High Temperature Equipment

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HIGH TEMPERATURE EQUIPMENT

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HIGH TEMPERATURE EQUIPMENT

PROCEEDINGS OF THE INTERNATIONAL CENTRE FOR HEAT AND MASS TRANSFER
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Preface

At the present time the application of high temperature heat exchangers has been spreading to energetics, metallurgy, and different areas of chemical technologies. That is why it was chosen as the subject of the Advanced Course, organized by the International Centre for Heat and Mass Transfer, in Dubrovnik, Yugoslavia.

At the meeting 12 lectures were submitted which covered the whole spectrum of the problems regarding design, manufacture, and operation of heat exchangers of this class. These lectures taken as a set, serve as an excellent introduction into the problem of high temperature heat exchangers for new users and researchers. At the same time they open a wide panorama of the problems encountered by researchers and point out ways to further technical progress in this field.

A. E. Sheindlin

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Heat Transfer Problems in High Temperature Heat Exchangers

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ABSTRACT

The report deals with construction types of high temperature heat exchangers, their circulation loops and temperature differences. Most attention is given to gas dynamics, convective heat transfer, convection and radiant heat transfer interaction in high temperature heat exchangers. The ways of heat transfer augmentation and efficient heat exchanger construction are discussed here.

1. INTRODUCTION

World consumption of fuel and power keeps increasing with very high ratios. The main part of heat consumed in industrial production is subject to transformation in heat exchangers of numerous construction types.

Improvement of power efficiency of thermal equipment involves high temperature processes.

Modern development of prospective power conversion methods, such as MHD-generators and gas turbines, involves considerations of more efficient fuel consumption through a high thermal efficiency of the equipment, by higher initial temperatures of the gases. This is achieved by preheating the supply air in heat exchangers.

Utilisation of the high temperature waste heat from industrial and power plants is now very actual. Waste heat means not only economic losses, but also thermal pollution of the environment, which may acquire hazardous levels with the further development of technology and power generation.

An important development for the improvement in thermal efficiency of industrial plants is presented by their aggregation with regenerators or recuperators which use exhaust combustion gas flows to preheat the supply air.

Consumption of natural gas as technological fuel in industrial furnaces and as power fuel in boiler equipment has been increasing in recent years. The rates of saving natural gas by air preheating may be as high as 30 to 40 %.

This estimate comes from a heat balance evaluation of an industrial plant with-and without preheating. The efficiency of preheating per unit amount of heat consumed is higher as compared to the efficiency of fuel combustion per unit amount of heat produced. This is because unit amount of heat produced by fuel combustion in an industrial furnace is only partially utilised in its working space, and the major part is exhausted. But unit amount of heat supplied with preheated air is totally utilised in the equipment, and does not produce any volume or temperature increase of the exaust combustion products.

Improvements of power efficiency and compactness of heat exchangers are closely related to their heat transfer augmentation. But the intensity of heat transfer, and efficiency of a heat exchanger is closely related to gas dynamics and hydraulic drag on its heat transfer surface, and to the approach to the process of high temperature heat transfer, implemented in the equipment.

2. TYPES OF HIGH TEMPERATURE HEAT EXCHANGERS

We consider high temperature heat exchangers with an over 800 K temperature of at least one of their heat carrier fluids.

According to the process of heat transfer involved, heat exchangers may be divided into direct contact and indirect contact heat exchangers. In direct contact heat exchangers heat is transferred by a contact of two immiscible fluids or of a gas and a bed of solid particles. The most common example of such arrangement is a cooling tower.

In indirect contact heat exchangers, heat is transferred from a hot gas to a surface and then to a cold gas. Such heat exchangers are sub-divided into recuperative and regenerative heat exchangers. In recuperators heat is transferred from one gas to another through a solid wall. In regenerators hot and cold gas flow alternatively through the same passages in a periodic manner.

2.1. High Temperature Regenerators

In a high temperature regenerator, heat is first accumulated by the solid wall from a hot gas flow (hot blast), which is then replaced by a cold gas flow (cold blast), and the accumulated heat is taken away from the wall to the cold gas. Thus, only transient heat transfer is possible in a regenerator. Air preheaters of blast furnaces and Martin furnaces are typical examples of high temperature regenerators.

Hot blast was known in metallurgy as early as the beginning of the 19 c. It was accomplished in cast-iron tubes. In 1857 an English engineer E. Cowper invented a stove with a stationary accumulating checkerwork and alternative hot and cold blast.

In modern high temperature regenerators heat is accumulated by their packings or checkerwork, or matrix. Fireclay bricks or silica bricks may constitute a continuous brickwork, or checkers of either in-line or staggered arrangement. Packings of ceramic plates or pebbles are used recently in many branches of industry.

Regenerators operate either by an alternative supply of hot and cold gas into the matrix (periodical flow or stationary regenerators), or by periodically moving the matrix in hot and cold gas flows (rotary and moving bed regenerators). Thus we distinguish three main types of regenerators:

1. Fixed matrix regenerators,
2. Moving bed regenerators,
3. Rotary regenerators.

Regenerators have some advantages over recuperators [1] - they are of simple construction and cheaper, and exceed recuperators by their compactness, or surface area density. They provide higher preheating temperatures (1700 K instead of 1200 K), higher reliability and lower maintainance cost. Relative disadvantages of regenerators are complicated, bulky and expensive auxilliary equipment (swich-over facilities, matrix drives a.o.), large heated surface areas are necessary, and leackage of about 5 % of combustion gas into the equipment, and hot air into the atmosphere.

2.2. High Temperature Recuperators

High temperature recuperators are sub-divided into three main groups: 1 - convective, 2 - radiant, 3 - combined.

Air-tube convective high temperature recuperators operate on internal tube flows of heated fluid, and the gas-tube ones - on external flows of heated fluid.

Air-tube convective recuperators as compared to gas-tube recuperators are seal-proof, they can be used to preheat gaseous fuel flows of higher excess pressures, more than 8 kPa. But they are less compact, especially for the low capacity units and their tubes are more sensible to abrasive wear by the solid suspended particles at flow velocities higher than 8 or 10m/s, and to fouling at flow velocities lower than 5 or 6 m/s.

Recuperators are made either of metal or of ceramics. Recuperators of fire-proof alloys may preheat air to 800° C, and the ceramic ones - to 1000°C and more, with heating gas temperature up to 1500°C.

Radiant recuperators are simpler in construction than convective ones, their heated surfaces are easily available for repair and cleaning, maximum temperatures of the heated surfaces are lower due to parallel flows, and so on. Relative disadvantages of radiant recuperators lie in their large size constructions, 10 to 15 times lower compactness, enabling to cool down the hot gas only to 700°C.

From a comparison of slot-type and tube-type radiant recuperators we note lower wall temperatures, in other words, better thermal behaviour of slot-type recuperators for similar hot gas and preheated air temperatures.

Combined radiant-convective recuperators are usually applied in combustion flows of more than 1000 to 1100°C. First a radiant section lowers the gas temperature to 800-1000°C, then it is supplied to a convective section, where the gas is cooled down to 400°C and lower. The air is first supplied to the convective section and heats up to 300-400°C, and then to the radiant section, where its temperature reaches 600-800°C.

The choise of the heat exchanger type depends on a number of factors: exhaust gas temperature, preheated air temperature, available space and material, a.o.

Modern intensification of technological processes connected with heating of metals and of other materials has resulted in an increase of exaust gas temperatures. This opens wide prospects for radiant recuperators which can operate at very high temperatures, up to 1400-1500°C for prolonged periods of time.

3. PRINCIPLES OF THERMAL DESIGN OF HIGH TEMPERATURE HEAT EXCHANGERS

The main task of thermal design for a heat exchanger is the necessity of determining its heated surface area. If the area is pre-given, heat transfer rates, final parameters of the fluids and the general thermal behaviour must be predicted.

Methods of these calculations are based on the analysis of temperature distribution in the apparatus. In recuperators temperature distribution on the longitudinal axis must be considered. For regenerators, time variation of the temperature is also very important, and their analysis is much more complicated than for recuperators. Some of the equations for recuperators also are valid to regenerators. Such are the notions of heat balance and temperature difference, efficiency and difference between parallel flow, counterflow and crossflow.

An important choice for a heat exchanger is that of the flow direction. According to flow direction there are three main groups of heat exchangers: parallel flow (Fig. 1a) with parallel flows of the two fluids in one direction, counterflow (Fig. 1b) with opposite flow directions of the fluids, and crossflow (Fig. 1c) with one of the fluid flows directed at a right angle to the other.

In the engineering practice different combinations of these three circulation types are used and form very complicated circulation fields.

Crossflow exhibits intermediate temperature distributions between the parallel flow and the counterflow cases.

Counterflow circulation has some advantages in heat trans-

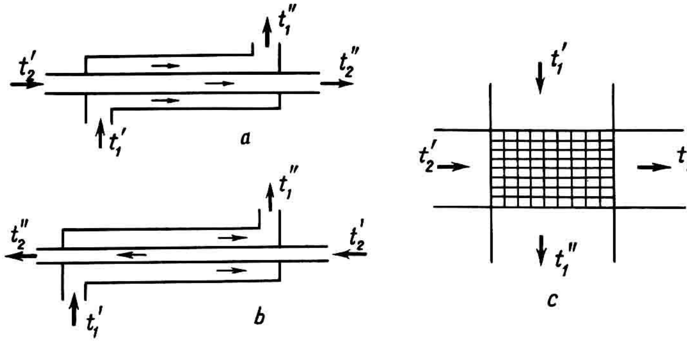

Fig. 1. Circulation paths in heat exchangers: a - parallel flow, b - counterflow, c - crossflow

fer efficiency over the parallel flow type:

1. For other similar conditions, amount of heat transfered is higher in counterflow heat exchangers, and their compactness can be improved.

2. For the same supply temperatures, higher preheating rates are achieved in counterflow circulation.

An advantage of parallel flow circulation is that the maximum wall temperature in it is lower for other similar conditions. This opens a possibility of using ordinary steel constructions at moderately high hot gas temperatures, up to 850°C.

However, operation experience with recuperators shows, that parallel flow is reasonable for recuperators of thin metal walls. For thicker walls, say, of cast iron, the very high inlet temperature gradients may cause fracture.

In counterflow, the preheating temperature approaches the maximum temperature of the hot gas.

In parallel flow, the temperature of the cold gas is always lower, than the minimum hot gas temperature. So maximum temperature differences are observed in counterflow, and minimum temperature differences - in parallel flow (Fig. 2).

Average temperature difference between the two gases $\overline{\Delta t}$ may be determined analytically from the equations of heat balance and heat transfer

$$\overline{\Delta t} = \frac{\Delta t_l - \Delta t_s}{\ln \dfrac{\Delta t_l}{\Delta t_s}} \tag{1}$$

where Δt_l - major temperature difference, Δt_s - minor tempe-

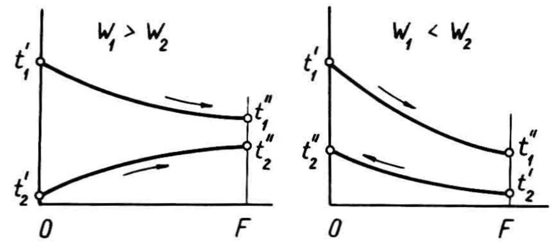

Fig. 2. Fluid temperature variation in parallel flow and in counterflow

rature difference.

This value is known as average logarithmic temperature difference. Eq. (1) applies both to counterflow and to parallel flow circulation.

For insignificant gas temperature variations along the heated surface, the arithmetic mean of the two limiting temperature difference values may be used

$$\overline{\Delta t} = \frac{\Delta t_1 + \Delta t_s}{2} \tag{2}$$

In heat exchangers of crossflow and combined circulation, the evaluation of the average temperature difference involves a complicated solution of

$$\Delta t_c = \overline{\Delta t}\,\varepsilon \tag{3}$$

where ε - correction factor to be found from corresponding charts.

Heat balance equations being esentially the same for recuperators and for regenerators, it is only reasonable to determine average temperature difference $\overline{\Delta t}$ for a regenerator in the same manner, as for a recuperator, only t''_1, t''_2 must be replaced by time-average outlet gas and air temperatures $\overline{t''_1}$, $\overline{t''_2}$ in eq.(1).

Thermal design for a heat exchanger consists of a simultaneous solution of the heat balance equations and of the heat transfer equation. For an ideal heat exchanger, when there is no heat loss into the environment, we have the following hot and cold gas heat balance equation

$$dQ = -G_1 c_{p1} dt_1 = G_2 c_{p2} dt_2 \tag{4}$$

$$Q = G_1 c_{p1}(t'_1 - t''_1) = G_2 c_{p2}(t''_2 - t'_2) \tag{5}$$

and with the so called water equivalent $W = Gc_p$

$$Q = W_1(t_1' - t_1'') = W_2(t_2'' - t_2') \tag{5a}$$

that is, the amount of heat supplied by the hot gas for a constant flow rate G and an average heat capacity c_p is proportional to its temperature change Δt.

Subscript 1 stands for the hot gas, 2 - for the cold gas, t_1', t_1'' - inlet and outlet hot gas temperature, t_2', t_2'' - inlet and outlet cold gas temperature.

Heat transfer surface area necessary to transfer a given amount of heat is found from the heat transfer equation

$$dQ = k\,\Delta t dF \tag{6}$$

or

$$Q = \int_0^F k\,\Delta t dF = k\,\Delta t F \tag{7}$$

where k - overall heat transfer coefficient, which is assumed constant for the whole surface area, $\Delta t \approx t_1 - t_2$ - difference between average hot and cold gas temperature.

In separate cases variable heat transfer coefficients and variable temperature differences may be observed on the surface. To perform calculation of local values of Δt_i and k_i from eq. (4) and (5), the surface must be split into several sections. The most difficult part of the calculation is that of determining the overall heat transfer coefficient k.

For a flat single-layer wall of thickness δ the overall heat transfer coefficient k is given by

$$k = \left(\frac{1}{\alpha_1} + \frac{\delta}{\lambda} + \frac{1}{\alpha_2}\right)^{-1} \tag{8}$$

for a cylindrical shell of internal diameter d_1 and external diameter d_2 by

$$k = \left(\frac{1}{\alpha_1 d_1} + \frac{1}{2\lambda}\ln\frac{d_2}{d_1} + \frac{1}{\alpha_2 d_2}\right)^{-1} \tag{9}$$

The values of thermal conductivity λ for walls of different materials are given in reference tables. From them, heat conduction through the wall is evaluated. Evaluation of the heat transfer coefficients α_1 and α_2 is more complicated. For forced convection heat exchange between a gas and a wall, the amount of heat is determined by the temperature difference and the gas flow velocity.

In most cases combined heat transfer of convection, conduction and radiation is observed. Usually convection heat transfer also includes an effect of conduction though convection is a predominant process. Thus a general expression is

$$\alpha = \alpha_c + \alpha_r \tag{10}$$

where α_c - coefficient of convective heat transfer with the account of conduction, α_r - coefficient of radiant heat transfer.

In a high temperature gas flow, the contribution of radiant heat transfer is very significant. For known absorption rates and temperature difference the component of radiant heat transfer may be found analytically.

The process of heat transfer under radiant-convective or radiant-conductive interaction, or combined heat transfer, is very difficult to define analytically.

Because of the periodic exposure of the heated surface in a regenerator to alternating hot and cold flows, important factors of their performance are the periods of hot blast τ_1 and cold blast τ_2.

A complete cycle of hot and cold blasts is

$$\tau = \tau_1 + \tau_2 \tag{11}$$

The periods of hot blast and cold blast are not necessarily strictly equal. Because regenerators operate in transient conditions only, the values of heat flux in the calculations must be related to cycle, rather than for a unit time.

$$Q_a = k_a(t_1 - t_2) \tag{12}$$

where k_a - average overall heat transfer coefficient per hot and cold cycle, t_1 - average temperature of primary heat carrier per hot blast, t_2 - average temperature of secondary heat carrier per cold blast.

In an ideal case, average temperature of the matrix surface is equal in hot blast and in cold blast. Then for an ideal regenerator

$$k_{ai} = \frac{1}{\frac{1}{\alpha_1 \tau_1} + \frac{1}{\alpha_2 \tau_2}} \tag{13}$$

where α_1 - total heat transfer coefficient per hot blast, α_2 - total heat transfer coefficient per cold blast.

For equal hot and cold blast periods

$$k_{ai} = \frac{\tau}{\frac{1}{\alpha_1} + \frac{1}{\alpha_2}} \tag{14}$$

We see that eq. (4) is proportional to the heat transfer equation for recuperators. Therefore in a general case, calculation of heat transfer coefficients α_1 and α_2 is analogic for regenerators and recuperators, and common prediction equations are valid. For a prediction of the heat transfer coefficient with the account of convection and radiation and in different cases, see [2, 3]. Note, that high temperature regenerators exhibit some specific features of heat transfer.

4. PECULIARITIES OF HIGH TEMPERATURE HEAT TRANSFER

Convective heat transfer coefficient α_c depends on a number of factors and variables. In a general case α_c is a function of the size, shape and temperature of the surface, of fluid dynamics, flow regime, temperature and physical properties of the fluid, and of some other parameters.

The relation in the dimensionless form is

$$Nu = f(Re, Pr) \tag{15}$$

where $Nu = \alpha l/\lambda$ - Nusselt number, $Re = ul/\nu$ - Reynolds number, $Pr = \nu/a$ - Prandtl number, α - heat transfer coefficient, l - characteristic dimension, λ - thermal conductivity, u - velocity, ν - kinematic viscosity, a - thermal diffusivity.

The exponential relation often used in practice is

$$Nu = cRe^{m}Pr^{n} \tag{16}$$

and for gases of constant Pr

$$Nu = cRe^{m} \tag{17}$$

The process of high temperature heat transfer involves a change of the gas temperature, and consequently, of its physical properties. This gives rise to the problem of accounting for the effect of gas physical properties in the boundary layer with a variable temperature, in other words, the problem of the choice of the so called reference of characteristic temperature to determine the gas physical properties in the similarity numbers. There exist several approaches to this choice.

Authors of different publications give the following reference temperatures: gas flow temperature t, wall temperature t_w, average boundary layer temperature t_m. Arithmetic mean temperature of the boundary layer is often chosen $t_m = 0.5(t + t_w)$.

A recommendation of including a temperature factor, i.e. the ratio of absolute flow temperature and wall temperature, in eq. (17) is often made. Then it becomes

$$Nu = cRe^{m}\left(\frac{T_f}{T_w}\right)^{p} \tag{18}$$

M. Mikheev [4] from his own experiments and an analysis of other works suggested the average gas flow temperature as the reference, while it is known or easily determined in engineering calculations.

From studies in high temperature flows, performed at the Institute of Physical and Technical Problems of Energetics, Academy of Sciences of the Lithuanian SSR [2] it was concluded, that bulk flow temperature t in a given cross section is a sufficiently accurate reference. This choice excludes the temperature factor $(T_f/T_w)^p$ from eq. (17).

In high temperature heat exchangers, air is heated up to 1200°C and combustion flow is cooled down from 1500°C to 300°C and lower. We know that the volume of a gas is proportional to its temperature

$$V_2 = V_1 \frac{T_2}{T_1} \tag{19}$$

This means that the hot gas volume decreases 3 to 4 times during cooling, and the volume of air increases accordingly during heating. In a regenerator, a corresponding change of the flow velocity occurs and is accompanied by a change in the heat transfer rate.

The choice of optimal flow velocity in a high temperature heat exchanger must be based on considerations of its thermal efficiency and economy. On the other hand, it must not exceed the admissible velocity of ash erosion.

Temperature variation of the hot gas and the wall involves a corresponding variation of the radiant component which must also be considered.

With respect to the processes of heat transfer in heat exchangers they may be divided into three groups:

1. Heat exchangers with predominantly convective heat transfer.

2. Heat exchangers with predominantly radiant heat transfer.

3. Heat exchangers with combined heat transfer.

Apparatus of the first group are used for heat recovery in power plants and industrial plants. They may be of different operation principles (recuperative and regenerative), different constructions (tubular, plate, pebble-bed). This group also includes heat exchangers used in nuclear power plants. Their design calculations are based either on the classical theory of similarity, or on the solution of differential equations of transfer.

Apparatus of the second group are mainly combustion furnaces of boilers, tubular furnaces in chemical and petrochemical production a.o. Thermal design of such equipment is based on a method of sectional evaluation of radiant heat transfer. As a rule the convective component in such apparatus is insignificant, though the heat transfer processes are sometimes very complicated.

There exists no distinct boundary between these two groups of heat exchangers, because both convection and radiation are ob-