The influence of supercooling of the main condensate at the outlet of the condenser on the operation of a cogeneration steam turbine plant

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Abstract. The article considers the issue of the influence of supercooling of the main condensate on the performance of a steam power plant. We describe the calculation method for a steam turbine condenser. Based on the mathematical model of a condenser, we propose a method for calculating supercooling degree of the condensate on its bottom with different air inflows in a turbo unit vacuum system. The degree of the condensate supercooling on the condenser bottom during the operation of the turbine unit in the heat and electricity production mode is determined in the example of the T-110/120-130 UTZ turbine unit. It is shown that an increase of air inflow into the vacuum system by four times leads to an increase in supercooling of the condensate from 3.5 to 11 °C. It has been established that an increase in the condensate supercooling leads to additional heat losses of up to 1.100 kW in the turbine unit per a cycle in the power production operating mode and up to 500 kW per a cycle in the heat production mode.

1. Introduction
In modern practice, the purpose of improving newly designed steam turbine condensing units is to obtain the most economically justified vacuum values for steam loads of the condenser. One of the ways to achieve this goal is to improve the system for removal of non-condensable gases [1-4].

Experience in operating steam turbine units of various types has shown that during their performance there is an increase in the number of difficult-to-remove defects, which lead to an increase in air inflows into the vacuum system [5–8]. In especially difficult air inflow conditions, condensation units of steam-generating turbines of an extended vacuum system are used during the housing-heating period.

According to the rules [9], in the range of the steam load of a condenser 40 - 100%, air inflows (kg/h) into the vacuum system should be no higher than the values determined by the formula (1)

\[ G_{air} = 8 + 0.065 \cdot N, \]

where \( N \) is the nominal electric power of a turbine unit during the heating period in condensation mode, MW.

If steam loads of a condenser are low, the volume of air inflow into the vacuum system is not regulated. They are usually 5-6 times higher than values calculated by the formula (1). Such inflows lead to an increase in condenser pressure. Under these conditions there is a significant supercooling of the main condensate (up to 4-6 °C), which leads to a decrease in the efficiency of the turbine unit.

In addition, [9] regulates the content of dissolved oxygen beyond the condensate pumps. At the same time, it is known that the magnitude of supercooling of condensate is directly related to the oxygen content [10, 11].
Thus, there is a problem to evaluate the efficiency of steam turbine unit operation in conditions of low steam loads (less than 40% of nominal) of their condensers under conditions of standard and high air inflows into the vacuum system.

2. Methodological basis of the study
Solving these problems is possible only on the basis of the corresponding mathematical models of research objects. There are several mathematical models for a condenser [12-16]. We use our own mathematical model of a condenser [17].

The development of this mathematical model is based on the existence of characteristics of a condenser at point \( t_c = f(D_c) \) (see fig. 1) as a piecewise linear function that has a well-defined break point with the coordinates \( t_c^* \) and \( D_c^* \). In fig.1 \( t_c \) is the saturation temperature at the pressure in the condenser, \( D_c \) is the steam consumption to the condenser. Similarly, the characteristic of the condenser can be represented at points \( \bar{\Delta} = f(\bar{Q}) \) (see. fig. 2) where \( \bar{\Delta} = \Delta / \Delta_{nom} \) is the present reduced disposable temperature difference; \( \bar{Q} = Q / Q_{nom} \) is the relative thermal load of the condenser.

It seems that during the operation of a condenser with steam loads corresponding to the left branch of the diagram, the pressure at the inlet to the ejector is exceeds the reachable value. It leads to an increase in pressure (due to air accumulated in its steam space) that provides dynamic equilibrium between the operation of the ejector and the condenser itself.

\[
\bar{t}_c = \frac{D_c}{D_{c, nom}} \left( t_{c, nom} - t_{1w} \right) + t_{1w}
\]

where \( D_c \) is the current steam consumption in the condenser; \( D_{c, nom} \) is the nominal consumption of steam in the condenser; \( t_{c, nom} \) is the saturation temperature at nominal steam pressure in the condenser.

It is considered that in these modes the air volume in the steam has a minimal effect on the process of steam condensation (almost pure steam is condensed).

The left branch of the diagram \( t_c = f(D_c) \), as shown in [5, 18], can be represented as

\[
\bar{t}_c = \frac{D_c}{D_{c, nom}} \left( t_{c, nom} - t_{c,0} \right) + t_{c,0}
\]
The determination of $t_{c0}$ is based on the assumption that there is a mode of operation of the condenser at a condenser’s heat load approaching zero. In this case, the pressure in the condenser can be determined using the well-known characteristics of the ejector when inflowing the steam-air mixture $p_{e_{mix}} = \varphi_1(G_{air})$ or dry air $p_{e_{dry_{air}}} = \varphi_2(G_{air})$ and the equation of hydraulic losses on the line condenser-ejector

$$p_{e0} = p_{e0} + \Delta p_0$$

where $p_{e0}$ is steam pressure in the condenser at $D_c = 0$; $p_{e0}$ is pressure in the receiving chamber of the ejector; $\Delta p_0$ is the pressure difference in the inlet of the condenser and at the entrance to the first stage of the ejector, it can be determined on the basis of direct measurements or adopted in accordance with the recommendations, for example, [18] $p_{e0} = (1.03...1.08)\cdot p_{e0}$.

With well-known characteristic of the ejector in dry air $p_{e_{dry_{air}}} = \varphi_2(G_{air})$, the calculation of $p_{e0}$ is performed by an iterative method for the equivalent air-steam mix [18] using the empirical formula

$$G_{eq} = \frac{G_{air}}{(0.18 + \varepsilon + 0.72)}$$

Here $\varepsilon = G_{air}/D_c$ is the relative air content of the air-steam mix, which can be defined as

$$\varepsilon = \frac{1}{1 + 0.622 \cdot \frac{p_{s0}}{p_{air0}}}$$

where $p_{s0}$ is the steam partial pressure corresponding to saturation temperature $t_{1w}$; $p_{air0}$ is the partial air pressure in the mode.

The value $t_{c0}$ is found from the value $p_{c0}$ calculated by equation.

The value $D_{c*}$ can be obtained on the basis of the analysis of variable modes characteristics presented in [18]. In [5, 17], it was shown that the left branch of the condenser diagram can be represented as a linear dependence

$$\bar{\Delta} = B \cdot \bar{\varnothing} + \bar{\Delta}_0$$

where $\bar{\Delta}_0 = \frac{t_{c0} - t_{tw}}{t_{nom} - t_{tw}}$ is available relative temperature difference at $\bar{\varnothing} = 0$ (see. fig. 2).

$B$ is the angular coefficient, which is the tangent of the angle of inclination to the generalized characteristic of variable modes at a point $(0, \bar{\Delta}_0)$

$$B = \frac{d\bar{\Delta}}{d\bar{\varnothing}} |_{\bar{\varnothing}=0} = (1 - \bar{\Delta}_0)^2$$

Thus, the initial part of the characteristics of the condenser will be

$$\bar{\Delta} = (1 - \bar{\Delta}_0)^2 \cdot \bar{\varnothing} + \bar{\Delta}_0$$

Since boundary modes with a constant flow of cooling water through a condenser are,... the reduced amount of heat corresponding to the transition to the free condensation mode after simple transformations of equation (9) is defined as

$$\bar{\varnothing}^* = \frac{\Delta^*}{\Delta_{nom}} = \bar{\Delta}^*$$

or

$$\bar{\varnothing}^* = \frac{1}{2 - \bar{\Delta}_0} \cdot D_{c_{nom}}$$
\[ D'_c = \frac{1}{2 - \Delta D} \cdot D_{c_{\text{nom}}} \]  

(11)

The corresponding values can be determined by the method of calculating the right arm of the diagram.

On the basis of the presented mathematical model, we have proposed a method for determining the condensate supercooling values on the condenser bottom with various inflows of the vacuum system of a turbine unit, as well as determining the effect of these inflows on the pressure in the condenser with its steam loads less than the boundary values.

3. Research results and discussion

Let us consider the use of the proposed technique on the example of a cogeneration turbine plant T-110/120-130.

We carried out several series of calculations in order to estimate the degree of condensate supercooling at the bottom of the condenser and the amount of heat required to compensate this supercooling in the regenerative heating system. We used the following limitations in these calculations:

- the condition of the condensate is determined by the conditions of equilibrium processes occurring in the regenerative heating zone under the tube bundles of the condenser;
- recirculation condensate and additional chemically demineralized water is supplied to the upper part of the condenser above its tube bundles;
- drains supplied to the hotwell do not have a significant effect on the temperature of the condensate entering the regenerative heating system;
- the minimum amount of condensate supplied by the condensate pump is 140 t/h;
- the influence of air inflows in the vacuum system within the limits of the condenser of the right branch of the characteristic is negligible.

The validity of the accepted restrictions is confirmed by the results of numerous experiments on existing equipment. The first series of calculations was aimed to define the characteristics of the capacitor \( t_c = f(D_c) \) (see. fig. 3).

We studied two levels of cooling water: nominal - 16000 t/h and minimum allowable - 8000 t/h with built-in bundles connected by cooling water for three levels of cooling water inlet temperature to the condenser (35, 20 and 5 °C). The article presents the results at a flow rate of cooling water 16.000 t/h.

The calculations have shown that with decreasing cooling water flow, the boundary values of the flow rate \( D_{c_{\text{nom}}} \) shift towards smaller values, i.e. the range of operation modes of the condenser, limited by the ejector, is reduced.

It should be noted that with a decrease in cooling water temperature and with an increase in air inflows into the vacuum system, the values \( D_{c_{\text{nom}}} \) and \( t_c \) increase significantly. With a fourfold increase in air inflow and at cooling water temperature \( t_{1_w} = 5 \) °C, most of the operating modes of the condenser become limited.

On the basis of the obtained characteristics of the condenser, we carried out the second series of calculations in order to obtain the dependences of supercooling of the condensate on the steam flow \( \delta t_c = f(D_c) \) within the right-hand branch (see. fig. 4).

In this case, the magnitude of supercooling of the condensate was determined as the difference between the saturation temperature at the pressure in the condenser (left branch of the characteristic) and the condensing temperature of pure steam, determined by the characteristic of the condenser (see the dotted continuation of the right branch of the characteristic).

The calculations have shown that the effect of air inflow on the degree of condensate supercooling is very large and leads to significant heat consumption for its heating in the regeneration system.

A characteristic feature of cogeneration turbine plant is the presence of a recirculation valve, which ensures the normal operation of the turbine plant in regimes with large heating steam extraction.
Figure 3. The calculated characteristics of the condenser T-110/120-130. There is one ejector EP-3-200. Cooling water consumption is \( W = 16000 \) t/h, heat exchange surface is \( F = 6200 \) m\(^2\), purity coefficient is \( a = 0.7 \).

\[ \begin{align*}
\text{---} & \quad \text{the condenser characteristic with standard air inflows } G_{air} = 15.8 \text{ kg/h;} \\
\text{---} & \quad \text{the characteristic of the condenser with air inflows } G_{air} = 63.2 \text{ kg/h;} \\
\text{---} & \quad \text{conditional characteristic of the condenser during the condensation of pure steam.}
\end{align*} \]

In these modes, a part of the main condensate in the \( G_{rec} \) volume is returned to the condenser through the recirculation valve, where the heat obtained in these heat exchangers is lost with cooling water. The remaining condensate in the amount of \( G_{reg} \) enters the regeneration line, and

\[ G_{reg} = G_{clp} + G_s. \]  

(10)

where \( G_{clp} \) is steam consumption to the condenser through the flow part of the turbine; \( G_s \) is the consumption of steam and water entering the condenser in addition to the flow part (standing drains, water additions, steam-air inflows, etc.).

In the turbine unit under consideration, the minimum condensate flow rate from the condenser is \( G_c = G_{reg} + G_{rec} \), equal to 140 t/h, and with an increase in the steam load of the condenser (\( G_{clp} \)) the recirculation valve closes fully, and the value \( G_c \) remains constant until \( G_{reg} \) exceeds these 140 t/h.

Figure 4. Dependence of condensate supercooling at the bottom of the condenser KG2-6200 of the turbine plant T-110/120-130. There is one EP-3-2 ejector in operation. Cooling water flow rate is \( W = 16000 \) t/h, heat exchange surface is \( F = 6200 \) m\(^2\), purity coefficient is \( a = 0.7 \).

\[ \begin{align*}
\text{---} & \quad \text{condensate supercooling with standard air inflows } G_{air} = 15.8 \text{ kg/h;} \\
\text{---} & \quad \text{condensate supercooling with air inflows } G_{air} = 63.2 \text{ kg/h.}
\end{align*} \]
On the basis of the obtained dependences $\delta t_c = f(D_c)$, we carried out calculations to obtain the dependences of the cost of heat to eliminate supercooling on the condensate consumption in the regenerative heating zone $Q_c = f(G_{\text{reg}})$ with two levels of air inflow — regular and quadrupled (see. fig. 5). The value $G_s$ was adopted on the basis of previous studies [19].

![Graph](image)

**Figure 5.** The dependence of the cost of heat to eliminate condensate supercooling at the outlet of the condenser of the T-110 -120 turbine plant on the condensate discharge at the outlet of the condenser. Cooling water consumption is $W=16000$ t/h, the heat exchange surface is $F=6200$ m$^2$.

- - the heat consumption at standard air inflows $G_{\text{air}}=15.8$ kg/h;
- - - - - - the cost of heat at the air inflows $G_{\text{air}}=63.2$ kg/h.

The data shown in Figure 5 makes it possible to establish that with sufficiently large but real air inflow and low cooling water temperatures, the cost of heat for eliminating condensate supercooling may increase to significant values. For example, with an increase in the air-inflow capacity by four times and at a temperature of cooling water $t_{1w} = 5$ °C, these costs increase from 520 to 1700 kW at $W=16000$ t/h and from 490 to 1400 kW at $W=8000$ t/h.

The elucidation of the effect of air inflow in operating modes of a cogeneration turbine of a turbine unit with closed regulating diaphragms of low pressure cylinders on the costs of heat to eliminate condensate supercooling in a regenerative heating zone is of particular interest.

Figure 6 shows the results of calculations of the cost of heat for the elimination of supercooling, depending on the air inflows into the vacuum system. The calculations were performed with steam consumption through a low-pressure cylinder $G_{\text{clp}}=20$ t/h, and cooling water flow rate $W=8000$ t/h.
Figure 6. The cost of heat to eliminate condensation supercooling in the system of regenerative heating zone of the T-110 / 120-130 turbine plant, depending on air inflows in the vacuum system. Cooling water flow rate is \( W = 8000 \text{ t/h} \), heat exchange surface of the condenser is \( F = 6200 \text{ m}^2 \). Regulating diaphragms of the low-pressure cylinder are closed with \( G_{clp} = 20 \text{ t/h} \).

\[ \begin{align*} &\text{the cooling water inlet temperature to the condenser is } t_{1w} = 35 ^\circ \text{C}; \\
&\text{the temperature of the cooling water at the inlet to the condenser is } t_{1w} = 20 ^\circ \text{C}; \\
&\text{the temperature of cooling water at the inlet to the condenser is } t_{1w} = 5 ^\circ \text{C}. \end{align*} \]

As it was shown in [19], the main losses of condensate heat in these modes occur in a condenser with recirculation condensate entering it and they are about 1.5-2 MW. Nevertheless, the cost of heat for eliminating supercooling with increasing air inflows increases significantly in these modes and amounts to hundreds of kilowatts. Thus, an increase in air inflows from a standard 15.8 kg/h to 70 kg/h at cooling water temperature \( t_{1w} = 5 ^\circ \text{C} \) leads to an additional expenditure of heat of 350 kW.

4. Conclusion
1. Based on the mathematical model of a condenser, we have proposed a method for calculating supercooling of condensate at the bottom with various air inflow volumes of the vacuum system of a turbine plant, as well as determining the effect of these air inflow volumes on the pressure in the condenser if its steam loads lower than the boundary ones.
2. Using the example of the proposed technique as applied to the T-110/120-130 UTZ cogeneration steam turbine units, the condensate supercooling values at the bottom of the condenser are determined when the turbine is operating on thermal and electrical plots. It is shown that the increase in air inflow into the vacuum system by four times leads to an increase in condensate supercooling from 3.5 to 11 °C.
3. Taking the T-110/120-130 UTZ turbine unit as an example, calculations showed that an increase in condensate supercooling leads to additional heat losses in the turbine plant cycle up to 1.100 kW during the operation modes according to the electrical schedule and up to 500 kW during cogeneration modes.

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