1. Introduction
Steam turbines are an important component of thermal and nuclear power plants. Because world energy demand has been increasing, developments of efficient and reliable steam turbines are a key issue for ensuring a stable supply of energy. Improved turbine efficiency also directly affects the reduction of stack emissions, including CO₂. The steam streaming through the rotor and stator gradually decreases in pressure and temperature and partially condenses in the steam turbine. The wetness loss caused by condensation is one of the primary losses, and nonequilibrium condensation is dominant in steam turbines. In this situation, the condensation starts at a supercooled temperature without a nucleus of water droplets. The conditions for condensation are described by Wilson point. In addition, CO₂ may condense in supercritical CO₂ compressors for a Brayton cycle. The accurate prediction of nonequilibrium condensation is also crucial for predicting the efficiency.

Nonequilibrium condensation in wet-steam flows has been investigated by a number of research groups. Most of the early research efforts conducted experiments for simple nozzle flows approximating flow passages in steam turbines. Moore et al. (1973) and Moses and Stein (1978) reported the experimental results under low-pressure conditions. Although experimental results for high-pressure wet-steam nozzle flows are still rare, Bakhtar and Zidi (1989) and Gyarmathy (2005) measured the static pressure in wet-steam nozzle flows under high-pressure condition. Numerical simulations of wet-steam nozzle flows and wet-steam flows through a two-dimensional turbine cascade have been reported by Bakhtar and Mohammadi Tochai (1980), Mohban and Young (1985), and Young (1992). Most of the existing nonequilibrium condensation models are based on the classical condensation model (Frenkel, 1955). Following such early studies, numerical studies for wet-steam flows have been further reported and 2-D and 3-D wet-steam flow simulations have become possible due to recent improvements in computing power. Our group simulated unsteady wet-steam flows through multi-stage stator and rotor blade rows accounting for nonequilibrium condensation (Miyazawa et al., 2016). As part of the trend of wet-steam flow studies, a Cambridge University group promoted a workshop for wet-steam nozzle flow simulation named the International Wet Steam Modeling Project (IWSPM) to revalidate the condensation models currently used in wet-steam flow simulations. As a part of this project, simple nozzle flow problems were used for the participants and then computational meshes for typical nozzles (Moore et al., 1973).
1973) (Moses and Stein, 1978) and the flow conditions were provided to the workshop participants. Finally, thirteen research groups, including our group, presented their results. Startzmann (2016) reported a summary of the results. This report suggests that nonequilibrium condensation models based on the classical condensation theory still have considerable errors in predicting the nucleation and growth rates of water droplets even if wet-steam flows through simple nozzle configurations are simulated. The homogeneous nucleation model (Frenkel, 1955) was originally derived using the assumption of an ideal gas. Debenedetti (1990) asserted that a condensation model that assumes an ideal gas could not accurately predict condensation under a supercritical condition in a high-temperature and –pressure environment. Bakhtar and Zidi (1989) further simulated the high-pressure nozzle problem using a finite-difference method and thermophysical models assuming a real gas. In that paper, both numerical and experimental results were in good agreement with each other. However, the thermophysical models used in the study were defined by polynomial equations with many complicated terms. Dykas et al. (2012) reported an experimental and numerical study considering a real gas equation of state for a wet-steam flow in a high-pressure nozzle. They continued to use a conventional condensation model based on classical condensation theory.

In this study, in accordance with the derivation process of the conventional model, we first derive a homogeneous nucleation model that accounts for the balance of the Gibbs free energy (GFE) in the liquid and gas states but does not use the ideal-gas assumption. The effect of a real-gas is adopted for the derivation process by using a real-gas equation of state (EOS). With this, a simple homogeneous nucleation model, which is applicable to high-pressure wet-steam flow problems, is newly derived. Next we simulate wet-steam flows under high-pressure conditions in a typical Laval nozzle (Gyarmathy, 2005) using the condensation model with the present homogeneous nucleation model. Finally, we show how the present model can improve the accuracy of predicting condensation under high-pressure conditions by comparison with the results obtained using the conventional condensation model and with the experimental data.

### Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
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<tbody>
<tr>
<td>$e$</td>
<td>total internal energy per unit volume</td>
</tr>
<tr>
<td>$G$</td>
<td>Gibbs free energy</td>
</tr>
<tr>
<td>$h$</td>
<td>enthalpy</td>
</tr>
<tr>
<td>$I$</td>
<td>homogeneous nucleation rate</td>
</tr>
<tr>
<td>$J$</td>
<td>Jacobian of transformation</td>
</tr>
<tr>
<td>$k$</td>
<td>turbulent kinetic energy</td>
</tr>
<tr>
<td>$k_B$</td>
<td>Boltzmann constant</td>
</tr>
<tr>
<td>$Kn$</td>
<td>Knudsen number</td>
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<tr>
<td>$n$</td>
<td>number density of water droplets</td>
</tr>
<tr>
<td>$p$</td>
<td>static pressure</td>
</tr>
<tr>
<td>$Pr$</td>
<td>Prandtl number</td>
</tr>
<tr>
<td>$R$</td>
<td>specific gas constant</td>
</tr>
<tr>
<td>$r$</td>
<td>radius of a water droplet</td>
</tr>
<tr>
<td>$r^*$</td>
<td>critical radius of a water droplet</td>
</tr>
<tr>
<td>$T$</td>
<td>temperature</td>
</tr>
<tr>
<td>$t$</td>
<td>physical time</td>
</tr>
<tr>
<td>$U_i$</td>
<td>contravariant velocities ($i = 1, 2$)</td>
</tr>
<tr>
<td>$u_i$</td>
<td>physical velocities ($i = 1, 2$)</td>
</tr>
<tr>
<td>$x_i$</td>
<td>Cartesian coordinates ($i = 1, 2$)</td>
</tr>
<tr>
<td>$\beta$</td>
<td>condensate mass fraction</td>
</tr>
<tr>
<td>$\delta_{ij}$</td>
<td>the Kronecker’s delta ($i, j = 1, 2$)</td>
</tr>
<tr>
<td>$\zeta_i$</td>
<td>general curvilinear coordinates ($i = 1, 2$)</td>
</tr>
<tr>
<td>$\phi$</td>
<td>compressibility factor</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density</td>
</tr>
<tr>
<td>$\kappa$</td>
<td>laminar thermal conductivity coefficient</td>
</tr>
<tr>
<td>$\kappa'$</td>
<td>turbulent thermal conductivity coefficient</td>
</tr>
<tr>
<td>$\tau_{ij}$</td>
<td>viscous stress tensors ($i, j = 1, 2$)</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>surface tension</td>
</tr>
<tr>
<td>$\Gamma$</td>
<td>mass generation rate</td>
</tr>
<tr>
<td>$\omega$</td>
<td>turbulent kinetic energy dissipation ratio</td>
</tr>
<tr>
<td>$l$</td>
<td>liquid phase</td>
</tr>
<tr>
<td>$V$</td>
<td>vapor phase</td>
</tr>
<tr>
<td>$0$</td>
<td>inlet value of total pressure or temperature</td>
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### 2. Numerical Methods

#### 2.1 Fundamental equations

Two-dimensional compressible Navier-Stokes equations, which include total density, the momentums, and total energy, are solved with equations for the number density of water droplets and condensate mass fraction of water droplets. These equations are further coupled with the SST turbulence model (Menter, 1994). The set of equations is defined in vector form by

$$\frac{\partial\mathbf{Q}}{\partial t} + \mathbf{L}(\mathbf{Q}) = \frac{\partial\mathbf{E}}{\partial t} + \frac{\partial\mathbf{F}}{\partial x_i} + \mathbf{S} + \mathbf{H} = 0 \quad (i = 1, 2)$$

(1)
where $Q, E_i (i = 1, 2), S,$ and $H$ are the vectors of unknown variables, the flux, the viscous term, and the source term, respectively, and are written in the following form:

$$ Q = J, E_i = J = \left[ \begin{array}{c} \rho \\ \rho u_1 \\ \rho u_2 \\ e \\ \rho \beta \\ \rho k \\ \rho \omega \\ \rho \omega \end{array} \right], S = \left[ \begin{array}{c} 0 \\ \tau_{ij} \\ \tau_{ij} \\ 0 \\ 0 \\ 0 \\ \sigma_{ij} \\ \sigma_{ij} \end{array} \right], H = \left[ \begin{array}{c} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{array} \right] $$

where $\sigma_{ij}, \sigma_{ij}, S_i,$ and $S_o$ are the dissipation and the source terms of $k$ and $\omega$, respectively. This study assumes that the water droplets are sufficiently small and that a homogeneous flow without velocity slip between the vapor phase and liquid droplets exists.

The convection terms in Eq. (1) are discretized by the compact MUSCL scheme (Yamamoto and Daiguji, 1993) and the flux difference scheme (Roe, 1981). The second-order central-difference method is used for the viscosity term. The LU-SGS scheme (Yoon and Jameson, 1988) is employed for the time integration.

### 2.2 Nonequilibrium condensation model

Wet-steam flows in steam turbines are governed by nonequilibrium condensation. The temperature and pressure decrease far below the saturation values after crossing the saturation line, reaching a supercooled condition. Condensation starts at a supercooled pressure and temperature at a point called the Wilson point, and the temperature and pressure turns and increases quickly toward the saturation line. This results in a rapid release of latent heat from the condensed water droplets. Such a quick change of temperature and pressure is induced by homogeneous nucleation.

Based on the classical nucleation theory (Frenkel, 1955), homogeneous nucleation assumes that nucleation starts from a small droplet that has an appropriate radius, called the critical radius. The critical radius is theoretically derived from the balance of the Gibbs free energy (GFE) between the liquid and vapor states.

The difference in GFE between the liquid and vapor phases is defined by the first and second laws of thermodynamics and the surface tension of a water droplet:

$$ \Delta G = n_{\text{liquid}}(g_{\text{lm}} - g_{\text{vm}}) + 4\pi r^2 \sigma $$

(2)

where $n_{\text{liquid}}, g_{\text{lm}}, g_{\text{vm}},$ and $r$ are the number of liquid molecules, the GFEs for one molecule in the liquid and vapor states, and the radius of the water droplet, respectively. Eq. (2) is identical to the energy required for phase change from uniform vapor to a water droplet. Assuming an isothermal field, the difference in GFE between the liquid and vapor states for one molecule is given by

$$ g_{\text{lm}} - g_{\text{vm}} = \int_{p_l}^{p} v_{\text{lm}} dp - \int_{p_l}^{p} v_{\text{vm}} dp $$

(3)

where $v_{\text{lm}}$ and $v_{\text{vm}}$ are the molecule volumes in the liquid and vapor states. If we assume $v_{\text{lm}} \ll v_{\text{vm}}$ and an ideal gas, Eq. (3) is simplified to

$$ g_{\text{lm}} - g_{\text{vm}} = -k_B T \int_{p_l}^{p} \frac{1}{p} dp $$

$$ = -k_B T \ln S $$

(4)

where $p_s$ is the saturated pressure and $S$ is the saturation rate defined by $S = p/p_s$. The volume of the water droplet is calculated from $v_{\text{lm}} n_{\text{liquid}} = 4\pi r^3/3$. Then Eq. (2) is rewritten as follows
\[ \Delta G = -\frac{4}{3} \pi r^3 \frac{k_B T}{v_{ml}} \ln S + 4\pi r^2 \sigma \]  
\((5)\)

The first term on the right hand side of Eq. (5) becomes always positive when \(S<1\) and be negative occasionally when \(S>1\).

Figure 1 shows a schematic of the relationship between \(r\) and \(\Delta G\). \(\Delta G\) decreases with increasing saturation rate and it has a maximum value which is the required energy for particle formation when \(S>1\). Differentiating Eq. (5) with respect to \(r\) results in the following relationship:

\[ \frac{\delta \Delta G}{\delta r} = -4\pi r^2 \frac{k_B T}{v_{ml}} \ln S + 8\pi r^* \sigma = 0 \]  
\((6)\)

where \(r^*\) is the critical radius of a water droplet at the maximum GFE and obtained from Eq. (6) as shown below

\[ r^* = \frac{2\sigma}{\rho_{l} R T \ln S} \]  
\((7)\)

and the value of \(\Delta G^*\) at \(r^*\) is derived from Eq. (5) as shown below

\[ \Delta G^* = \frac{4}{3} \pi r^* \sigma \]  
\((8)\)

Because the condensation model based on an ideal gas with the derivation of Eq. (4) restricts the application to low-pressure conditions, the original homogeneous nucleation model with Eq. (7) may not be applicable to wet-steam flows under high-pressure conditions. In this paper, we propose a simple form of the critical radius considering a real-gas EOS under high-pressure conditions.

A real-gas EOS based on a virial-type equation can be defined by the simple form:

\[ p = \rho_{\text{real}} RT = \phi \rho_{\text{ideal}} RT \]  
\((9)\)

where \(\phi\) is a compressibility factor; \(\phi = \rho_{\text{ideal}} / \rho_{\text{real}}\). The values of \(\rho_{\text{ideal}}\) and \(\rho_{\text{real}}\) are the densities obtained from the ideal-gas EOS and from the real-gas EOS. Using \(\phi\) with an assumption of pressure independence, Eq. (4) is reconstructed as

\[ g_{ml} - g_{lm} = -\phi k_B T \int_{p_{ml}}^{p} \frac{1}{p} dp + v_{ml} (p - p_{ml}) \]  
\((10)\)

The critical radius \(r^*\) is finally derived in the same manner as in Eq. (7) as follows:

\[ r^* = \frac{2\sigma}{\phi \rho_{l} R T \ln S - \Delta p v_{l}} \]  
\((11)\)

The calculation procedure for \(\phi\) with a real-gas EOS is shown in the next section.
The homogeneous nucleation model employed in this study is given by

\[ I = \frac{q_c}{1 + \theta} \left( \frac{2\sigma}{\pi m^2} \right) \frac{m^2}{\rho} \exp \left( -\frac{\Delta G^*}{k_B T} \right) \]  

(12)

where \( q_c \), \( \theta \), and \( m \) are the condensation coefficient, Kantrowitz's non-isothermal correction term, and molecular mass for water liquid, respectively. In this study, the critical radius \( r^* \) composing \( \Delta G^* \) in Eq. (12) is represented by Eq. (11) instead of Eq. (7).

The mass generation rate \( \Gamma \) in the source term of Eq. (1) is composed of the homogeneous nucleation rate \( I \) in Eq. (12) and the averaged growth rates of the water droplets \( d\vec{r}/dt \). In this study, a simplified equation proposed by Ishizaka et al. (1995) is employed:

\[ \Gamma = \frac{4}{3} \pi \rho_i \left( h_{in}^3 + 3\pi \rho_i nr^* \frac{d\vec{r}}{dt} \right) \]  

(13)

The average growth rate of water droplets \( d\vec{r}/dt \) is calculated using the Gyarmathy model customized for wet-steam flows as follows:

\[ \frac{d\vec{r}}{dt} = \frac{\lambda}{\rho_i h_{in}} \left( 1 - \frac{r^*}{r} \right) (T_i - T) \]  

(14)

where \( h_{in} \) is the heat of formation for the mixed gas. \( \lambda \) represents the modified thermal conductivity coefficient considering the mean free path of water droplets as follows:

\[ \lambda = \frac{\kappa}{r} \left( 1 + 4Kn \right) + 3.78(1 - \nu) \frac{Kn}{Pr} \]  

(15)

and \( \nu \) is the Young modification parameter (Young, 1982) defined by

\[ \nu = \frac{RT}{h_{in}} \left( \alpha - 0.5 - 2\frac{q_c}{2q_c} \left( \gamma + 1 \right) \frac{C_p T_i}{h_{in}} \right) \]  

(16)

In this paper, \( \alpha = 7.0 \) is employed.

### 2.3 Real-gas EOS and thermophysical properties

The governing equations in Eq. (1) are solved with a real-gas EOS. In this study, we employed the EOS programmed in the Reference Fluid Thermodynamic and Transport Properties Database (REFPROP) developed by the National Institute for Standards and Technology (NIST). In accordance with the EOS for wet-steam flows derived by Ishizaka (1995), and using the compressibility factor \( \phi \), the virial-type EOS is further modified to the following equation:

\[ p = (1 - \beta)\phi \rho RT \]  

(17)

Actually, the pressure and temperature in Eq. (17) are obtained from REFPROP based on density \( \rho \) and internal energy \( \varepsilon \). The compressibility factor in the critical radius Eq. (11) is calculated by Eq. (17). However, the direct calculation may require significant amount of computational time. Recently we developed a numerical method for simulating supercritical-fluid flows (Yamamoto, 2011). A thermophysical database was employed in this method, in which the thermophysical properties were calculated once from the database and then stored in several look-up tables as a function of temperature and pressure. Therefore, the present model is also applicable to the numerical method. This method dramatically reduced computational costs.

Figures 2(a) and 2(b) show the pressure and the temperature plotted on a \( \rho-\varepsilon \) diagram for water vapor. When the pressure and temperature are located in the vapor-state region beyond the saturation line, those values are obtained from the \( \rho-\varepsilon \) diagram directly. However, these values in wet-steam flows occasionally decrease far below the saturation line, reaching a supercool condition. In this case, these values cannot be directly obtained from the \( \rho-\varepsilon \) diagrams.
because a vapor condition continues to exist as a supercool condition. These values are then linearly extrapolated from values near the saturation line until condensation starts. Thermophysical properties such as viscosity, thermal conductivity, and isobaric specific heat are also calculated from REFPROP in each iteration. Figures 3(a) and 3(b) show the viscosity and the thermal conductivity plotted on a $p$-$T$ diagram. The thermophysical properties are interpolated from the 801x801 look-up table produced from these values on the $p$-$T$ diagram. If the value is located at the outer area of the look-up table, the thermophysical properties are directly calculated by REFPROP.

3. Numerical Simulation of High-Pressure Wet-Steam Flows

3-1 Numerical results using the conventional model

We first simulated wet-steam flows in a Laval nozzle experimentally studied by Gyarmathy (2005) using an ideal-gas EOS and a convectional condensation model based on Eq. (12) coupled with Eq. (7) to calculate the nucleation rate and the critical radius with the assumption of an ideal gas. Two cases from the experiments, CASE 1 ($P_0 = 10.07\text{MPa}$, superheated condition) and CASE 2 ($P_0 = 10.07\text{MPa}$, $T_0 = 615.35\text{K}$), were considered for flow conditions here. The EOS for an ideal gas derived by Ishizaka (1995) and the conventional homogeneous nucleation model in Eq. (12) based on the critical radius in Eq. (7) were employed. The value of $T_0$ in CASE 1 was set to 773.15K to simulate the superheated condition.

Figure 4 shows the computational mesh with 201 x 81 grid points. Dry steam flowed into the nozzle from the inlet at the left and the Mach number reached 1.0 at the throat. The flow expanded downstream of the throat and accelerated to supersonic speed at the outlet. Under these conditions, nonequilibrium condensation could occur in the nozzle for
CASE 2.

Figures 5(a) and 5(b) show the static pressure and static temperature contours for CASE 1. The pressure and temperature in the direction of the nozzle outlet. Expansion fans are generated downstream of the nozzle throat and reflect at the opposite wall. The pressure at the outlet reaches 0.65 MPa and the difference in temperature between the inlet and the outlet is 325 K.

Figures 6(a) and 6(b) show the corresponding results for CASE 2; Figure 6(c) shows the condensate mass fraction. Condensation starts at the nozzle throat and the mass fraction increases to 0.10 at the outlet. A weak condensation shock induced by the release of the latent heat of condensation is observed in Fig. 6(a). The increase in temperature results in a higher value at the outlet compared to that in Fig. 5(b).

![Computational grid for the 4/B nozzle](image)

Fig. 4 Computational grid for the 4/B nozzle.

![Static pressure and temperature contours](image)

Fig. 5 Numerical results for CASE 1 with the ideal-gas assumption.

![Static pressure and temperature contours](image)

Fig. 6 Numerical results for CASE 2 with the ideal-gas assumption.

![Mass fraction of water droplets contours](image)

Fig. 7 plots the normalized pressure distributions along the center of the nozzle obtained by the present computations for CASE 1 and CASE 2 compared with those of the experimental results. The position x = 0 corresponds to the nozzle throat. Both the results coincide with each other until the location where condensation starts. These results are also in good agreement with the experiments. After condensation begins, the calculated pressure for CASE 2 is slightly higher compared to that for CASE 1 because the release of latent heat causes an increase in temperature and
consequently the pressure increases. However, the pressure for CASE 2 after the start of condensation is still underestimated compared to the experimental results. It indicates that the conventional nonequilibrium condensation model and the ideal-gas EOS cannot evaluate the pressure increase accurately, even though the onset of condensation is well predicted.

![Fig.7 Normalized pressure distribution in the nozzle with the ideal-gas assumption.](image)

### 3-2 Numerical results using the present model

The same problems addressed in the previous section were simulated by a method using the real-gas EOS and the proposed condensation model based on Eq. (12) coupled with Eq. (11) to calculate the nucleation rate and the critical radius. The mass generation rate is the same with the conventional model shown in Eq. (13). The same computational grid was used and the same flow conditions were specified.

Figures 8(a), 8(b), and 8(c) show the contours of static pressure, static temperature, and condensate mass fraction, respectively. The static pressure and temperature contours at the throat are slightly different from those of Figs. 6(a) and 6(b), while the pressure and temperature contours in the other regions are almost the same as those of Figs. 6(a) and 6(b). A typical difference in the results is that the maximum value of the condensate mass fraction is 0.15 at the outlet and this value is relatively higher than that of Fig. 6(c).

Figure 9 shows the normalized pressure distributions obtained using (1) the real-gas EOS with the present condensation model for CASE 1, (2) the real-gas EOS with the present condensation model for CASE 2, and (3) the real-gas EOS with the conventional condensation model for CASE 2. They are compared with each other and with the experiments. As was true for the cases shown in the previous section, three numerical results coincide with each other until the location where condensation starts. The results of both cases using the real-gas EOS for CASE 2 show a pressure increase that is closer to that of the experimental results when compared to those assuming the ideal-gas EOS. The location of the start of condensation obtained by using the real-gas EOS with the present condensation model is also in good agreement with the experiments, while the location obtained by using the real-gas EOS with the conventional condensation model is shifted to an upstream region.

Figure 10 shows the calculated mean radius of the water droplets compared with the values from experiments. The results calculated using the ideal-gas EOS are relatively underestimated compared to the values from the experiments, while both the results using the present condensation model and the conventional model with the real-gas EOS reach values close to those of the experiments.

Figures 11 and 12 show the distributions of temperature and nucleation rate in the nozzle obtained using the ideal-gas EOS with the conventional condensation model and using the real-gas EOS with the present model, respectively. Both cases indicate the same trend of temperature decrease before the start of condensation. The location of the start of the temperature increase due to the release of latent heat for the ideal-gas case shifts upstream as compared with that for the real-gas case. Because the present condensation model calculates a critical radius that is
relatively larger than that of the conventional model as shown in Eq. (7) and Eq. (11), the difference results in the delay of the start of condensation. Explaining this another way, the conventional condensation model underestimates the critical radius. The rate of increase in the temperature after condensation starts is also different. The ideal-gas case calculates the temperature increase to be somewhat smaller than that of the real-gas case.

Fig. 8 Numerical results assuming a real gas with the present condensation model.

Fig. 9 Normalized pressure distribution in the nozzle with the real-gas assumption.

Because the mass fraction of the water droplet indicates the total value of the latent heat, the temperature at the outlet for the ideal gas is lower than that for the real-gas case. The nucleation rate obtained from the ideal-gas case in Fig. 12 is higher than that of the real-gas case. A higher nucleation rate increases the number density of the water droplets. Therefore, the mean radius of the water droplets in the ideal-gas case is underestimated compared to that of the real-gas case as is shown in Fig. 10.
4. Conclusions

A simple equation for the critical radius of a water droplet in condensation was newly derived by assuming a real-gas EOS and compressibility. Using a condensation model with the critical radius equation, wet-steam flows in a nozzle under high-pressure conditions were numerically investigated. After that, the results obtained using an ideal-gas EOS with the conventional condensation model and those obtained using a real-gas EOS with the present model were compared with the experiments. The results obtained are summarized as follows:

1) Numerical simulations with the ideal-gas EOS and the condensation models could not accurately obtain the pressure increase induced by the release of the latent heat due to condensation, which was observed in the experiments by Gyarmathy (2005) for a high-pressure wet-steam nozzle flow.

2) Numerical simulations with the real-gas EOS could capture the pressure increase induced by the release of the latent heat due to the condensation in the high pressure conditions. The location of the onset of nonequilibrium condensation obtained by the proposed nucleation model with the critical radius equation and the assumption of a real-gas was in good agreement with that of the experimental results, which was not achieved by the conventional nucleation model based on the ideal-gas.

3) Because the conventional nucleation model underestimated the critical radius of a water droplet, condensation started earlier than that obtained with the present nucleation model. In addition, the outlet temperature was underestimated by the conventional model due to the difference of the mass fraction of water droplets. These evaluations resulted in a decrease in the mean radius of the water droplets.
To improve the efficiency and the reliability of turbines and other components, the accurate prediction of pressure field and the condensation is crucial. The proposed model is certainly valuable not only for simulating wet-steam flow in a steam turbine, but also for supercritical CO₂ flow with condensation in high pressure conditions, which is expected for future power plants.

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