Analysis of Deep Heat Recovery From Flue Gases

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Abstract. The condensing mode of operation steam boiler using as fuel blast furnace gas is analyzed. The heat transfer in the flue of the boiler is calculated, the maximum depth of the cooling of the flue gases is defined, the scheme of installation of the heat exchanger in the flue tract is developed. The optimal operating parameters of heat exchanger are defined.

1. Introduction

The main direction of energy development is currently improving the efficiency of natural fuel use. The majority of boilers currently operate on natural gas and have an efficiency of 88-93%. In the thermal balance of gas-fired boilers are the biggest losses with exhaust gases (5-10%), therefore the only area of substantial increase of efficiency in the boilers is to reduce these losses. It is possible to do by reducing the flue gas temperature.

Currently, the minimum flue gas temperature for boilers is equal to 120-130 °C, in this temperature of leaving gases eliminates the condensation in the flues and smoke stacks, natural draught increases. However, in many boilers the actual temperature of the flue gas is equal to 180 – 200 °C. In hot-water boilers to further reduce the flue gas temperature is generally not provided. Reserves of natural fuel are reduced, deterioration of the environment. Therefore, the task of reducing the flue gas temperature becomes very important.

There are various technologies deep heat of products of combustion. The temperature of exhaust gases it is possible to reduce to 65-70 °C due to the use of the latent heat of vaporization of water vapor. Therefore, the efficiency (gross) boiler unit increases by 2-6 %. Heat of the flue gases can go on heating the raw water or chemically purified water, preheating combustion air, preheating of reverse network water in the hot water supply system, the technological needs of enterprises, etc.

As the heat exchanger can be used in heat exchangers contact and surface types.

The contact type heat exchangers are compact, low metal content and relatively low energy consumption for operation. The flue gases are cooled to 40 °C, while 60 – 90% of water vapor condensates. The main disadvantage is the saturation of the heated water with carbon dioxide, this water can cause corrosion (pH=3.5-5.0), this limits its further application.

This drawback is deprived of the heat exchanger of surface type. On the other hand, these devices are more metal-intensive. The heating surface is performed mostly from bimetallic tubes with transverse fins. Use surface heat exchanger allows flue gases to cool to a lower temperature (less than 40 °C).

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The main problem in the heat exchanger of the condensing type associated with the harmful effects of aggressive condensate on the contact surface with him. The resulting during the cooling of the flue gas condensate absorb O₂ and CO₂ and becomes higher corrosion properties. This leads to a gradual destruction of not only the heat exchanger but also the flue system and smoke stack. Therefore it is
necessary to limit the cooling of the flue gas dew point temperature or preheating the gases before the smoke stack. In practice, the most widely used scheme with bypass: one part of the flue gases passed through the heat exchanger and the second past him. At the entrance to the smoke stack both streams are mixed, providing a temperature that does not allow corrosion. The amount of flue gases passing through the heat exchanger must ensure that it is condensing its mode, as it increases the heat transfer coefficient, leads to a reduction of the size of the heat exchangers and the cost of their installation and operation. Thus, the range of performance of the condensing heat exchangers in real conditions is significantly limited. It is therefore necessary to determine the conditions in which the heat exchanger will operate with the greatest intensity, will provide maximum heat power.

2. Optimal operation of heat exchanger

It is necessary to determine the conditions in which the work of the condensing heat exchanger of the type will be the most effective. As an example, consider shell and tube heat exchanger designed for heat recovery of exhaust flue gas of boilers E-220-9,8-540 GD. The original boiler is designed to operate on a mixture of natural gas (40%) and blast furnace gas (60%). In consequence, the boiler is transferred to the combustion of blast furnace gas is mainly (98%). Causing the temperature of the flue gases after the boiler has increased from 165 to 192 °C. the scheme of the heat exchanger shown in Fig. 1. For cooling was used untreated water.

![Diagram of the system heat recovery of flue gases](image)

**Figure 1.** Diagram of the system heat recovery of flue gases: 1 – boiler; 2 – induced-draft fan; 3 – smoke stack; 4 – heat exchanger.

The calculation was performed with the following initial data:
- fuel is a mixture of natural gas (2%) and blast furnace gas (98%) of the following composition: CH₄=0.20%, C₂H₆=0.01%, H₂=10.16%, N₂=38.13%, CO₂=21.68%, CO=26.81, H₂O=3.01%;
- excess air coefficient past of the boiler 1.3;
- dry gas flow past of the boiler 99.34 kg/c;
- flue gas temperature past of the boiler 192 °C;
- flue gas temperature at the entrance to the smoke stack 162 °C;
- the water temperature at the inlet of the heat exchanger 10 °C.
For the given parameters the moisture content of the flue gases past of the boiler $d'' = 0.05 \text{ kg/kg}$, and the dew point temperature $t_p = 40.5 \degree \text{C}$. Take the temperature of the water at the outlet of the heat exchanger is $5 \degree \text{C}$ below the dew point temperature $t_{w,\text{max}} = 50 \degree \text{C}$.

It is also necessary to determine the mass flow rate of flue gases and the temperature past of the heat exchanger. The temperature of the mixture of two streams that passed through the heat exchanger and the flue tract, at the point II was equal to the minimum possible temperature $t_{\text{sm}} = 50 \degree \text{C}$. This temperature should be as small as possible, which corresponds to the maximum degree of heat recovery. On the other hand, it is necessary to eliminate the possibility of condensation of water vapor from the flue gases in the flue path and in the smoke stack. Calculation of heat transfer in the smoke stack and the flue tract was performed, the minimum possible temperature is equal $76 \degree \text{C}$.

The equation of heat balance

$$
\left( ct' + \left( r + c_s t' \right) d' \right) (G_e - G) + \left( c t'' + \left( r + c_s t'' \right) d'' \right) G = \left( c t_{w,\text{in}} + \left( r + c_s t_{w,\text{in}} \right) d_{w,\text{in}} \right) G_e
$$

Material balance equation for moisture

$$
d_{w,\text{in}} G_e = d' \left( G_e - G \right) + d'' G
$$

Here $c$ and $c_s$ – mass specific heat of dry flue gases and water vapor, respectively, $J/(\text{kg} \cdot \text{K})$; $t'$, $t''$ and $t_{w,\text{in}}$ – the temperature of the flue gases at the inlet, the outlet of the heat exchanger and mixes, respectively, $\degree \text{C}$; $r$ – the latent heat of vaporization of water, $J/\text{kg}$; $d'$, $d''$ and $d_{w,\text{in}}$ – the moisture content of the flue gases at the inlet, the outlet of the heat exchanger and mixture, respectively, $\text{kg/kg}$; $G_e$ – the mass flow rate of dry flue gases at the boiler outlet (at point I), $\text{kg/s}$; $G$ – the mass flow rate of dry smoke through the heat exchanger, $\text{kg/s}$.

Let’s denote the proportion of flow through the heat exchanger of the total flow rate of the flue gas as $n = G/G_e$. The mode of condensation in the heat exchanger starts at a fraction of the flow through the heat exchanger $n = n_{\text{max}}$, in this case $d' = d''$. After transformation of (3) with (4) we obtain

$$
n_{\text{max}} = \frac{t' - t_{w,\text{in}}}{t' - t''}. \tag{3}
$$

Consequently, the proportion of flow through the heat exchanger must satisfy the following relation $n < n_{\text{max}}$, in order to use in condensing mode the heat of the phase transition.

On the other hand, the minimum share of the flow through the heat exchanger $n_{\text{min}}$ is determined from the following conditions: temperature of heating medium (flue gas) at the outlet of the heat exchanger $t''$ can not be lower than the temperature of the heated medium (water) at the entrance $t_{w,\text{in}}$. The water vapor in the flue gas at the outlet of the heat exchanger is in a saturated condition, and $t_{w,\text{in}} = t''$. Find the minimal proportion of $n_{\text{min}}$ using the equations (1) and (2).

$$
n_{\text{min}} = \frac{c + c_s d' \left( t' - t_{w,\text{in}} \right)}{c_s t_{w,\text{in}} \left( d'' - d' \right) + \left( c + c_s d' \right) t' - \left( c + c_s d'' \right) t''}, \tag{4}
$$

Thus, a shell and tube heat exchanger will operate in condensing mode, if it will carry part of the flue gases in the range from $n_{\text{min}}$ to $n_{\text{max}}$ from consumption of dry smoke. Thus the heat capacity of flue gas will be

$$
Q_d = G \left[ c \left( t' - t'' \right) + \left( r + c_s t' \right) d' - \left( r + c_s t'' \right) d'' \right]. \tag{5}
$$

The minimum flow rate of the heated medium $G_{w,\text{min}}$ at a given maximum temperature at the outlet of the heat exchanger $t'_{w,\text{max}}$:

$$
G_{w,\text{min}} = \eta \frac{Q_d}{c_s \left( t'_{w,\text{max}} - t' \right)}, \tag{6}
$$
Here $\eta$ – the efficiency of the heat exchanger, taking into account the heat loss to the environment; $c_s$ – the specific mass heat of water, J/(kg⋅K).

Maximum temperature of heating water is limited to 3...5 °C below the dew point of the flue gas $t'_p$ at the inlet of the heat exchanger. In this case, the whole surface of the pipe in the apparatus from the flue gases will have a temperature below the dew point, which should facilitate the intensification of heat transfer by condensation of unsaturated flue gas, that is

$$t'_{p,\text{max}} = t'_p - (3...5)$$

(7)

3. The results of the study

In order for the heat transfer in the heat exchanger was the most effective, you must comply with the following requirements:

- maximum heat output should correspond to equation (5). It determines the minimum temperature of the gases at the outlet of the heat exchanger;
- the minimum flow rate of the heated medium, which is supplied to the heat exchanger, is determined from the equation (6);
- the heating medium fed to the heat exchanger in an amount of from $n_{\min}$ to $n_{\max}$ the total consumption of dry flue gases. The shares $n_{\min}$ and $n_{\max}$ shall be determined by equations (3) and (4).

In the given conditions the minimum and maximum share of consumption by dry weight of flue gases going through the heat exchanger, are $n_{\min} \approx 0.656$ and $n_{\max} \approx 0.766$, which is consistent with the data [1].

Since the flow of heating medium through the heat exchanger is limited in such a narrow range, the thermal capacity of the heat power is also limited. In this case, maximum flow is limited by the limit mode condensation in the heat exchanger, i.e. the minimum degree of cooling of flue gases to their dew point.

Obviously fundamentally possible to operate the heat exchanger at high flow rates, i.e. at higher thermal load. But then the increase in consumption must correspond to a proportional increase in the temperature of flue gases at the outlet of the heat exchanger. This is necessary to maintain a given temperature level of the mixture. However, then the device will not work in condensing mode and the useful heat output by flue gas will be constant and, moreover, the smallest possible values. Thus, only the use of condensing mode allows to significantly increase the efficiency of utilization of heat of flue gases.

It should be noted that a significant expansion of the range of the flow rate through the heat exchanger can be achieved by reducing the temperature of the mixture. It is also possible to cool to 100% of the flue gas. However, you must provide technical measures to prevent condensation of water vapor from the flue gases on the surfaces of the flue and smoke stack. For this purpose, use a special coating or heating of flue gases. But the economic feasibility of such measures needs further study.

Thus, in real conditions of operation of boiler units use recuperative heat exchangers for cooling the outgoing flue gases in the regime of condensation is due to a limitation of the share of the flue gases going through the heat exchanger. Moreover, the possible range of the share of consumption of smoke is quite narrow (no more than ~10%) that determines the limit of heat power of the heat exchanger.

In Fig. 2 shows the dependence of the temperature and moisture content of the gas at the outlet of the heat exchanger of the proportion of flue gases passing through the heat exchanger. In Fig. 3 shows the effect of the share of the flue gases in heat power and minimum water flow. It is seen that a linear temperature change with the change of the flow rate corresponds to a nonlinear change of the other parameters of the heat exchanger.
Figure 2. A modification of physical parameters of flue gases at the outlet of the heat exchanger

Figure 3. A modification of heat power in the heat exchanger
4. Conclusion
A method of optimizing operating parameters of a condensing heat exchanger of the recuperative type. On its basis defines the basic operating parameters of the heat exchanger for the operating conditions of the boiler E-220-9,8-540 GD. It is shown that the most efficient operation of heat exchangers in condensing mode. In real conditions the organization of condensation associated with the restriction of the share of the flue gases going through the heat exchanger. The possible range of the share of consumption of smoke is quite narrow (no more than ~10%) that determines the limit of heat power of the heat exchanger.

5. References
[1] Byakin I G, Shatskikh Y V and Melnichuk A E 2010 The fifth Russian national conference on heat transfer vol 4 (Moscow) p 227