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Efficiency of Application the Condensing Heat Utilizers in the Existing Boiler's Unit in Heat Power Station

Normuminov J.A.

Assistant, Dean of Heat Power Engineering, Tashkent State Technical University, Tashkent, Uzbekistan

ABSTRACT: The system of integrated heat supply from steam boilers using condensation heat exchangers based on thermal testing of the boiler unit TGMP-114S in the Syrdarya HPS, which is the largest heat power station in Uzbekistan with a capacity of 3200 MW, was considered. It has been calculated that the increasing of the efficiency the heat supply system for this boilers is equal to 3.4%.

KEYWORDS: Classification, Data Mining, Machine Learning, Predictive analysis, Social Networking Spam, Spam detection.

I.INTRODUCTION

A perspective reserve for improving the efficiency of boiler plants in HPS is the use of the heat of condensation of the vapors contained in the exhaust gases [1]. First of all, this refers to the maintenance of the heating system and hot water supply system at heat power stations. To create a condensation mode of operation a steam boiler (vapor condensation mode from flue gases), it is necessary that the heat exchange surfaces with which the exhaust gases come in contact have a temperature below the dew point. If the boiler room has a DHW circuit, then this mode of operation of the boiler can be ensured by supplying heat energy from the outgoing flue gases through the condensing heat exchanger from cold water supply system.

II. DESCRIPTION OF SCHEME

The most effective use of cold water in the condensing utilizer will be in service:
- two-pipe heating system with open water intake for DHW supply;
- four-pipe heating system.

As for the more commonly used open systems in which heating of cold water in condensing utilizer is possible, they usually use HPS boilers as a heat source (in which steam from steam turbine selections is used to heat heating water). When the cold make-up water is heated in the condensing utilizer, steam extraction decreases, which reduces the efficiency of the regenerative cycle of the HPS. To do this, its are equipped with powerful equipment of water softening and deaeration.

Figure 1 presents the scheme of the boiler unit with condensation heat utilization [2]. The boiler unit includes a steam boiler 1, a water economizer 2, a feedwater deaerator 3 with a nozzle 4 for evacuation of steam, connected by pipe 5 to the main duct 6, a condensation surface heat exchanger-heat exchanger 7 of the heat of combustion products, a shell-and-tube heat exchanger 8 for heating the network water, directed to the feed pipe 9 of the heat supply system, collecting condensate tank 10 with a pump 11, smoke exhauster 12, water cleaning system 13, surface heat exchanger 14. In the main gas duct, a fee is additionally equipped with collector 15 condensation of water vapor from the gate 16 and the hydraulic-separation eliminator device 17

The technical result is an increase in the efficiency of the boiler plant by increasing the volumes of combustion products passed through surface of the condensing heat utilizer due to the use of a recuperative heat exchanger on the suction side of the exhauster for heating the outgoing flue gases, which eliminates corrosion processes in the flue. When this occurs, the content of harmful nitrogen oxides NO_x in the waste products of the combustion the gas fuel due to the removed condensate decreases.

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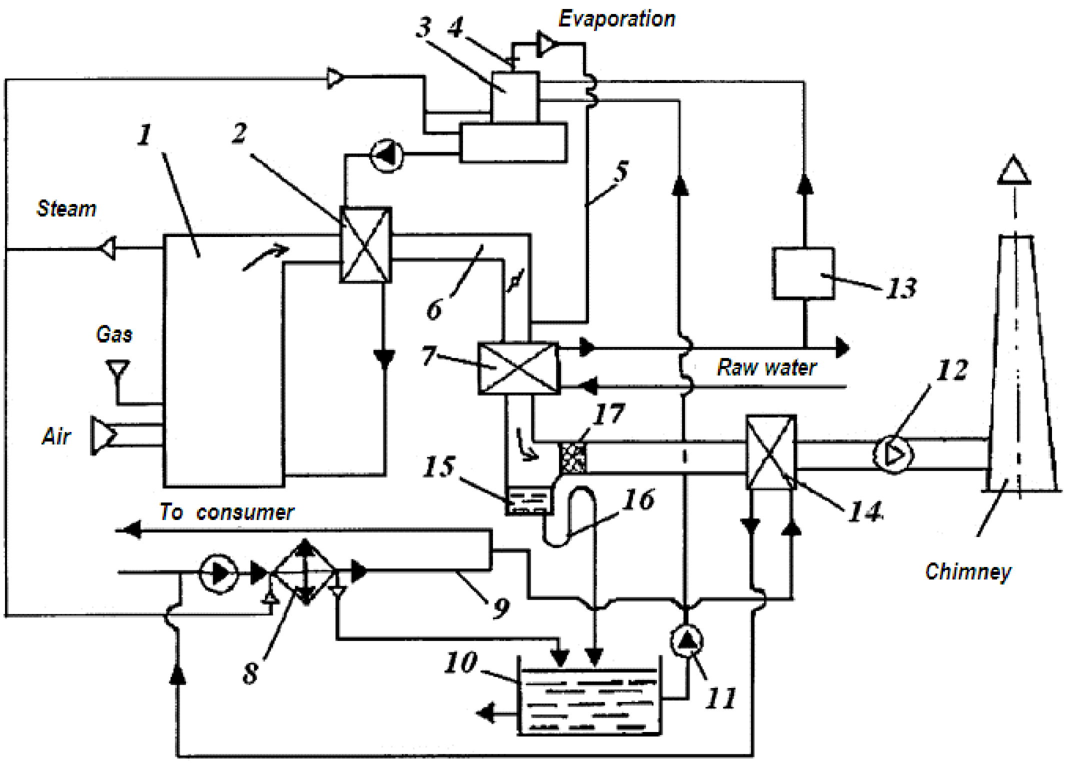

Fig1. Diagram of a boiler plant with a condensing heat exchanger

The use of open systems allows to significantly increase the efficiency of boilers, but this is due to restrictions on water cooling. In the case of closed systems, cold water is successively heated in the CU to 30÷40 ° C, and after softening in the preheater it enters the DHW supply pipe. The need for additional pipes is due to the fact that the DHW water heated in the boiler room must be transported via pipeline to the places of consumption. This increases the cost of the heating system and requires a compulsory feasibility study.

In both cases, the heated water both at the inlet and at the exit from the heat utilizer has a temperature below the dew point of the flue gas t_p, averaging 55 ° C (when burning gas fuel), which ensures the vapor condensation on the entire heat exchange surface of the heat exchanger and its high efficiency. Most of the existing heat supply systems in the Uzbekistan are open two-pipe systems, in which boiler power is supplied by water from a return pipeline with a temperature significantly higher than water supply.

From figure 1 it can be seen that the area of effective condensation mode of use of a heat utilizer, corresponding to the temperature of water heating below the dew point of gases (55 ° C), expands with an increase in the calculated temperature of the network water. It is seen that the effective area of the condensing heat exchanger use mode corresponding to the heating water temperature below the dew point of gas (55 ° C), expands with increasing temperature, the calculated water network.

A feature of closed systems is that the heating of cold water for hot water supply is carried out near the places of consumption of hot water. The useful feature of cold water is that it is possible to maximally efficiently cool the flue gases of the boiler, in this case it is used only indirectly - to cool the return water returned to the boiler. The temperature of the return network water is much higher than cold. In addition, it varies significantly during the heating season: from 60-70 ° C at maximum loads to 30-35 ° C at minimum. Yet for a considerable duration of the heating season it is below tp. The variability of the return water temperature raises the question of the possibility and feasibility of using condensing utilizer in boilers serving closed two-pipe heat networks. For a proper assessment, it is necessary to perform calculations based on the method of heat and moisture calculation of the condensing utilizer, which has significant differences from thermal calculations in dry heat exchangers.

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III. METHODOLOGY

A. SURVEY OF PUBLICATIONS

A lot of publications [1 ÷ 3] are devoted to the issue of KU in boiler equipment, but none of them addresses the fundamental issue of the general method of calculating the process. Separate special cases are described, empirical data relating to a narrow range of parameters studied are given. This is explained as follows.
First, it is impossible to determine the average difference Δi_{av} from the two extreme values of the difference in enthalpy, as is customary when calculating the average logarithmic temperature difference in dry heat exchangers. In condensing utilizer there is no linear connection between the enthalpies of the heating (exhaust gases) and the heated coolant (water), which exists in dry heat exchangers. Therefore, using the cumbersome step method of calculation using the *I-d* diagram [3].
Secondly, using the mass transfer coefficient instead of the heat transfer coefficient. Meanwhile, it was established [5, 6] that when water evaporates in a gas stream, a relationship takes place to determine the heat capacity of a gas, which allows instead of using dependencies for. In [4], it was established that this ratio with small deviations is observed for processes with vapor condensation. Thus, it remains to be determined. To solve this problem in [5,6] proposed a calculation equation:

$$\Delta i_{av} = i_0 - 4{,}19 \cdot B \cdot t_{out} + \left\{2{,}093 \cdot B \cdot \left(t_{out}^2 - t_{in}^2\right) - 351 \cdot \left(e^{0{,}0535 \cdot t_{out}} - e^{0{,}0535 \cdot t_{in}}\right)\right\} / \left(t_{out} - t_{in}\right)$$

here, $B = G_w / G_{dg}$ - the flow of water and dry gases; t_{out}, t_{in} - water temperature (initial and final); i_o - the initial enthalpy of gases.
Using the example of a four-pipe heating system scheme, we will determine the feasibility of using condensing utilizer of flue gas heat in steam boilers with a closed scheme.
The calculation procedure was carried out as follows. Known are the consumption of exhaust gases G_{eg}, the its initial temperature t_{eg}, the moisture content x_{eg}, the initial temperature of the heated water is t_{w1}. To calculate the average value Δi_{av}, it is required to know the water flow rate G_w and its final temperature t_{w2}.The values of the final enthalpy of gases are given. ,where is the enthalpy of the saturated vapor-gas mixture at the initial water temperature in the table of water vapor [3], Δ is the excess of the enthalpy of gases in relation to the theoretically minimum value. The values of the final enthalpy of gases are given

$$i'_{eg} = i''_w + \Delta \ ,$$

Here, i''_w – the enthalpy of the saturated vapor-gas mixture at the initial water temperature in the table of water vapor [3], Δ is the excess of the enthalpy of gases in relation to the theoretically minimum value

B. Calculation of the characteristics of condensation heat exchangers

The transfer of heat energy from the flue gases to the heated water in the heat exchangers of traditional dry heat exchange occurs due to the temperature difference between the heat carriers. The amount of transmitted energy is calculated by the known formula

$$Q = k \cdot F \cdot \Delta t_{cp} ,$$

here k is the heat transfer coefficient, F is the heat exchange surface area, and Δt_{av} is the average temperature difference during the process, defined as the log average value between the initial and final temperature difference, based on the linear relationship between the temperature of the gas and water.

In contact heat exchangers, the enthalpy of the layer is determined by a similar formula, which uses water temperature and moisture content corresponding to the saturated state of the vapor-gas mixture at water temperature and is a tabular value.In the recuperative heat exchanger, in addition to the heat transferred by lowering the temperature of the gases, heat of condensation is added. The amount of heat contained in the gases is equal to the sum of the temperature and humidity components, which is expressed by the enthalpy [4]

From [2] it can be seen that the area of effective condensation mode of use of a heat utilizer, corresponding to the temperature of water heating below the dew point of gases (55 ° C), expands with an increase in the calculated temperature of the network water. From Table 1, it is seen that the effective area of the condensing heat exchanger use mode corresponding to the heating water temperature below the dew point of gas (55 ° C), expands with increasing temperature, the calculated water network.

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$$i = c \cdot t + i'' \cdot x,$$

here c, t, i'', x are, respectively, the heat capacity, temperature, the enthalpy of saturated vapor and the moisture content of the gases. Therefore, to calculate the condensing utilizer parameters for calculating the heat transfer process, we use the difference between the enthalpies of heat transfer media — gases and heated water.

The tolerance is that the gases do not have direct contact with water, because on its way there is a gas layer covering the surface of the water and containing vapors emitted by water. Water temperature determines the vapor concentration. Thus, in this case, the heat and mass transfer of gases occurs not with the water itself, but with a layer saturated with water vapor that envelops it.

In condensation recuperative heat exchangers, gases are in contact with the outer surface of the pipe, inside of which there is water. The temperature of this surface, although slightly higher (by 1 °C) is the water temperature, but still below the dew point of the gases, and as a result, water vapor condenses on the surface of the pipe.

At [3], in particular, describing air conditioning processes, it was assumed that a vapor-gas layer with a moisture content corresponding to the saturation state at the temperature of the outer wall is formed at the surface of the pipes. Thus, the process of heat and mass transfer in the condensation and contact heat exchanger is similar. The exception is that in the condensation heat exchanger the temperature of the pipe wall is slightly higher than that of the water in the contact.

IV. RESULTS OF CULCULATIONS

The results of the calculation of the main indicators of the operation of the heat exchanger are shown in figure 2. At the figure 2 it can be seen that the area of effective condensation mode of use of a heat utilizer, corresponding to the temperature of water heating below the dew point of gases (55 °C), expands with an increase in the calculated temperature of the network water. The capacity of the heat exchanger increases accordingly.

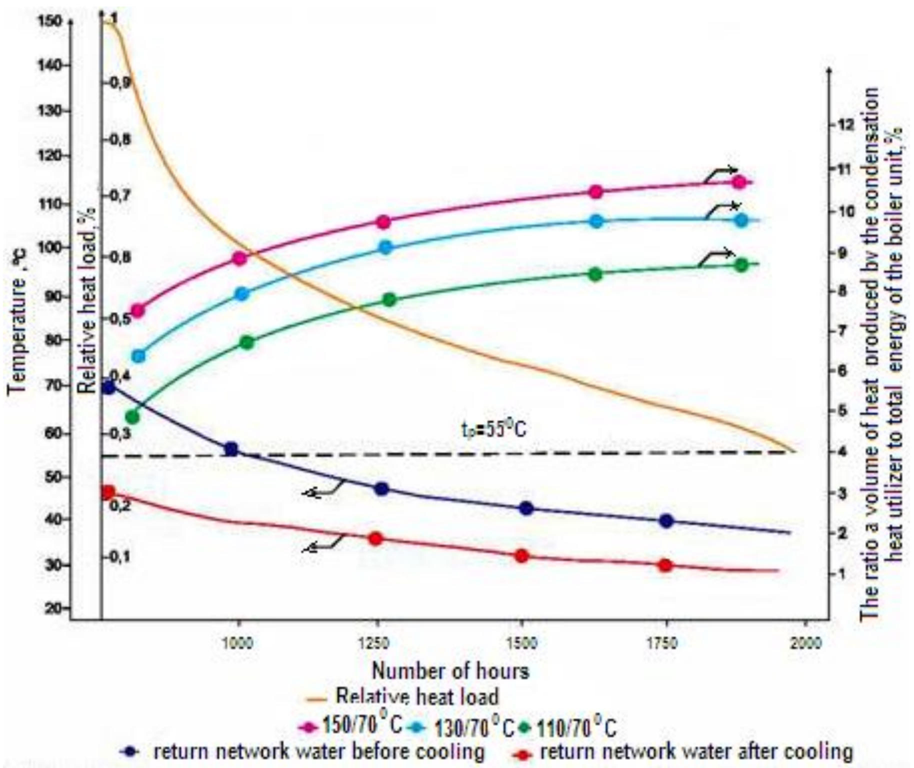

Fig 2: Change in temperature of the return network water after cooling in the condensation heat exchanger, the ratio a volume of heat produced in the it and produced in the boiler during the heating season

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Next, determine the performance of condensing utilizer

$$Q = G_w \cdot \left(i_{eg} - i'_{eg}\right),$$

Values $t_к$ are selected and calculated.

According to the formula, determine Δi_{av}. At the end, the required pipe surface is calculated.

A. Calculation in relation to the boiler unit TGMP-114S

Heating capacity of the boiler at the break point:

Q = (0, 41 × 0.8 + 0.2) × 314 × 10^6 × 0.239 = 15.3 × 10^6 kJ / h

Consumption of combustible gas

B = 15.3 × 106 / 40.1 × 10 × 0.95 = 40.3 t / h

Flue gas consumption:

G_{eg} = 40.3 × 13.8 = 556.14 t / h,

network water:

G_c = 75 × 106 / 4.19 × (130–70) = 298.5 t / h

Let us assume that the enthalpy of gases at the exit of the utilizer

$$\Delta i_{eg} = i_{eg} - (i''_w + \Delta i)$$

here i''_w is the enthalpy of the saturated gas-vapor mixture at the temperature of the supply water at the inlet to the utilizer, which is the theoretical limit for the cooling of gases, Δi is the excess of the enthalpy of the cooled gases relative to the theoretical limit.

The temperature of the network water at the utilizer outlet from the heat balance equation

298.5 × 4.19 × ($t_к$– 35.5) = 556.14 × (499–198), $t_к$ = 3.45 + 35.5 = 38.95 °C

Heating capacity:

Q = 298.5 × 10^6 × 3.45 = 10298.7 × 10^6 kJ / h.

Enthalpy difference between heat carriers: at the entrance to the utilizer

499–153 = 346 kJ / kg,

at the output 70 kJ / h was adopted, the average difference according to the simplified log average

(346–70) / ln (346/70) = 180 kJ / kg

(when calculated by the exact formula Δi_{av} = 214 kJ / kg).

The heating surface is

F = 10298.7 × 106/180 × 150 = 178 m^2,

the heat flux density is

q = 10298.7 × 106/178 = 57.8 × 103 kJ / m × h.

B. Calculation of indicators at the maximum load of heating.

Consumption of combustion products

G = 75 × 106 / 40.1 × 10 × 0.95 × 13.8 = 32.6 t / h

δτ = 0.25 × (130–70) × 15 = 70–5 = 55 ° C,

which coincides with the dew point of gases 56 ° C, the utilizer operates in the mode without vapor condensation.

The temperature of the cooled gases at the utilizer outlet is determined from the relationship:

Q = 3260 × (180 – t_{eg}) = 150 × 17.8 × [(180 – t_k) + (t_{eg}-40) × 0.5].

It takes t_k = 45 ° C, then the selection of t_{eg} = 101 ° C.

Q = 32600 × (180–101) = 2575000 kJ / h.

Increase of efficiency

Δη = 2575000 × 100/75 × 106 = 3.4%

The ratio of the condensing utilizer heating surface (62.5 m^2) to the heating surface of the boiler was about 30%; in VTI cast iron economizers installed behind steam boilers, this ratio is close to 100%

V. CONCLUSION.

The above calculations refer to the modes close to the limiting cooling of the flue gases of the boiler, at which their temperature decreases to 80 ÷ 85 ° C, and the moisture content decreases 2 ÷ 3 times. The reduction of moisture emissions into the atmosphere is a positive factor, but the low temperature of the flue gases worsens the conditions for dispersion of harmful emissions. In addition, deep cooling of gases requires significant heat exchange surface areas.

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The best solution in this situation could be the development of technical schemes, in particular [7], which provide for a certain limitation of the degree of gas cooling with a decrease in heat exchanger performance and, consequently, a decrease in thermal energy savings. Thus, the use of heat from the flue gases of gasified boilers in condensing utilizer can increase the efficiency of boilers with an extremely deep cooling of gases by 3.4%. The optimum temperature for gas cooling and, accordingly, the saving of thermal energy and the increase in efficiency are selected for technical and economic reasons

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