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Calculation of the optimal parameters of the heat transfer surface of the condensing heat exchanger in boiler units

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ABSTRACT: It is shown that for the transition to the mode of condensation of vapors from flue gases in the operation of the boiler unit, it is necessary that the heat exchange surfaces of the condensing heat exchanger be at a temperature below the dew point. An algorithm for calculating the thermal parameters of the finned surface of heat exchangers has been developed, which makes it possible to speed up the operation of calculating the geometric characteristics of a condensing heat exchanger and improve the accuracy of the results. The optimal parameters of the heat transfer coefficient are found, based on the values of the step between the fins and their height.

KEY WORDS: condensing heat exchanger, dew point, finned heat exchanger.

I.INTRODUCTION

Increasing the efficiency of heat power plants largely depends on the utilization of the heat of flue gases of HPP discharged into the environment and the development of modern methods and effective connection to the thermal circuit of boiler units of heat exchange plants for deep cooling of these gases to the dew point, called condensing heat recovery units.

At a flue gas temperature of 40^0C at the outlet of the heat exchanger, according to the test results [1], from the same volume of combustion products in the surface heat exchanger, condensate from water vapor is obtained by 25% more than in the contact economizer of standard boilers, and therefore the same amount of heat is utilized and transferred consumer.

The calculation of the optimal design parameters of condensing heat exchangers in terms of heat transfer requires the use of reference tables on the thermophysical characteristics of water and flue gases, graphs to clarify the emissivity of CO_2, water steam. Some calculation steps constitute approximate iteration algorithms.

II. LITERATURE SURVEY

The heat transfer coefficient for the turbulent regime of water flow was determined by the model of M.A. Mikheev [2]. Due to the fact that the number of rows along the flow for boilers of different capacities varies, the third row was chosen to calculate the heat transfer coefficient from flue gases to pipes:

the calculation steps are approximate iterative algorithms.

To simplify the calculation, a number of assumptions are made:

- the pipe wall temperature is approximately equal to the water temperature and less than the arithmetic mean flue gas temperature by 20 °C,
- transition from a cross-flow system of movement of heat carriers to a counter-current one - transfer of a cylindrical wall to a flat wall to calculate the heat transfer coefficient.

All this allows not to exceed the permissible error in the result.

III. DESCRIPTION OF SCHEME

The installation of the required heat output is assembled using flanges from several heat exchanger modules connected in series or in parallel by gases, shown in Fig.1.

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The algorithm for thermal calculation of the thermal characteristics of heat exchangers is based on the balance of heat during its exchange between flue gases and cooling water:

$$Q_2 = Fk_t \Delta t_l \leftrightarrow Q_w = G_w c_{pw}\left(t_{w2} - t_{w1}\right). \tag{1}$$

where F is the area of the heat transfer surface of the heat exchanger, m^2; k_t is the heat transfer coefficient, W/m; Δt_l - temperature gradient through the heat exchanger, ^{0}C; G_w is the flow rate of cooling water through the heat exchanger, kg/s; c_{pw} - heat capacity of water, W·s/(kg K); t_{w1}, t_{w2} - water temperature at the inlet and outlet of the heat exchanger.

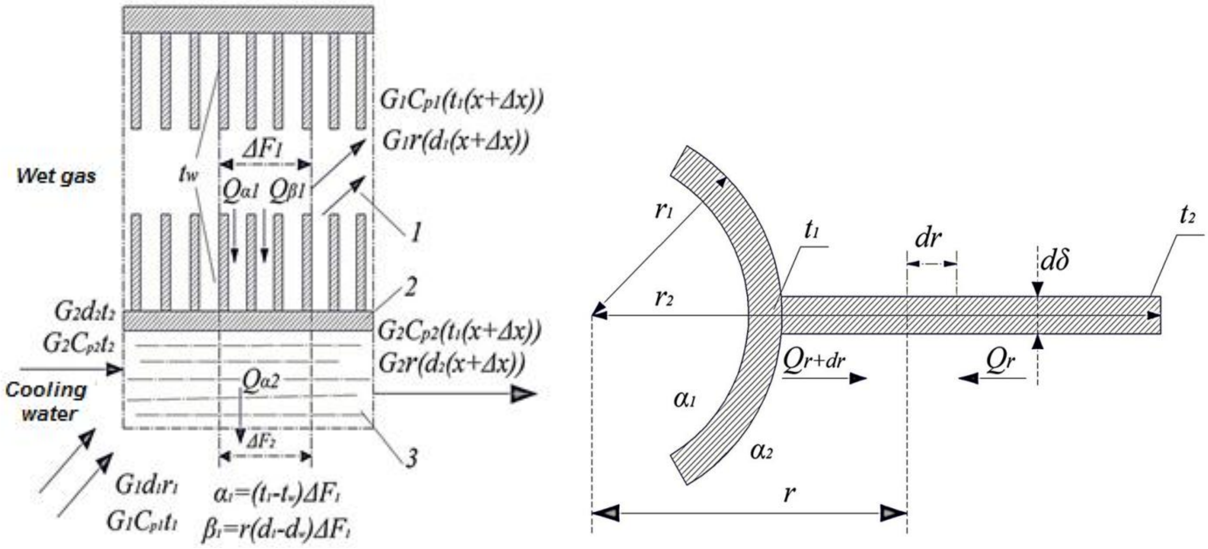

Fig.1 The section of a finned heat exchanger - utilizator.

1 - hot coolant (moist gas); 2 - ribbed wall separating heat exchange media; 3 - cold coolant (liquid).

IV. METHODOLOGY

A. Survey of publications

For the existing turbulent regime of water flow in the condensing heat exchanger, the calculation of the heat transfer coefficient was carried out according to the formula of M.A. Mikheev [2]. As is customary in the classical theory of heat transfer, when changing the number of rows along the flow for boiler units of various capacities, the third row is chosen to calculate the heat transfer coefficient α_{g3} steam-gas mixture to the pipes of the condensing heat exchanger:

$$\mathrm{Nu}_g = 0{,}41 \cdot \varepsilon_s \,\mathrm{Re}_g^{0,6}\, \mathrm{Pr}_g^{0,33} \rightarrow \alpha_{g3} = \lambda_g \,\mathrm{Nu}_g / d_2 \tag{2}$$

A number of assumptions were made in the calculation:
– the temperature of the pipe wall is approximately taken to be equal to the water temperature with a difference of 20 0C with the arithmetic mean flue gas temperature,
– transition from a cross-flow system of movement of heat carriers to a counter-flow one,
– simplification of a cylindrical wall to calculate the heat transfer coefficient to a flat one.
All this allows not to exceed the permissible error.
The following techniques are proposed to speed up the operation of calculating the geometric characteristics of the condensing heat exchanger and improve the accuracy of the results. Taking into account that the temperature difference between flue gases and water in the condensing heat exchanger is very small, then for the dependence of

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thermophysical properties on temperature, the interpolation by the Lagrange polynomial of the 2nd degree was performed [3]:

$$L^{(2)}(t)=\frac{(t-t_1)(t-t_2)}{(t_0-t_1)(t_0-t_2)}y_0+\frac{(t-t_0)(t-t_2)}{(t_0-t_1)(t_1-t_2)}y_1+\frac{(t-t_0)(t-t_1)}{(t_2-t_0)(t_2-t_1)}y_2$$

Selected by three values of temperature and the corresponding thermophysical index. They are denoted by indices 0,1,2.
Emissivity index of 3 atomic gases [3]:

$$\varepsilon_g=1-\exp(-k_g r_n p l_r),\ k_g=\left(\frac{0{,}78+1{,}6r_{\mathrm{H_2O}}}{0{,}316\sqrt{r_n p l_r}}-1\right)\left(1-0{,}37\frac{T_{11}}{1000}\right),$$

where $r_n=r_{\mathrm{RO_2}}+r_{\mathrm{H_2O}}$ - in sum, the value of the parts of the volume of 3 atomic gases and water vapor; kg is the radiation transmittance of the outgoing flue gases.
Radiant energy heat transfer coefficient [3]:

$$\alpha_r=5{,}13\varepsilon_g T_{g1}^3\left[1-(T_{sz}/T_{g1})^n\right]/(1-T_{sz}/T_{g1})$$

The calculation of the average temperature gradient was carried out according to the multiple cross flow formula [3]:

$$\Delta t_c=\frac{-P\cdot R(t_{11}-t_{21})}{m\ln\{1+R\ln(Z_{PR})\}},\quad Z_{PR}=(R-1)/\left\{R-\left[(1-PR)/(1-P)\right]^{1/m}\right\} \qquad (3)$$

where the generalizing coefficients P and R are found from the formulas [3]:

$$P=(t_{11}-t_{12})/(t_{11}-t_{21});\ R=(t_{22}-t_{21})/(t_{11}-t_{12})$$

B. Calculation of the characteristics of condensation heat exchangers

In table. 1 shows the results of calculating the thermal characteristics of the condensing heat exchanger, taking into account the parameters of the heat exchangers, and also takes into account the quality of the calorific value of natural gas in boilers

Table 1

Calculation results	Q, MW	Heat transfer and heat transfer coefficients, W/(m^2 ·K)				Δt, °C	F, m^2	n, amount	l_{z1}, m
		mixtures α_2	heat exchanger α_1,	edges α_r	pipes k,				
Krasnoshchekov [4]	42.2	4505	80.9	9.7	86.5	443	1067	85.6	76.8
Calculation	41.9	4807	82.2	11.4	92.4	440.1	1019.8	84.5	74.6

When making calculations applied known thermodynamic characteristics of water, efficiency coefficient of the heat exchanger, physical properties of flue gases.
In the course of applying the algorithm in describing dependencies between specific volume of water and its enthalpy interpolation equations [2] were used:

$$v=D_w p_1^4+C_w p_1^2+B_w p_1+A_w,\ p_1=5-p/10,\quad (\mathrm{m^3/kg});$$

$$I_w=K_w p_1^4+G_w p_1^2+F_w p_1+E_w,\qquad (\mathrm{kJ/kg}).$$

The intensity of heat transfer from flue gases to water is low, so transverse finning is required from the outside of the heat exchanger to increase the heat exchange area of condensing heat exchangers. Increasing compactness in contrast to smooth pipes.
When developing such heat exchangers, their rational geometric parameters are determined. For this purpose, the requirements for the smallest overall dimensions are established for the heat exchanger, taking into account the dimensions of the gas duct of the boiler unit. Comparison of the results is shown in table. 2.

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Table 2

Calculation results	Flue gas and water enthalpies at inlet and outlet, kJ/kg				Qg, kJ/kg	t_{w2}, °C	Δt, °C
	Ig_1	Ig_2	Iw_1	Iw_2			
Claus [5]	9350	7056	1198	1381.6	2624	314	135.9
Estimated data	9482.1	7069.3	1242.2	1350.4	2204.2	309.3	139.2
Calculation results	Heat transfer coefficients from gas to water, fins, pipes, heat transfer of the heat exchanger W/(m^2 ·K)				F_t, m^2	F_{pt}, m^2	N_{pt}, amount
	α_{re}	α_k	α_s	k_t			
Claus [5]	10.8	81.6	91.7	66.8	5102	1.279	5.2
Calculation data	9, 6	82.3	89.6	64, 3	5307	1, 316	6.4

To calculate the developed heat exchange surfaces, the thermal calculation method was used, which consists of the flow balance equations heat through the finned surface of the heat exchanger.
The amount of heat given off by the gases to the heat exchanger includes the heat transferred by the fins and the intermediate part of the pipe among the fins:

$$Q_{pc} = Q_p + Q_c = nQ_{p1} + \alpha_k 2\pi r_1 t_1 \left(L_t - n\delta_p\right) \qquad \textbf{(4)}$$

The heat transferred from each fin is:

$$Q_{p1} = -2\pi r_1 \lambda \delta_p \left(\frac{dt}{dr}\right)_{r=r_1} = 2\pi r_1 \lambda \delta_p m \vartheta_1 \psi \,,$$

here are the generalized thermal parameters of the heat exchanger [6]

$$\psi = \frac{I_1(s_2)K_1(s_1) - I_1(s_1)K_1(s_2)}{I_0(s_1)K_1(s_2) + I_1(s_2)K_0(s_1)} \,, m = \sqrt{2\alpha / \delta_p \lambda} \,.$$

The edge efficiency is found by the formula [6]:

$$E_p = \psi \frac{2r_1^2}{m\left(r_2^2 - r_1^2\right)} . \qquad \textbf{(5)}$$

In well-known calculations [6], not entirely accurate graphs are used to determine the heat transfer coefficient of a bimetallic pipe α_k , which results in an error in the calculations E_p .

V. RESULTS OF CULCULATIONS

For computer processing of iterative calculations, an algorithm was compiled for calculating the characteristics of condensing heat exchangers, on smooth pipes of which fins are installed. Coefficient α_k was determined based on the step values between the ribs and their heights.
From the thinning of the fin naturally reduces the overall utilization of the thermal energy of the flue gases Q_{pc}, by the fin, however calculations show that, in this case, the specific heat transfers of the rib increases. On figure2 marked: thickness of the rib 1 - δ_p = 5 mm; 2- δ_p = 1 mm. At the same time, its step s_p varied in the range: variant 1 – from 21 to 4 mm; option 2 - from 23 to 4 mm. Was taken value of the speed of the outgoing flue gases - 9 m/s. The heat transfer coefficient varied within the following limits: option 1 – 36.8 ÷55.2 W/(m^{2}K), the second option - 37.6 ÷ 55.6 W / (m^{2}K), the value of the heat transfer coefficient of the bimetallic pipe α_t = 42.4 W/(m^{2}K^2).
Installing more fins increases the efficiency of the pipe E_T and lowers a little E_p . With a decrease in the thickness of the ribs, both the efficiency of the rib decreases E_p , and pipes E_T (Fig. 3).

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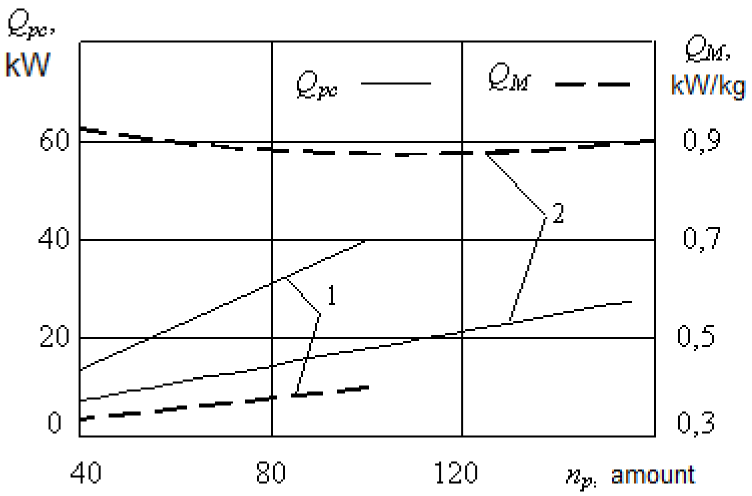

Fig. 2. Dependence of absolute Q_{pc}, and specific Q_m of heat output on the number of fins at different thicknesses: $1-\delta_p$ = 5 mm; 2-δ_p = 1 mm.

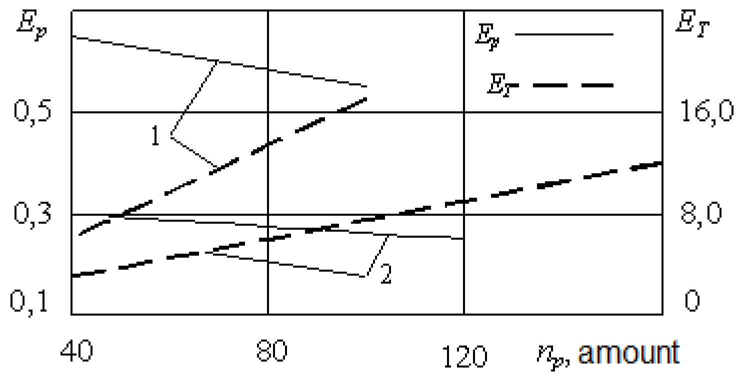

Fig. 3. Dependence of the efficiency of one fin and pipe on their number

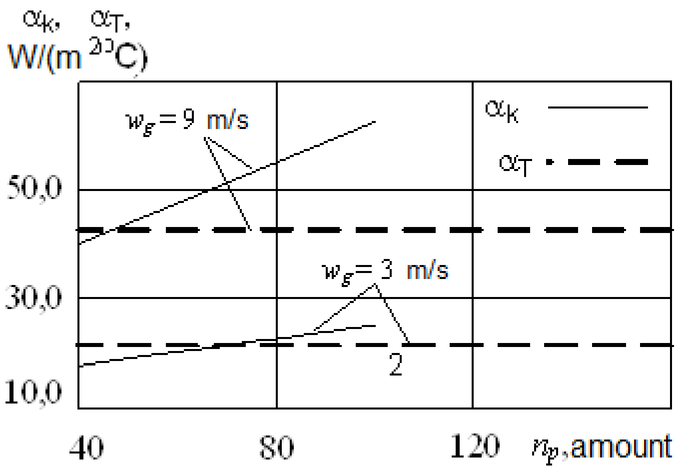

Fig. 4. The dependence of the heat transfer coefficients of the bimetallic α_k and smooth-walled α_T pipes on the number of fins at gas velocities

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VI.CONCLUSION

Reducing the flue gas velocity w_g by 3 times at δ_p = 5mm lowered the value of the heat transfer coefficient α_k , α_T (Fig. 4), and for a pipe E_T decreased to 6.4.
Despite the theoretical results, technically, the installation of a larger number of fins with a minimum step between them complicates the manufacture and maintenance of this element of the condensing heat exchanger.
At the same time, within the values of the heat exchangers actually used in practice, the rates of passage of flue gases for the values of the fin thickness cannot be achieved in the total and specific heat removed to the maximum extent by the heat exchange surface of the condensing heat exchanger of the boiler.

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