Mathematical Model of a Steam Turbine Condenser

M. Shavdinova*, **

Kazakh-German University, Almaty, 050010 Kazakhstan

*e-mail: shavdinova@dku.kz

**e-mail: nirvana-18@mail.ru

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Abstract—The main purpose of the condenser of a steam turbine unit is to condense the exhaust steam of the turbine and maintain the steam pressure behind the last stage under nominal conditions not higher than the calculated, which are determined on the basis of technical and economic indicators. Faults in the condensers lead to disruptions in the operation of the steam turbine, to its failures and breakdowns, loss of heat, water, and steam at the station, etc. In order to ensure the reliable operation of the condenser and improve the efficiency of its equipment, a mathematical model of the T-110/12.8 turbine condenser was developed, verified, and tested in relation to the conditions at the Almaty Power Plant (ALES) of CHPP-2. It is shown that the model can be used to build new standard characteristics of the condenser at steam flow rates in the condenser above the nominal values and supply of circulating water to the main bundles and raw water to the built-in bundles. The model takes into account the peculiarities of determining the steam pressure in various zones of the condenser when cooling water is supplied to the main and built-in bundles of tubes. With the help of the developed module of the monitoring system for the technical condition of the condensing unit of the T-110/12.8 AIES CHPP-2 steam turbine, it was found that the deviation of the actual condenser performance (including steam pressure) from the standard (calculated) values occurs mainly due to contamination of the condenser tubes. The influence of air suction in most modes is insignificant. The mathematical model allows one to change the input parameters, build dependency graphs, conduct a computational experiment, and calculate performance indicators. A feature of the mathematical model of the condenser is that it is built according to a technique that takes into account the peculiarities of determining the steam pressure in various zones of the condenser when cooling water is supplied to the main and built-in bundles. Based on the results of an industrial experiment, a comparison was made of the values of standard, calculated, and actual condenser pressures for each mode.

Keywords: condenser, steam turbine, mathematical model, average heat-transfer coefficient, steam flow rate, cooling water flow rate, diagnostic model

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The steam turbine condenser, designed to condense the exhaust steam, is a horizontal shell-and-tube heat exchanger consisting of main and built-in tube bundles. For example, in the KG2-6200 condenser installed at AIES CHPP-2, the main bundles are cooled by water from circulation pumps, and the built-in bundles are cooled by tap (raw) water.

There are various mathematical models for calculating the condenser of a steam turbine that were developed in their research by such specialists as K.E. Aronson, S.I. Khaet, G.G. Shklover, A.G. Shempelev, P.V. Eaglin, R. Laskowski, A. Rusovich, and others.

In [1], a mathematical model of a steam condenser operating under changing conditions is presented. After selecting a list of independent parameters that affect the water temperature at the outlet of the condenser, using the Buckingham theorem, a functional relationship was established between two dimensionless quantities. In [2, 3], a two-dimensional mathematical model of a condenser is proposed, in which the tube bundle is considered as a porous layer. A two-pass capacitor with a power of 50 MW was subjected to analysis. The authors of [4] developed a program for measuring flows on the side of the tube and on the side of the casing as well as heat transfer in the power plant condenser (steam temperature, its pressure along the perimeter of the tube bundle, distribution of water pressure at the inlet to the tube sheet, flow rate and temperature of steam at the tube board). In [5, 6], data are given on the existing mathematical models of condensers that take into account the heating of make-up water in built-in tube bundles while passing cooling water and the practical use of these models. Heat savings when heating make-up water in built-in bundles when operating in the heat-production mode is provided by an external increase in the heat load of turbine extractions, which leads to a decrease in the load on peak hot water boilers.
Diagnostic models of steam turbine condensers have also been developed. For example, a model is presented in [7] that allows determining the timing of the replacement of condenser piping systems and one in [8] that, based on statistical analysis, can be used to assess the state of steam turbine condensers and predict their residual life. The authors of [9, 10] describe a diagnostic model of a condenser that takes into account the separate influence of air suction and contamination of the condenser’s heat-exchange surfaces on the steam pressure in the steam turbine condenser. In [11], a diagnostic model is proposed that allows one to determine the state of technological subsystems of steam turbines (rotors, bearings, automatic control, and turbine protection systems as well as other components of the turbine unit, condenser unit equipment), eliminate malfunctions of equipment elements, and formulate recommendations on methods and timing eliminate defects and reduce the risk of their development.

Using all the described mathematical and diagnostic models, the calculation of the main tube bundle was carried out. This paper presents a mathematical model that allows obtaining the values of steam- and cooling-water flow rates for the main and built-in condenser tube bundles as well as diagnosing the condition of the equipment.

The objective of this study is to ensure reliable and efficient operation of the steam turbine condenser. To do this, it is necessary to solve the following tasks:

1. choose the appropriate calculation method;
2. create a mathematical model of the condenser, taking into account the operation of the main and built-in tube bundles;
3. compare the values of standard, calculated, and actual absolute steam pressures in the condenser for each mode;
4. analyze the causes of discrepancies between the standard, calculated, and actual pressure values in the condenser;
5. carry out diagnostics of the state of the condenser using the method of taking into account the separate influence of air suction and contamination of the heat-exchange surface on the steam pressure in the condenser.

The practical significance of this work lies in the fact that the proposed method allows obtaining new standard characteristics of the capacitor. The developed mathematical model can be used at the station for diagnosing the state of the condenser, when conducting an energy audit, in studies to improve the efficiency of equipment, and in the educational process.

METHODS FOR CALCULATION OF STEAM TURBINE CONDENSERS

At the first stage of developing a mathematical model, the requirements for a mathematical model of a capacitor are formulated. It should

1. ensure the receipt of values of steam flow rates and flow rates of cooling and raw water in the main and built-in tube bundles as well as values of cooling water subcooling to saturation temperature;
2. contribute to the construction of standard characteristics of the capacitor;
3. become the basis for the development of a diagnostic model that takes into account the separate influence of air suction and pollution on the steam pressure in the condenser;
4. be easy to use and versatile, i.e., applicable to various kinds of steam turbine condensers.

At the second stage of developing a mathematical model, it is necessary to choose a calculation method. There are various methods for calculating capacitors: All-Russian Technological Institute (ATI), Heat Exchanger Institute (HEI) (United States), Leningrad Metal Plant (LMP), Ural Turbine Plant (UTP), Kaluga Turbine Plant (KTP), and Ural State Technical University (USTU-UPI). In [12], a description of each technique is presented.

Using the ATI, HEI, LMP, and UTP methods, it is possible to estimate the average heat-transfer coefficient for the entire heat-exchange surface of the condenser from the integral operating and design characteristics of the equipment. However, it is impossible using these methods to calculate the separate effect of air suction and tube contamination on the steam pressure in the condenser. The KTP and USTU-UPI methods make it possible to carry out this calculation [12, 13]; however, with any change in one of the process parameters, it is necessary to refine the values of the thermophysical properties of water and the condensate film depending on temperature. There is no such drawback in the ATI and HEI (United States) methods.

To select the appropriate method, a verification thermal calculation of the capacitor was carried out according to the methods of ATI, KTP, USTU-UPI, and HEI (United States). The calculated dependences of the pressure in the condenser $p_{\text{cond}}$ from the steam consumption in it $D_{\text{cond}}$ are constructed. The calculations were carried out on the example of the KG2-6200 condenser of the T-110/120-12.8-5 turbine unit of AES CHPP-2. Analyzing the obtained dependences (Figs. 1–3), one can see that the discrepancy between the standard pressure values and its calculated values obtained by the ATI method does not exceed 14%. 13% for HEI (United States), and 23% for KTP and USTU-UPI.

The KTP and USTU-UPI methods give higher calculation errors compared to the standard values, but they can be used, for example, when developing a mathematical model with various methods of heat-transfer intensification. Thus, according to the KTP method, a model of a condenser was created on which heat-transfer intensification was carried out with the application of annular grooves on tubes [12].
Although the ATI and HEI methods also have errors, it is preferable to use them to develop a mathematical model of a capacitor in order to diagnose the state of equipment at the plant. However, according to [13–15], the ATI method can be used in the calculation of capacitors only in the range of steam loads from 50 to 100% of the nominal. At lower steam flow rates, the ATI method gives discrepancies with experimental data and standard characteristics.

**MATHEMATICAL MODEL OF A CAPACITOR**

The third stage in the development of a mathematical model is its implementation. The mathematical model of the condenser consists of a thermal calculation of the main and built-in condenser tube bundles and was developed in the Microsoft Excel program. The initial data for creating a condenser model are the nominal flow rate of the exhaust steam ($D_{\text{cond, nom}}$, kg/s), cooling temperature ($t_{cw1}$, °C), and raw water ($t_{rw1}$, °C) at the inlet to the condenser and the geometric characteristics of the equipment. The design of the capacitor is shown in Fig. 4.

The main bundle of tubes is calculated according to the ATI method, and the built-in bundle is calculated according to the method proposed by Metro-Vickers since it is easy to use and recommended by ATI for calculating network water heaters (the built-in bundle is considered to be the heating stage of network water). The disadvantage of the Metro-Vickers method is that it is integral, i.e., it is impossible using it to calculate the heat-transfer coefficients from the steam side and from the steam-air mixture, and it is impossible to take into account some factors (the effect of the specific steam load of the condenser on the heat-transfer coefficient, tube diameter, thermal resistance of the tube wall, cleanliness of the heat-exchange surface).

Heat-transfer coefficient of the main beam $k_{Mb}$ (index $Mb$ is from English: main bundle), W/(m² K), according to the ATI method, is determined by the formula [16]

$$k_{Mb} = 4070a\Phi_w\Phi_z\Phi_\delta,$$

where $a$ is the coefficient of contamination of the heat-exchange surface; $\Phi_w$, $\Phi_z$, and $\Phi_\delta$ are multipliers that take into account the influence of the speed of the cooling water, its temperature at the condenser inlet, the number of strokes, and the specific steam load of the condenser.

Heat-transfer coefficient of built-in bundle $k_{Bb}$ (index $Bb$ is from English: built-in bundle) according to the Metro-Vickers method is calculated as follows [17]:

$$k_{Bb} = \frac{1}{k_0 + \left(R_{rf} - R_{rf}^{1.65}\right)} \beta_{saf} \beta_d \beta_{aircool},$$

where $k_0 = 1096\sqrt{\frac{w_\text{w}}{\text{W}} \frac{t_{cw1} + t_{cw2}}{2}} + 17.8$ is the heat-transfer coefficient of a pure condenser, W/(m² K); $w_\text{w}$ is the average speed of the cooling water in the condenser tubes, m/s; $t_{cw1}$, $t_{cw2}$ are raw water temperature.
MATHEMATICAL MODEL OF A STEAM TURBINE CONDENSER

at the condenser inlet and outlet, °C; \( R_t \) is thermal resistance of the tube wall made of brass LO-70-1, m² K/W; \( R_{st,68}^{L68} \) is thermal resistance of the wall of a tube made of L68 brass and having an outer diameter of 19 mm and a thickness of 0.75 mm, m² K/W; \( \beta_{sa} = 0.85 \) is safety factor; \( \beta_d = 0.974 \) is coefficient for tubes whose diameter exceeds 19 mm, in this case for tubes with a diameter of 24 mm; \( \beta_{aircool} = 0.94 \) is the correction for the fraction of the air cooler surface area in the total surface area of the built-in condenser bundle.

Absolute steam pressure in the condenser \( p_{cond} \) depends on the steam flow in it \( D_{cond} \), cooling circulating water flow \( G_{cw1} \), and its temperature \( t_{cw1} \) at the inlet to the condenser, i.e. \( p_{cond} = f(D_{cond}, G_{cw1}, t_{cw1}) \).

The flow rate of circulating cooling water into the main bundle is determined from the heat balance by the formula

\[
G_{cw1} = \frac{D_{cond} \Delta h_{s.c}}{c_w (t_{cw1} - t_{cw2})},
\]

where \( D_{cond, Mb} \) is the steam flow rate in the main condenser bundle, kg/s; \( \Delta h_{s.c} \) is enthalpy difference between steam and condensate, kJ/kg (for T-100/120-12.8 turbine \( \Delta h_{s.c} = 2304.5 \) kJ/kg); \( c_w \) is heat capacity of cooling water, kJ/(kg K); \( t_{cw1} \) is temperature of the cooling water at the outlet of the main condenser tube bundle, °C.

The consumption of raw cooling water in the built-in bundle is also calculated from the heat balance:

\[
G_{cw2} = \frac{D_{cond, Bb} \Delta h_{h.c}}{c_w (t_{cw2} - t_{cw1})},
\]

where \( D_{cond, Bb} \) is the steam flow rate in the built-in condenser bundle, kg/s; \( t_{cw1} \) and \( t_{cw2} \) are the temperature of the cooling raw water at the entrance to the built-in bundle and at the exit from it.

To set the steam consumption values for the main \( D_{cond, Mb} \) and built-in \( D_{cond, Bb} \) capacitor beams, introduce coefficients \( b_1 \) and \( b_2 \):

\[
b_1 = \frac{F_{Mb}}{F}, \quad b_2 = \frac{F_{Bb}}{F},
\]

where \( F_{Mb}, F_{Bb} \) are the surface area of the tubes of the main and built-in bundles, respectively, m²; \( F \) is the total area of the heat-exchange surface of the condenser, m².

Then, \( D_{cond, Mb} = \frac{D_{cond}}{b_1}, \quad D_{cond, Bb} = \frac{D_{cond}}{b_2} \).

The results of calculations of the flow rates of circulating water are presented in Table 1. After determining the standard values of pressure in the condenser for each mode (steam flow to the condenser), the values \( G_{cw} \) and \( G_{ww} \) clarify.

The main bundle receives circulating cooling water with a temperature \( t_{cw1} \) at the condenser inlet and \( t_{cw2} \) at its outlet, while the built-in receives raw water with \( t_{cw1} \) and \( t_{cw2} \), respectively. The calculation of the main and built-in beams of the condenser is carried out by dividing the segment in half until the pressure values in the
main and built-in beams of the condenser are equal to each other \( p_{\text{cond}_Wb} = p_{\text{cond}_Bb} \).

Saturation temperature, °C, for the main and built-in beams is calculated by the formulas

\[
\begin{align*}
\Delta h_{\text{s.c}} &= 2304.5 \text{ kJ/kg; for turbine T-100/120-12.8-3,} \\
\Delta h_{\text{s.c}} &= 4.514 \times 1000 \text{ kJ/kg; for condenser KG2-6200,} \\
\Delta h_{\text{s.c}} &= 0.81 \times 1000 \text{ kJ/kg.}
\end{align*}
\]

For turbine T-100/120-12.8-3, \( \Delta h_{\text{s.c}} = 2304.5 \text{ kJ/kg; for the condenser KG2-6200,} \ F_{\text{Mb}} = 5240 \text{ m}^2, \ F_{\text{Bb}} = 940 \text{ m}^2, \ c_w = 4.179 \text{ kJ/(kg K), then}

\[
\begin{align*}
t_{\text{Mb}} &= t_{w_1} + \frac{\Delta h_{\text{s.c}} D_{\text{cond}_Mb}}{c_w G_{wMb}} + \frac{\Delta h_{\text{s.c}} D_{\text{cond}_Mb}}{c_w G_{wMb} \times 1000} - 1; \\
t_{\text{Bb}} &= t_{w_1} + \frac{\Delta h_{\text{s.c}} D_{\text{cond}_Bb}}{c_w G_{wBb}} + \frac{\Delta h_{\text{s.c}} D_{\text{cond}_Bb}}{c_w G_{wBb} \times 1000} - 1.
\end{align*}
\]

The calculation is carried out until equality \( t_{\text{Mb}} = t_{\text{Bb}} \) is achieved.

Table 1. Results of calculations of the flow rate of circulating water in the condenser

| \( D_{\text{cond}} \), kg/s | \( D_{\text{cond}_Mb} \), kg/s | \( D_{\text{cond}_Bb} \), kg/s | \( G_{wMb}, \) kg/s | \( G_{wBb}, \) kg/s |
|--------------------------|--------------------------|--------------------------|-----------------|-----------------|
| 91.26                    | 77.37                    | 13.89                    | 4451            | 294             |
| 90.17                    | 76.45                    | 13.72                    | 3921            | 244             |
| 84.48                    | 71.62                    | 12.86                    | 3982            | 221             |
| 72.56                    | 61.53                    | 11.03                    | 3509            | 203             |
| 73.67                    | 62.45                    | 11.20                    | 3443            | 206             |
| 81.81                    | 69.37                    | 12.45                    | 3956            | 237             |
| 93.37                    | 79.17                    | 14.20                    | 3852            | 326             |
| 49.03                    | 41.56                    | 7.45                     | 2433            | 171             |
| 67.09                    | 56.87                    | 10.20                    | 3726            | 238             |
| 62.78                    | 53.23                    | 9.56                     | 3668            | 471             |
| 66.12                    | 56.06                    | 10.06                    | 3905            | 308             |
| 33.56                    | 28.45                    | 5.11                     | 1981            | 176             |
| 59.20                    | 50.20                    | 9.00                     | 4313            | 253             |
| 79.87                    | 67.70                    | 12.14                    | 5090            | 200             |
| 79.42                    | 67.34                    | 12.08                    | 4843            | 239             |
| 47.20                    | 40.03                    | 7.17                     | 2430            | 141             |

In the first approximation, the values are set as \( D_{\text{cond}_Mb}^{(n)} \) and \( D_{\text{cond}_Bb}^{(n)} \) for each mode (with different steam flow rates in the condenser); as a result of calculations, we obtain \( t_{sMb}, t_{sBb} \). Next, find \( t_{sMb}, t_{sBb} \) already at \( D_{\text{cond}_Mb}^{(n+1)}, D_{\text{cond}_Bb}^{(n+1)} \). After determining the average values of steam consumption on the main \( D_{\text{cond}_Mb}^{\text{av}} \) and built-in \( D_{\text{cond}_Bb}^{\text{av}} \) and capacitor bundles' output values are again \( t_{sMb}, t_{sBb} \). The calculation is carried out until equality \( t_{sMb} = t_{sBb} \) is achieved.

According to the received values \( t_{sMb}, t_{sBb} \), using directories, find pressure values mainly \( p_{\text{cond}_Mb} \) and built-in \( p_{\text{cond}_Bb} \) beams of the condenser, and also determine the values of subcooling of the cooling water to the saturation temperature.
RESULTS AND DISCUSSION

Using the results of an industrial experiment, a comparison was made of the values of standard, calculated, and actual steam pressures in the condenser for each mode (Figs. 5–7). The calculated pressure values were obtained using the developed mathematical model, and the standard characteristics taking into account the built-in beam. Heat-transfer surface condition coefficient was adopted as \( \alpha = 0.75 \). The figures show the steam flow rates corresponding to the heating and nonheating seasons (the heating season in Almaty lasts from October 15 to April 15). The heating season includes steam costs 59.20, 66.12, and 72.56 kg/s.

The difference between the actual and calculated values of the steam pressure in the condenser, as well as between the actual and standard values, varies in the range from 0.06 to 6.70 kPa; there is no difference between the calculated and standard ones.

The reasons for the discrepancy between the calculated or standard data and the actual ones may be contamination of the heat-exchange surfaces and air suction into the condenser. On the basis of the mathemat-
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Fig. 7. Steam pressure dependence in the condenser $p_{\text{cond}}$ from its expense $D_{\text{cond}}$. Steam pressure in the condenser: 1—estimated; 2—actual.

Fig. 8. Joint characteristic of the condenser and ejector at $D_{\text{cond}} = 49.03$ kg/s. Steam pressure in the condenser: 1—actual; 2—settlement; 3—pressure in the condenser, taking into account the influence of air suction. Area: I—II—Job condenser; II—III—joint operation of the condenser and ejector.

Table 2. Results of calculations of the combined effect of air suction and contamination of the heat-exchange surface on the steam pressure in the condenser

| Parameter                                      | Steam flow to condenser $D_{\text{cond}}$, kg/s |
|-----------------------------------------------|-----------------------------------------------|
| Steam pressure in the condenser, kPa:         |                                               |
| actual                                        | 9.11                                          | 13.93                                         |
| calculated                                    | 6.74                                          | 6.41                                          |
| taking into account the influence of air suction | 5.45                                          | 6.25                                          |
| Change in steam pressure in the condenser, kPa: |                                               |
| due to increased air intake $\Delta p_a$      | $-1.285$                                      | $-0.160$                                      |
| due to contamination of the heat-exchange surface $\Delta p_{pol}$ | 3.66                                          | 7.68                                          |

If the difference between the calculated steam pressure and the pressure according to the joint characteristic of the condenser and ejector $\Delta p_a > 0.5$, Excel displays the following diagnostic message: “Vacuum deviation in the condenser due to increased air suction is large. Carry out a search for suction points in the technological model, a diagnostic model was also created based on a technique that makes it possible to assess the effect of air suction and condenser contamination on the steam pressure in it [7–10]. Using this technique, a software package was developed, implemented in Microsoft Excel, using which calculations were made for several values of steam flow into the condenser when it was combined with the EPO-3-200 ejector. The calculation results are presented in Table 2.
vacuum system.” If the difference between the actual pressure and pressure according to the joint characteristic of the condenser and ejector $\Delta p_{pol} < 0.5$, the following message is displayed: “Vacuum deviation in the condenser due to deposits on the tubes is large. The condenser needs to be cleaned.”

Figure 8 shows the joint characteristic of the condenser and the ejector (section $II-III$), which can be constructed in the case when the ejector affects the condenser at certain steam flow rates in it.

It follows from Table 2 that the deviation of the actual steam pressure from the normative one is due to the contamination of the heat-exchange surface. At steam flow rates in the condenser of more than 27.78 kg/s and air suctions of up to 0.008 kg/s, the influence of the latter on the steam pressure in the condenser may be absent but may appear when $D_{cond} \leq 16$ kg/s.

In the process of research, we found the approximate thickness of the deposits $\delta$: 0.8 mm at $D_{cond} = 47.2$ kg/s, 1.4 mm at $D_{cond} = 49.03$ kg/s.

The built-in condenser bundle is designed to heat tap water used to feed the network. Consequently, more steam is condensed in this bundle, thereby increasing the heat generation (combined generation of electricity and heat).

Figure 9 shows the dependence of the specific steam loads $d_{cond}$ of the (1) main and (2) built-in beams from the steam flow in the condenser.

Fig. 9. Dependence of the specific steam loads $d_{cond}$ of the (1) main and (2) built-in beams from the steam flow in the condenser.

1. build a new standard characteristic of the capacitor;
2. profile the tubes of the built-in condenser bundle (when using profiled tubes, the heat output increases, i.e., there is an increase in the amount of heat for hot water supply and heating, therefore, fuel consumption is reduced).

CONCLUSIONS

(1) The developed mathematical model of the steam turbine condenser makes it possible to obtain the values of steam and cooling water flow rates in the main and built-in beams for any mode as well as to calculate new standard characteristics. Using data on the flow rates of steam and cooling water entering the condenser, it is possible to adjust their values, thereby increasing the efficiency of the steam turbine.

(2) The mathematical model was tested on the KG2-6200 condenser of AIES CHPP-2. The discrepancy between the actual and calculated steam pressure values in the condenser ranges from 0.06 to 6.70 kPa. As a result of further research, it was revealed that the cause of a significant pressure difference is contamination of the condenser tubes.

(3) The mathematical model of a capacitor can be used in diagnosing the state of equipment, for timely troubleshooting in operation, in developing ways to improve the efficiency of the installation, and in conducting energy audits and for educational purposes.

(4) Based on the results obtained in the work, the following recommendations were formulated for the operating personnel of AIES CHPP-2: to build a new standard characteristic and to profile the tubes of the built-in condenser bundle.
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