NEW REFRACTORY CRUCIFORM FOR IMPROVED ENERGY EFFICIENCY OF REGENERATIVE GLASS FURNACE

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SEPR has a 35 year experience of innovation in regenerator for glass furnaces. Special products like ER5312RX (fused alumina) have been developed for protecting top of the packing and increasing regenerator life. For improving fuel savings, Type 4, a corrugated surface product, has been used widely in the industry for a long time. Today a new solution, Type 8, is introduced for the glass furnace to reach even higher range of energy efficiency. The development of this new cruciform is the result of advanced CFD modeling and experimental trials obtained on half scale regenerator testing rig. A numerical model that reproduces the heat transfer regimes encountered in a regenerator is now in place and used to optimize the packing for each application. This paper presents the technical benefit of this new solution: Type 8 has a higher specific surface area and a shape enhancing drastically the heat exchange. Due to these characteristics the Type 8 product, dedicated to the top of the packing, makes the optimization of the regenerator design more versatile. First feedback from full scale industrial experience will be detailed.

Introduction

During the past few years, the sharp increase of already high energy prices as well as tougher environmental regulations drove the pressure on Glassmakers to reduce fuel consumption to lower levels. Regenerative furnaces are a very competitive solution to produce glass economically. However the regenerator chamber performance is not yet maximized despite the numerous improvements in checker designs in the last 35 years. During that period of time, SEPR has developed a wide range of materials and shapes to adapt the regenerator packing to the evolving constraints of the glass furnaces: corrugated checker (Type 4) for enhanced heat transfer, large flue size (Type 6) to have modular solution in the lower courses, higher corrosion resistant material (ER5312RX) for top course checkers. The superiority of such cruciform designs over other available designs has been also discussed. Although the theoretical energy efficiency of a regenerator lies around 75 to 80%, depending on the type of fuel and process parameters such as fraction of excess air in the combustion, the actual performances cannot in practice reach the maximal efficiency. The new cruciform checker “Type 8” allows the thermal performance to go beyond the usual limits and get closer to the theoretical value while keeping the regenerator chamber dimensions as compact as possible. A complete thermal characterization of the new checker and its behavior in a regenerator packing based on laboratory tests and industrial on-site measurement, as well as an extensive numerical modeling study, was performed. This paper presents the technical benefits of this new solution and the results of laboratory experiments, an industrial case analysis based on numerical simulation and first industrial experience.

Development approach

Principles

The heat regeneration theory has been described extensively by different authors. It stipulates that the efficiency of the regenerator depends on the following dimensionless factors called reduced length Λ and reduce period Π.
where $b$ is the heat transfer coefficient (in W/m$^2$/K), $S_p$ the specific surface area (in m$^2$/m$^3$), $\dot{m}$ the mass flow rate (in kg/s), $C_{pg}$ the heat capacity of the gas (in J/kg/K), $C_{ps}$ the heat capacity of the solid (in J/kg/K), $M$ the mass of the checkerworks (in kg) and $\tau$ the time between flow inversions (in s). Both reduced quantities are different whether the air phase or the flue gas phase is considered. For simplicity, we propose to illustrate the principle with the ideal case of a balanced and symmetrical regenerator, i.e. for which $\Lambda_a = \Lambda_f$ and $\Pi_a = \Pi_f$. In this case the efficiency can be simply derived from these two factors. Figure 1 presents the diagram that gives the efficiency function of $\Lambda$ and $\Pi$.

For a typical regenerator, the reduced period $\Pi$ is always lower than unity due to the relatively short air or flue gas period duration and the large amount of thermal mass of the checker works. In order to improve the efficiency, the reduced length must be increased, that is, for a given furnace at a given set of gas and air flow rates, the same as increasing the product $bS_p$, which is the density of heat exchange expressed in W/m$^3$/K.

We propose in the present paper a new cruciform that increases both the heat transfer coefficient $b$ and the specific surface area $S_p$: the Type 8 cruciform. The surface area is locally increased by parting the usual square channel in 2 rectangular smaller channels created by another refractory partition. In other worlds, the packing is made denser in the section with more refractory material for heat storage and more surface area for heat exchange. It is worth mentioning that in addition to the increase of specific surface area, such a design increases the heat transfer coefficient as well. Two reasons can explain this effect. First, the surfaces of the denser section are corrugated. Corrugations are known to increase heat exchange as well as increasing surface area, as schematized in figure 2, as the evolution heat transfer coefficient $b$ around an obstacle, at x=0 on the axis in the figure, is locally increased with regards to the case without obstacle $b_o$. 
The second reason is intrinsically due to the rectangular shape of the channel. In forced laminar convection for instance, the heat transfer in a rectangular channel only depends on its aspect ratio $\gamma$ defined by the ratio of channel width by its length. The following correlation for the Nusselt number $Nu_{Dh}$ has been established by Morini and correlates well with the previous experimental work by Shah:

$$Nu_{Dh} = 8.235 \left(1 - 2.042\gamma + 3.075\gamma^2 - 2.438\gamma^3 + 1.008\gamma^4 - 0.165\gamma^5\right)$$

For a flue channel of 150x150mm made of cruciform checkers with smooth, 30mm thick walls, the expected increase of heat transfer coefficient from square shape to rectangular is very important as shown in the figure 3. The remaining potential improvement from rectangular to parallel plates is marginal.

However, the physics involved in actual glass furnace regenerators cannot be confined into forced convection only. The heat transfer modes are complex and vary a lot depending on the regenerator design as well as on where and when they take place. Transient working conditions, radiation from hot gases, temperature dependant physical properties, all contribute to generate high temperature gradients that strongly promote the mix of forced and natural convective flow regimes.

Based on all these considerations, the analysis of the heat transfer requires the new checker to be placed in the top course of the regenerator packing. The resulting Type 8 checker is a corrugated cruciform with asymmetrical wing lengths, as shown in figure 4. The height is 420mm as Type 3 and Type 4 and it is available in ER5312RX material that is suitable for the top course of glass furnace regenerator.
The following sections present the experimental and the numerical validation of the concept of regenerator with the new checker described above. In this paper the energy efficiency $\eta_{\text{regen}}$ is the ratio of the energy recovered by air $E_{\text{air}}$ during a period by the available energy in the flue gases $E_{\text{flue}}^{\text{max}}$. The definition of these energy quantities is given by the following relations:

$$E_{\text{air}} = \int_{T_{\text{air, outlet}}}^{T_{\text{air, inlet}}} m_{\text{air}} C_{\text{p, air}} dT dt ; \quad E_{\text{flue}}^{\text{max}} = \int_{T_{\text{flue, outlet}}}^{T_{\text{flue, inlet}}} m_{\text{flue}} C_{\text{p, flue}} dT dt ; \quad \eta_{\text{regen}} = \frac{E_{\text{air}}}{E_{\text{flue}}^{\text{max}}} ,$$

where $T_{\text{flue, inlet}}$ is the inlet temperature of the flue gases in the regenerator, $T_{\text{air, inlet}}$ is the inlet air temperature and $T_{\text{air, outlet}}$ is the air preheating temperature exiting the regenerator.

**Experimental set up**

The experimental set up used for the regenerator study is based in CEA-GRETh laboratory in Grenoble, France. It has been described in previous publications and its principle is shown on the sketch of figure 5. The set up can test two regenerator packing configurations simultaneously in two chambers and in the same flow rate and temperature conditions, close to an actual glass furnace working point. The test chamber has a usable height of 5m and a 6x6 channels cross section of 150x150mm flues. The gas temperature is measured at the top and the bottom section with aspiration pyrometers. The solid temperature is monitored by thermocouples placed in the checker in the central channel.
For the purpose of the present study we have compared two packing geometries: the reference that consists in the combination of smooth cruciform (Type 3) in the lower section and corrugated cruciform (Type 4) in the upper section, and the innovative packing solution made of the combination, from bottom to top sections, of Type 6, Type 3, Type 4 and Type 8. The comparison between these packing designs is summarized in the table 1 as well as the results of the measurements. For very similar working conditions (gas flow rates and inlet temperatures) the innovative packing preheats the air 80°C higher than the reference and therefore the energy recovery is almost 14% higher. It is also important to note that this performance increase is achieved in this particular case at a similar total packing mass, which prefigures the fact that the solution could be very economical.

<table>
<thead>
<tr>
<th></th>
<th>Reference Types 3-4</th>
<th>New Solution Types 6-3-4-8</th>
</tr>
</thead>
<tbody>
<tr>
<td>Weight (tons)</td>
<td>5.17</td>
<td>5.06</td>
</tr>
<tr>
<td>Surface area (m²)</td>
<td>17.2</td>
<td>18.7</td>
</tr>
<tr>
<td>Air flow (Nm³/h)</td>
<td>866</td>
<td>845</td>
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<tr>
<td>Fumes flow (Nm³/h)</td>
<td>1205</td>
<td>1205</td>
</tr>
<tr>
<td>Air inlet (°C)</td>
<td>153</td>
<td>155</td>
</tr>
<tr>
<td>Air outlet (°C)</td>
<td>934</td>
<td>1014 (+80)</td>
</tr>
<tr>
<td>Flue gas inlet (°C)</td>
<td>1179</td>
<td>1151</td>
</tr>
<tr>
<td>Flue gas outlet (°C)</td>
<td>658</td>
<td>571</td>
</tr>
<tr>
<td>Energy Efficiency</td>
<td>44%</td>
<td>50% (+13.6%)</td>
</tr>
</tbody>
</table>

Table 1: Packing descriptions and laboratory results based on gas temperature measurements

Furthermore, the laboratory set up gives insight on how the regenerator works. The local heat exchange intensity can indeed be derived from the checker temperature measured at different levels along the height of the central channel of the packing. The local exchanged power, figure 6, shows that the increase of performance is due to a higher heat exchange in the top section where the Type 8 cruciform checkers have been installed.

![Figure 6: Exchanged power (in kW/m) vs packing height in the laboratory set up](image)
Numerical Modeling

Knowing the results from the laboratory set up, it is interesting to predict the actual gain of energy efficiency in industrial cases. For this purpose a numerical model has been developed and validated with previous results of the experimental set up described above. It consists in a single channel model described in the framework of the ANSYS-Fluent® CFD code, to take into account the physics of the problem: gas radiation, effect of transition between different checker designs, turbulent convective heat transfer, and with a specific development to account for natural convection contribution in vertical channels that has been described in a previous publication. The basic principle is a transient calculation that alternates a period during which air flows upwards, and a period in which flue gases flow downwards. The convergence is reached when at each checker level the energy released to the air and the energy gained from the flue gases are equal. The model reproduces well the different flow regimes encountered in a glass furnace regenerator as illustrated in figure 7 in the case of a large regenerator (0.20 Nm/s in air, 15m high).

Industrial Validation

In order to validate the concept, we need to find a furnace on which it would be possible to compare a pair of chambers with Type 8 and a pair of chambers without Type 8. For an end fired furnace, there is only one pair of chamber, hence the comparison can only be done between the situations before and after a cold repair. The furnace would need to be set to equivalent working conditions (pull rate, temperatures), which is difficult to obtain in the industrial reality. Furthermore it is well known that furnace efficiency is higher at the beginning of a campaign so to carry out a fair comparison we would have had to find two similar furnaces that were repaired at the same time and working in the same conditions. Such opportunity is of course very unlikely, not to say impossible. The alternative was to test the solution on a cross fired furnace and to insure that two pairs of chambers would work as close as possible in the same conditions (gas consumption, air excess and inlet temperature levels). Again in the industrial world it is almost never the case. However we found a good candidate and two pair of chambers were installed for a side by side comparison. These two
chambers were engineered to work at the same fuel consumption. Based on the model presented earlier, the comparison between the solution without Type 8 and the solution with Type 8 gives an increase of energy efficiency over 3.5% (table 2). Two cases have been considered: Type 8 is substituted to Type 4 over 23% of the height of the regenerator in the first case, and 27% in the second case. The improvement between these two cases was quite marginal, therefore is has been chosen to conduct the trial with the top 23% of the chamber to be installed with Type 8 for the tested configuration.

<table>
<thead>
<tr>
<th>Calculated heat balance (MWh/channel)</th>
<th>Standard</th>
<th>Type 8 - 23% of the height</th>
<th>Type 8 - 27% of the height</th>
</tr>
</thead>
<tbody>
<tr>
<td>Available Energy (nominal assumption)</td>
<td>9.13</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Regenerated Energy</td>
<td>5.90</td>
<td>6.22</td>
<td>6.27</td>
</tr>
<tr>
<td>Energy efficiency</td>
<td>64.7%</td>
<td>68.2%</td>
<td>68.8%</td>
</tr>
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Table 2: Calculation comparison between standard and new configuration cases based on nominal conditions of the furnace

Similarly to the laboratory set up, we were able to evaluate the performance of the packing in two ways. First, the heat balance of the complete chamber is based on the flow rates and the temperature at the top and bottom. The integrated energy values are summarized in the table 3. It is almost impossible to get the same exact working conditions between two pair of chambers. This being stated, the different measurements done on the furnace gave quite similar flow rates for air and for flue gases in the two test chambers. The inlet temperature of the air and the flue gases was slightly different and that explains the difference in available energy in the two chambers. The resulting energy efficiency is greatly in favor of the solution with Type 8. It confirms the trend of the lab scale results as well as the order of magnitude that is predicted by the numerical model.

<table>
<thead>
<tr>
<th>Measured Heat Balance (MWh/channel)</th>
<th>Standard</th>
<th>With Type 8</th>
</tr>
</thead>
<tbody>
<tr>
<td>Available Energy</td>
<td>12.45</td>
<td>11.89</td>
</tr>
<tr>
<td>Regenerated Energy</td>
<td>8.58</td>
<td>8.90</td>
</tr>
<tr>
<td>Energy efficiency</td>
<td>69.0%</td>
<td>74.8%</td>
</tr>
</tbody>
</table>

Table 3: Measured heat balance of the two test chambers

Second, the instrumentation of the checkers was carried out for the two test chambers of the furnace. One central channel was equipped during the construction of the chambers with thermocouples at different height levels. The thermocouples are located in the wing of a specially machined cruciform and connected to the outside of the chamber through a dedicated opening. The cables are protected inside a specific groove (in red in figure 8) between two cruciform layers in order to minimize their disturbance to the furnace operation. The thermocouples survived the start up of the furnace and were working well after the furnace conditions and pull rate were stabilized so that relevant temperature profiles can be derived from the measurements.

The resulting exchanged power profiles from the checker instrumentation are shown in figure 9. They are taken at the same time as the heat balance reported in table 3. They
strongly confirm the trend observed during the laboratory tests showing a significant increase of heat exchange intensity especially in the Type 8 section.

Figure 8: Instrumentation of the checkers with thermocouples: for square Type 3 channels (left) and for rectangle Type 8 channels (right)

Figure 9: Exchange power profile in the 2 instrumented chambers standard chamber (red) and innovative packing with Type 8 (green)

Type 8 cruciform fulfills the expectations of higher thermal performance for a glass furnace regenerator. While the top of the packing where the new cruciform Type 8 is proposed is generally not an area of heavy clogging but recognizing that the size of the Type 8 channel represents approx 40% of the corresponding Type 3 channel, one could argue that the smaller flue size would likely favor more rapid clogging of the packing than regular cruciform channels. Exhaust gases from glass furnaces are indeed prone to carry particles that could deposit on the checkers or to transport chemical species that could condensate in the lower part of the regenerator (e.g. alkali sulfates). The trial on the cross fired furnace is too recent to firmly conclude in this case but the issue of the plugging of the channels by carry over has been investigated in another test on an end fired furnace. A patch of Type 8 channels (4x12 channels / located transversally at 1/3 of the length of the chamber near the target wall) has been installed in the top course of a Type 4 regenerator chamber. The furnace has been running for 5 years and observations did not reveal any difference in
erosion or damage between the standard flues (Type 4) and Type 8 flues. No plugging of the top courses of the packing was observed.

Concerning the risk of condensation on Type 8 cruciform checkers and subsequent risk of plugging, it can be theoretically reduced by installing the Type 8 outside of the condensation zone. For sodalime silica glass operating under normal conditions, the last bottom course of Type 8 should be located above the 1150°C isotherm and not below where the sodium sulfates start condensing (800 – 1150°C). Numerical modeling once again helps the designer of the regenerator to choose the right amount of Type 8 for his furnace. In typical furnace, the change from a Type 3 – Type 4 configuration to a Type 8 configuration leads to a modification of the temperature profile in the checkers. Such a change is illustrated in figure 10. Past experience in numerical modeling based on industrial conditions shows that between 15% and 25% of the height of the regenerator can be substituted by Type 8 and having the smaller flues working at temperature above the condensation range.

![Figure 10: Typical temperature profile in the checkers between a standard solution and a solution with Type 8 substitution in the top courses](image)

**Conclusion and perspectives**

A new cruciform has been developed for the top section of glass furnace regenerator. Type 8 cruciform allows reaching higher energy efficiency for a given chamber without increasing the size of the regenerator. The principle has been verified on an experimental set up and is under validation in an industrial furnace. The data measured on the furnace has confirmed both the observation made on the laboratory scale experiment and the prediction of the numerical simulation.

Finally, the Type 8 cruciform brings flexibility to the design of the regenerator allowing more combination to optimize its thermal performance:

- higher heat exchange capacity in a given regenerator envelope for a given flue size
- lower cost penalty than chamber size increase, with a higher performance
- better performance – cost compromise with the combination of Type 8 and larger overall flue size (from 140x140mm to 160x160mm)

- better performance – cost compromise with a design with large Type 6 flue size in the bottom section and Type 8 flue size in the upper section.

Figure 11: View of the top course of a chamber with Type 8 rectangular channels while the furnace is in production

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viii  D. Lagarenne, Thèse INSA Lyon, Récupération d'énergie par les régénérateurs de chaleur des fours de verrerie (1990).