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# Determining steam condensation pressure in a power plant condenser in off-design conditions

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**Abstract** The paper presents formulas which can be used to determine steam condensation pressure in a power plant condenser in off-design conditions. The mathematical model provided in the paper makes it possible to calculate the performance of the condenser in terms of condensing steam pressure, cooling water temperature at the condenser outlet, and condenser effectiveness under variable load conditions as a function of three input properties: the temperature and the mass flow rate of cooling water at the condenser inlet and the mass flow rate of steam. The mathematical model takes into account values of properties occurring in reference conditions but it contains no constant coefficients which would have to be established based on data from technical specifications of a condenser or measurement data. Since there are no such constant coefficients, the model of the steam condenser proposed in the paper is universally applicable. The proposed equations were checked against warranty measurements made in the condenser and measurement data gathered during the operation of a 200 MW steam power unit. Based on the analysis, a conclusion may be drawn that the proposed means of determining pressure in a condenser in off-design conditions reflects the condenser performance with sufficient accuracy. This model can be used in optimization and diagnostic analyses of the performance of a power generation unit.

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**Keywords:** Thermal power plant; Steam condenser; Condenser model; Steam condensation pressure; Reference parameters; Steam condenser effectiveness

## Nomenclature

| | | |
|---|---|---|
| $A$ | – | heat transfer area, $m^2$ |
| $c_F$ | – | coefficient of resistance of sediment |
| $c_M$ | – | coefficient of the type of tube material |
| $c_T$ | – | coefficient indicating the effect of cooling water temperature at the condenser inlet |
| $c_w$ | – | water specific heat, J/(kg K) |
| $d_o$ | – | tube outer diameter, mm |
| $\dot{m}_s$ | – | mass flow rate of steam, kg/s |
| $\dot{m}_w$ | – | mass flow rate of cooling water, kg/s |
| NTU | – | umber of transfer units |
| $p$ | – | pressure, kPa |
| $\dot{Q}$ | – | flow rate of the heat transferred, W |
| $r$ | – | latent heat, J/kg |
| $R$ | – | individual gas constant for steam, J(kg K) |
| $s''$, $s'$ | – | specific entropy of saturation for steam and water, J(kg K) |
| $t$ | – | temperature,°C |
| $T$ | – | absolute temperature, K |
| $U$ | – | overall heat transfer coefficient, W/($m^2$ K) |
| $w_w$ | – | average velocity of the water in the condenser tubes, m/s |
| $v''$, $v'$ | – | specific volume of saturation for steam and water, $m^3$/kg |

### Greek symbols

| | | |
|---|---|---|
| $\varepsilon$ | – | effectiveness of the steam condenser |

### Subscripts

| | | |
|---|---|---|
| $i$ | – | inlet |
| $o$ | – | outlet |
| $r$ | – | reference parameter |
| $w$ | – | water |
| $s$ | – | steam |
| 2 | – | cooling water |

# 1 Introduction

Mathematical models have been developed to analyze thermal systems and their components. Depending on particular processes, the models can be related to steady or transient (dynamic) states. Mathematical models have been created mainly to determine the performance of thermal systems and their components. Mathematical modeling can be used at the design stage when a so-called design point, referring to a structure of the thermal system, is designed or to assess the performance of a system in off-design conditions. At the design stage, a structure of the system is created and characteristic properties of each component, such as geometry of the boiler, heat exchangers, and the turbine, are determined. For off-design conditions, the operation of a known structure of the system and geometry of machines and equipment is analyzed in conditions that are different from design conditions. In this context the research focuses on how a change in operating conditions, particularly the load, of generation sources affects the performance of a power generation unit. For instance, in order to ensure an optimal operation of a unit, it is examined how properties of the cooling system [1–4] (temperature and the cooling water mass flow rate) affect the performance of the condenser, the cooling system, and the whole thermal system of the unit, or how steam properties (pressure, temperature) influence the efficiency and power output of the system [5–8].

Mathematical modeling employs mainly three fundamental equations, namely the mass balance, energy balance, and moment of momentum balance, which are used to work out pressure drops. As far is the turbine is concerned, the energy balance is utilized to find the power output and efficiency of the turbine, and in the case of heat exchangers to find the flow rate of the heat transferred and the effectiveness of the heat exchanger. For the turbine, the Fugel-Stodola equation can also be used as a relation between the ratio of outlet and inlet pressures to the rate of mass flowing through a group of stages. For the heat exchanger, we also have the Péclet equation [9, 10] which is used to determine the heat flow rate as the product of the overall heat transfer coefficient, heat transfer surface, and logarithmic mean temperature difference between fluids.

When the performance of heat exchangers in off-design conditions is analyzed, geometry of the heat exchanger (heat transfer surface area, length of the heat exchanger, and properties of the tube assembly) is given and temperatures and mass flow rates of fluids at the heat exchanger inlet are known. Temperatures of fluids at the heat exchanger outlet are to be found. We have two equations: the energy balance and the Péclet equation; these can be used to calculate temperatures of fluids at the heat exchanger outlet. Due to the implicit form of temperatures of fluids at the heat exchanger outlet, the calculations have to be performed iteratively.

The greatest inaccuracy in heat exchanger modeling occurs when the overall heat transfer coefficient is determined. The main reason for this inaccuracy is that the overall heat transfer coefficient takes into account heat transfer coefficients that are calculated from approximate equations established on the basis of a dimensional analysis. In addition, these equations are non-linear and are functions of dimensionless numbers, such as the Reynolds number and Prandtl number. In order to determine dimensionless values, one has to find out thermodynamic properties of fluids, such as density, thermal conductivity, kinematic and dynamic viscosity, and specific heat. The thermodynamic properties are most often established for the average temperature of fluids, taking into account temperatures at the heat exchanger inlet and outlet; this is why in order to find out temperatures of fluids at the heat exchanger outlet, one has to make iterative calculations.

Depending on the complexity of the heat exchanger, its model may comprise about ten to twenty equations, most frequently non-linear and implicit ones, which have to be solved iteratively [11–13]. In addition, geometrical data of a particular heat exchanger have to be provided, such as heat transfer surface area, length of the heat exchanger, pitch, tube diameter, shell diameter, number of tubes, and material properties including thermal conductivity of a tube. It is not always possible, however, to provide exact geometrical data. To reduce these difficulties in determining the performance of heat exchangers, approximate mathematical models have been developed. The approximation may relate to the overall heat transfer coefficient [14], effectiveness of the heat exchanger [15–20], or temperatures of fluids at the heat exchanger outlet [21–24].

The power plant condenser is most often a shell-and-tube heat exchanger. Steam flowing from the low-pressure part of the turbine condenses on the outer surface of tubes. Cooling water is flowing inside the tubes to remove heat of condensing steam and transfer it to the environment or a cooling tower. In the case of a steam condenser there are five variables: cooling water temperature at the condenser inlet and outlet, the cooling water mass flow rate, steam pressure (temperature), and the steam mass flow rate. We have two equations, namely the energy balance and Péclet equation; therefore, by means of the condenser model we can determine two output properties. In most cases these properties are pressure (temperature) of condensing steam and temperature of cooling water at the condenser outlet. The input properties of the condenser model include: the temperature and mass flow rate of cooling water, and the mass flow rate of steam. In addition to these three properties, the steam condenser performance is also influenced by thermal resistance of inert gases and fouling [25, 26]. Since there is a high level of vacuum in the condenser and the device is not leakproof, air is sucked into it. The presence of air elevates pressure in the condenser and inhibits steam flow to the heat transfer surface, which results in deterioration of heat transfer conditions. Air is removed from the condenser by means of steam or water jet ejectors, and vacuum pumps have been used in recent years for this purpose. Systems designed to remove inert gases usually operate failure-free [27, 28]. As impurities contained in cooling water precipitate and build up on the inner surface of tubes, thermal resistance of fouling becomes higher. Sponge balls are used to clean the inner surface (the heat transfer surface) of tubes to prevent sediment from building up on it [25]. The influence of inert gases and sediment on the surface of tubes was taken into account in the model indirectly as the condenser performance in reference conditions.

The HEI (Heat Exchange Institute) model [29–31] is an example of an approximate model designed to calculate the heat transfer coefficient for a condenser. In this model, the overall heat transfer coefficient is given as a product of a certain constant, the root of cooling water velocity, and coefficients related to the effect of cooling water temperatures, tube material, tube outer diameter, and the deposit on the tube's surface. The overall heat transfer coefficients are also approximated by means of a quadratic or linear function of temperature and the mass flow rate (velocity). The dimensionless number of heat transfer units (NTU) [32], which is equal to the product of the heat transfer coefficient and heat transfer surface area divided by the lower heat capacity rate of both fluids, is also approximated using a dimensional analysis.

Approximate equations for the effectiveness of a heat exchanger have also been developed. One of the first such equations is the one proposed by Beckman in the form of products of temperatures at the heat exchanger inlet and mass flow rates which are raised to the power of certain exponents [15]. Other approximate equations for the condenser effectiveness can be found in [17–20]. These equations also contain constant coefficients which have to be determined using measurement data or manufacturer specifications of a given heat exchanger. Black box models are most frequently used to develop approximate equations for fluid temperatures at the heat exchanger outlet. These models are designed to examine how each input variable affects the output ones, and approximation is based on the resulting curves.

Approximate models often contain constant coefficients which need to be calculated from measurement data or technical specifications of a particular heat exchanger. The coefficients in approximate equations take different values for various types of heat exchangers, which causes some difficulty in their application. This is why as far as approximate equations are concerned, attempts have been made to include reference parameters (which are most commonly parameters relevant to the nominal state) and to reduce the number of constant coefficients with little compromise on accuracy of the model.

The application of reference parameters in an approximate equation for the performance of a condenser is shown in [33]. The proposed equation is less accurate for small cooling water mass flow rates.

In [34], in order to determine cooling water temperature at the condenser outlet and steam temperature (pressure), two equations are given in the form of functions of only inlet variables, that is temperature and mass flow rate of cooling water at the condenser inlet, the steam mass flow rate, and values of these properties in reference conditions. These two proposed equations contain no constant coefficients, which makes them universal at the expense of a relatively small error of the model. Reference parameters were taken for the average cooling water temperature at the condenser inlet. Cooling water temperature at the condenser outlet was calculated from the first equation. This equation was formulated from the energy balance considering the reference parameters and assuming that the ratio of actual heat of condensing steam to that in reference conditions is approximately equal to one. The second approximate equation for the terminal temperature difference that makes it possible to calculate steam temperature (pressure) was proposed based on curves which were generated by a condenser simulator for a wide range of changes of input properties. The approximate equation for the terminal temperature difference as a function of the steam mass flow rate containing reference properties was proposed based on the analysis of the condenser characteristics. Since the reference parameters are taken for the average cooling water temperature and it is assumed that the terminal temperature difference depends only on the steam mass flow rate (which has the greatest impact), the inaccuracy of the model increases for cooling water temperatures that are considerably different from the reference value.

The aim of this paper is to present a highly accurate approximate equation for the condensing steam pressure as a function of the cooling water mass flow rate and temperature at the condenser inlet, containing reference parameters and no constant coefficients which would have to be obtained from technical documentation or measurement data of a particular heat exchanger. Since there are no such constant coefficients, the proposed equation can be applied universally. The paper presents an approximate model of a condenser that can be used to calculate the condensing steam pressure (temperature) being the key property influencing the performance of a power plant condenser. In addition, an equation for cooling water temperature at the condenser outlet is given. The proposed equations were checked against guarantee measurements made in the condenser and measurement data gathered during the operation of a 200 MW steam power unit. The proposed model takes into account the initial thermal resistance of fouling and inert gases by means of reference parameters.

# 2 Mathematical model of a steam condenser

## 2.1 Equation for cooling water temperature at the condenser outlet

Using the energy balance of the condenser based on data of the current operating point and reference data (as reference parameters, it usually takes nominal parameters), we obtain the cooling water temperature at the condenser outlet that can be given as

$$t_{2o} = t_{2i} + \frac{\dot{m}_s}{\dot{m}_{sr}} \frac{\dot{m}_{wr}}{\dot{m}_w} \left( t_{2or} - t_{2ir} \right) . \tag{1}$$

In off-design conditions, both the steam enthalpy at the outlet of the low pressure part of the turbine (at the condenser inlet) and the steam dryness fraction change. Changes in these properties are rather small [8], which allows us to conclude that the ratio of heat of condensing steam at the current operating point to the one in reference conditions is approximately equal to one. This assumption was verified as correct in [34]. In order to determine cooling water temperature at the condenser outlet, one needs values of cooling water temperature ($t_{2i}$) and mass flow rate ($\dot{m}_w$) at the condenser inlet, the steam mass flow rate ($\dot{m}_s$), and values of these properties in reference conditions.

## 2.2 Equation for the steam condenser effectiveness

The condenser effectiveness is calculated from the energy balance and the Péclet equation:

$$\dot{Q} = \dot{m}_w c_w \left( t_{2o} - t_{2i} \right) , \tag{2}$$

$$\dot{Q} = U A \Delta T_{\ln} , \tag{3}$$

where the logarithmic mean temperature difference of a condenser takes the form

$$\Delta T_{\ln} = \frac{t_{2o} - t_{2i}}{\ln \dfrac{t_s - t_{2i}}{t_s - t_{2o}}} . \tag{4}$$

On rearranging Eqs. (2) to (4), the effectiveness of the condenser can be given as

$$\varepsilon = \frac{t_{2o} - t_{2i}}{t_s - t_{2i}} = 1 - \mathrm{e}^{-\mathrm{NTU}}, \tag{5}$$

where NTU is

$$\mathrm{NTU} = \frac{UA}{\dot{m}_w c_w} . \tag{6}$$

The overall heat transfer coefficient according to the HEI standard [29–32] has the form

$$U = 6.47878(441.325 - d_o)\sqrt{w_w} c_T c_M c_F , \tag{7}$$

where: $c_T$ – coefficient indicating the effect of cooling water temperature at the condenser inlet, $c_M$ – coefficient of the type of tube material, $c_F$ – coefficient of the resistance of sediment.

Assuming $C_1$ to be constant and equal to

$$C_1 = 6.47878(441.325 - d_o) c_M c_F \tag{8}$$

the equation for the overall heat transfer coefficient can be given as

$$U = C_1 \sqrt{w_w}\, c_T , \tag{9}$$

where $c_T$ according to the HEI standard has the form

$$c_T = 1.395 - \mathrm{e}^{-\frac{t_{2i}}{22.61}} - \frac{t_{2i} - 21}{166} . \tag{10}$$

The dependence of $c_T$ as a function of cooling water temperature according to Eq. (10) and its power function approximation are shown in Fig. 1. It is proposed to approximate $c_T$ by means of a power function according to the formula

$$c_T = C_2 \left(t_{2i}\right)^{0.22} . \tag{11}$$

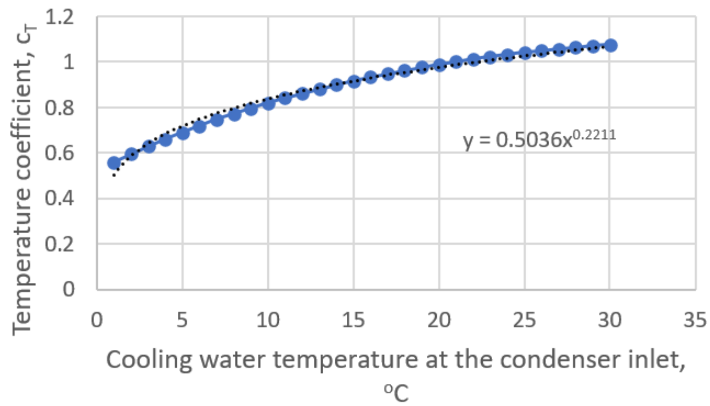


Figure 1: The dependence of $c_T$ as a function of cooling water temperature according to Eq. (10) and its power function approximation.

Taking into account the equation of continuity for the cooling water mass flow rate, the ratio of the overall heat transfer coefficient to its value in reference conditions can be given as

$$\frac{U}{U_r} = \frac{\sqrt{w_w}}{\sqrt{w_{wr}}}\frac{c_T}{c_{Tr}} = \frac{\sqrt{\dot{m}_w}}{\sqrt{\dot{m}_{wr}}}\left(\frac{t_{2i}}{t_{2ir}}\right)^{0.22}. \tag{12}$$

Taking into account Eqs. (5) and (6), the overall heat transfer coefficient in reference conditions can be also written in the form

$$U_r = -\frac{\dot{m}_{wr}c_w}{A}\ln(1-\varepsilon_r). \tag{13}$$

On considering Eqs. (12) and (13), the number of transfer units can be expressed as

$$\mathrm{NTU} = -\ln(1-\varepsilon_r)\sqrt{\frac{\dot{m}_{wr}}{\dot{m}_w}}\left(\frac{t_{2i}}{t_{2ir}}\right)^{0.22}. \tag{14}$$

Thus, the effectiveness of the condenser can be given as

$$\varepsilon = 1-\exp\left[\ln(1-\varepsilon_r)\sqrt{\frac{\dot{m}_{wr}}{\dot{m}_w}}\left(\frac{t_{2i}}{t_{2ir}}\right)^{0.22}\right]. \tag{15}$$

The condenser effectiveness is thus given as a function of temperature and the mass flow rate of cooling water at the condenser inlet and their values in reference conditions.

### 2.3 Equation for condensing steam pressure

Once the condenser effectiveness is calculated from Eq. (15), steam condensation temperature can be determined from Eq. (5) in the form

$$t_s = t_{2i} + \frac{(t_{2o} - t_{2i})}{\varepsilon} = t_{2i} + \frac{(t_{2o} - t_{2i})}{1 - \exp\left[\ln(1-\varepsilon_r)\sqrt{\frac{\dot{m}_{wr}}{\dot{m}_w}\left(\frac{t_{2i}}{t_{2ir}}\right)^{0.22}}\right]} . \tag{16}$$

In saturation conditions pressure of condensing steam is a function of its temperature, $p_s = f(t_s)$. Pressure of condensing steam as a function of temperature can be calculated using approximate equations available in a number of software applications, such as the IF97 formulation in MS Excel worksheet XSteam_Excel_v2.6.xls [35], or from the Clapeyron–Clausius equation

$$(v'' - v')\, dp_s = (s'' - s')\, dT_s . \tag{17}$$

At phase transition the difference between specific entropy of saturated steam and that of saturated water is expressed as

$$s'' - s' = \frac{r}{T_s} . \tag{18}$$

Assuming that the specific volume of saturated steam greatly exceeds that of saturated water ($v'' \gg v'$) and that for the saturated steam we can use the Clapeyron ideal gas equation ($v'' = RT_s/p_s$), Eq. (17) can be transformed into

$$\frac{dp_s}{p_s} = \frac{r}{R}\frac{dT_s}{T_s} . \tag{19}$$

On integrating Eq. (19) between the current and reference parameters and assuming a constant value of phase transition heat, we obtain

$$p_s = p_{sr} \mathrm{e}^{\frac{r_r}{R}\left[\frac{1}{T_{sr}} - \frac{1}{T_s}\right]} . \tag{20}$$

Figure 2 compares pressure calculated from Eq. (20) and according to the approximation formulation IF97 implemented in XSteam_Excel_v2.6.xls as a function of saturation temperature. The difference (the relative error) is 2% or less. The reference parameters were taken for the reference steam pressure ($p_{sr}$) of 4 kPa, which corresponds to the reference steam temperature ($t_{sr}$) of 28.96°C or ($T_{sr}$) of 302.11 K. The smallest relative error is at the point of reference conditions and is increasing up to a maximum of 2% when steam properties are farther from this point.

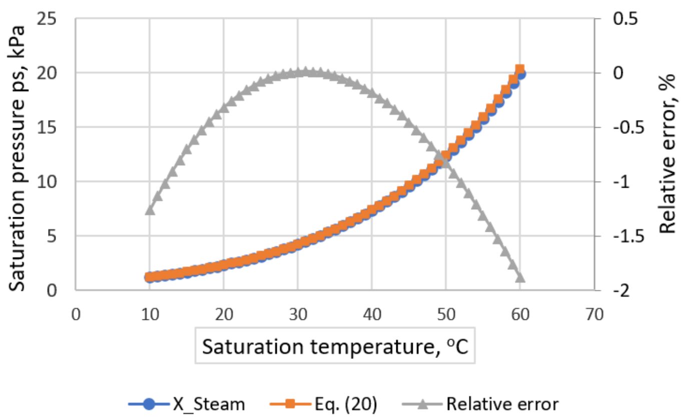


Figure 2: Saturation pressure as a function of saturation temperature according to Eq. (20) and XSteam_Excel_v2.6.xls.

For prescribed values of the steam mass flow rate at the condenser inlet ($\dot{m}_s$), the cooling water mass flow rate ($\dot{m}_w$), and cooling water temperature at the condenser inlet ($t_{2i}$), Eqs. (1), (15), (16), and (20) are used to calculate cooling water temperature at the condenser outlet ($t_{2o}$), the condenser effectiveness ($\varepsilon$), steam condensation temperature ($t_s$), and condensing steam pressure ($p_s$).

## 3 Results

The proposed equations were first checked against measurement data collected from a 200 MW power generation unit under variable load conditions. Table 1 lists data available from five measurements. The following measurements were taken in the condenser: cooling water temperature at the condenser inlet and outlet, cooling water mass flow rate, power output of the unit, steam mass flow rate, and pressure of steam condensing in the condenser. Table 1 also lists values of condensing steam temperature and the condenser effectiveness according to measurement data.

Properties (condensing steam pressure and cooling water temperature at the condenser outlet) calculated from the proposed equations including relative errors are listed in Table 2.

Figure 3 compares cooling water temperatures at the condenser outlet according to measurements and calculated from the proposed Eq. (1) in variable load conditions of the power unit.

Table 1: Properties measured in the condenser.

| Property | Parameter | Unit | Measurement 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|---|---|---|
| Power output of the unit | $P$ | MW | 140 | 160 | 180 | 200 | 225 |
| Steam mass flow rate at the condenser | $\dot{m}_s$ | kg/s | 82.042 | 90.46 | 101.188 | 112.64 | 127.37 |
| Cooling water mass flow rate | $\dot{m}_w$ | kg/s | 8048.8 | 8243.9 | 8123.2 | 8073 | 8104.1 |
| Mean water temp. at the condenser inlet | $t_{2i}$ | °C | 7.91 | 10.14 | 8.665 | 8.82 | 10.555 |
| Mean water temp. at the condenser outlet | $t_{2o}$ | °C | 13.54 | 16.17 | 15.48 | 16.42 | 19.04 |
| Pressure of steam in the condenser | $p_s$ | kPa | 2.1 | 2.4 | 2.5 | 2.7 | 3.2 |
| Temperature of steam in the condenser | $t_s$ | °C | 18.27 | 20.41 | 21.08 | 22.34 | 25.16 |
| Condenser effectiveness | $\varepsilon$ | – | 0.543 | 0.587 | 0.549 | 0.562 | 0.581 |

Table 2: Properties calculated from the proposed equations including relative errors.

| Property | Parameter | Unit | Measurement 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|---|---|---|
| Power output of the unit | $P$ | MW | 140 | 160 | 180 | 200 | 225 |
| Pressure of steam in the condenser, measured | $p_s$ | kPa | 2.10 | 2.40 | 2.50 | 2.70 | 3.20 |
| Pressure of steam in the condenser, calculated | $p_{s\_c}$ | kPa | 2.04 | 2.43 | 2.43 | 2.65 | 3.20 |
| Mean water temp. at the condenser outlet, measured | $t_{2o}$ | °C | 13.54 | 16.17 | 15.48 | 16.42 | 19.04 |
| Water temp. at the condenser outlet, calculated | $t_{2o\_c}$ | °C | 13.41 | 16.06 | 15.39 | 16.35 | 19.04 |
| Relative error of steam pressure | $\delta p_s$ | % | 2.76 | –1.33 | 2.97 | 1.79 | 0.00 |
| Relative error of cooling water temp. at the condenser outlet | $\delta t_{2o}$ | % | 0.94 | 0.66 | 0.58 | 0.41 | 0.00 |
| Condenser effectiveness calculated from the equations | $\varepsilon_c$ | – | 0.555 | 0.565 | 0.564 | 0.570 | 0.581 |
| Temperature of steam in the condenser calculated from the equation | $t_{s\_c}$ | °C | 17.82 | 20.63 | 20.59 | 22.04 | 25.16 |

Figure 4 compares pressures of steam in the condenser according to measurements and calculated from Eqs. (15) to (17) in variable load conditions of the power unit.

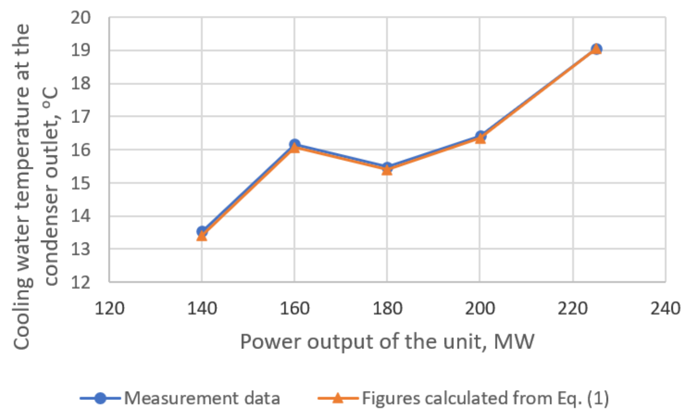


Figure 3: Cooling water temperature at the condenser outlet: measurements *vs.* calculations according to Eq. (1).

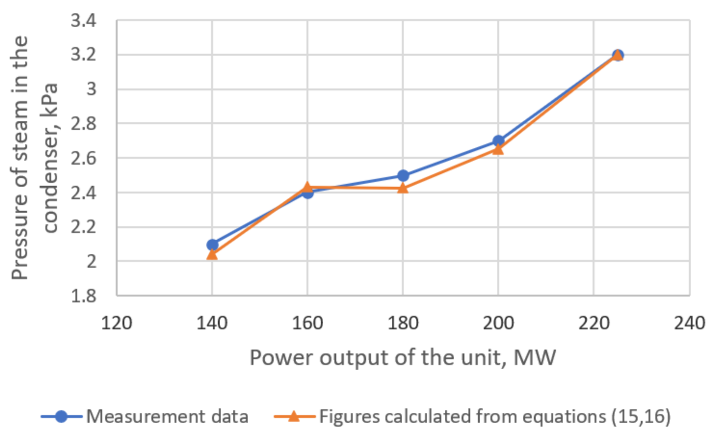


Figure 4: Pressure of steam in the condenser: measurements *vs.* calculations according to Eqs. (15) to (16).

Condensing steam pressure and cooling water temperature at the condenser outlet depend on the load of the power unit. As the load becomes lower and thus steam is fed to the condenser at a lower mass flow rate, there is a drop in cooling water temperature at the condenser outlet and in condensing steam pressure. On the other hand, when cooling water temperature at the condenser inlet is higher, condensing steam pressure and cooling outlet water temperature are on the rise, which can be seen in the case of measurement 2. The proposed equations reflect quite close the condenser performance: in both cases the curves nearly coincide. Major discrepancies occur for steam pressure, but this stems from the fact that the measurement of condensing steam pressure is prone to greater errors than the measurement of cooling water temperature at the condenser outlet due to a large volume of space where steam is condensing.

The proposed equations were also checked against operational data of a 200 MW power generation unit. The authors had hourly measurements spanning one year of the power plant operation. During a year the unit operated for about 7000 hours; during the rest of the year the unit was shut down and maintenance activities were carried out. Based on measurement data, cooling water temperature at the condenser outlet was calculated from Eq. (1). The difference between measured cooling water temperatures at the condenser outlet and those calculated from the equation is shown in Fig. 5.

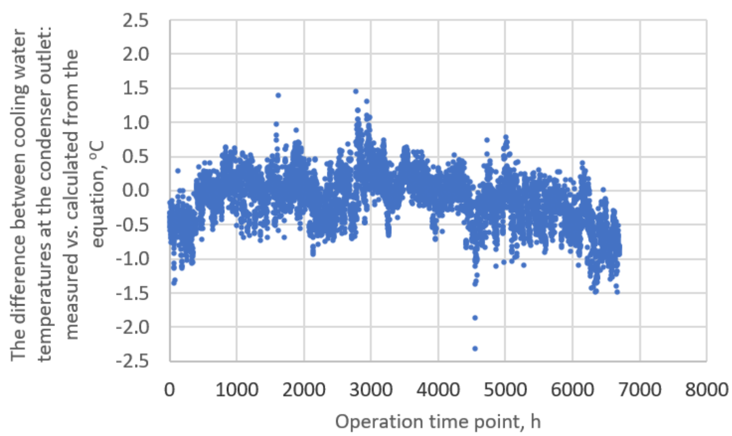


Figure 5: The difference between cooling water temperatures at the condenser outlet: measured *vs.* calculated using Eq. (1).

The difference between measured condensing steam temperatures and those calculated using Eqs. (15) and (16) is shown in Fig. 6.

The difference between measured cooling water temperatures at the condenser outlet and those calculated from the equations is in the range of $\pm 1.0°$C. This is also the case with the difference between condensing steam temperatures measured and calculated from Eqs. (15) and (16). For some data this range is slightly exceeded. Temperature is measured in power plants to an accuracy of $\pm 1.0°$C.

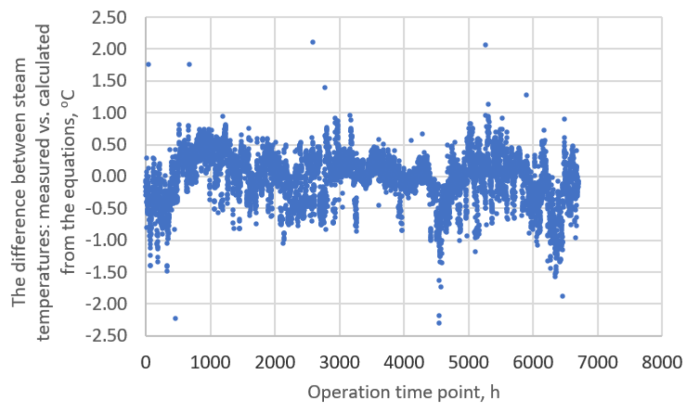


Figure 6: The difference between steam temperatures: measured *vs.* calculated from Eqs. (15) and (16).

## 4 Conclusions

The paper proposes a formula to calculate steam condensation pressure (temperature) in a power plant condenser in off-design conditions. The input properties of the model are: the mass flow rate of steam and the temperature and mass flow rate of cooling water at the condenser inlet. In order to determine the performance of the condenser in terms of condensing steam pressure and cooling water temperature at the condenser outlet in off-design conditions, one has to provide the input properties and their corresponding values in reference conditions. Ideally, the reference properties should be taken as nominal operating parameters of the power unit.

The proposed equations contain no constant coefficients which would have to be established based on data from technical specifications of a condenser or measurement data. Since there are no such constant coefficients, the proposed equations describing the condenser performance in off-design conditions are universally applicable.

The proposed equations were checked against guarantee measurements and measurement data during the operation of a 200 MW steam power unit. As the analysis indicates, the proposed equations proved to be highly accurate. The proposed model can be used in the event of off-design operating conditions of a condenser and for diagnostic purposes to assess the technical condition of a condenser.

In addition to these three key inlet properties (the mass flow rate of steam and the temperature and mass flow rate of cooling water at the condenser inlet), the condenser performance is also influenced by thermal resistance of inert gases (air) and fouling, and the steam dryness fraction. In off-design conditions, the steam dryness fraction at the condenser inlet changes only little. For instance, for the condenser under consideration the steam dryness fraction varied between 0.92 and 0.94, which makes the ratio of heat of condensing steam at the current operating point to the one in reference conditions approximately equal to one. This condition was used to devise Eq. (1). The proposed model takes into account the initial (reference) thermal resistance of inert gases and fouling. If vacuum conditions in the condenser deteriorate or the fouling thermal resistance increases, the condenser performance will drop since steam in the condenser will condense at a higher pressure. The proposed mathematical model can be therefore used not only to analyze the operation of a power unit in off-design conditions but also as a diagnostic tool to show a difference between the pressure reading and the pressure calculated from the proposed condenser model. No increase in fouling thermal resistance was observed in the condenser considered in the paper since the condenser was constantly cleaned by means of sponge balls on the cooling-water side. The inert gas removal system also operated as intended for measurement data included in the analysis.

The paper proposes a relatively simple model of a condenser in off-design conditions. By combining heat flow equations, measurement data and parameters in reference conditions, the model proved to be satisfactorily accurate. This model can be used in optimization and diagnostic analyses of the performance of a power generation unit.

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