

EVALUATION OF HEAT TRANSFER COEFFICIENTS DURING THE WATER VAPOR CONDENSATION CONTAINED IN THE FLUE GAS

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Abstract. Is shown the influence of the heat transfer coefficient from the wet flue gas to the heat exchange surface on the overall heat transfer coefficient in the gas-air heat exchanger with the water vapor condensation. Experimental data are compared with calculations based on the mathematical model of the condensing heat exchanger.

1 Introduction

Thermal design of the heat exchanger surface is a classical task in heat engineering. But heat transfer from two-component medium when changing the aggregate state one of the components has certain complexity. Such task is necessary to solve when the mathematical model for flue gas heat recovery condenser is developing. [1]. Installation scheme is shown in Figure 1.

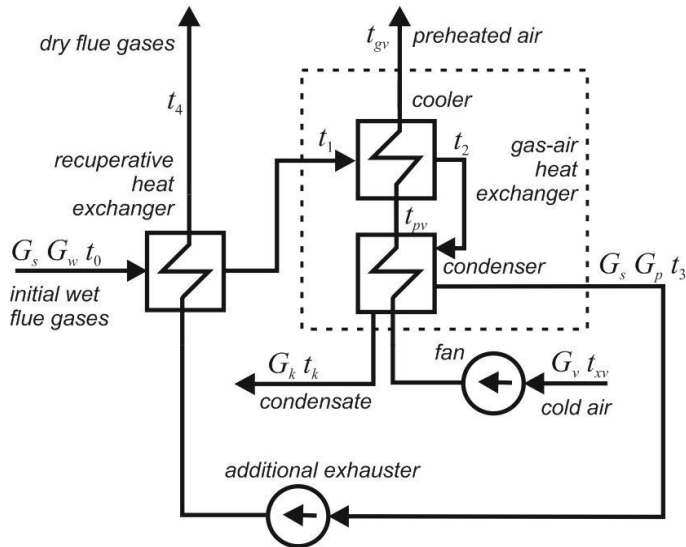


Figure 1. Deep heat recovery of flue gas, installation scheme.

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Initial flue gases are pre-cooled in the recuperative gas-gas surface plate-type heat exchanger by heating the dry flue gases. Then they are fed to the gas-air surface plate-type heat exchanger. Structurally, it is a single unit conventionally divided by two parts: cooler and condenser. This heat exchanger heats the cold air supplied by the fan. The heating carried out by cooling the flue gases and using the latent heat of water vapor condensation. Drained flue gases for heating are fed to a heat exchanger by means of the additional exhauster to compensate aerodynamic resistance into the flues, heat exchanger and condenser.

2 Research methodologies

The mathematical model of the installation includes the heat and material balance equation, as well as heat and mass transfer equations. The input variables are the cold air temperature t_{xy} and the initial flue gas mass flow with initial temperature t_0 , which consist of a dry flue gas G_s and water vapor G_w . Heat exchange surface area can be calculated if set in the first approximation the heat transfer coefficients of each heat exchanger. Based on the calculated heat exchange surface areas the heat exchangers can be designed. The geometrical parameters of heat exchange surface areas allow to clarify the heat exchangers heat transfer coefficients and to adjust the streams temperature. The heat transfer coefficient is calculated as follows [2]:

$$k_t = \frac{1}{\frac{1}{\alpha_1} + \frac{\delta_s}{\lambda_s} + \frac{1}{\alpha_2}}, \quad (1)$$

where α_1 – coefficient of heat transfer from the hot medium (flue gases) to the heat exchange surface, δ_s – sheet thickness, λ_s – thermal conductivity of sheet, α_2 – coefficient of heat transfer from the heat exchange surface to the heated medium.

For heat exchangers where both gaseous media do not have a phase transition (no water vapor condensation), the heat transfer coefficients are calculated by the usual method and depend mainly on the gas velocity. This is true for calculation the heat transfer coefficient of the recuperative heat exchanger and the cooler. When calculating the condenser heat transfer coefficient the coefficient of heat transfer from the heat exchange surface to the air flow is calculated in the same way. For air flow under atmospheric pressure criterial equation is

$$\text{Nu} = 0,018 \cdot \text{Re}^{0,8},$$

where $\text{Re} = w \cdot D / \nu$, w – air velocity, D – equivalent diameter of the channel and ν – air kinematic viscosity. The coefficient of heat transfer from the heat exchange surface to the air flow is

$$\alpha_2 = \text{Nu} \cdot \lambda / D.$$

Air velocity is limited to a quadratic increasing aerodynamic drag of the air tract. When the air velocity above 18 m/s the heat exchanger resistance exceeds 1 kPa. It is undesirable due to the fan characteristics to supply air. Turbulent airflow begins at speeds of over 3 m/s. For an acceptable heat transfer the air velocity must be 13÷18 m/s, and the heat transfer coefficient is 57÷75 W/(m²·K).

The main difficulty is the determination of the heat transfer coefficient from the wet flue gas to the heat transfer plate surface α_1 during the water vapor condensation contained in them. Proceeding from the relation (1), with known thermal resistance of the heat transfer surface and the heat transfer coefficient from the surface to the air flow α_2 , it is possible to assess the impact α_1 to overall heat transfer coefficient k_t (Figure 2).

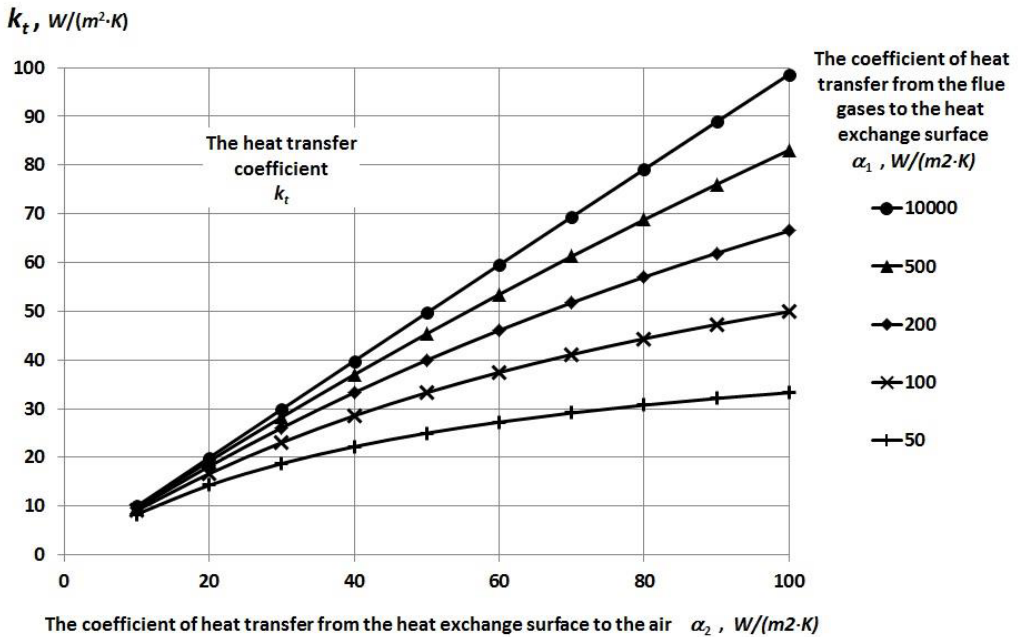


Figure 2. The α_1 influence on the heat transfer coefficient.

For small values α_1 ($\alpha_1 \ll \alpha_2$) the heat transfer coefficient depends mainly on the heat transfer from the condensable vapor in the flue gas stream α_1 . For large values α_1 ($\alpha_1 \gg \alpha_2$, $\alpha_1 > 500$), the heat transfer coefficient to a greater extent depends on the heat transfer from the air α_2 . In the considered heat recovery condenser α_2 less than $100 W/(m^2 \cdot K)$. Due to condensation, the heat transfer from the wet flue gas is much higher if turbulent regime is provided.

For today is not developed a universal method of calculating the heat transfer coefficient from the flue gas (or vapor mixture at low vapor content) with water vapor condensation to the heat transfer surface. All known methods are based on the experimental data results analysis and offer the equations in rather narrow range of heating media physical parameters. Choosing and applying the calculation method for functionally similar heat recovery units requires analysis and justification. Simulation of the process carried out by the Moscow Energy Institute scientists, led by professors Y. Kuzma-Kichta, A. Sedlov, A. Solodov, A. Garjaev [3, 4, 5]. In the thesis D. Buhonov gives an overview and analysis the known calculation methods of condensing heat exchange systems and their applicability to the heat recovery units calculation. Deviations of heat transfer coefficients according to known techniques are 70÷500%. These authors offer its own method for calculating the heat transfer coefficient from steam-air mixture to the heat transfer surface [6]. Another technique is described by Professor A. Kudinov from Samara State Technical University [7, 8, 9]. This technique is based on the experimental data analysis by testing the operated heat recovery units. Corresponding coefficients were calculated for the criterial equation derived by similarity principle. The same methods apply by other authors [10, 11].

Calculated by these methods, heat transfer coefficients from vapor condensed flue gases to the surface of described above heat recovery condenser showed a very large difference (from 52 to $800 W/(m^2 \cdot K)$).

3 Results

In Tomsk CHP-3 was created an experimental installation for flue gas deep heat recovery. Five experiments were performed, each lasting 30 to 50 minutes, to determine the experienced heat transfer

coefficients. The error in determining the heat transfer coefficient in this experiment series is within 15%. Table 1 summarizes the processing experiment results compared with the results calculated by mathematical model of heat transfer process in a condenser. The input data for the mathematical model were the experimental parameters and the geometric characteristics of the experimental installation.

Table 1. Comparing the experimental data with calculated by the mathematical model.

Parameter	The average values by experiments	Calculated values by the mathematical model
Excess air ratio α	2,1	2,1
The average air temperature entering the condenser t_{av} , °C	3,8	3,8
The average air temperature at the condenser outlet t_{gn} , °C	29,7	25,1
The average wet flue gas temperature entering the condenser t_1 , °C	56,5	56,5
The average dry flue gas temperature at the condenser outlet t_3 , °C	37	35,8
The moisture content of initial wet flue gases, grams per kilogram of dry gas, g/kg.d.g.	74,4	74,4
The moisture content of dried flue gases, g/kg.d.g.	36,5	38,1
The volumetric flue gas flow rate, m^3/c	0,108	0,100
The theoretical flow rate of condensate, kg/s	0,0046	0,0042
The volumetric air flow rate, m^3/c	0,493	0,4900
The thermal power transferred to the air from the flue gas Q , kW	13,00	13,00
The average temperature difference Δt , °C	23,0	26,34
The overall heat transfer coefficient k , $W/(m^2 \cdot K)$	61,8	67,30
The air velocity in the condenser cross section w , m/c	22,35	19,33
The heat transfer coefficient from the heat transfer surface to the air α_2 , $W/(m^2 \cdot K)$	82	73,70
The heat transfer coefficient from wet flue gases to the heat transfer surface α_1 , $W/(m^2 \cdot K)$	251	798,1

As can be seen from the experimental results, the heat transfer coefficient $k = 62 \pm 6 \text{ W}/(m^2 \cdot K)$. The calculated heat transfer coefficient from the heat transfer surface to the air $\alpha_2 = 82 \pm 4 \text{ W}/(m^2 \cdot K)$ and a known plate thermal resistance allow to determine the heat transfer coefficient from the wet flue gases to the heat exchange surface by the equation (1). The last one considering the error was found to equal $\alpha_1 = 251 \pm 50 \text{ W}/(m^2 \cdot K)$.

A mathematical model of heat transfer in the condenser is built on determining the overall heat transfer coefficient by calculating the heat transfer from the wet flue gases to the sheet cooled by air on the other side. The heat transfer coefficient α_1 calculated by Kuzma-Kichta equations in the mathematical model gives a result $\alpha_1 = 798 \text{ W}/(m^2 \cdot K)$.

4 Conclusion

- A determining influence on the overall heat transfer coefficient in the gas-air heat exchanger with heating air and water vapor condensation in the flue gases has the heat transfer coefficient from the sheet to the air α_2 . When air velocity comparable to flue gas the heat transfer coefficient from them to the sheet is higher $\alpha_1 > \alpha_2$ due to heat exchange intensification by the water vapor condensation.
- Existing methods of calculating the heat transfer coefficient from the wet flue gases to the heat exchange surface with the water vapor condensation do not allow to obtain results that would provide a good agreement with experimental results. Should be carried theoretical and experimental study of heat transfer between the flue gas (or gas mixture) and the cooling surface, when contained therein water vapor are condensing.

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