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RECUPERATIVE HEAT EXCHANGER FOR HIGH TEMPERATURES

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În industria actuală, există o mare varietate de schimbătoare de căldură ; având în vedere reducerea distribuţiei transferului termic între purtători cu temperaturi de până la 1800^{0}C şi presiuni până la 5 bari, se propune un nou schimbător, unul recuperativ gaz- gaz, în contracurent, construit din beton . Funcționarea sa poate fi extinsă chiar la agenţi sub 100^{0}C . Investiţia în schimbător se recuperează în 4 luni de funcțiune, costul fiind 20 % din cel al unui schimbător metalic echivalent (până la 500^{0}C) şi doar cu 30 % mai mare. Pentru temperaturi înalte, schimbătorul propus nu are rival. Există o arie largă de aplicare.

In the actual industry, there is a great variety of heat exchangers and in order to reduce dramatically the distribution of heat transfer between carriers with temperatures up to 1,800˚C and pressures up to 5 barr, we propose a new one – a recuperative gas-to-gas heat exchanger in counterflow, made by concrete. Its operation can be extended down below 100˚C. The investment in this exchanger is recovered in less than 4 months of operation, being 20% of the amount requested by an equivalent metallic exchanger (up to 500˚C) and its size is only 30% larger. For higher temperatures, there is no rivalry. It can find a wide range of applications.

Keywords: recuperative heat exchanger, concrete, high temperatures

1. Preliminaries

The proposed heat exchangers might be surprinsing for specialists, because it is made with concrete, a material considered unsuitable for gas-to-gas heat exchangers. However, a closer investigation finds it usable in a wide range of temperatures, with different types of cement.

It is well known that, in the majority of cases, the global thermal resistance of a wall against gases on both faces is higher than 40^{-1} K·m^2 / kW **[1][2]**, while the one of a concrete wall 15 mm thick is 12^{-1} K·m^2 / kW. Therefore the heat transfer through the surface of a metallic wall is comparable to the one through a

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concrete wall whose surface is 3 time larger. Larger surface is responsible for a cost increase of 20%, but on the other hand, material cost makes the concrete heat exchanger 10 times cheaper than an equivalent metallic one. Considering the thermal insulation, joint and base, a total cost reduction of 5 times is achievable. Therefore investment and maintenance costs favour concrete, a reliable material, compatible with gases which are corrosive to steel.

Ever since 1950, "Beton Kalendar " pointed out that "concrete is to large extent insensitive to heat; several tests established that its compression strength changes only slightly up to 500˚C and decreases only slowly at higher temperatures" **[3]**. The refractory concrete can be used up to 1,800˚C, being insensitive to CO, H_2 etc; at such temperatures the concrete heat exchanger has even higher efficiency. Its tensile strength is lower but sufficient to assure the integrity of the structures provided the temperature differences between confining heat carrier does not exceed 300 K (several times higher than the economical difference) to avoid cracks caused by differential thermal expansion. We point out that the specialised literature provides numerous cheap and efficient techniques of corrosion control for concrete in contact with wet gases containing SO_2 to be cooled below the acid dew point **[4],[5]**.

2. The description of the proposed heat exchanger and its mathematical model

In a concrete block of paralelipiped shape, developed vertically, channel rows, parallel to the vertical faces, are formed in the casting process. Two gas streams flow in opposite directions through alternate rows exchanging heat. Distributors and collectors are placed at the ends of the channels to separate the streams and connect them to the sources and the exhausts of the heating and heated carriers. So the carriers are in counterflow and maintain a moderate temperature difference at each normal section of the channels.

The unit thermal stress of the concrete should not exceed the admissible limits : in a cross-section normal to the rows, the temperature difference between opposite flows should not exceed 40 K; at this amplitude, it corresponds a temperature difference ΔT between thermal carriers greater than 200 K.

The construction of the heat exchanger is simplified by a convenient grouping of modular units. In most applications, the height of the solid does not exceed 15 m, which leads to the compression stress of the modest value of 4 dN / cm^2. The concrete permeability to gas can be adjusted even at pressure differences of some barrs. This is the case of some air preheating plants (as the cowpers of blust furnances), where the container of the exchanger will be metallic, air tight, with internal pressure equal to the one of the more compressed gas.

The use of concrete as building material for heat exchangers imposes adequate technology for forming numerous vertical channels separated by unusually thin walls. We have developed different practical versions; but in what follows we configure ourselves to a mesh of square channels; this simplifies the mathematical model and the calculations.

The heating gas A and the cold gas B flow through parallel rows of channels like in figure 1. Heat flows across a wall from A to B and along a rib perpendicular to the wall. Consider an element limited by a control surface of square section around a node like in figure 2.

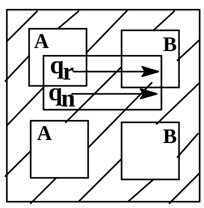

Fig. 1. Square channels of the heat exchanger

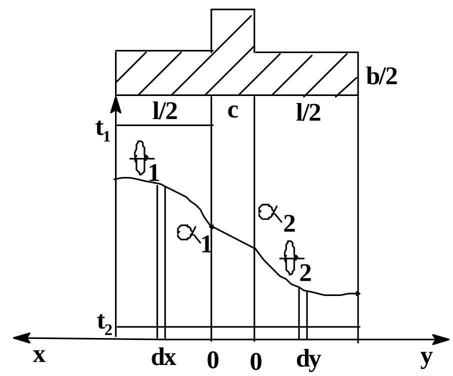

Fig. 2. Temperature distribution in a module

The surface cuts the wall at one half of its lenght and the rib at one half of its cross section. We apply to both the branches of the element the heat conduction mechanism of the bar and the convective heat transfer mechanism of the wall, respectively.

Suppose the temperature θ to be uniform across the bar and stationary, only varying along the x and y directions, with the distribution given by the following second order differential equations system :

$$\frac{\partial^2\theta_1}{dx^2} - \frac{2\alpha_1}{\lambda b}(\theta_1 - t_1) = 0$$

$$\frac{\partial^2\theta_1}{dy^2} - \frac{2\alpha_2}{\lambda b}(\theta_2 - t_2) = 0 \tag{1}$$

where the constant coefficients are indicated in figure 2. By integration, it follows:

$$\theta_1 = t_1 + A_1 e^{(a_1 x)} + B_1 e^{(-a_1 x)}$$

$$\theta_2 = t_2 + A_2 e^{(a_2 y)} + B_2 e^{(-a_2 y)} \tag{2}$$

where denote :

$$a_1 = (\frac{2\alpha_1}{\lambda b})^{1/2} \text{ and } a_2 = (\frac{2\alpha_2}{\lambda b})^{1/2} \tag{3}$$

And A_1, B_1, A_2, B_2 are arbitrary constants which will be determined from the boundary conditions. Firstly, the symmetry requires the condition :

$$\frac{\partial\theta_1}{dx}\Big|_{x=l/2} = 0 \text{ and } \frac{\partial\theta_2}{dy}\Big|_{y=l/2} = 0$$

whence one gets: $B_1 = A_1 e^{(a_1 l)}$ and $B_2 = A_2 e^{(a_2 l)}$.

Secondly, the absence of heat sources and wells leads to the equation:

$$\frac{\partial\theta_1}{dx}\Big|_{x=0} = -(-\frac{\partial\theta_2}{dy})\Big|_{y=0} = \frac{1}{c}(\theta_1\Big|_{x=0} - \theta_2\Big|_{y=0}) \tag{4}$$

Then:

$$c(a_1A_1 - a_1A_1e^{(a_1l)}) = -c(a_2A_2 - a_2A_2e^{(a_2l)}) = t_1 + A_1 + A_1e^{a_1l} - t_2 - A_2 - A_2e^{a_2l} \tag{5}$$

By solving this system, one gets:

$$A_1 = \frac{t_1 - t_2}{ca_1(1-e^{a_1l}) - (1+e^{a_1l}) - (1+e^{a_2l})\frac{a_1(1-e^{a_1l})}{a_2(1+e^{a_1l})}} \tag{6}$$

and

$$A_2 = -A_1\frac{a_1(1-e^{a_1l})}{a_2(1-e^{a_2l})} \tag{7}$$

The heat exchanged through a quarter of rib ($0 < x <$ l/2) is:

$$q_r = \int_0^{l/2}\alpha_1(t_1 - \theta_1)dx = -\alpha_1\int_0^{l/2}(A_1e^{a_1x} + B_1e^{-a_1x})dx = \frac{\alpha_1}{a_1}A_1(1-e^{a_1l}) \tag{8}$$

According to relation (6), one obtains:

$$q_r = \frac{\lambda b(t_1 - t_2)}{2C},$$

Where

$$C = c + \frac{e^{a_1l}+1}{a_1(e^{a_1l}-1)} + \frac{e^{a_2l}+1}{a_2(e^{a_2l}-1)} \tag{9}$$

The heat transfered through one half of the wall is:

$$q_w = (t_1 - t_2)\frac{l}{2}(\frac{1}{\alpha_1} + \frac{c}{\lambda} + \frac{1}{\alpha_2})^{-1} \tag{10}$$

Since each heat carrier flows through one half of the total number of channels, the flux density exchanged by each channel per unit height is:

$$q_0 = 2(q_r + q_w) = (t_1 - t_2)\left[\frac{\lambda b}{C} + lx(\frac{1}{\alpha_1} + \frac{c}{\lambda} + \frac{1}{\alpha_2})^{-1}\right] \quad (11)$$

3. A numerical example

Consider a heat exchanger which heats 10.000 kg / s (2.78 kg / s) of air from 0˚C to 100˚C, with gases effectively cooled from 150° to 70˚C. That means temperatures of the gas air are t_1 = 110 K and t_2 = 50 K. Disregarding optimization, we choose:

a) channel section: 20 x 20 mm (hence *l* = 20 mm);
b) comcrete wall thickness: *b* = *c* = 10 mm;
c) velocity in channels: 5 m_n / s;
d) mean temperature of gases and air: 110˚C and 50˚C respectively;
e) kinematic viscosity of air at 50˚C: v_a = 1.86 x 10^{-5} m^2 / s;
f) kinematic viscosity of hot gas at 110˚C: v_g = 2.52 x 10^{-5} m^2 / s;
g) specific heat of air and combustion gas: c_p = 1.02 kJ / kg·K;
h) thermal conductivity of air and combustion gas: λ_g = 0.0272 W / m·K;
i) thermal conductivity of concrete: λ_c = 1.40 W / m·K;
j) Prandl number of air and combustion gas: Pr = (cp·μ) / k = 0.71;

With these data, the mean velocity of air is: 5 x (323/273) = 5.92 m/s and Reynolds number is:

$$\mathrm{Re} = \frac{5.92 \cdot 0.020}{1.86 \cdot 10^{-5}} = 6{,}366 \quad (12)$$

By applying the usual formula for our exchange regime, we get Nusselt number:

$$Nu = 0.0209 \cdot \mathrm{Re}^{0.8} \cdot \mathrm{Pr}^{0.45} \cdot (1 - \frac{6 \cdot 10^5}{\mathrm{Re}^{1.8}}) = 18.10 \quad (13)$$

whence: $$\alpha_1 = \frac{Nu \cdot \lambda}{l} = 18.10 \cdot \frac{0.0272}{0.020} \approx 24.61 W / m^2 \cdot K \quad (14)$$

The thermal load of the exchanger is Q = 2.78 x 1.02 x 100 = 283.56 kW. The flow rate of the hot thermal carrier, by allowing for 2% heat loss, is:

Φ_{hg} = 1.02 x 283.56/1.02 x (150-70) ≈ 3.55 kg / h (15)

Its mean velocity in the channels will be (5 x 3.55) / 2.78 x (383/273) ≈ 8.96 m/s. Then the Reynolds and Nusselt numbers are respectively Re = 7,120 and:

$$Nu = 0.0263 \cdot \mathrm{Re}^{0.8} \cdot \mathrm{Pr}^{0.35} \cdot (1 - \frac{6 \cdot 10^5}{\mathrm{Re}^{1.8}}) = 26.2 \quad (16)$$

The difference in parameters from the previous formula takes into account that now the gas is cooling down. Thus, one obtains the coefficient of laminar heat transfer $\alpha_2 = 41.1$ W / m^2·K. Then by (14) and (3), one gets:
$a_1 = [(2 \times 24.61)/(1.40 \times 0.01)]^{1/2} \approx 59.29$ m^{-1} and $a_2 = [(2 \times 41.1)/(1.40 \times 0.01)]^{1/2} \approx 76.63$ m^{-1}.

Since $e^{a_1 l} = e^{(59.29 \times 0.02)} \approx 3.27$and $e^{a_2 l} \approx 4.63$, we get by (9):

$$C = 0.01 + \frac{1}{59.29} \times \frac{4.27}{2.27} + \frac{1}{76.73} \times \frac{5.63}{3.63} \approx 0.062 \qquad (17)$$

By (11), one obtains: $\frac{1}{\alpha_1} + \frac{c}{\lambda} + \frac{1}{\alpha_2} = \frac{1}{24.61} + \frac{0.01}{1.40} + \frac{1}{41.1} \approx 0.07$ and:

$$q_0 = 60 \times (\frac{1.40 \cdot 0.01}{0.062} + \frac{0.02}{0.07}) \approx 30.69 W/m \qquad (18)$$

The flow-rate of air passing through one channel will be $\Phi = 0.02^2 \times 5 \times 3{,}600 = 7.2$ m$_n^3$ / h and consequently, the number of channels for air will be 10,000/7.2 = 1,390. The total numer of vertical channels is rounded off to N = 2,800. The height of the exchanger will be $Q/(q_0 \times N) = [(283.56/(30.69 \times 2{,}800)] \times 10^3 \approx 3.3$ m and need a total surface of about 4 m^2 (including the insulation and carcass). The total volume of concrete is under 15 m^3 (a third part being light perlitic concrete for thermal insulation). The whole investment in this concrete, heat exchanger is recovered in less than 4 months of continuous operation.

The calculation of gases and air through exchanger requires pressure drops of about 400 Pa. In the previous, we have remarked the contribution of the thermal radiation at the heat transfer. As another example, consider a preheater for 100,000 m$_n^3$ / h air from 0 to 1,200°C, at 3 bars, heated with burning gases (at atmospheric pressure), usefully cooled from 1,300° to 100°C; suppose that both air and gas circulate by channels with a square section 20 x 20 mm, at a maximum velocity of gas 30 m / s. Such a preheater will have a height of about 45 m, with the thermal load of 140 GJ / h ≈ 39 MW and needs about 25,000 channels, the thickness of separating walls being of about 15 mm; the necessary surface is less than 40 m^2. The efficiency of the heat transfer is about 90%, inaccessible to the existent regenerative cowpers.

4. Considerations regarding the exchanger materialization.

We have considered several possibilities to establish a suitable technology for using concrete as a structural material for heat exchangers, in order to carry out numerous vertical channels with unusually thin walls. The channels in the heat exchanger are obtained by casting concrete in a parallelipiped solid. The channel cross-section can be either circular or square (with rounded corners). The carriers move in counterflow in the exchanger, inlets and outlets of both being applied to the ends of the perforated solid via collectors attached to the exchanger by means of distributors in the form of vertical rectangular slots. One of our successful solutions to form the channels used metallic rods lubricated and guided by a tube plate at the top and by the pre-formed channels at buttom. By turning a few degrees back and forth the rods during hardening of the concrete, it was possible to withdrow then without any damage to the walls no matter they were as thin as 2 mm. Simple drivers were applied to the upper ends of the rods. This type of exchanger is suitable for clean gases, or containing solid suspensions with little adherence to smooth surfaces.

More and better solutions are available and were actually carried out, applicable to the case of gases with aggressive components or solid suspensions of the flying ash type not easily trapped by purifing plants.

The contribution of the exchanger does not require any special equipment and after supplying and prepairing the necessary materials, it can be rapidly produced (for instance, that exemplified in point **3.**, was made in only a week).

5. Conclusions

The concrete heat exchanger above presented has a large number of applications, such as:

- preheating of comburent air by exhaust gases from all existent furnances and boilers (increasing their efficiency with 4 – 7%, without affecting their internal structure);
- recovering the heat of all gaseous combustion products, even those wet or containing SO_2 and ashes;
- recovering waste, wet and relatively hot gases exhausted by drying plants, workshops and livestock shalters to heat air for drying and ventilation;
- heating compressed air for blast furnances (replacing cowpers with the advantage of static operation and simplified automation); in this case, the refractory concrete heat exchangers are technically and economically superior to those actually in operation, even at temperatures up to 1,400°C.

The design and the operation of conventional and concrete heat exchangers are comparable. The reliability of the concrete exchangers is not inferior and besides, one can adopt a modular construction and standardization. We add also the implicit ecological advantages.

By many persons considered well-advised, the concrete is regraded a material suitable for large constructions but unfit for “lace” structures or “indentation” or to stand temperatures for above 80°C (a limit admitted by most of the builders), but the achievement of the first heat exchangers made of concrete has disaproved this opinion, confirming our theoretical and techno-economical assumptions.

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