THERMAL DESIGN OF INDIRECT HEATED OVENS WITH NATURAL DRAFT

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Abstract. *Indirect heating in industrial ovens prevents direct contact of the combustion gases with the load deposited in the cavity of the oven. This is important, for instance, to the food industry, where flue gases may contain carcinogenic substances. In conventional industrial ovens with natural draft, a heat exchanger is used between flue gases and air present in the cavity of the oven. The riser tubes that draft flue gases from the combustion chamber also works as a heat exchanger, since a fan forces external convection with air inside the cavity. This configuration frequently implies in a compromise between heat exchange and combustion quality. To find more effective heat exchanges without an investment in heat exchange surfaces of prohibitive costs, it's necessary augment the pressure drop. Meanwhile, with heat exchanges improved, the natural draft force of the riser tube diminishes, in result of the reduction of mean temperature of the gases. In this work, were implemented a simulation model of oven with this characteristics, based on analytical and empiric correlations for the calculus of the combustion estequiometry, pressure drop and thermal exchanges in tube bundle. Also through a similar model, is proposed the use of furcated thermosyphon as a device of exchange that reduces the effects of that compromise between heat exchange and combustion quality.*

Keywords: *bakery oven, indirect heating, mathematical modeling, heat exchanger*

1. INTRODUCTION

Conventional industrial ovens with natural draft and indirect heating protect the load deposited in the chamber of the oven from the combustion gases. This type of oven prevent the contamination of the products put into the cavity, needful for the foods industry, preserving the quality of the products.

In this equipment, is used an adiabatic combustion chamber with natural draft performed by small riser tubes, see Fig. 1. The heating of the oven is done by this small riser tubes inserted in the interior of the cavity, where a ventilation system improve the heat exchange of the air in the chamber with the flue gases in the riser tubes.

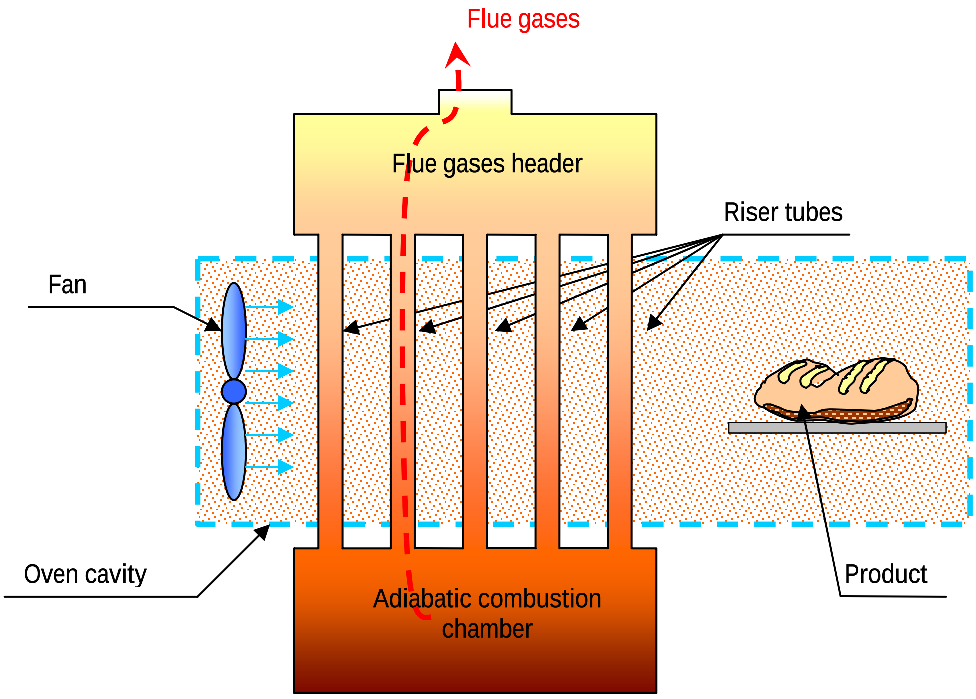

Figure 1. Schematic diagram of the riser tubes, combustion chamber and fan.

Some attention is necessary because the draft of the riser tube depends directly of the specific volume of the gases in the entrance and in the outlet of these riser tubes. How much

1.1. OBJECTIVE

The objective in this work were create a thermal model for ovens, based in numerical, analytical and empiric correlations, for use this model as a tool for designing, in the way to get the best compromise between heat exchanges, riser tube draft, pressure drop and combustion estequiometry.

2. METODOLOGY

It was elaborated a thermal model based in existing correlations in literature and selected an existing oven for the case study. Using the experimental data of the oven, the model was calibrated. Based in the calibrated model, modifications were discussed in this oven.

The thermal oven look for evaluate the potency of heat exchanger and the air excess from a design proposed for the combustion chamber (potency, fuel consumption) and for the riser tubes (diameter, length, number of riser tubes and setting)

This model was implemented on EES package (Engineering Equation Solver) to do a complete thermal analysis, through empiric correlations and analytical equations. The Fig. 2 illustrates a diagram with the steps taken in the calculation of the mathematical model.

The selected oven for case study is made in Brazil and used in food industry. It was accomplished tests in laboratory to evaluate the oven performance, and to check the mathematical model. Among other performance parameters characteristics LEVANTADAS in laboratory, it were determinated the air excess and the composition of the flue gases. The experiments done won’t be described in this article.

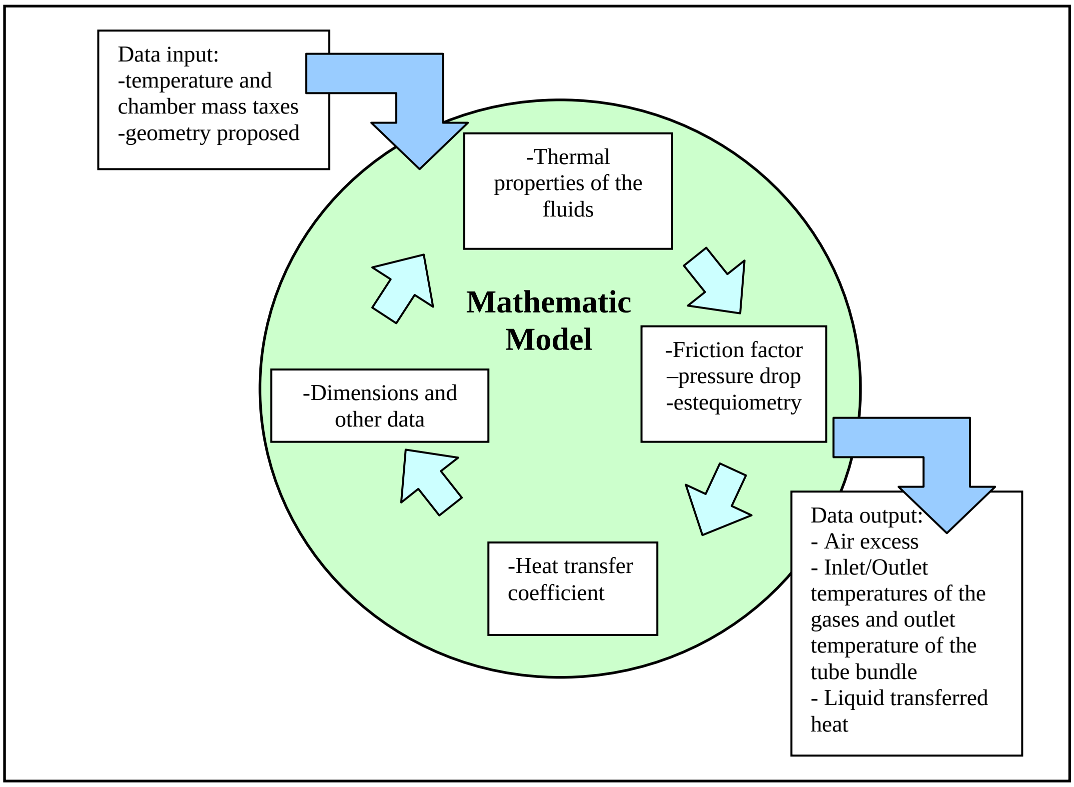

Figure 2. Step diagram.

2.1. EQUIPMENT DESCRIPTION

The selected equipment for case study is a foods industry oven. In the thermostat, it’s possible to adjust temperatures from 50 to 250 °C. The oven counts with a PLC control module that accomplishes the operation with LPG (Liquefied Petroleum Gas) and controls security devices.

In the opposite side to the cover of the oven are concentrated the devices for combustion and heat exchange, as display Fig. 3.

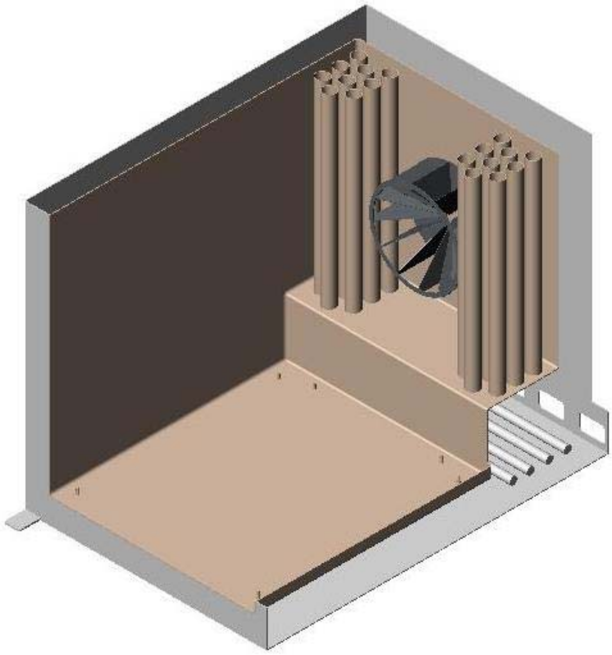

Figure 3. Place of heat exchanger, fan and burners.

2.2. THERMAL MODEL

The thermal model considers the heat exchange modeled as a shell and tubes heat exchanger type with cross flow, as well the estequiometry and riser tube draft. This model connects the most important phenomena that occurs in the oven during its operation.

Some of the assumptions are:

- Complete mixture of air excess and flue gases in the combustion chamber.
- Uniform distribution of flue gases in all the ascension ducts.
- The combustion quality is evaluated in function of resulting air excess. However, for the estequiometry calculus, complete combustion is assumed.
- Steady state conditions.
- Thermosyphon internal resistance negligible.
- Uncompressible viscous flow inside the ducts.
- It was considered heat loss all the energy that won't be transferred to the flue gases. Consequently, the energy transferred directly from the combustion chamber to the cavity is also considered heat loss. The calculation of this heat loss is necessary only for determinate the burning efficiency.

The analysis was based in a tube-shell conventional heat exchanger, using the LMDT method. This method is the most appropriated because it has the known of the inlet and outlet temperatures of the hot side (flue gases in the riser tubes), and the inlet temperature of the cold side (in the cavity). The outlet temperature of the hot side can be set by the energy balance equation.

The used equations for the GLP combustion are the same of Borges (1994) in a model that can use other fuels besides this gas, including solid fuels, and it models the complete combustion estequiometry, if necessary. For this case, it was assumed that the combustion would be complete anyway. The combustion quality is evaluated in function of the resulting air excess. It was considered that air excesses lesser than 1,5 would generate incomplete combustion for the type of burner (flute) used in the oven.

The mathematical modeling for the pressure drop in ducts was gathered from Borges (1994) and the drag coefficients were obtained from tables of ASHRAE FUNDAMENTALS 1997.

It was used a model of external convection in tube bundle with known geometry and with correlations obtained form Zukauskas (1972, apud Lienhard, 2000) where the convective heat transfer coefficient of a single tube depends of the geometric configuration of the whole bundle. In case of a few tubes, correction coefficients are used.

In this case, the used correlation has validity for Reynolds numbers between 10^3 and $2x10^5$. For values of Reynolds number under 10^3, the thermal heat exchange would be ineffective, and for values above $2x10^5$ it's practically impossible to have under conditions of operation of this type of oven.

As it is treated of a foods oven, the use of fins in the tubes (riser tubes) is not possible for hygiene criteria, because, the oven needs to be cleaned after the use, for removal of the possible incrustation coming from the foods cooking. This question needs a smooth surface without sharp edges.

The model uses correlations for internal convection heat exchange in tubes interior (see Fig. 4), separated for laminar and turbulent boundary layers, as well for fully developed internal flow and entrance flow, depending of the conditions of the flow given by the pressure drop model and riser tube draft.

The equation for radial heat conduction in tubes with steady state conditions were used in the model.

The thermal resistances determination is necessary to the net heat transferred calculation. The internal convective resistances, external convective resistances and conduction resistances can be calculated as Eqs. (1), (2) and (3), respectively.

$$R_{internal,conv} = \frac{1}{h_{internal,conv}.\pi.d_i.l} \tag{1}$$

$$R_{external,conv} = \frac{1}{h_{external,conv}.\pi.d_e.l} \tag{2}$$

$$R_{conduction} = \frac{\ln(r_2/r_1)}{2.\pi.l.k} \tag{3}$$

Where $h_{internal,conv}$ is the internal convection coefficient, $h_{external,conv}$ is the external convection coefficient, d_i is the internal diameter of the riser tubes, d_e is the external diameter of the riser tubes, l is the riser tubes length, r_2 is the external radius of the riser tubes, r_1 is the internal radius of the riser tubes and k is the metal conductivity of the riser tubes.

The overall heat transfer coefficient $U.A$ is obtained by the sum of the inverse of the thermal resistances, as Eq. (4), the Fig. 4 shows the thermal resistances and the geometrical data of the riser tubes.

$$U.A = \frac{1}{\sum R_t} \tag{4}$$

Where R_t is each one of the thermal resistances discussed previously.

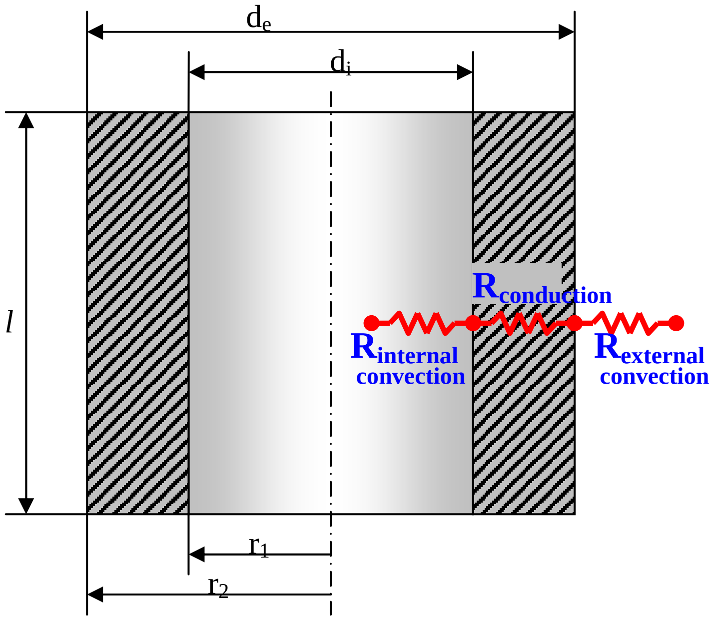

Figure 4. Schematic view of the riser tube and thermal resistances.

In this case, the thermosyphon internal thermal resistances of boiling, condensing and conduction were not considered. The thermosyphon work fluid is water, and it has a small thermal resistance, so, for pre-design effects they are negligible. These internal resistances are of the magnitude of 10^{-3} K/W.

3. EQUIPMENT TESTS

The equipment tests were accomplished in the Heat Pipe Laboratory (Laboratório de Tubos de Calor - LABTUCAL), in the department of Mechanical Engineering at UFSC (Federal University of Santa Catarina). To adequate the Liquified Petroleum Gas (LPG) supply to the oven, a LPG line duct with 2 cylinders with 45 kg in parallel were installed. During the operation with gas, the door of the tests room was keep opened with an extractor fan to renovation of the ambient air.

4. RESULTS

4.1. THE BASIS CASE

The oven in its original geometry have an internal diameter of 38,1 mm (1½ in), with length approximately of 0,686 m, the tube bundle is arranged at 30°, and its transversal pitch is approximately 57,75 mm. The material is stainless steel and the riser tubes thin is 1,5 mm. The heat exchanger is compound by two bundles of tubes, the width is 0,246 m. Inside each one of the riser tubes there was a twisted metal ribbon to raise the pressure drop and improve the heat transfer.

With cavity inlet temperature of 200°C (573,15 K), by tests, the average air excess measured was 167%, or E=2,67.

Through the analysis by the mathematical model, the air excess obtained were 167,5% (E=2,675), so it can be told that the model is reliable and precise, with an error approximately of 0,2%. The Fig. 5 shows the inlet/outlet data screen of the EES implemented model.

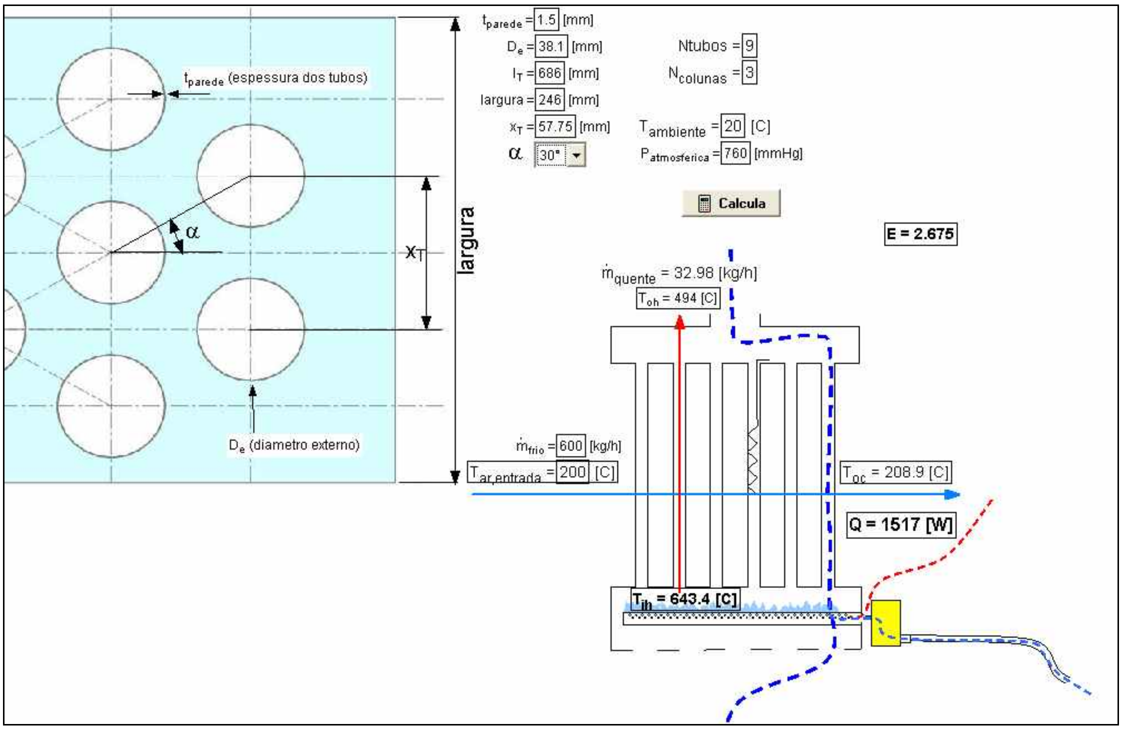

Figure 5. Input/output data screen of the mathematical model.

All the analysis of the system were based on only one of the heat exchangers, because the 2 exchangers have the same operation conditions. So, to determinate the heat transferred, just multiply for 2.

4.2. DESIGN IMPROVEMENT

A first improvement can be done by reduction of the riser tubes external diameter to 25,4 mm (1 in), avoiding the use of the pressure dropping device (ribbons), therefore, to improve the thermal performance, are added more tubes, consecutively the transverse pitch reduces.

The new bundle configuration, with 15 tubes of 1 in (25,4 mm) without the pressure drop devices, had a substantial improvement in the thermal performance without commit the air excess, see Fig. 6.

Through the plot in Fig. 7, we can see the number of tubes effect on air excess, and with the plot in Fig. 8, we see the net heat as a function of the number of tubes.

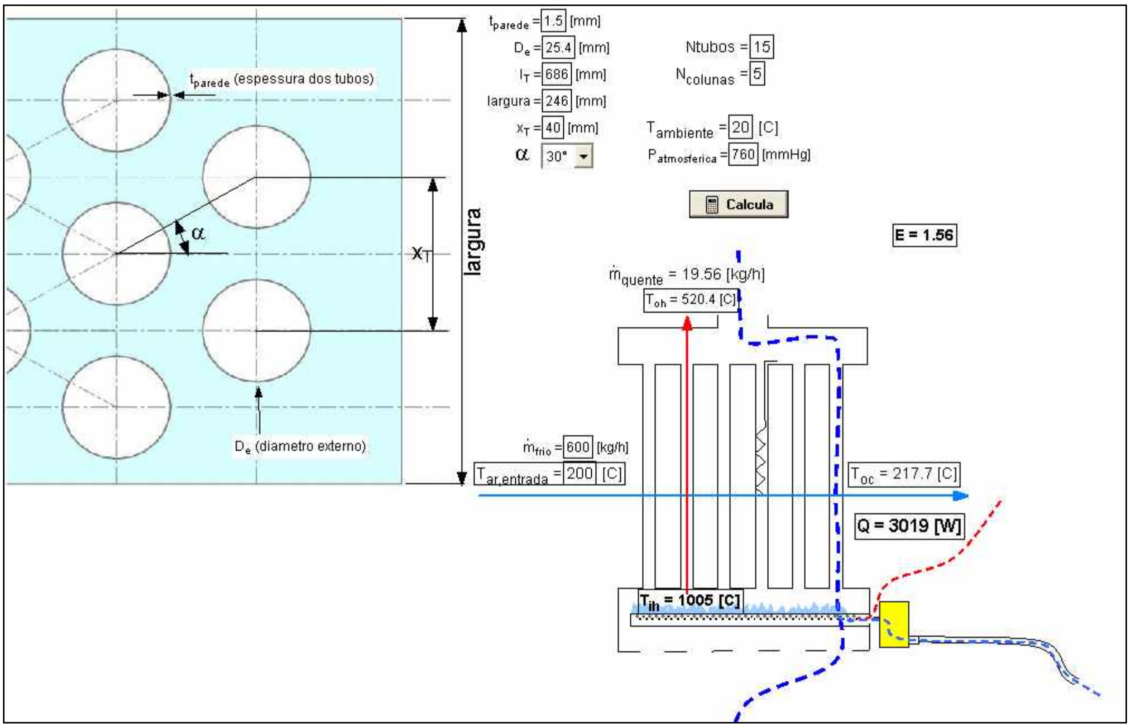

Figure 6. Proposed configuration for the tube bundle.

With this setting, it is observed that the supplied heat to the chamber's air have its value doubled, and the air excess falls significantly.

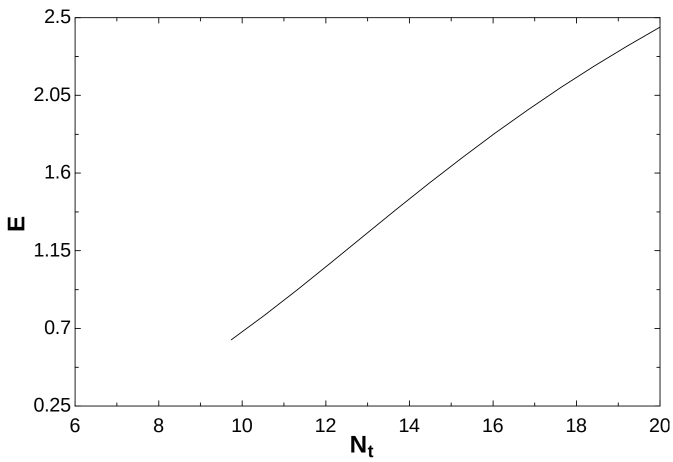

Figure 7. Diagram of air excess versus number of tubes, where E is the air excess and N_t is the number of tubes.

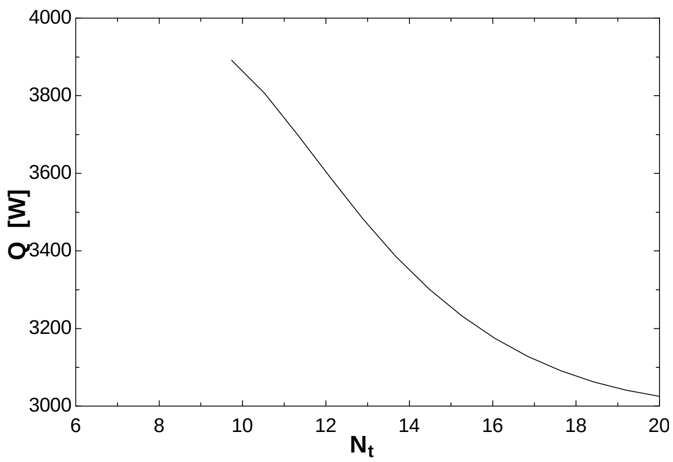

Figure 8. Diagram of heat transferred versus number of tubes, where Q is the heat transferred to the chamber and N_t is the number of tubes.

4.3. THE MODEL WITH THERMOSYPHONS
O MODELO COM TERMOSSIFÕES

In this situation, it was used the same number of tubes of the original (standard) configuration, just for saving tubes. Thus, in the condenser side, we have nine tubes arranged at 90° (aligned). These tubes are organized in three columns with three tubes, each one column is linked to a single evaporator, then it will be three evaporators.

This type of lay out also can be found in literature, called branch thermosyphon, or furcated thermosyphon.

The Fig. 9 shows the input/output data interface of the implemented program.

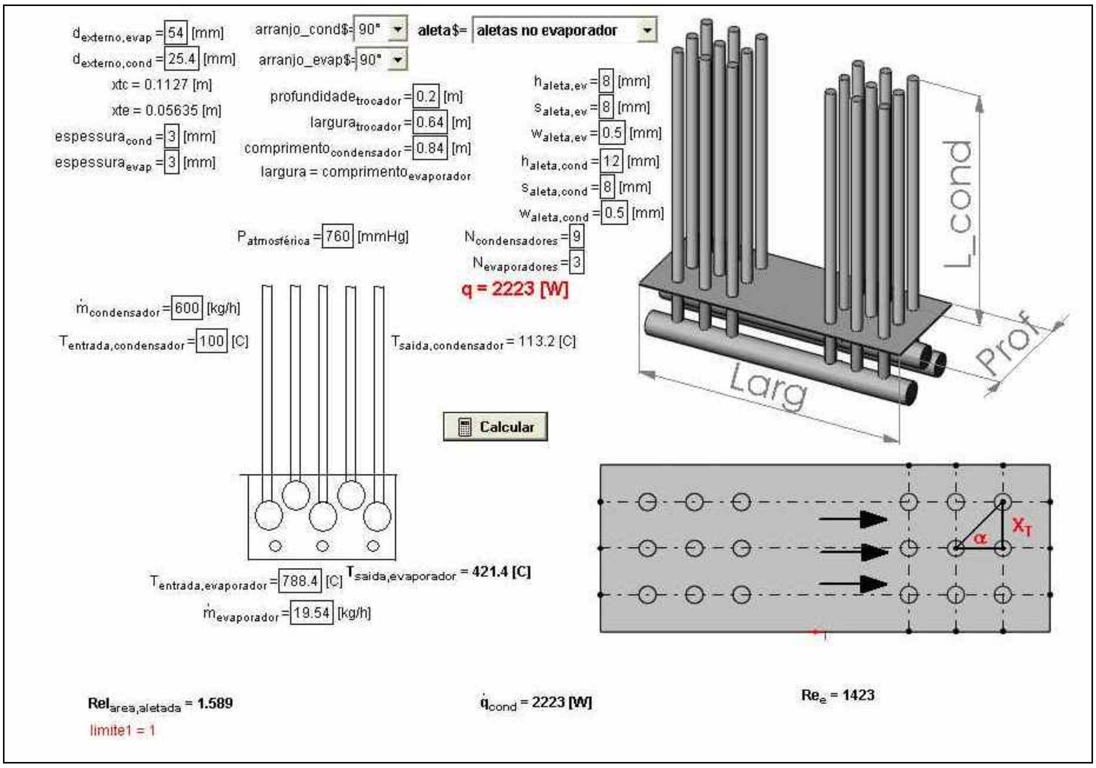

Figure 9. Input/output data screen of the oven with thermosyphon model.

In this study with thermosyphon, it's observed that the riser tube, consecutively the natural draft, does not depend necessary of the heat exchanger. The riser tube can be situated after or before the heat exchanger.

The natural draft buoyancy force from the chimney is proportional to the density diference between ambient air and chimey gases, therefore proportional to the temperatures diference. The greater the heat exchange, the lower is the chimney buoyancy force, when the chimney is installed after the heat exchange section. Good results are obtained when a small chimney is installed after the combustion chamber and before the heat exchanger.

The limiting factor is the evaporator's area, because in the condenser a turbulent regime is preserved by the fan and still having a great heat exchange area. For the evaporator the use of fins is necessary, reminding that the evaporator isn't in the cavity of the oven.

Is observed that the transferred heat is highest than the standard configuration, for the same conditions of operation. The question of natural draft is resolved with the correct design of the riser tube, insuring the gases flux necessary to the proposed model.

5. RECOMMENDATIONS AND FINAL CONSIDERATIONS

The program considers several physical phenomena that rule the heat exchange between the riser tubes, the oven cavity, the combustion chamber and external air. As the mathematic model was checked with measurements made in a real oven and the error shown was small, is evidenced the precision of the model for analysis and design of oven with the same working principle.

In the analyzed case is observed that a reduction of the tube diameter results in an increase on the pressure drop, resulting in an improvement on the operation conditions. The optimization can be implemented in the mathematical model to get better configurations for a given oven, considering aspects like net heat transferred, pressure drop and costs.

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