decrease of 44.5% compared to the current temperature of 146°C. The energy lost by convection ($\bar{h}$ = 6.49 W/m^2.K) and radiation (ε = 0.9) through the roof of the furnace for this configuration is 2.68 kW and 2.92 kW respectively, totalizing 5.6 kW. This value indicates a reduction of approximately 63.5% compared to the current energy loss of 15.35 kW.

5. CONCLUSIONS

The energy balance implementation concluded that the current heating kiln presents a low thermal efficiency, especially in the soaking chamber, where over 57% of energy from the combustion leaves the chamber by combustion gases to the chimney, while to the pre-heating chamber the energy loss is more than 50%. It is identified some deficiency on the kiln design, as volume, size of chamber and position of chimneys that difficult the heat exchange between the gas and plates, reducing the thermal efficiency.

The first proposal to change the design of the furnace can generate a NG savings of more than R$ 54,286.00 per year, with a pay back of 5 months, not being considered in this value the maintenance costs reduction and the heat loss in the current soaking chimney. The thermal efficiency of the furnace for the new design was estimated at 36.9%. The second proposal, which includes the installation of a recovery heat exchanger in the furnace new design chimney, can generate savings of R$ 17,052.00 per year in NG, with a pay back of 17 months and the thermal efficiency of this new conception can reach 40.5%. The NPV methods allowed the two proposals, however, indicating a faster return on investment in the first.

Another deficiency is the high temperatures recorded on the external surface of the soaking chamber that wastes energy and harms the working environment. The analysis of various configurations for the side walls of the furnace showed that "N°3", (Table 5) has more efficiency and lower cost of deployment. For the roof, that presents the greatest loss of energy is proposed the configuration "N°1", in which there is a reduction of approximately 63.5% (from 15.35 kW to 5.6 kW) for the energy lost to the current configuration.

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7. RESPONSIBILITY NOTICE

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