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Evaluating Condenser Performance Based on Pressure Transition Temperature

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1. Introduction

Thermal power plants including fossil and nuclear generally adopts steam Rankine cycle as a power conversion system. A condenser is one of main components in the Rankine cycle, which condenses the exhausted steam from turbines. As cycle efficiency is mostly determined by the turbine work, the condenser is operated in a vacuum condition to maximize the enthalpy difference. Thus, it is important to evaluate the condenser performance in terms of saturation pressure when designing a power conversion cycle for a power plant.

There are several methods to analyze condenser performance. Firstly, the simplified approach was used in recent studies [1], which assumes that condenser saturation temperature has constant difference with the cooling water temperature. The simplified approach is based on empirical data without any physical meaning. Secondly, the Heat Exchanger Institute (HEI) suggested the standards when designing a steam surface condenser [2]. The HEI standards has established look-up table of overall heat transfer coefficient based on their test results. However, the specific design of the condenser such as tube bundle configuration is left for the engineers. Lastly, two-dimensional and three-dimensional numerical models were suggested by several researchers [3, 4]. These approaches have limitation in elucidating the condenser performance along with various operating and environmental conditions.

Recently, preceding study of our research team [5] found that there are limiting criterion for condenser performance which is determined by cooling water temperature and cooling duty. For given cooling duty and cooling water flow rate conditions, there exists a pressure transition temperature (PTT) above which the condenser cannot fully condense the incoming heat. The uncondensed steam then builds up pressure inside the condenser, which may result in deteriorated cycle efficiency. Thus, accurate estimation of the PTT can provide reasonable methodology to evaluate condenser performance.

The objective of this study is to develop a methodology to estimate the PTT for given thermal-hydraulic conditions based on experimental results. The analytically estimated PTT was compared with the experimental results.

2. Experimental Setup

2.1. Experimental facility

Figure 1 shows a schematic of experimental facility capable of simulating condenser condition with a representative condenser tube having diameter of 0.0254 m and length of 0.5 m. Steam is generated from the boiler with immerged cartridge heaters. The pressure reducing valves are installed at the inlets to simulate exhausted steam from the turbine. The cooling water circulates the closed loop by the frequency-drive centrifugal pump. The temperature of the cooling water is controlled using a preheater and a cooling coil inside the surge tank. The test chamber was maintained in a vacuum condition using a vacuum pump.

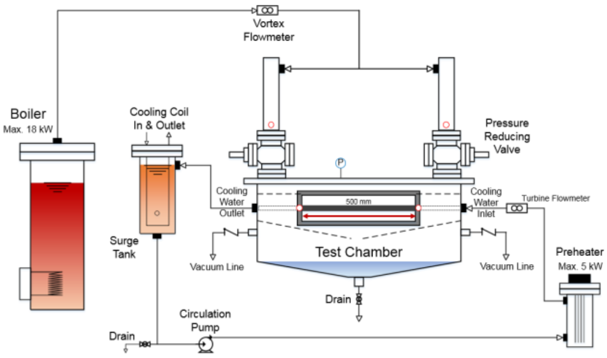

Fig. 1. Schematic of the experimental facility

The cooling duty for the condenser tube was adjusted by power input for the heaters. Vortex and turbine flowmeters were utilized to measure steam and cooling water flow rates. Vacuum pressure transmitter was installed to measure saturation pressure inside the test chamber. 4-wire resistance temperature detector (RTD, Pt100) sensors were installed at the inlet and outlet of the chamber to measure bulk water temperature. The signals from the instrumentation were saved as a digital data through a data acquisition system.

2.2. Data reduction

From the measurement data of cooling water flow rate and bulk temperature at the inlet and outlet, the average condensation heat flux was calculated as Eq. (1).

$$q''_c = \frac{\dot{m}_w c_p (T_{out} - T_{in})}{\pi d_o L} \tag{1}$$

$\dot{m}_w$, c_p, T, d_o, and L are cooling water mass flow rate, specific heat of water, bulk water temperature, external diameter of condenser tube, and length of the tube, respectively.

The condenser performance was evaluated in terms of overall heat transfer coefficient. Log mean temperature difference (LMTD) analysis and thermal resistance calculation method were used to derive the overall heat transfer coefficient. While the LMTD analysis was based on experimental data, the thermal resistance calculation was based on heat transfer correlations.

The LMTD is a parameter determining the driving force in a heat exchanger and is calculate as shown in Eq. (2). T_{sat} is saturation temperature of the steam at measured steam pressure. Consequently, the experimental overall heat transfer coefficient was calculated as shown in Eq. (3).

$$\Delta T_{LM} = \frac{\left(T_{sat} - T_{w,in}\right) - \left(T_{sat} - T_{w,out}\right)}{\ln\left(\dfrac{T_{sat} - T_{w,in}}{T_{sat} - T_{w,out}}\right)} \tag{2}$$

$$U_{\exp} = \frac{q_c''}{\Delta T_{LM}} \tag{3}$$

The correlation based overall heat transfer coefficient was calculated based on thermal resistance between the steam and the cooling water as shown in Eq. (4).

$$\frac{1}{U_{th} A_o} = \frac{1}{\bar{h}_c A_o} + \ln\left(\frac{d_o}{d_i}\right)\frac{1}{2\pi L k_{wall}} + \frac{1}{h_w A_i} \tag{4}$$

The three terms of the right-hand side each refer to thermal resistance by steam condensation, wall conduction, and water convection, respectively. $\bar{h}_c$ is a mean condensation heat transfer coefficient for filmwise condensation suggested by Nusselt as shown in Eq. (5) [6]. h_w is convective heat transfer coefficient of the cooling water and was calculated using the Dittus-Boelter correlation shown in Eq. (6). Thermophysical properties of the water were evaluated at mean temperature. The effect of non-condensable gases was considered negligible as the vacuum pump was continuously operated during the experiments.

$$\bar{h}_c = 0.728 \frac{k_f}{d_o}\left[\frac{\rho_f \left(\rho_f - \rho_g\right) g h_{fg} d_o^3}{\mu_f k_f \left(T_{sat} - T_{wall,avg}\right)}\right]^{1/4} \tag{5}$$

$$h_w = 0.023 \frac{k_w}{d_i} \mathrm{Re}_w^{0.8} \mathrm{Pr}_w^{0.4} \tag{6}$$

2.3. Experiment conditions and procedure

The experiments to evaluate the pressure transition temperature (PTT) were carried out for average condensation heat flux of 34 kW/m^2 with cooling water velocity of 1 and 2 m/s. The vacuum pressure was maintained at 23 kPa during the experiments.

Experiments were conducted with following steps. First, the water insider the boiler was heated up at atmospheric condition in an isolated state from the test chamber. Before releasing the steam, the test chamber was vacuumed until the pressure reaches lowest achievable condition and the cooling water velocity was set to desired value. The power level of the boiler was set to equivalent value corresponding to 34 kW/m^2. After all readings of temperature measurement reached steady state, the data were saved for 3 min and time averaged. The cooling water inlet temperature was increased by 2°C for each step. The experiments were continued until the condensation heat flux decrease below 25 kW/m^2.

3. Results and Discussions

3.1. Pressure transition temperature

From the previous study [5], the PTT was defined as a threshold temperature above which the condenser fails to reject the designed cooling duty. As the experiments in this study were conducted with fixed cooling duty condition, the cooling water temperature where the average condensation heat flux started to diminish was determined as the PTT.

As shown in Fig. 2, the average condensation heat flux maintains a same value before cooling water temperature reaches certain value. The PTTs were 52.41°C and 54.15°C for cooling water velocity of 1 and 2 m/s, respectively. After this point, it showed decreasing trend. This result indicates that the designed cooling duty of 34 kW/m^2 requires at least 11.59°C and 9.85°C temperature difference to be removed by the given condenser conditions.

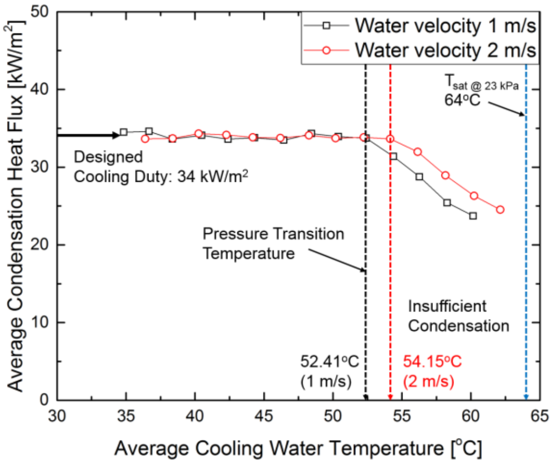

Fig. 2. Average condensation heat flux along with average cooling water temperature.

The temperature difference between the saturated steam and the PTT means the minimum temperature difference (MTD) required to reject the designed cooling duty for given thermal-hydraulic conditions. If the cooling water temperature is lower than the PTT, the condenser has sufficient thermal margin and it has chance to lower steam saturation temperature to maximize the cycle efficiency. On the other hand, if the water temperature exceeds the PTT, the condenser shows insufficient condensation and the uncondensed steam builds-up pressure. The pressure increases until the temperature difference between the steam and cooling water again satisfies the MTD. If we assume that condenser is operated to make saturation temperature as low as possible, the saturation temperature of the steam always satisfies the MTD condition with the cooling water temperature. Thus, accurate evaluation of the PTT can provide physically meaningful relationship between the steam and cooling water.

3.2. Overall heat transfer coefficient

The overall heat transfer coefficients obtained by LMTD analysis and thermal resistance calculation were compared as shown in Fig. 3. It is noticed that overall heat transfer coefficient obtained based on heat transfer correlation fails to predict the data obtained from the experimental data before the cooling water temperature reaches the PTT. After the PTT however, the estimations are in good agreement with the experimental data within 10%. Faster cooling water velocity condition showed larger uncertainties as the temperature difference between the inlet and outlet became smaller. As the condensation heat transfer correlation suggested by the Nusselt assumed there are plenty of steam around the sub-cooled surface, it cannot be applied for the region below the PTT. Thus, the theoretical approach can be applied starting from the PTT.

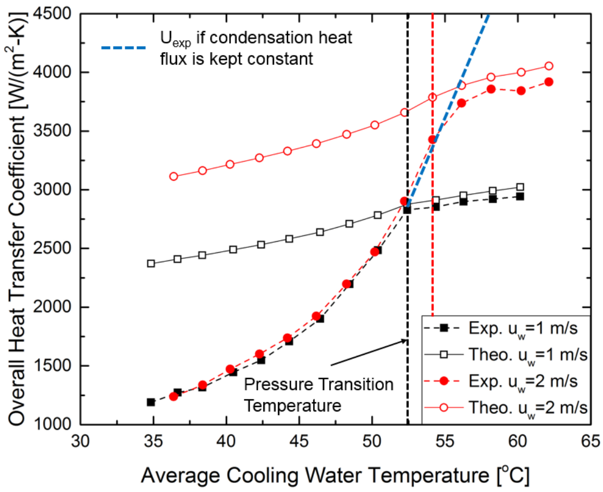

Fig. 3. Comparison of overall heat transfer coefficients obtained by LMTD and thermal resistance analysis

If we assume that condensation heat flux is constant for entire region of water temperature, the overall heat transfer coefficients obtained by the LMTD U_{exp} and thermal resistance analysis U_{th} only coincides at a single data point which can be determined as the PTT value like shown in blue dotted line in Fig. 3. Figure 4 shows the algorithm to estimate the PTT for given condenser condition using the assumption. Like the experiment procedure, starting from the water temperature at room temperature with fixed condensation heat flux, it increases until the overall heat transfer coefficients obtained by the LMTD and thermal resistance analysis become equivalent. Figure 5 shows the comparison of the PTT between the experiments and estimation using the algorithm. It shows good agreement with relative error less than 1.5%.

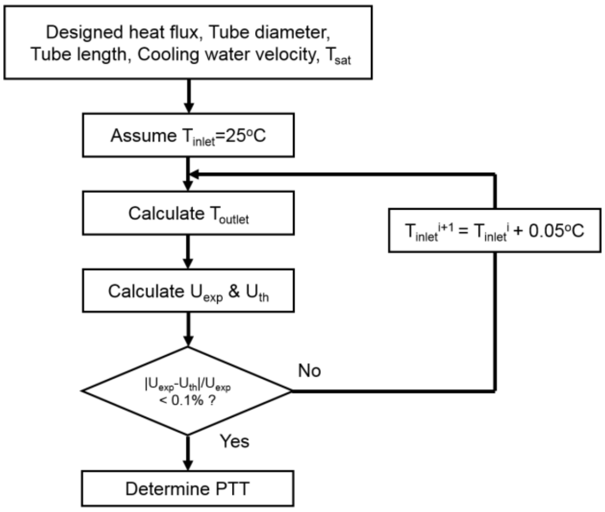

Fig. 4. Algorithm to estimate the PTT for given condenser condition.

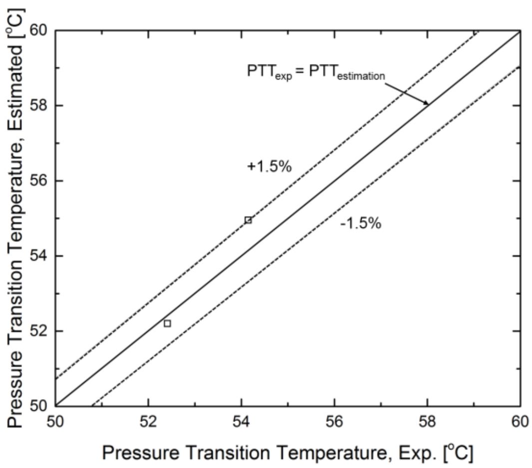

Fig. 5. Comparison of the PTT between the experiments and estimation using the algorithm.

The methodology obtained in this study cannot be directly applied to the power plant scale condensers. The real condenser has additional characteristics not considered in the experiments such as inundation due to tube bundle, fast entering velocity of exhausted steam, and fouling resistance due to chemical reaction with sea water. If we can consider these features to the thermal resistance analysis, the condenser performance can be evaluated using the methodology based on the PTT. As the PTT is dependent not only on thermal-hydraulic conditions but also the geometrical condition, it is expected to overcome the limitations of preceding models for condenser performance. Development of a model to estimate the PTT in power plant scale is currently an on-going research. The model is expected to be utilized for design and multi-objective optimization of condenser system.

4. Conclusion

In this study, a new methodology to evaluate condenser performance was suggested in terms of pressure transition temperature (PTT). Experiments were carried out to evaluate the PTT in a vacuum condition. From overall heat transfer coefficient analysis, an algorithm to estimate the PTT was derived. Major findings are summarized as follows:

- ✓ The PTT was clearly shown in the experiments showing the decreasing trend of condensation heat flux after certain water temperature.
- ✓ Overall heat transfer coefficient obtained by thermal resistance analysis using heat transfer correlations failed to predict experimental data until the water temperature reaches PTT.
- ✓ By adding the additional characteristics of power plant scale condenser system to thermal resistance calculation, suggested methodology is expected to be used in design and optimization studies for condenser system.

Nomenclatures

q''_c	Average condensation heat flux [W/(m^2-K)]
$\dot{m}_w$	Cooling water mass flow rate [kg/s]
c_p	Specific heat of water [J/(kg-K)]
$T_{w,in}$	Bulk water temperature at the inlet [°C]
$T_{w,out}$	Bulk water temperature at the outlet [°C]
d_o	External diameter of tube [m]
d_{in}	Internal diameter of tube [m]
L	Length of tube [m]
ΔT_{LM}	Log mean temperature difference [°C]
T_{sat}	Saturation temperature of steam [°C]
$U_{\exp}$	Overall heat transfer coefficient based on LMTD analysis [W/(m^2-K)]
U_{th}	Overall heat transfer coefficient based on thermal resistance analysis [W/(m^2-K)]
$\overline{h}_c$	Mean condensation heat transfer coefficient [W/(m^2-K)]
A_o	External area of the tube [m^2]
A_i	Internal area of the tube [m^2]
k_{wall}	Thermal conductivity of tube material [W/(m-K)]
k_f	Thermal conductivity of saturated water [W/(m-K)]
ρ_f	Density of saturated water [kg/m^3]
ρ_g	Density of saturated vapor [kg/m^3]
h_{fg}	Latent heat of water [J/kg]
μ_f	Dynamic viscosity of saturated water [Pa-sec]
$T_{wall,avg}$	Average external wall temperature of tube [°C]
h_w	Convective heat transfer coefficient [W/(m^2-K)]
Re_w	Reynolds number of cooling water [-]
Pr_w	Prandtl number of cooling water [-]

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