MODERN ANODE BAKE FURNACE DEVELOPMENTS

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SUMMARY

Improving the efficiency and the performance of anode baking furnaces is of paramount importance in producing prebaked anodes. In order to predict the properties of a design modification, a theoretical model has been developed to simulate the baking curve of a furnace. To visualize the gas flow in the flues a scale model of the flues was built.

Introduction

In the production of prebaked anodes, two major steps can be distinguished: The production of the so-called green anodes, and the baking process. From the technical point of view both steps are more or less equally important: substandard green anodes will never result in good calcined anodes; on the other hand, inadequate heat-treatment of even the best green anodes will also result in poor anode performance.

From the economic point of view, however, baking is much more important than the manufacture of green anodes: energy consumption is two to three times higher; depending on the degree of automation, the labour costs in the baking step can also be about twice the labour costs required for making the green anodes. About the same ratios apply to other cost elements, such as maintenance and depreciation.

Anodes may be baked in different types of furnaces : apart from a small amount of anodes calcined in tunnel furnaces, most anodes are heat-treated in either open or closed ring-type furnaces. Aluchemie operates six open ring-type anode baking furnaces, each with a design capacity of 38,000 metric tons per annum. For this reason, this paper will discuss developments in open type furnaces only. Considering the economic impact, it is easy to understand that various papers have been presented describing methods to improve furnace performance. Of particular interest are the articles by B.J. Racunas, describing the thermal balance of an open ring-type furnace (1), M. Adams, describing a system of computer control (2), H. Onder, discussing the energy consumption (3), and A.N. Stuart, discussing the use of fluid coke as a packing material (4). All these papers deal with existing furnaces that remain unchanged in essence. The rebuilding of a furnace, however, offers the possibility of realizing modifications to the design of a given furnace. Introducing such modifications by means of full scale "trial and error" entails an unacceptable risk. In order to be able to predict the effect of such modifications in advance, investigations concerning the properties of a furnace were initiated. In a first step, we restricted ourselves to the following major aspects:

1. optimum flue design to ensure homogeneous heat transfer, and
2. influence of furnace geometry on the heating curve.

These problems are discussed in the following sections.

Homogeneity of the temperature distribution in a flue

Our objective was to find such a flue design, that all parts of the packed anodes are heated in the same way. A homogeneous heat transfer cannot be expected if the velocity distribution of the hot flue gases is very irregular.

Therefore, a series of experiments was started, partly in cooperation with the Netherlands Organization for Applied Scientific Research (TNO), to visualize the velocity distribution in a flue. This was done with a horizontal model of the flue, in which a layer of water flows to simulate the actual gas flow. It appeared that this method is very well suited to discover important differences in the flow pattern in various flues. As an example, figures 1 and 2 show the velocity distribution in two flues of the same overall dimensions, with three baffles in the case of figure 1 and with five baffles in the model shown in figure 2.

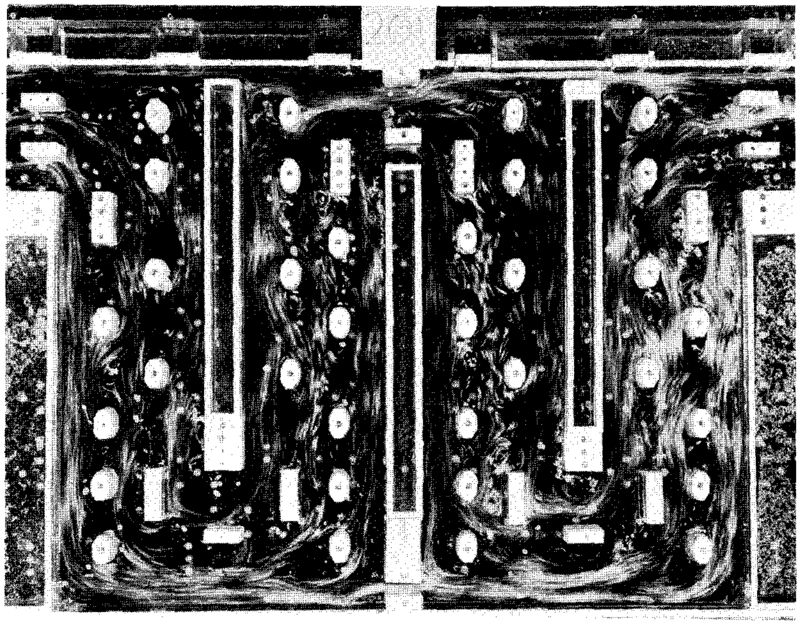

Figure 1 : Model of a flue with 3 baffles.

The figures (each) show a complete section with all the baffles and supporting bricks, with a scale factor of about 0,2. The traces consist of small pieces of white paper floating on the water surface. With exposure times of about 0,5 seconds flow direction and velocity in different places can be made clearly visible. Overall gas flow direction is in all cases from the right to the left.

It is difficult to judge whether an optimum design in a 2-dimensional model will also mean optimum performance in a full-scale flue, considering the fact that we have not succeeded in making the Reynolds figures in the model and in the full-scale flue exactly equal. To solve that problem, two full-scale flues were built, the first with normal refractory and the second with one transparent side wall. With the aid of this set-up, we were able to prove that the results obtained with the water flow model correspond quite well to the flow pattern in a full-scale flue.

Our experience is that the above-mentioned experiments can serve to distinguish between good and poor flues, but that they do not give sufficient information to distinguish between good and better, or poor and worse ones.

We have to take into account, that there are different heat sinks in different locations. In order to find the optimum design, measurements of the temperature distribution have to be made, either in an experimental set-up or in practice. With a given temperature distribution, the model can be very helpful in the search for configurations that will "push" the energy towards places with insufficient temperature.

The conclusion is, therefore, that the model is very helpful in comparing quite a few different flue designs within a short period of time at low cost.

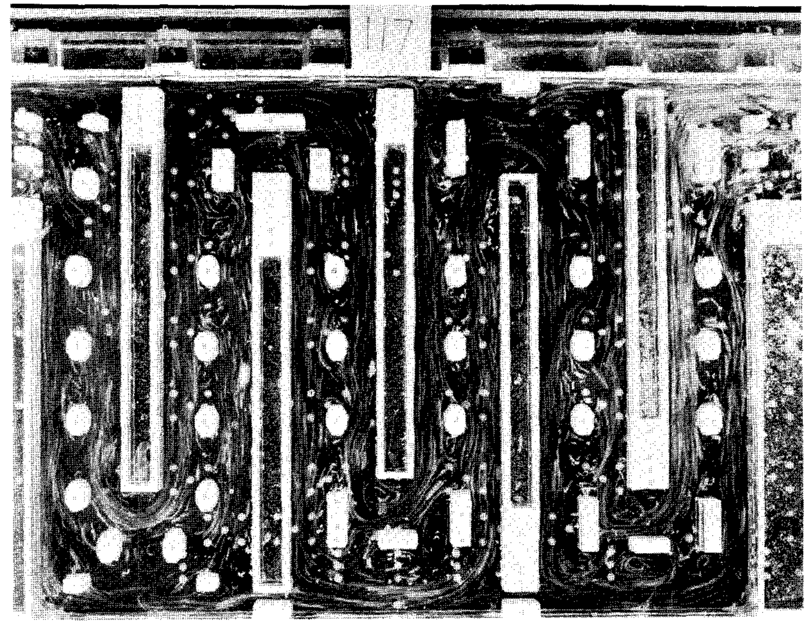

Figure 2 : Model of a flue with 5 baffles.

Simulation of a baking curve

Introduction

It can be expected that the baking curves of two points situated at a distance of one pit length from each other are the same, apart from being shifted over one fire cycle.

Our assumption is, that the same can be said for any two points: we assume that the baking curves of two points situated a distance D apart, are shifted over a time D/U. Here, U is the "furnace velocity":

$$U = \frac{\text{pit length}}{\text{fire cycle.}} \qquad (1)$$

In figure 3a three measured baking curves are shown. From figure 3b it can be seen that they coincide fairly well, if they are shifted over the proper time. This means that a furnace can be considered as a continious heat exchanger, in which flue gases flow in the negative direction and the furnace "flows" with velocity U in the positive direction.

In our mathematical model, the so-called furnace flow is divided into nine layers as shown in figure 4, each layer being characterized by a mean temperature. The heat exchange between these layers and the flue gases can be approximated with some proper differential equations, supplemented with an overall heat balance. Additionally, some relations are needed to deal with phenomena such as, for instance, air leakage and pitch evaporation. These relations were found by postulating model equations and determining the coefficients in these equations experimentally.

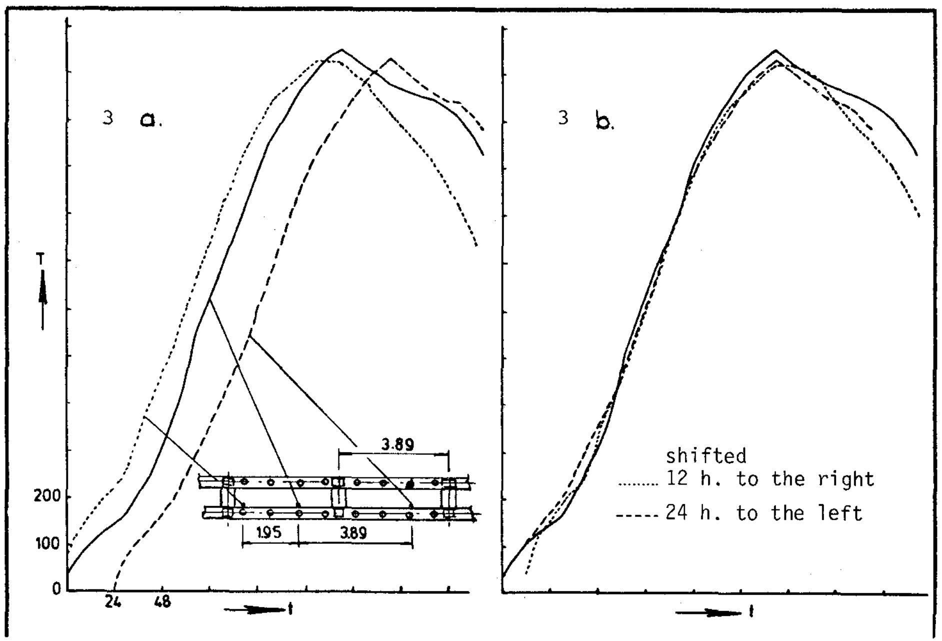

Figure 3 : Some baking curves, measured at different positions in a furnace.

1. first anode layer
2. second anode layer
3. packing material
4. flue wall
5. baffles and support bricks
6. bottom layer (anode)
7. bottom layer (flue)
8. top layer (packing material)
9. top layer (flue)

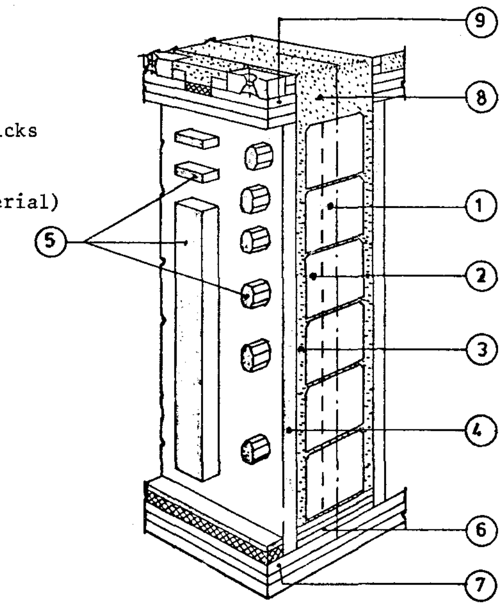

Figure 4 : Division of the "furnace flow" into nine layers.

The basics of the calculations are shown in figure 5 : anodes, packing material and refractory bricks as shown in figure 4 are replaced by a package of parallel layers each with a different thickness and with different thermal proporties. In order to obtain more accurate results, the anode layer is split up into two layers. It can be shown that the behaviour of the temperature in a given layer is governed by an equation of the type

$$C_i \frac{dT_i}{dt} = A_i \left(\bar{T}_{i-1} - \bar{T}_i\right) + B_i \left(\bar{T}_{i+1} - \bar{T}_i\right) \qquad (2)$$

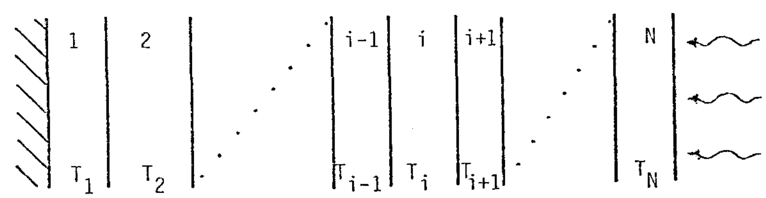

Figure 5 : Schematic model of the furnace.

T_i is the mean temperature of the layer with the same index and A_i, B_i and C_i are constants that are dependent on the thicknesses and on the thermal proporties of the layers with the indices i-1, i and i+1.

To calculate the influence of the other parts of the furnace, that is of the bottom and the top layers, is quite difficult. Some primary calculations, however, have shown that in a "normal" furnace with sufficient insulation of the bottom and a top layer of sufficient thickness, the overall influence of these parts is not too great. Therefore, it seems reasonable to simulate a few subordinate effects by means of comparatively simple equations rather than making the overall calculation much more complicated. Especially for the study of transient phenomena it would be necessary to modify the approach in such a way that the furnace is split up into a much larger number of different sections. However, this would mean that the range of necessary calculations would be much longer. Tests with the model described in the next paragraph make it clear, that for our purposes the simplified model is good enough. Another advantage is, that all the calculations including the graphic print-out of the results could be executed with a rather small desk computer of the Olivetti type P 6060.

Boundary conditions and calculation procedure

Figure 6 shows a simplified result of the calculations. The meaning of the different curves is:

- T_a : the mean temperature of the anode
- T_f : the temperature of the flue gases
- M : the mass flow of the flue gases
- P : the vacuum or pressure in the flue
- Q : the heat flux obtained by the burning of natural gas.

The result as shown in figure 6 represents a certain baking furnace defined by some eighty parameters such as : dimensions of the flues, thermal properties, pitch content in the anodes, drag coefficients, etcetera.

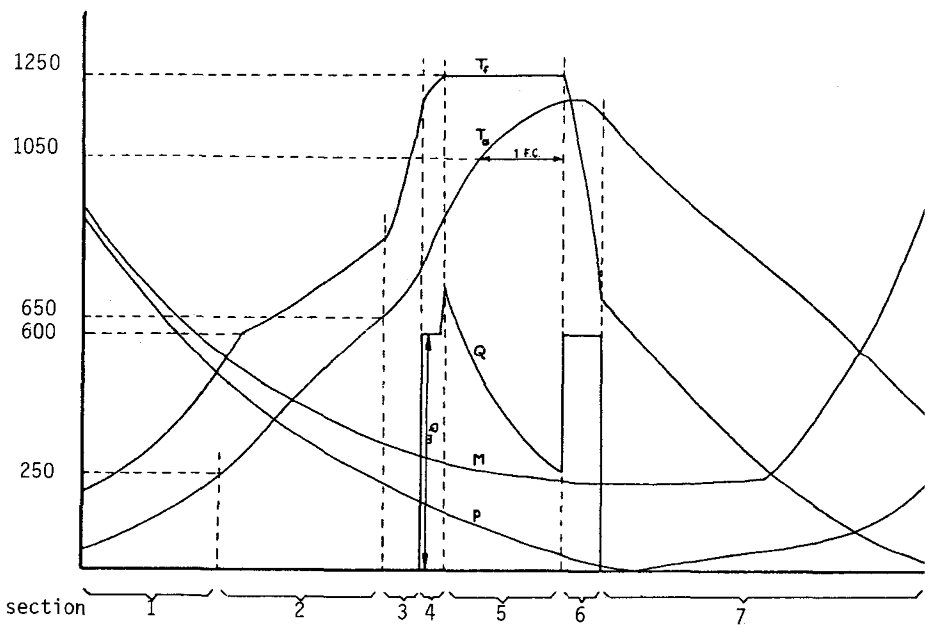

Figure 6 : Division of a baking curve into seven sections.

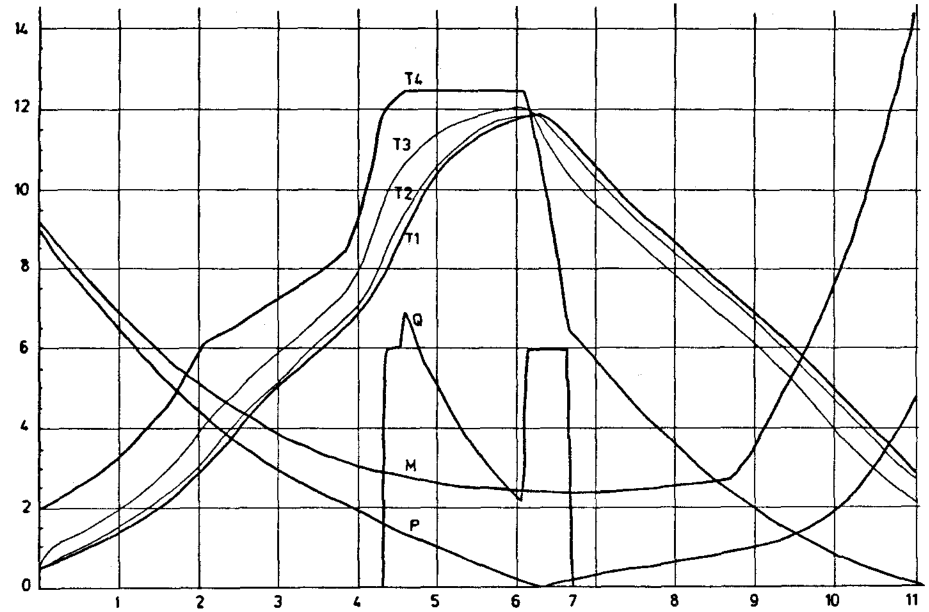

Figure 7 : The result of a representative calculation.

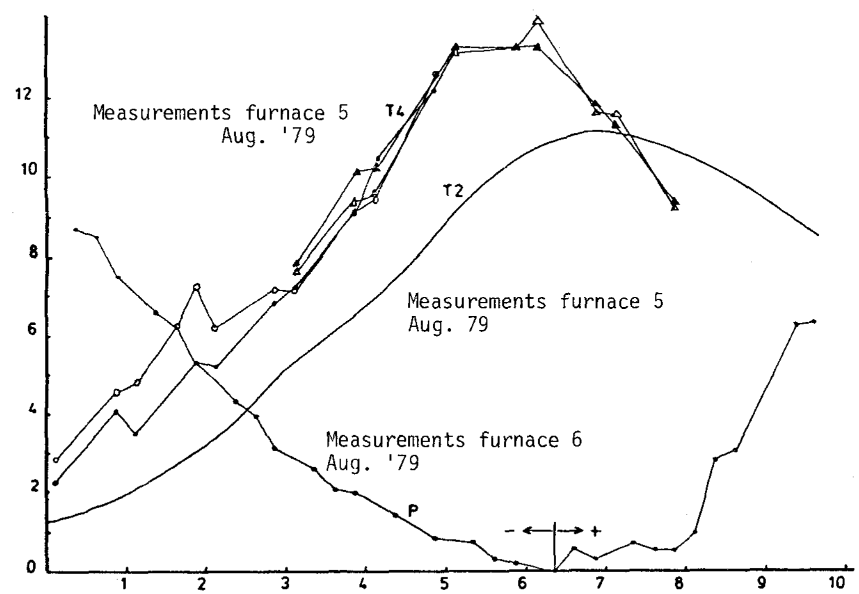

Figure 8 : Some representative measurements at a furnace.

T_a and T_f show the usual behaviour of flue gas and anode temperature with the typical influence of pitchburning visible in the flue gas temperature. M shows the decreasing quantity of flue gases from the exhaust manifold to a point in the cooling section. The increasing quantity on the right hand side is due to the open lids, provided in the flues for the placing of coolers or exhaust manifold. p shows the amount of vacuum in the heating section resp. the amount of pressure in the cooling section. Both, vacuum and pressure are shown on the positive side of the x-axis for reasons of the computer drawing.

The calculations are carried out in the positive direction, starting at the flue gas exhaust manifold. First the initial conditions for T_a, T_f and either p or M must be chosen. p and M cannot be selected independently since the operation of the furnace requires that the pressure passes through zero immediately behind the fire section. The calculations are carried out in seven separate sections as indicated below the x-axis in figure 6.

1. <u>The preheating section</u>

   The end of this section is reached when the anode has reached the temperature at which the pitch fumes start escaping.

2. <u>The pitch fume section</u>

   The escape of pitch fumes is governed by the temperature of the anode. The end of this section is reached when the anode has reached a particular temperature. The fumes will start burning as soon as the flue gas temperature exceeds a particular level.

3. <u>The intermediate section</u>

   In the calculations it is assumed that the flues can only endure a particular maximum amount of external heat input Q_m and a particular maximum flue gas temperature. In general, the fire section begins before the flue gases have reached this maximum temperature. For this reason, at every step in the intermediate section, an estimate is made of the maximum temperature that would result if the fire section were te begin here.

4. <u>The first fire section</u>

   In this section, a constant heat input Q_m is given. The end of this section is reached when the temperature of the flue gases is equal to a particular maximum flue gas temperature, for instance 1250°C.

5. <u>The second fire section</u>

   The flue gas temperature is maintained here at the above-mentioned temperature, for instance 1250°C. This means that the heat input is variable and must be calculated. The end of this section is reached when the temperature of the anode has exceeded a particular value, (e.g. 1050°C) during at least a given period (e.g. one firing cycle).

6. <u>The third fire section</u>

   In this section, the air that comes from the cooling section is heated to the temperature needed in the second fire section viz. 1250°C in our example. The choice of the end of this sixth section is more or less arbitrary, and determines the length of the seventh and final section. However, we want the air heated to 1250°C as quickly as possible, so that a high and constant heat input (Q_m) is specified.

7. <u>The cooling section</u>

   In our calculations we assumed that the lids are removed from the air entry holes in the flues, starting a particular distance from the third fire section. (We assumed two pit lengths). Furthermore, we assumed that there is only one cooling air supply manifold. The end of the cooling section is reached when the flue gas temperature equals the temperature of the ambient air (20°C).

<u>Results</u>

Figures 7 and 8 show the results of a calculation and the results of some representative measurements respectively. The calculated pressures and temperatures appear to be in fair agreement with reality. The same is true for the specific fuel consumption, not shown in figure 7. The meaning of the indices is the same as in figure 6; in addition, the temperature of the packing material T2 and the brickwork T3 is given.

The effect of decreasing the flow of flue gases at the exhaust manifold, without changing the furnace geometry, is shown in figure 9. By operating a baking furnace under low vacuum it was possible to prove that the model will correctly predict the real situation, also in this case.

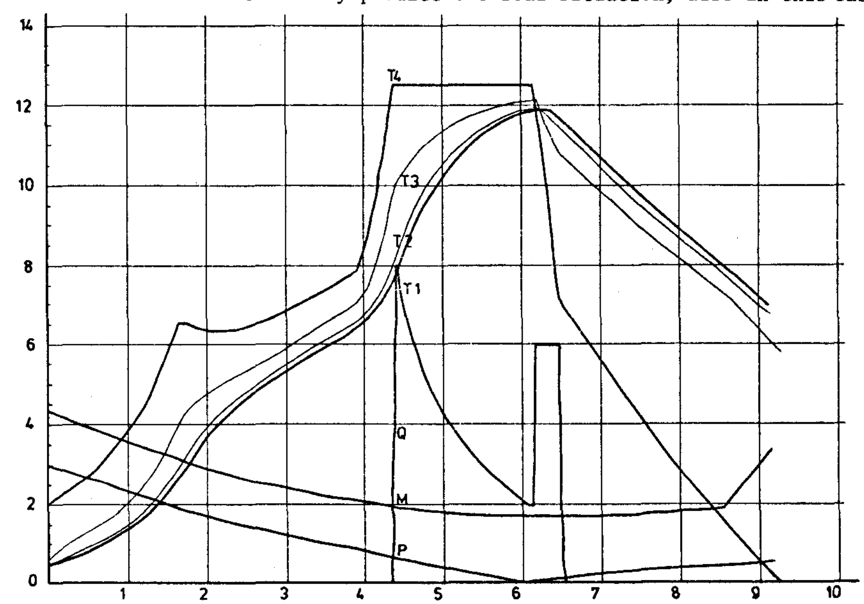

Figure 9 : The baking curve for an low exhaust vacuum.

Outlook

In conclusion it can be said that the model has the potential to predict the most important features of a particular furnace. By varying some parameters describing the geometry of the furnace, it is possible to calculate the effects of a change in the design. By varying other parameters (such as the vacuum at the exhaust manifold), the most economical way of operation can be determined. It is true that there are limitations; nevertheless the model seems to be an excellent tool for process evaluation and a valuable aid in the search for optimum furnace geometry.

With these results our efforts for the moment are not directed at further improving the described model, but at studying the problems that are created by the so-called "cross-over", that is the connection between the two parts of the baking furnaces. We feel that in the case of the cross-over,as well as in the flue design, model techniques can be helpful in optimizing the furnace design.

Conclusion

The large investments required for the overhaul of a furnace, or even for the construction of a new furnace justify a search for methods to predict the consequences of a modification to the existing design. In the previous sections, we have presented some aids for predicting whether a new furnace design will perform well or poorly, namely:

- a scale model of a flue can be used to determine a design in which the velocity distribution is good;
- with the mathematical model it is possible to predict the influence of changing parameters on furnace operation;
- last but not least, it is possible to predict the change in the parameters of furnace operation due to changes in the flue geometry;

We hope that, with this paper, we have contributed to more efficient operating and design practices for open ring flue furnaces.

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