

Numerical Study on Flow Field Characteristics and Efficiency Losses in Nuclear Steam Turbines Based on the Non-equilibrium Condensation Model

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Abstract

The steam turbine is one of the most critical energy conversion components in power generation systems. Unlike thermal power plants, nuclear steam turbines operate in the wet steam region in most of their stages, with the inlet steam often near saturation, resulting in more severe non-equilibrium condensation. This study aims to investigate the non-equilibrium condensation flow characteristics and efficiency of a nuclear steam turbine. Firstly, a suitable condensation model was selected and validated by comparing it with the existing experimental data on nozzles. Then, a thermodynamic analysis of non-equilibrium condensation in a nine-stage nuclear steam turbine was performed using the Euler-Euler method. Due to the saturated inlet conditions and a maximum supercooling of only 7 K, nucleation was suppressed by pre-existing droplets, and condensation was dominated by droplet growth. Furthermore, the effects of steam extraction ports and shaft seals on condensation were examined. Results indicate that although extraction promotes nucleation, the extracted flow, accounting for approximately 17.4% of the inlet mass flow, removes part of the droplets, thereby reducing overall moisture content. Finally, internal efficiency and various types of losses were analyzed. The turbine efficiency decreased from 91% to around 80%, with wet steam loss being the most significant, reaching about 15% at the last stage. The findings provide insights for improving the design and operation of nuclear steam turbines, enhancing their economic performance and operational reliability.

Full Text

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Abstract

The steam turbine represents one of the most critical energy conversion components in power generation systems. Unlike thermal power plants, nuclear steam turbines operate predominantly in the wet steam region across most of their stages, with inlet steam often near saturation conditions, resulting in severe non-equilibrium condensation effects. This study investigates the non-equilibrium condensation flow characteristics and efficiency performance of a nuclear steam turbine.

First, an appropriate condensation model was selected and validated against existing experimental nozzle data. Subsequently, a thermodynamic analysis of non-equilibrium condensation in a nine-stage nuclear turbine was performed using the Euler-Euler method. Due to saturated inlet conditions and a maximum supercooling of only 7 K, nucleation was suppressed by pre-existing droplets, with condensation dominated by droplet growth. Furthermore, the effects of steam extraction ports and shaft seals on condensation behavior were examined. Results indicate that although extraction promotes nucleation, the extracted flow—accounting for approximately 17.4% of the inlet mass flow—removes a portion of the droplets, thereby reducing overall humidity. Finally, internal efficiency and various loss mechanisms were analyzed. Turbine efficiency decreased from 91% to approximately 80%, with wet steam loss representing the most significant component, reaching about 15% in the final stage. These findings provide valuable insights for improving the design and operation of nuclear steam turbines, enhancing both economic performance and operational reliability.

Keywords: Non-equilibrium condensation; nuclear power turbine; flow characteristics; efficiency calculation; wet steam

Nomenclature

Symbol	Description	Units
a	thermal diffusivity	m^2/s
c	specific heat capacity at constant pressure	$\text{J}/(\text{kg} \cdot \text{K})$
h	static enthalpy	J/kg
$h_{\{fg\}}$	latent heat of vaporization	J/kg
J	nucleation rate	$1/(\text{m}^3 \cdot \text{s})$
k_B	Boltzmann constant	J/K
Kn	Knudsen number	-
L	latent heat of condensation	J/kg
l	mean free path	m
m_1	monomolecular mass	kg
m	droplet mass with critical radius	kg
N	droplet number	$1/\text{m}^3$
p	pressure	Pa
Pr	Prandtl number	-
$\dot{m}$	mass flow rate	kg/s
q_c	condensation coefficient	-
Q	heat flux	W/m^3
R	gas constant	$\text{J}/(\text{kg} \cdot \text{K})$
r	droplet radius	m
r_c	critical radius	m
S	supersaturation	-
S_m	mass sources	$\text{kg}/(\text{m}^3 \cdot \text{s})$
S_E	energy sources	W/m^3
T	temperature	K
u, v, w	velocity components	m/s
v	specific volume	m^3/kg
v_l	molecular volume	m^3
Greek		
α	volume fraction	-
α_{th}^*	thermal accommodation coefficient	-
β	correction factor in Young model	-
γ	heat capacity ratio	-
Δh	specific enthalpy drop	J/kg
	non-isothermal correction factor	-
λ	thermal conductivity	$\text{W}/(\text{m} \cdot \text{K})$
	dynamic viscosity	$\text{Pa} \cdot \text{s}$
	kinematic viscosity	m^2/s
	density	kg/m^3
σ	surface tension	N/m
σ_∞^*	planar surface tension	N/m

Symbol	Description	Units
Superscript		
*	critical	
Subscript		
0	stagnation state	
1	stator outlet	
2	rotor outlet	
<i>g</i>	gas phase	
<i>l</i>	liquid phase	
<i>sat</i>	saturation state	
<i>sc</i>	supercooling state	
<i>tot</i>	total number	

1. Introduction

With global population growth, economic expansion, and rising living standards, the demand for clean energy and sustainable environmental solutions continues to increase [1, 2]. As a clean energy source unaffected by weather or environmental conditions, nuclear power represents an inevitable trend in the future development of the power industry [3]. The steam turbine is the most critical energy conversion device in nuclear power generation units, transforming the thermal energy of the working fluid into mechanical energy and playing a vital role in the power system [4]. When wet steam expands through turbine flow passages, it does not condense immediately upon crossing the saturated vapor line but continues to expand within a humidity range of 0–4%. At the extremum point, droplets begin to form, leading to non-equilibrium condensation.

Based on studies of thermal power plant turbines [5, 6, 7, 8, 9], steam typically approaches saturation temperature only in the low-pressure stages. After expanding through the flow passage, condensation begins, leading to liquid droplet formation. However, in nuclear power turbines, steam at the turbine inlet is often already saturated or even contains a certain degree of moisture. Consequently, most or even all turbine stages operate in the wet steam region, making the condensation phenomenon in nuclear power turbines significantly more severe.

Due to the large physical size of steam turbines, experimental observation is challenging. In the early stages before computers were widely adopted for scientific computing, experimental observation served as the primary research method. Klassen [10] investigated the effect of partial admission on the performance of a single-stage steam turbine and found that efficiency decreases as the admission percentage decreases. White and Young [11] conducted detailed experimental and theoretical studies on non-equilibrium condensing steam flow in transonic

stator cascades, deriving thermodynamic losses due to irreversible condensation directly from experimental measurements for the first time. Bakhtar et al. [12, 13] discussed the application of classical nucleation theory in predicting droplet condensation in nozzles and two-dimensional planar cascades, analyzing flow parameter variations during condensation under changing physical and geometric conditions. Dykas et al. [14, 15] performed experimental studies on non-equilibrium spontaneous condensation in transonic steam flow within nozzles and cascades, measuring pressure distributions in the wet steam region. Cai et al. [16, 17, 18] employed optical probes and light extinction methods to comprehensively investigate flow characteristics of fine and coarse wet steam droplets inside steam turbines. Wang [19] presented a method based on light scattering technique capable of measuring steam wetness fraction, mean water droplet diameter, and full size distribution.

Since accurately measuring key liquid-phase parameters in wet steam flow is challenging, most experimental studies in the literature focus on averaged parameters [11, 12, 20, 21, 22], limiting their applicability in guiding turbine design. Today, computational fluid dynamics (CFD) research on wet steam two-phase condensation flow has matured, making significant contributions to the study of non-equilibrium condensation. Liang et al. [23] investigated condensation characteristics of wet steam in Moses-Stein nozzles and Dykas blade cascades by varying inlet superheat and found that higher superheat effectively suppressed the onset of non-equilibrium condensation. Gerber et al. [24] proposed a novel multiphase CFD model to predict phase transition processes in the low-pressure section of thermal power turbines under non-equilibrium conditions. Hric and Halama [25] implemented a non-iterative steam equation of state in CFD solvers and validated it under selected transonic wet steam flow conditions, successfully simulating actual steam thermodynamic properties. Yu et al. [26, 27, 28] studied the impact of blade roughness on nucleation stage performance and spontaneous nucleation processes using both single-stage and multi-stage models. Yu [29] modified the nucleation model and computed non-equilibrium condensation flow in Moses-Stein nozzles and three-dimensional cascades, demonstrating that the revised model exhibited good consistency with previous experimental results. Zhang et al. [30] developed an improved condensation model to study the relationship between nucleation processes and boundary conditions in non-equilibrium flows and proposed several novel blade cascade dehumidification structures [31, 32, 33].

Based on the above research, the study of wet steam two-phase flow in steam turbines has developed rapidly, with relatively well-established fundamental work. However, the flow field characteristics and operating conditions of nuclear steam turbines differ significantly. Most previous studies have focused on two-dimensional nozzles, planar cascades, and steam turbines used in fossil-fuel power plants, while research specifically targeting nuclear steam turbines remains limited. Additionally, since droplet formation occurs as early as the inlet stages of nuclear power turbines, most nuclear turbines are equipped with steam extraction ports to reduce moisture and maintain efficiency, yet previous

literature has rarely explored this aspect in detail. Lastly, while many studies have calculated efficiency losses in steam turbines, they have not classified losses caused by condensation or quantified their respective contributions.

This study first validates the accuracy of the condensation model by comparing it with existing experimental data from the Gyarmathy nozzle and the IWSEP nozzle. Then, using a nine-stage nuclear power turbine as the research object, the Euler-Euler method is employed to conduct a thermodynamic analysis of the non-equilibrium condensation phenomenon in nuclear power turbines. Finally, the study analyzes efficiency loss in nuclear power turbines and compares parameter differences between models with and without steam extraction ports and shaft seals. The findings of this study are of great significance for reducing steam losses in nuclear power turbines and improving their economic efficiency.

2.1 Governing Equations

From a thermodynamic perspective, during rapid expansion, single-phase steam surpasses the saturation line and enters an unstable state. When the degree of superheat reaches a certain level, these unstable conditions lead to spontaneous and uniform condensation phenomena [34]. The non-equilibrium condensation flow is simulated using the Euler-Euler two-fluid model. Given the high degree of superheat during rapid expansion, this study focuses on handling extremely small droplets, allowing the assumption of a no-slip condition between steam and droplet velocities. Additionally, mass and heat exchange between droplets and surrounding steam are achieved through source term definitions. The governing equations are summarized as follows: the expansion of steam flow leads to droplet formation in a uniform non-equilibrium condensation process, with the governing equations presented below.

The gas and liquid phases each have their respective mass and energy conservation equations, with heat and mass transfer between phases realized through source terms. The mass conservation equations for the gas and liquid phases are represented by:

$$\frac{\partial \rho_g}{\partial t} + \frac{\partial(\rho_g u_i)}{\partial x_i} = -S_m \quad (1)$$

$$\frac{\partial \rho_l}{\partial t} + \frac{\partial(\rho_l u_i)}{\partial x_i} = S_m \quad (2)$$

Here, t denotes time (s); x, y, z represent Cartesian coordinates; u, v, w are velocity components in the x, y, z directions, respectively (m/s). ρ_g and ρ_l refer to the densities of vapor and liquid phases (kg/m³), while α_g and α_l denote the corresponding volume fractions.

J represents the nucleation rate ($1/(\text{m}^3 \cdot \text{s})$), and m is the critical nucleus mass (kg).

The term S_m represents mass exchange caused by droplet growth, which can be calculated from interfacial density and droplet growth rate:

$$S_m = Jm^* + 4\pi\rho_l \int_0^\infty r^2 n(r) \frac{dr}{dt} dr \quad (3)$$

For the gas phase, the energy conservation equation in tensor form can be expressed as:

$$\frac{\partial(\rho_g h_{tot,g})}{\partial t} + \frac{\partial(\rho_g u_i h_{tot,g})}{\partial x_i} = \frac{\partial}{\partial x_i} \left(\lambda_g \frac{\partial T_g}{\partial x_i} \right) + \frac{\partial p}{\partial t} + \frac{\partial(\tau_{ij} u_i)}{\partial x_i} + S_E \quad (4)$$

Here, $h_{tot,g}$ denotes the total specific enthalpy of the gas phase (J/kg); λ_g is the thermal conductivity of the gas phase (W/(m · K)); T_g represents gas temperature (K); p is pressure (Pa); and τ denotes shear stress (Pa). The source term in the energy equation consists of two components: one accounting for energy transfer associated with mass exchange, and the other corresponding to energy transfer caused by temperature difference between gas and liquid phases:

$$S_E = S_m h_{fg} + Q_l A_{gl} \quad (5)$$

Here, $h_{tot,l}$ denotes the total specific enthalpy of the liquid phase (J/kg), and Q_l represents the interfacial heat transfer coefficient per unit area between gas and liquid phases (W/m²), calculated using:

$$Q_l = \frac{\lambda_g}{r} \frac{1}{1 + 3.18Kn} \quad (6)$$

Here, Kn denotes the Knudsen number, representing the ratio of molecular mean free path to characteristic length scale; T_l denotes liquid phase temperature (K).

For the liquid phase, the energy conservation equation is given in a form defining its temperature. For small droplets, a simple algebraic expression based on capillary action was proposed [35]:

$$T_l = T_{sat} - T_{sc} \frac{r^*}{r} \quad (7)$$

Here, T_{sat} denotes saturation temperature under local pressure (K); T_{sc} represents vapor supercooling (K); and R denotes critical droplet radius (m).

Additionally, a control equation governs the number of droplets in the liquid phase:

$$\frac{\partial N}{\partial t} + \frac{\partial(Nu_i)}{\partial x_i} = J \quad (8)$$

2.2 Nucleation and Droplet Growth Models

The nucleation model provides a mathematical framework describing the initial formation of microstructure during phase transition. In condensation, nucleation is the first step where, due to thermodynamic instability, gas molecules aggregate to form small liquid clusters or “nuclei” under supersaturated conditions. The key to nucleation modeling lies in calculating the change in free energy during nucleation, which typically involves cluster size and the corresponding energy barrier. Nucleation models are generally based on Classical Nucleation Theory (CNT) [36], which posits that a cluster can only exist stably and continue growing once its size reaches a certain critical threshold. Based on capillary phenomena, the classical nucleation model [36] describes how gas-phase molecules fluctuate to form a stable liquid droplet cluster that then begins growing into a visible droplet. The free energy change in the model is a function of cluster size, with its maximum representing the energy barrier:

$$\Delta G^* = \frac{16\pi\sigma^3 v_l^2}{3(k_B T_g \ln S)^2} \quad (9)$$

Here, k_B represents the Boltzmann constant ($1.3806488 \times 10^{-23}$ J/K); v_l denotes the average volume of a single molecule in the liquid phase (m^3); σ represents surface tension of a flat surface (N/m); T_g is gas-phase temperature (K); and S is the supersaturation ratio, defined as p/p_{sat} .

Correspondingly, the critical radius of the condensation nucleus is defined as:

$$r^* = \frac{2\sigma v_l}{k_B T_g \ln S} \quad (10)$$

Here, R_g represents the gas constant (J/(kg · K)). Therefore, the nucleation rate in classical nucleation theory can be expressed as:

$$J_{CNT} = \frac{p}{k_B T_g} \sqrt{\frac{2\sigma}{\pi m_1}} \exp\left(-\frac{\Delta G^*}{k_B T_g}\right) \quad (11)$$

Here, m_1 denotes the mass of a single molecule (kg).

Classical nucleation theory has certain limitations, and currently no theory can quantitatively predict the nucleation process for all working fluids across wide condensation temperature ranges. Consequently, scholars have continuously developed corrections based on classical nucleation theory. The most widely used model, proposed by Kantrowitz, considers non-isothermal effects during the nucleation process [37]:

$$J_{CNT,K} = \frac{q_c}{1 + \eta} J_{CNT} \quad (12)$$

Here, q_c denotes the condensation coefficient, which varies widely across studies and is typically taken as 1; p and p_{sat} represent pressure and corresponding saturation pressure at liquid-phase temperature, respectively.

The Kantrowitz factor, η , is defined as:

$$\eta = \frac{2(\gamma - 1)}{\gamma + 1} \frac{L}{R_g T_g} \left(\frac{L}{R_g T_g} - \frac{1}{2} \right) \quad (13)$$

Here, γ represents the specific heat ratio of the gas, and L denotes the latent heat of condensation (J/kg).

After nucleus formation, steam molecules continue condensing onto the surface of critical clusters. This stage is defined as the droplet growth stage, where liquid phase mass increases. Simultaneously, due to significant latent heat release into the gas phase, steam supercooling rapidly decreases, preventing further nucleation. Gyarmathy [38] proposed that the growth rate is primarily related to the rate of latent heat diffusion to surrounding water vapor. Based on energy balance, the following equation is established:

$$\frac{dr}{dt} = \frac{\lambda_g (T_{sat} - T_g)}{\rho_l h_{fg} r} \frac{1 - r^*/r}{1 + 1.5 Pr_g \frac{\gamma - 1}{\gamma} \frac{h_{fg}}{R_g T_g}} \quad (14)$$

Here, Pr_g denotes the Prandtl number of the gas; h_{fg} is the latent heat of vaporization (J/kg); α_{th}^* represents the thermal accommodation coefficient, characterizing the heat capacity of gas molecules rebounding from the droplet surface and typically taken as 1 [46]; and γ is the specific heat ratio of the gas.

Kn is the Knudsen number, defined as:

$$Kn = \frac{l}{2r} \quad (15)$$

Here, l denotes the mean free path of gas molecules (m). Generally, supercooled gas condensation flow belongs to the continuous flow regime when $Kn < 0.01$,

the free molecular flow regime when $Kn > 4.5$, and the transitional regime when Kn is between 0.01 and 4.5.

In 1982, Young et al. [39] further improved the growth model based on Gyarmathy's work [38], employing the Prandtl number and two calibration factors:

$$\frac{dr}{dt} = \frac{\lambda_g(T_{sat} - T_g)}{\rho_l h_{fg} r} \frac{1 - r^*/r}{1 + \beta Kn + \alpha Kn^2} \quad (16)$$

where ψ is the correction factor, defined as:

$$\psi = \frac{c_{pg}(T_{sat} - T_g)}{h_{fg}} \frac{\rho_g}{\rho_l} \quad (17)$$

Here, c_{pg} represents the specific heat of steam at constant pressure ($J/(kg \cdot ^\circ C)$); γ is the specific heat ratio of the gas; ρ_l denotes liquid phase density (kg/m^3); and h_{fg} is the latent heat of vaporization (J/kg). In Young's [39] droplet growth model, parameters β and α were introduced. While many researchers set $\beta = 0$, a few have chosen $\beta = 2$ to achieve improved accuracy. The parameter α characterizes the relationship between condensation and evaporation coefficients q_c and q_v based on Taylor series expansion, and is generally taken as 0 or 9.

From the nucleation rate expression, a 10% variation in surface tension may result in an order of magnitude difference of 10^{10} in the nucleation rate [40]. Therefore, surface tension accuracy is crucial for ensuring reliable nucleation rate calculations. The Benson model is employed to correct surface tension [41]:

$$\sigma = \sigma_\infty \left(1 - \frac{\beta}{r}\right) \quad (18)$$

2.3 State Equation

In the ANSYS CFX software platform, the IAPWS-IF97 formulation [42] is employed to calculate thermophysical properties of wet steam during condensation. This formulation is constructed based on pressure and temperature and can also determine fluid states through inversion methods using other property combinations, such as pressure-enthalpy or entropy-enthalpy pairs. It plays a crucial role in solving the governing equations of wet steam flow.

2.4 Geometry and Mesh of the Steam Turbine

The steam turbine investigated in this study is a nine-stage nuclear steam turbine. Blade geometry was modeled using NX, while the flow passage was constructed using the Geometry module in ANSYS. Computational meshes for each stage were generated using the automatic topology feature in TurboGrid, with local mesh refinement applied near wall boundaries to resolve the boundary layer accurately. The established physical model is shown in Figure 1 [Figure 1: see original paper], and detailed inlet and outlet boundary conditions for the nine-stage turbine are listed in Table 1 .

Figure 1. Physical model of the nine-stage steam turbine.

Table 1. Nine-stage steam turbine boundary conditions.

Parameter	Pressure (MPa)	Temperature (°C)	Dryness
Inlet			
Outlet			
Extraction port 1			
Extraction port 2			
Front shaft seal			
Rear shaft seal			

The rotor and stator blades adopt twisted profiles with significant radial variation. To ensure mesh quality during grid generation, 57 flow passages are distributed along the blade height, with approximately 33% concentrated in the mid-span region to achieve mesh refinement at the hub and tip. An HOH-type mesh topology is employed, where the O-type grid primarily refines the area surrounding blade surfaces. The boundary layer thickness is set to 1×10^{-5} m. A single-stage mesh is illustrated as a representative example in Figure 2 [Figure 2: see original paper].

Figure 2. Mesh structure.

Grid independence was verified using three mesh densities: coarse (992,256 cells), medium (5,694,080 cells), and fine (10,137,600 cells), as shown in Figure 3 [Figure 3: see original paper]. The horizontal axis values from 1 to 18 represent stator and rotor blades separately: odd numbers correspond to the 1st through 9th stage stator blades, and even numbers correspond to the 1st through 9th stage rotor blades, respectively. The same convention applies throughout this paper.

Figure 3. Mesh independence verification for the nine-stage operating condition.

The influence of mesh density on simulation accuracy was assessed by comparing liquid phase mass fraction distributions. Results from the three mesh density

levels exhibited nearly identical curves, indicating mesh independence. Therefore, considering computational cost and accuracy, the medium-density mesh with approximately 5.72 million cells was selected for subsequent simulations.

2.5 Solution Scheme

In this study, the computational grid was generated using ICEM software. Simulations were conducted on the ANSYS CFX platform, with steam properties obtained from the IAPWS-IF97 formulation [42] included in CFX. The turbulence model employed was the first-order upwind SST $k-\omega$ model [43, 44]. Various condensation models were developed and implemented using compiled user-defined expressions in CFX. The convergence criterion was set to 10^{-6} , with boundary conditions specified as total pressure and total temperature at the inlet, static pressure at the outlet, and a no-slip condition at the walls.

3. Model Validation

The core aspects of the physical model, phase transition process, source term expressions, and turbulence-condensation coupling methods in the three-dimensional model remain consistent in the two-dimensional cases, with only geometric dimensions being simplified. By maintaining consistency in key physical processes, the two-dimensional model can effectively reduce computational complexity, facilitating preliminary evaluation and debugging. The nozzle represents a classic model in wet steam condensation studies, with abundant experimental and numerical data available for reference, enabling convenient comparative analysis of key physical quantities such as condensation initiation location and liquid phase mass fraction. Therefore, despite dimensional differences, two-dimensional cases remain representative and effective for validating the physical plausibility and numerical accuracy of the three-dimensional mathematical model.

Due to the absence of nucleation models suitable for wide temperature ranges and full-scale droplet growth models, coupled with the complexity of turbine models and long computation times, this section employs high- and low-pressure nozzles with shapes similar to turbine flow passage structures to validate the model. This validation aids in selecting the mathematical model, particularly the condensation model, and verifying its accuracy. The results provide a reference for subsequent simulation and calculation of the nine-stage multi-stage turbine.

3.1 Validation of the Condensation Model in the Gyarmathy Nozzle

Gyarmathy's high-pressure nozzle [45] is used to evaluate non-equilibrium condensation behavior under high-pressure conditions in turbine CFD simulations.

Gyarmathy's [38] model provides an opportunity to study nucleation and droplet growth rate behavior in condensation flows under high-pressure conditions. The high-pressure nozzle is designed as a semi-Laval nozzle with a throat height of 10.00 mm. Key dimensions include a contraction section length of 30 mm and an expansion section length of 100 mm. The inlet and outlet heights of the supersonic nozzle are 19.99 mm and 17.91 mm, respectively. The computational grid and corresponding boundary conditions are shown in Figure 4 [Figure 4: see original paper] and Table 2 .

Figure 4. Gyarmathy [45] nozzle mesh.

Table 2. Introduction to high-pressure conditions.

Parameter	Value
Inlet Pressure (MPa)	
Inlet Temperature (K)	

This study employed a structured numerical grid with refined mesh scheme applied to the boundary layer. Grid independence was evaluated using 6,786, 10,400, and 41,292 cells. Based on the dependency study, the 10,400-cell grid was selected as the optimal computational grid for subsequent numerical studies to optimize computational resource utilization, as shown in Figure 5 [Figure 5: see original paper]. The axis p/p_0 represents the ratio of static pressure in the nozzle to total pressure at the inlet, and the radius represents the generated droplet radius.

Figure 5. Mesh independence test of the nozzle.

Six different combinations of condensation models were evaluated against experimental data from three operating conditions of the Gyarmathy [45] nozzle to compare the effectiveness of various mathematical models. Results are presented in Figure 6 [Figure 6: see original paper] through Figure 8 [Figure 8: see original paper]. In the figures, CNT represents the classical nucleation model [36], while CNTK denotes the non-isothermal correction nucleation model [37]. Gyarmathy, Young0, and Young9 correspond to the Gyarmathy [38] droplet growth model and Young's [39] droplet growth models with coefficients of 0 and 9, respectively.

Figure 6. Results of different mathematical models for case 1. (a) Pressure distribution (b) Radius distribution

Figure 7. Results of different mathematical models for case 2. (a) Pressure distribution (b) Radius distribution

Figure 8. Results of different mathematical models for case 3. (a) Pressure distribution (b) Radius distribution

Figures 6 through 8 reveal two main characteristics. First, regarding static pressure, computed results show good agreement with experimental data. Except for

case 1, the overall pressure ratio curve shifts downward. Experimental data show a throat pressure ratio of 0.62, and most mathematical models align with this value. In each simulated operating condition, the Wilson point location within the nozzle shifts slightly upstream compared to experimentally determined positions, especially noticeable in case 1. Second, concerning average droplet size, under conditions with low inlet parameters, computed droplet sizes are closer to experimental values, indicating higher simulation accuracy for low inlet parameters. Since the classical nucleation model lacks correction parameters, its computed results are larger, making nucleation easier, with the nucleation point occurring earlier than in the non-isothermal correction nucleation model. This results in larger discrepancies from experimental data, although the trend of the classical nucleation model curve is more similar to the droplet size curve. Comparing different droplet growth models, the Gyarmathy [38] droplet growth model yields a numerically lower curve, with better matching to experimental droplet radius compared to other models. Overall, simulation results using the classical nucleation model [36] and the Gyarmathy [38] droplet growth model are closer to experimental data.

3.2 Validation of the Condensation Model in the IWSEP Nozzle

A low-pressure nozzle was used to select and validate models within the low-pressure range. The International Wet Steam Experiment Project (IWSEP) nozzle study [46] on steam condensation flow provides both experimental and numerical simulation data, serving as a valuable reference for numerical simulations of low-pressure stages in steam turbines. The nozzle geometry is shown in Figure 9 [Figure 9: see original paper], where $x = 0$ mm represents the nozzle throat. Operating conditions for each case are listed in Table 3 .

Figure 9. IWSEP nozzle [46] structure.

Table 3. Introduction to low-pressure conditions.

Parameter	Value
Inlet Pressure (kPa)	
Inlet Temperature (K)	

The IWSEP nozzle mesh and mesh independence verification are shown in Figure 10 [Figure 10: see original paper]. The nozzle wall mesh was refined to improve accuracy. Numerical simulations of different liquid phase mass fractions were conducted using three mesh resolutions: high (14,602), medium (5,600), and low (2,337) elements.

Figure 10. Mesh and mesh independence test of the IWSEP nozzle. (a) Mesh of the low-pressure nozzle (b) Mesh independence test

The liquid phase mass fraction curves remain nearly identical across different

mesh resolutions. The medium-resolution mesh with 5,600 elements was selected for subsequent simulations to optimize computational resources.

Using data from case 1, 12 different condensation models were compared by analyzing static pressure distribution along the nozzle bottom wall, liquid phase mass fraction, and droplet diameter along the nozzle centerline. Benson's surface tension correction model was applied to the low-pressure nozzle. Simulation results are presented in Figure 11 [Figure 11: see original paper] and Figure 12 [Figure 12: see original paper], where curve labels are consistent with those in Section 3.1.

Figure 11. Calculation results of different models for Case 1 without surface tension correction. (a) Pressure distribution (b) Mass fraction distribution (c) Droplet diameter distribution

Figure 12. Calculation results of different models for Case 1 after Benson surface tension correction. (a) Pressure distribution (b) Mass fraction distribution (c) Droplet diameter distribution

The nucleation points shifted downstream for different models, and pressure curves moved upward. The numerically simulated liquid phase mass fraction was lower than that in the original literature, with liquid phase formation onset occurring downstream of the nozzle throat. Among different condensation models, Benson's [41] surface tension correction model best agreed with the original literature. Similar to the high-pressure nozzle, the Gyarmathy [38] droplet growth model also exhibited good fitting trend for the low-pressure nozzle. Between the two nucleation models, each had strengths and weaknesses across different parameter distributions. Based on original experimental results, the non-isothermal correction nucleation model [37] was chosen as the preferred option.

4.1 Analysis of Non-Equilibrium Condensation Flow in Nuclear Steam Turbines

This section focuses on analyzing thermodynamic characteristics of primary droplets during wet steam condensation inside the nuclear steam turbine. As shown in Figure 13 [Figure 13: see original paper], inlet steam is in a saturated state and carries an initial moisture content of 0.76%. Therefore, no distinct jump in moisture content is observed in contour plots; instead, it gradually increases along the flow direction. Under high-pressure conditions, the degree of supercooling is relatively low, suppressing the nucleation rate and leading to a more stable condensation process. This implies that even when employing a non-equilibrium model, supercooling does not accumulate significantly, resulting in phase change characteristics relatively close to those predicted by equilibrium models.

Figure 13. Wetness distribution at 50% span.

As illustrated in Figure 14 [Figure 14: see original paper], droplet number decreases inversely with increasing droplet diameter, indicating droplet coalescence during growth. The liquid mass fraction is determined by both droplet size and droplet number. As droplets grow larger, the likelihood of collision and coalescence increases, causing small droplets to merge into larger ones from inlet to outlet. While droplet number decreases, continuous condensation of wet steam causes the liquid mass fraction to steadily rise along the flow path.

Figure 14. Droplet evolution along the streamline.

The distributions of temperature, supercooling, and nucleation rate along the 50% spanwise section under non-equilibrium conditions were analyzed and are shown in Figure 15 [Figure 15: see original paper]. Since inlet steam is saturated, condensation does not occur immediately under non-equilibrium conditions. Instead, steam continues to expand, resulting in a certain degree of supercooling that causes steam temperature to fall slightly below saturation temperature. At blade junctions, expansion and compression effects are more pronounced, leading to regular fluctuations in local temperature.

Figure 15. Temperature and nucleation rate distribution along the streamline.

As shown in Figure 16 [Figure 16: see original paper], low levels of nucleation rate appear near the pressure side of blade throats in the first and second stages, though magnitudes remain relatively small. This may be attributed to initial moisture content at the inlet, indicating a near-saturated state. The presence of pre-existing droplets and limited supercooling suppresses nucleation activity, resulting in nucleation rates too low to trigger significant nucleation events.

Figure 16. Nucleation distribution. (a) The throat region of the first stage (b) The throat region of the second stage

The distribution of static pressure along axial chord at different blade heights for the first and second stage rotor and stator blades, where nucleation rates are relatively high, is analyzed and shown in Figure 17 [Figure 17: see original paper] and Figure 18 [Figure 18: see original paper]. Overall, static pressure first decreases along the axial direction, then slightly recovers near the blade trailing edge, and finally drops sharply at the outlet. This trend reflects steam acceleration, expansion, and pressure recovery within blade passages. Comparing different blade heights, pressure at 80% span is generally higher, possibly due to vortex or secondary flow effects near the blade tip. The 50% span shows intermediate pressure levels, while the lowest pressure is observed at 20% span, particularly near the rotor blade inlet. This is likely due to stronger boundary layer effects near the blade root, resulting in locally lower pressure. Compared to stator blades, pressure variation across different spans is more pronounced in rotor blades, which may be attributed to more intense flow dynamics. As steam enters blade passages, flow acceleration leads to static pressure decrease. In the mid-blade region, local expansion or flow separation may occur, causing temporary pressure increase. Near the blade trailing edge, strong expansion leads to sharp static pressure drop.

Figure 17. Pressure distribution along blade surfaces at different relative heights in the first stage. (a) Stator (b) Rotor

Figure 18. Pressure distribution along blade surfaces at different relative heights in the second stage. (a) Stator (b) Rotor

4.2 Influence of Extraction Ports and Shaft Seals on Non-Equilibrium Condensation

This section compares flow field parameters related to condensation in the steam turbine before and after removing extraction ports and shaft seals, investigating the influence of extraction and sealing structures on non-equilibrium condensation within the turbine. The modified model without extraction ports and shaft seals is shown in Figure 19 [Figure 19: see original paper].

Figure 19. Steam turbine model without extraction ports and shaft seal.

First, detailed flow characteristics before and after removing extraction ports and shaft seals are presented. As shown in Table 4, compared to the model without these features, their presence leads to approximately 2.9% increase in inlet mass flow. However, the extracted flow accounts for about 17.4% of inlet flow, resulting in 12.3% reduction in outlet flow.

Table 4. Mass flow rate comparison before and after removing extraction ports and shaft seal.

Parameter	With extraction ports and seal	Without extraction ports and seal
Inlet mass flow rate (kg/s)		
Outlet mass flow rate (kg/s)		
Shaft seal (kg/s)		
Extraction port 1 (kg/s)		
Extraction port 2 (kg/s)		

Parameter	With extraction ports and seal	Without extraction ports and seal
Total outlet mass flow (kg/s)		

As shown in Figure 20 [Figure 20: see original paper], removing extraction ports and shaft seals results in higher total mass flow rate throughout the steam path. This slows steam expansion, delays condensation, and leads to slightly higher temperature and pressure levels in the flow field. When extraction ports and shaft seals are present, partial steam extraction intensifies expansion in the flow path and triggers earlier non-equilibrium condensation. This shifts the condensation region upstream, promotes droplet formation, and causes reductions in both temperature and pressure. Additionally, some formed droplets are removed along with extracted steam, leading to lower liquid mass fraction compared to the model without extraction, which contributes to improved turbine efficiency and safety. The nucleation rate comparison further supports this: although nucleation locations remain nearly unchanged, the nucleation rate in the model without extraction is approximately five orders of magnitude lower. This indicates that pressure drop caused by extraction ports may locally increase supersaturation, thereby promoting nucleation, which is consistent with the analysis above.

Figure 20. Flow field parameters distribution. (a) Pressure/Mass fraction/Temperature (b) Nucleation rate

Due to presence of extraction ports in the 4th and 6th stages, liquid mass fraction distribution along blade height was investigated for both stages, as shown in Figure 21 [Figure 21: see original paper] and Figure 22 [Figure 22: see original paper]. In the figures, 20%, 50%, and 80% represent cross-sections at 20%, 50%, and 80% blade height, respectively.

Figure 21. Liquid mass fraction distribution at different blade heights of the 4th stage.

Figure 22. Liquid mass fraction distribution at different blade heights of the 6th stage.

The distribution of liquid mass fraction along blade height indicates that at 20%, 50%, and 80% blade height sections, liquid mass fraction under extraction conditions is significantly lower than without extraction. The presence of extraction ports reduces liquid mass fraction. The minimum liquid mass fraction is observed at 80% blade height, and humidity at the trailing edge decreases with increasing blade height. This is attributed to extraction ports being located near the blade tip, where they have the most pronounced effect on wet steam.

A further comparison of Mach number distributions at the 4th and 6th stages is shown in Figure 23 [Figure 23: see original paper]. Cases with steam extraction ports exhibit higher Mach numbers than those without. Inclusion of extraction ports alters mass flow rate distribution along the flow path, leading to more pronounced expansion near the extraction region. Steam extraction causes local reduction in pressure and density, accompanied by temperature drop, which lowers the speed of sound and significantly increases Mach number. From stator outlet to rotor inlet, Mach number gradually decreases from hub to tip, while at rotor outlet, it increases. A sharp drop in Mach number is observed in both cases at hub and tip regions. However, at the rotor outlet tip, presence of the extraction port moderates the extent of this drop.

Figure 23. Mach number distribution at different blade heights of the 4th and 6th stages. (a) 4th stage (b) 6th stage

4.3 Efficiency Analysis and Loss Mechanisms

Efficiency is a key indicator of turbine performance. This section investigates stage internal efficiency of the nine-stage nuclear steam turbine and the proportion of various loss types. The thermodynamic process line [47] of the turbine stage is shown in Figure 24 [Figure 24: see original paper].

Figure 24. Stage thermodynamic process line [47].

Regarding Figure 24 and considering practical conditions of this study, the stage internal efficiency equation is simplified as:

$$\eta_{stage} = \frac{\Delta h_{actual}}{\Delta h_{isentropic}} = \frac{h_0 - h_2}{h_0 - h_{2s}} \quad (19)$$

In the equation, h_0 represents total enthalpy at the inlet, h_2 denotes static enthalpy at the outlet. The terms f_{hd} , h_{dd} , x_{hd} , and $2c_{hd}$ represent friction loss, leakage loss, wet steam loss, and residual kinetic energy loss, respectively.

Figure 25 [Figure 25: see original paper] presents calculated internal efficiency for each stage, showing that stage efficiency decreases progressively with increasing stage number. This trend aligns with typical turbine operating characteristics: as steam expands through successive stages, both kinetic and thermal energy diminish while flow losses increase. Moreover, as stage number increases, droplets continue to grow and coalesce, eventually causing erosion and impingement on blades, further reducing efficiency. As shown in Figure 1, the flow passage at the last stage exhaust approximates a diffuser nozzle. Due to this special structure, velocity decreases with increasing cross-sectional area, leading to corresponding reduction in residual kinetic energy loss. Consequently,

a slight efficiency increase is observed in the last stage, with overall efficiency reaching approximately 80% at the outlet.

Figure 25. Internal efficiency of each stage.

To investigate the proportion of internal stage losses, relative shares and distribution of various losses are presented in Figure 26 [Figure 26: see original paper]. Due to structures such as steam seals in the turbine, friction and leakage losses account for a relatively small portion of total enthalpy. In contrast, residual velocity loss and wet steam loss contribute more significantly. Among these, the proportion of residual velocity loss generally increases with stage number. As wet steam expands through flow passages, its velocity continues to rise, leading to corresponding increases in residual velocity losses, which are velocity-dependent. Meanwhile, as stage number increases, droplet number grows, and wet steam losses also rise. Notably, wet steam losses are more significant in low-pressure stages than in high-pressure ones. As wet steam losses increase and represent a larger share of total losses, overall stage losses grow progressively, resulting in gradual efficiency decrease.

Figure 26. Distribution of various losses. (a) Stage-wise distribution curves (b) Relative proportion

5. Conclusion

This study focuses on non-equilibrium condensation of wet steam in steam turbines, specifically targeting a large-scale nine-stage nuclear steam turbine. By comparing nucleation and droplet growth models in both high- and low-pressure nozzle configurations, a non-equilibrium spontaneous condensation flow model applicable to varying steam humidity conditions was developed. Thermodynamic analysis of primary droplets, investigation into the impact of extraction ports on condensation, and evaluation of stage internal efficiency and loss mechanisms were conducted, revealing development characteristics of wet steam within the turbine. The main conclusions are summarized as follows:

1. The condensation model's applicability for nuclear steam turbines was evaluated through numerical comparisons with existing experimental data from the Gyarmathy high-pressure nozzle and the IWESP low-pressure nozzle. The condensation model was selected and validated based on its accuracy and consistency with experimental results.
2. The non-equilibrium condensation process in a nuclear steam turbine was investigated. Due to presence of liquid droplets at the inlet boundary of the nine-stage turbine, wet steam is initially in a saturated state. The maximum degree of supercooling under non-equilibrium conditions is only 7 K, which is relatively low. Pre-existing droplets serve as condensation nuclei, suppressing the nucleation process and resulting in condensation occurring primarily through droplet growth.

3. The influence of steam extraction ports and shaft seals on wet steam condensation within the turbine was investigated. Extracted flow through steam extraction ports and shaft seals accounts for approximately 17.4% of inlet mass flow rate, leading to 12.3% reduction in outlet flow. Although extraction ports promote nucleation due to localized pressure drops, extracted steam also removes a portion of liquid droplets, thereby reducing overall steam humidity. This reduction in moisture content contributes to improved economic efficiency and operational safety of the steam turbine.
4. Stage efficiency and various loss types were analyzed. After undergoing nine stages of expansion, thermodynamic losses accumulate progressively, decreasing turbine efficiency from 91% to approximately 80%. Among different loss components, wet steam loss accounts for the largest proportion. Both wet steam loss and residual velocity loss increase with stage number, reaching about 15% in the final stage.

This study can serve as a reference for design and operation of large-scale nuclear steam turbine stages, as well as for understanding thermodynamic processes within turbine stages, thereby contributing to improved safety and economic performance of these turbines.

Credit Author Statement

Xiaoqin Du: Writing - review & editing, Resources, Investigation, Project administration.

Zhuojun Jiang: Writing - original draft, Visualization, Validation, Methodology, Conceptualization, Investigation, Formal analysis, Software, Data Curation.

Wan Sun: Writing - review & editing, Methodology, Conceptualization, Supervision, Project administration, Funding acquisition.

Zhuhai Zhong: Writing - review & editing, Resources.

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Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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