Cross verification of a simplified model of pure steam condensation in a tube bundle using experimental and CFD-simulation data

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Abstract. This paper presents an analysis of a simplified model of pure steam condensation in a bundle of smooth horizontal tubes. The model was based on classical recommendations for calculating the condensation process in tube bundles. Two different approaches were used to calculate the condensation heat transfer coefficient in the model. First, the inundation effect was not considered, and the average heat transfer coefficient was calculated for condensation on a single cylinder using the Nusselt equation. Second, condensation drainage which is asymmetric to the vertical diameter of the tube was considered. The results of these calculations were verified using available experimental data. The analysis of distributions of condensation local characteristics was presented. The results of detailed CFD simulation using a previously developed author’s model was employed for additional cross-verification of closing equations used in the simplified model.

1. Introduction

Condensers of steam are one of the key elements of many technological processes. Despite abundant literature devoted to condensation enhancement via heat exchange surface modification, condensation occurs on bundles of smooth horizontal tubes of various configurations in most systems [1]. This is largely because in such systems, the efficiency of heat transfer is also controlled by the processes on the cooling agent side. For the design of existing condensers, the methods of VTI [2], HEI [3], BEAM [4] etc. are used, which were developed based on the experimental results of tests on a number of condenser constructions. These methods do not fully reflect the influence of all factors on the condensation process. The modernization of these approaches is the so-called differential condenser calculation method [5], which integrates well with modern computer fluid dynamic (CFD) codes in which the condenser can be modeled by a porous medium with effective properties [6,7]. The authors of a recent review [1] note that they were struck by the continuous change in the closing equations between the publications of different authors and even between the publications of the same authors in which porous media approach was used. Moreover, the reasons for choosing correlations, as a rule, are not discussed in detail. All this leads to the need to develop or choose physically based closing equations.

A simplified model describing the condensation of pure steam inside a tube bundle of smooth horizontal pipes has been tested. The validity of the used closing equations is verified by a comparison with experimental data [8] on integral characteristics and local CFD modeling data [9-11].
2. Problem statement and mathematical model description

Figure 1 shows the design of the condenser selected for simplified model verification. The configuration reproduces the first three sections of the condenser from the experimental work [8], which is being developed as a small-scale model of a condenser for a high-temperature gas-and-steam turbine plant operating on combined fuel [12]. The tube bundle consists of a brass tube $d_{out}/d_{in} = 22/20$ mm with the length being 200 mm; a tube bundle pitches $S_1 = 30$ mm and $S_2 = 64$ mm. The device is characterized by a decreasing cross section area along the steam flow path. In [8], the condensation from both pure moving steam and steam from a moving vapor–gas mixture was studied.

![Figure 1. Tube bundle configuration.](image)

For the simplified model for calculating pure steam condensation, the input parameters are pressure in the condenser $p_0$; steam mass flow rate $G_0$ and cooling water temperature $T_{cw,in}$ at the inlet; cooling water mass flow rate $G_{cw}$; sequence of cooling water flow through the condenser tubes within each section; the number of tube in each vertical row, which is necessary to calculate the inundation effect. All condenser sections are connected in parallel to flow cooling water. Figure 1 shows the order in which the tubes are connected in a series in each section. Each tube is characterized by its own cooling water temperature $T_{cw}$. The iteration stopping criterion establishes a constant temperature of water in the tubes, which is an indirect criterion for the stabilization of all other characteristics in the iterative process. As an initial approximation, the temperature in each tube was assumed to be equal to the temperature of the cooling water at the section inlet. At the specified temperature values, the wall heat flux of each tube $q_w$ was estimated. The cooling water heat transfer coefficient was determined by the Petukhov–Kirillov equation [13], and the heat transfer coefficient for condensation by the Nusselt equation for a single cylinder. Further, the calculation consisted of a cyclical iteration over the following steps.

First, determine the cooling water temperature in each tube using the heat balance equation:

$$
T_{cw,i}^{n+1} = T_{cw,i}^{n} + \frac{q_{w,i} + q_{w,i-1}}{2} \frac{F}{G_{cw} c_{p,cw}}, i > 1
$$

$$
T_{cw,i}^{n} = T_{cw,in} + \frac{q_{w,i}}{2} \frac{F}{G_{cw} c_{p,cw}}, i = 1
$$

where $i$ is the tube number, $F$ is the single tube outer surface area and $c_{p,cw}$ is the cooling water heat capacity.

Second, determine the heat transfer coefficients from the tube wall to the cooling water $\alpha_{cw}$ using new values of cooling water temperatures. Use the outer surface of the tubes for all heat transfer coefficient calculation.
Third, determine the temperature of the outer surface of each tube $T_w$ by solving the equation:

$$
\left(\frac{1}{\alpha_{cw}} + R_w\right)^{-1}(T_w - T_{cw}) = \alpha_{cond}(T_{sat} - T_w)
$$

(2)

$$
R_w = \frac{d_{out} \ln(d_{out}/d_w)}{2 \lambda_w}
$$

(3)

where $\lambda_w$ is thermal conductivity of the wall material, $\alpha_{cond}$ is condensation heat transfer coefficient and $T_{sat}$ is saturation temperature.

Fourth, calculate the heat flux density $q_w$:

$$
q_w = \left(\frac{1}{\alpha_{cw}} + R_w + \frac{1}{\alpha_{cond}}\right)^{-1}(T_{sat} - T_{cw})
$$

(4)

Two different approaches were used to calculate $\alpha_{cond}$. First, the inundation effect was not considered, and the average heat transfer coefficient was calculated using the Nusselt equation for a single cylinder:

$$
\alpha_{cond} = 0.728 \sqrt{\frac{\lambda^3 g \Delta \rho h_{LG}}{v(T_{sat} - T_w) d_{out}}}
$$

(5)

where $\lambda$ is condensate thermal conductivity, $g$ is gravity acceleration, $v$ is condensate kinematic viscosity, $h_{LG}$ is the latent heat of condensation and $\Delta \rho$ is the density difference between condensation and vapor phase.

In the second approach, condensate drainage asymmetric to the vertical diameter of the tube was considered. The proposed model is based on CFD modeling data [9–11] and experimental studies [14], which indicate that when cross flow condensation occurs on a horizontal cylinder, the condensate drainage point shifts from the lower cylinder point by 10–15° in the direction of steam flow. At the same time, the steam dynamic influence at the parameters considered in this paper is not enough to condensate entrainment by the steam flow in the form of individual drops. Based on the literature results, an asymmetric drainage model was used. In this model, for the windward half of the tube perimeter, the average heat transfer coefficient was determined by the equation (5), and for the other half of the tube perimeter – by integrating the differential equation for the condensate flow in the film, taking into account the condensation mass flow rate forming on the upper row tube $G_{l,in}$:

$$
G_{l,out} = \left(\frac{4}{3} A \cdot C \cdot (T_{sat} - T_w)\right)^{3/4}
$$

(6)

$$
A = \frac{\lambda d_{out}}{2 h_{LG}} \left(\frac{g \Delta \rho}{3v}\right)^{1/3}
$$

(7)

$$
\alpha_{cond} = \frac{(G_{l,out} - G_{l,in})h_{LG}}{\pi (d_{out}/2) (T_{sat} - T_w)}
$$

(8)

where $G_{l,out}$ is the condensation mass flow rate drainage from the second tube half and $C$ is a constant defined by numerical integration and equal to 2.587.

Below, the used models will be referred to as “without inundation effect” and “with asymmetric drainage”.

For all variants under consideration, the dynamic effect of steam flow on the condensate film was taken into account by modifying the heat transfer coefficient according to the recommendation [15]:
\[ \frac{\alpha_{\text{cond}}}{\alpha_{\text{Nu}}} = 1 + 9.5 \cdot 10^{-3} \Rev^{11/4} \Nu^{5} \]  \hspace{1cm} (9)

\[ \Rev = \frac{u_{c} d_{\text{out}}}{v_{c}}, \quad \Nu_{\text{Nu}} = \frac{\alpha_{\text{Nu}} d_{\text{out}}}{\lambda} \]  \hspace{1cm} (10)

where \( \alpha_{\text{Nu}} \) is the heat transfer coefficient for motionless steam calculated according to the equations described earlier, \( v_{c} \) is vapor kinematic viscosity and \( u_{c} \) is the steam velocity related to the total cross section of the row in which the tube is located.

The results of calculations based on the proposed simplified model were compared with experimental data \[8\] and results of CFD modeling \[9–11\] for steam condensation on a bundle of smooth horizontal tubes from a moving vapor–gas mixture. In \[9–11\], the equations of single-phase hydrodynamics, energy and diffusion are used for the external flow of a vapor–gas mixture. The boundary conditions (BC) for these equations on the surface of the condensate film in the approximation of a small film thickness (compared to the tube diameter) are set on the outer surface of the tubes. The parameters required for setting the BC are determined from the solution of one-dimensional equations for the condensate film for each tube. The condensation mass flow rate from the upper tubes was taken into account under the assumption that the condensation from the upper tube drainage points moves vertically down and falls on the lowest tube.

Table 1 shows the pure steam vapor condensation regimes selected for the simplified model verification.

**Table 1.** Regime parameters from \[8\]. The cooling water velocity is 1.5 m/s.

| Regime Numbers | 1  | 2  | 3  | 4  |
|----------------|----|----|----|----|
| \( p_{0} \), kPa | 6.88 | 8.22 | 9.34 | 10.88 |
| \( T_{cw,\text{in}} \), °C | 30.0 | 31.7 | 30.9 | 31.4 |
| \( G_{0} \), kg s\(^{-1}\) | 0.021 | 0.037 | 0.042 | 0.050 |

**3. Results**

Table 2 shows the heat transfer coefficients for the first section which were calculated using simplified and CFD models and obtained experimentally in \[8\]. "CFD-model without inundation effect" indicates simulation results in which it was assumed that the condensation removes from the tubes surface in location, which is determined during simulation, but no condensation deposition on the other tubes occurs. The data for the second and third sections is almost identical to the data for the first section.

**Table 2.** Heat transfer coefficient for the first section, W/(m\(^{2}\)K).

| Regime Numbers | 1  | 2  | 3  | 4  |
|----------------|----|----|----|----|
| The simplified model without inundation effect | 4860 | 4980 | 4960 | 5020 |
| The simplified model with asymmetric drainage | 4460 | 4670 | 4690 | 4790 |
| CFD-model without inundation effect | 4770 | 4900 | 4900 | 5040 |
| CFD-model with inundation effect | 4200 | 4450 | 4510 | 4530 |
| Experimental data \[8\] | 4640 | 4750 | 4810 | 4690 |

The heat transfer coefficient was determined by the following formulas:

\[ k = \frac{q}{\Delta T_{LMTD}} \]  \hspace{1cm} (11)

\[ \Delta T_{LMTD} = \left( T_{cw,\text{out}} - T_{cw,\text{in}} \right) \left( \ln \left( \frac{T_{sat} - T_{cw,\text{in}}}{T_{sat} - T_{cw,\text{out}}} \right) \right)^{-1} \]  \hspace{1cm} (12)
All data agree within the experimental error. The experimental data lie between the calculated data obtained both without and with inundation effect. This is largely because when pure steam condenses, the heat transfer coefficient is limited by the processes on the cooling water side, and the integral characteristics are almost insensitive to the details of the inundation effect calculation. A small increase in heat transfer coefficients with an increase in steam mass flow rate from regime 1 to regime 3, obtained experimentally, is reproduced by all models.

Figure 2 shows the distribution of the average condensation heat transfer coefficient on the first-row tubes. The calculation results of the proposed simplified model and CFD modeling data agree, but calculation without considering the inundation effect can lead to a 50% error in calculating the condensation heat transfer coefficient. At the same time, considering the influence of steam flow in the simplified model leads to a better agreement with the CFD modeling data when the inundation effect is not included as the equation (9) was obtained in [15] by generalizing the experimental data for a single tube located inside a model tube bundle, and the tube was not irrigated by condensation from the upper uncooled tubes.

![Figure 2. Distribution of the heat transfer coefficient for condensation on the first vertical row of tubes, considering the steam flow effect. Calculation: 1 is without inundation effect and 3 is with asymmetric drainage. CFD modeling data: 2 is without inundation effect and 4 is with inundation effect.](image)

The value of the heat transfer coefficient obtained without inundation effect increases due to the cooling conditions of the tube bundle (sequential connection of tubes for cooling water flow within each section of the condenser). Numbers of tubes in Figure 2 correspond to the numbering of the first-row tube from top to bottom and do not coincide with the tube numbers in Figure 1 (the last ones are shown in Figure 2 in brackets). As the tube number increases, the difference between the saturation temperature and the wall temperature decreases, making the heat transfer coefficient increase. A small non-monotony occurs as the 12 and 13 tubes in the first row of tube are 34 and 32 tubes for cooling water flow. For asymmetric drainage, the change in the heat transfer coefficient is non-monotonic; after a sharp drop on the second tube (by 20%), further changes are insignificant.

The effect of steam flow at the condensation process is illustrated in Figure 3. Without considering the steam flow (equation (9) is not used), the values of heat transfer coefficients for 20% below the values are obtained using the CFD simulation.
Figure 3. Distribution of the ratio of heat transfer coefficient during condensation on the first-row tubes to the heat transfer coefficient on the top tube, calculated with simplified model without taking into account steam flow: 1 is without inundation effect and 2 is with asymmetric drainage. For the CFD modeling data, 3 is without inundation effect and 4 is with inundation effect.

Often [1] to account for the inundation effect, the following dependencies are used:

\[
\frac{\alpha_n}{\alpha_i} = \left( \frac{\sum_{i=1}^{n} \Delta G_{i,j}}{\Delta G_{j,s}} \right)^{-s}
\]  

(13)

where \(\Delta G_{i,j}\) – the condensate mass formed on the \(i\)-th tube. Figure 4 shows the dependence of the exponent \(s\) on the tube number obtained from processing data for asymmetric drainage.

Figure 4. The dependence of \(s\) on the tube number.

Accounting for the inhomogeneity of cooling of the first-row tubes was made by calculating \(\alpha_i\) by the Nusselt equation using the wall temperature of the current tube. The value of \(s\) decreases with the tube number. The literature recommends the following for choosing the value of the indicator \(s\): 0.07 [16], 0.223 [17] and 0.16 [18], as well as a linear change from 0 for the upper tube to 0.37 on the lower
one [7]. If the first constant values fit into the range shown in Figure 4, then a comparison with [7] shows that the dependence obtained in [7] work is “inverted”, relative to the dependence in Figure 4.

4. Conclusion
A simplified model has been developed for calculating the condensation of moving steam on a bundle of smooth horizontal tubes. The proposed model, considering the asymmetric flow of condensation from the upper tube, describes the experimental data [8] and the results of CFD modeling [9–11] with deviations of 10–15%. The obtained dependence of the relative heat transfer coefficient with the growth of the tube number from top to bottom is consistent with the model from [18]. Not taking into account the inundation effect can lead to a 50% error in calculating the condensation heat transfer coefficient. The steam movement leads to an increase in the heat transfer coefficient by 20% even for the case of relatively low pressures and steam velocity in a narrow cross-section that does not exceed 24 m/s. In the future, the authors will try to include the presence of non-condensing impurities in a simplified model.

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