Numerical Study on Moisture Removal Methods for Steam Turbine Stator Blades: Postprint

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Abstract

This paper addresses the wet steam formation phenomenon in the last stage of steam turbines, analyzing the governing equations and physical models for non-equilibrium condensation flow of wet steam. Computational analysis was performed on the Moses and Stein nozzle to characterize the wet steam formation process, and comparison with experimental results demonstrates the validity of the numerical calculation model and method. Finally, numerical simulations were conducted on a steam turbine stator blade to analyze the distribution of wetness and subcooling, a novel moisture removal structure for stator blades was proposed, and computational analysis of the moisture removal effectiveness was performed. The results indicate that as the channel diameter increases, the average wetness at the stator blade exit decreases, and moisture losses are reduced.

Full Text

Preamble

Numerical Study on Dehumidification Methods for Steam Turbine Stator Blades

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Abstract: This paper investigates wet steam formation phenomena in the final stage of steam turbines by analyzing the governing equations and physical models for non-equilibrium condensation flow in wet steam. Numerical calculations were performed on the Moses and Stein nozzle, revealing characteristics of wet steam formation that show good agreement with experimental results, thereby validating the numerical model and methodology. Finally, numerical simulations were conducted on a steam turbine stator blade to analyze the distribution of humidity and supercooling, and a novel stator dehumidification structure was proposed with its effectiveness evaluated through computational analysis. The results demonstrate that as the channel diameter increases, the average humidity at the stator outlet decreases and wetness losses are reduced.

Keywords: Stator dehumidification; Non-equilibrium; Droplet formation; Two-phase flow

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0 Introduction

Steam turbines are critical energy conversion devices that play a significant role in national economies. However, in the final stages of steam turbines, steam expansion work causes the steam temperature to drop below the saturation temperature, resulting in non-equilibrium condensation and the formation of numerous droplets. These droplets are detrimental to stage performance, not only reducing turbine stage efficiency but also causing severe erosion of rotor blades as they continue to grow.

The generation of droplets carried by wet steam in turbine flow passages and their subsequent flow with the steam constitute a highly complex phase change and two-phase flow problem. The difficulty in studying wet steam lies in the fact that humidity is not uniformly distributed along either the circumferential or radial directions of the blades, and heat and mass transfer occur continuously between droplets and steam, involving intricate phase change phenomena. Stodola and Wilson pioneered investigations of this phenomenon in 1920 using supersonic Laval nozzles, followed by similar studies from Yellott et al. in 1930. Bakhtar and Zidi experimentally examined nucleation phenomena in two-dimensional nozzles with different expansion rates. With advances in computer technology and maturation of two-phase flow theory, numerical simulation has become a viable method for investigating wet steam flow in steam turbines. By establishing mathematical models for wet steam condensation flow, researchers can accurately determine wet steam parameters such as droplet radius, droplet number, and humidity.

Current dehumidification methods for steam turbines are primarily classified as internal or external. External dehumidification improves steam parameters through heat regeneration to achieve moisture removal, while internal dehumidification is divided into three categories: stator blade dehumidification, rotor blade dehumidification, and diaphragm dehumidification between stator and rotor blades. This study focuses on stator blade dehumidification. Existing stator dehumidification methods primarily involve manufacturing hollow blades and using cavities with purging, suction, or wall heating to remove water films deposited on vane surfaces or large droplets concentrated at trailing edges, thereby increasing steam dryness. However, these methods feature complex structures and fail to fundamentally eliminate wet steam formation.

A properly designed stator dehumidification structure can remove most water droplets from wet steam, thereby improving turbine efficiency and safety. This paper first analyzes wet steam formation within steam turbines, then employs a two-phase flow model for numerical calculations on the Moses and Stein nozzle, comparing results with experimental data to ensure computational reliability. Finally, a novel dehumidification structure is proposed and evaluated.

1 Wet Steam Phase Transition Process and Mathematical Model

The transformation from dry to wet steam in turbine stator passages involves three primary stages. In the first stage, saturated steam crossing the saturation line undergoes condensation, generating a small number of extremely fine droplets. In the second stage, as the steam continues to expand, supercooling increases, causing the fine droplets from the first stage to suddenly grow larger. However, due to their limited number, these droplets absorb heat from the steam, reducing their radius and further increasing supercooling. In the third stage, continued steam expansion gradually reduces supercooling, causing the nucleation rate to decrease and eventually stabilize, while the droplet number approaches a constant value and droplet radius increases slightly (0.1-1 m). Throughout this process, a non-equilibrium two-phase flow model can be used for calculations.

The motion of small droplets formed in wet steam within turbines follows the steam flow. For this study, we assume no relative motion between droplets and steam, meaning they share a common velocity. Therefore, the governing equations include both continuous-phase steam and discrete-phase droplets.

The control equations can be expressed as follows. The continuous-phase mass equation is:

$$\frac{\partial \rho_c r_c}{\partial t} + \frac{\partial (\rho_c u_i r_c)}{\partial x_i} = -\sum (S_d + m^* r_c J_d)$$

where the right-hand side represents the source term, indicating the total mass of all droplets formed from steam.

The discrete-phase droplet mass equation is expressed as:

$$\frac{\partial \rho_d r_d}{\partial t} + \frac{\partial (\rho_d u_i r_d)}{\partial x_i} = S_d + m^* r_c J_d$$

Each droplet corresponds to a mass equation:

$$\frac{\partial \rho_d N_d}{\partial t} + \frac{\partial (\rho_d u_i N_d)}{\partial x_i} = \rho_d r_c J_d$$

In the above equations, J_d represents the nucleation rate (the number of droplets formed per unit volume of steam per unit time), r_c is the steam volume fraction, which relates to the individual droplet volume fraction r_d as:

$$r_c + \sum r_d = 1$$

m^* is the mass generation rate in the non-equilibrium condensation process. In classical nucleation theory, m^* is determined by both nucleation and droplet growth (or evaporation). The sum of their mass increase constitutes the mass generation rate, which can be written as:

$$m^* = \frac{4}{3}\pi \rho_l J_d r^{*3} + 4\pi \rho_l \eta r^2 \frac{\partial r}{\partial t}$$

where r is the average droplet radius, ρ_l is the droplet density, r^* is the critical droplet radius, and η is the number of droplets per unit volume.

Considering non-isothermal effects, we adopt the nucleation rate expression J_d modified by Kantrowitz from classical nucleation theory:

$$J_d = \frac{q_c}{1+\theta}\sqrt{\frac{2\sigma}{\pi M_m}}\frac{\rho_c^2}{\rho_l}\exp\left(-\frac{16\pi\sigma^3}{3kT\rho_l^2(RT\ln S)^2}\right)$$

where q_c is the condensation coefficient (typically 1), k is the Boltzmann constant, M_m is the mass of a single water molecule, σ is the droplet surface tension, ρ_l and ρ_c are the densities of droplets and steam at temperature T, respectively, and θ is the non-isothermal correction coefficient:

$$\theta = \frac{2(\gamma-1)}{\gamma+1}\frac{h_{cl}}{RT}\left(\frac{h_{cl}}{RT} - 0.5\right)$$

where h_{cl} is the latent heat of phase change at pressure p, and γ is the specific heat ratio.

After formation of condensation nuclei, heat and mass transfer continuously occur between the small droplets and surrounding steam. Droplets grow when their radius exceeds the critical radius and evaporate when smaller. The critical radius expression is:

$$r^* = \frac{2\sigma}{\rho_l R T_l \ln S}$$

where S is supersaturation, defined as the ratio of actual vapor pressure to saturation pressure at equilibrium:

$$S = \frac{P}{P_{sat}(T)}$$

where P is steam pressure and P_{sat} is the saturation pressure of water vapor at that temperature.

Supercooling characterizes the degree to which gas temperature falls below saturation temperature:

$$\Delta T = T_{sat} - T$$

Droplet growth involves two mechanisms: mass transfer from steam condensation and heat transfer between droplets and steam in the form of latent heat. This energy transfer relationship, first proposed by J.B. Young, can be written as:

$$\frac{dr}{dt} = \frac{\lambda}{\rho_l h_{cl} r}(T_d - T) + \frac{\gamma + 1}{2\gamma} \frac{C_p(T_d - T)}{h_{cl}\rho_l\sqrt{2\pi RT}}$$

where T_d is the droplet temperature.

Given our earlier assumption of common velocity for all phases, we only need to solve the momentum equation for the continuous phase (steam), with interphase momentum changes represented by the source term S_m:

$$\frac{\partial(r\rho u_j u_i)_c}{\partial x_j} = -r_c \frac{\partial p}{\partial x_i} + \frac{\partial(r\tau_{ji})_c}{\partial x_j} - \sum S_{m,d}$$

For the continuous-phase energy equation, we present it in terms of total enthalpy:

$$\frac{\partial(r\rho h)_c}{\partial t} + \frac{\partial(r\rho u_j h)_c}{\partial x_j} = \frac{\partial}{\partial x_j}\left(r_c \lambda \frac{\partial T}{\partial x_j}\right) + \frac{\partial(r\tau_{ji} u_i)_c}{\partial x_j} + \sum S_{h,d}$$

where S_h includes the effects of mass and heat transfer. According to Gyarmathy' s research, when droplet diameters are smaller than 1 m, the droplet temperature can be calculated using:

$$T_d = T_{sat}(p) - \Delta T \left(\frac{r^*}{r}\right)$$

2 Model Validation and Analysis

Steam turbine stator blades represent a special type of nozzle. Due to measurement difficulties, their internal flow characteristics can be studied through nozzle analysis. This paper validates the correctness of the two-phase flow model by studying the Moses and Stein nozzle, providing a foundation for subsequent dehumidification calculations.

2.1 Moses and Stein Nozzle Calculation and Results Analysis

The nozzle geometry is shown in [Figure 1: see original paper]. The converging-diverging nozzle wall consists of two circular arcs: a subsonic region with radius R=0.053 m and a supersonic region with radius R=0.684 m, smoothly connected. The expansion rate of this nozzle is 8230 $\sec^{-1}$, determined by the formula $e_x = -(u/p)(dp/dx)$. The throat is located at x=0.0822 m with a square cross-section of 10$×$10 mm.

Moses conducted multiple experimental runs; this paper selected seven cases for numerical calculation, with boundary conditions and measured data presented in . Grid number significantly affects numerical result reliability. Grid independence was verified by comparing calculations with different grid numbers against experimental values, with the final mesh shown in [Figure 2: see original paper].

This section first presents numerical calculations for Experiment No. 193 compared with experimental measurements in [Figure 3: see original paper]. The results show excellent agreement, demonstrating the correctness of the computational model and method.

To explain droplet formation in steam, parameters including droplet mass fraction, supercooling, droplet number, diameter, and nucleation rate along the nozzle centerline were extracted, as shown in [Figure 4: see original paper].

The figure can be divided into three regions: Region I before the nozzle throat, Region II from the throat to the Wilson point location, and Region III thereafter. To better analyze [Figure 4: see original paper], we first explain the Wilson point, discovered and proposed by Wilson, which represents the point of maximum supercooling. Based on the previous equations, maximum supercooling corresponds to maximum nucleation rate.

In Region I, steam expands continuously before reaching the nozzle throat. However, due to high steam temperature and low supercooling, the nucleation rate is low, producing only a small number of droplets and minimal steam humidity. [Figure 4: see original paper] shows that nucleation does not occur at low supercooling; the nucleation rate only increases significantly when supercooling reaches approximately 18 K. Steam carrying a small number of droplets passes through the throat into Region II, where expansion continues. Reduced expansion rate causes minimal change in saturation temperature, while droplet radius decreases due to heat absorption, lowering steam temperature and increasing supercooling to its maximum at the Wilson point. Both nucleation rate and droplet number increase with supercooling, but steam humidity remains nearly constant due to decreasing droplet diameter. In Region III, increased steam expansion rate causes saturation temperature to decrease more rapidly, while the numerous droplets from Region II grow larger, releasing latent heat that raises steam temperature, reduces supercooling, and decreases nucleation rate until equilibrium is reached.

To further validate the mathematical model and computational method, multiple experimental cases were calculated, with Wilson point locations and corresponding supercooling values compared in [Figure 5: see original paper]. The predicted Wilson point positions show excellent agreement with experimental values, with a maximum error of approximately 2.5%. For supercooling at the Wilson point, calculated values differ from experimental values by less than 4 K, which is acceptable.

Through numerical calculations on the Moses and Stein nozzle and comparison with experimental results, this study analyzed droplet formation mechanisms and influencing factors, validated the mathematical model and computational method, and provided theoretical support for subsequent dehumidification analysis.

3.1 Model Introduction

This section examines the final stage stator blades of a particular steam turbine. With 60 blades total, only one flow passage was calculated. The hub and casing diameters are 762 mm and 910 mm, respectively, with the computational model shown in [Figure 6: see original paper]. Boundary conditions employed total pressure and total temperature at the inlet (26.5 kPa, 340.15 K) and static pressure at the outlet (6.62 kPa). Grid independence was verified prior to calculation.

3.2 Computational Results Analysis

Two metrics are generally used to evaluate humidity distribution in flow passages: supercooling (which causes droplet formation) and humidity within the passage (the result of droplet formation). Based on these characteristics, this study investigated dehumidification methods. [Figure 7: see original paper] and [Figure 8: see original paper] present the supercooling and humidity distributions at 50% span for the original blade.

Comprehensive analysis of [Figure 7: see original paper] and [Figure 8: see original paper] reveals that the location of maximum supercooling does not coincide with maximum humidity. Supercooling peaks near the throat region downstream of the blade passage, while humidity begins to increase in the region after the throat, consistent with the characteristics described in [Figure 4: see original paper].

3.3 Dehumidification Structure and Computational Results Analysis

Based on the analysis in Section 3.2, two dehumidification approaches are possible: (1) reducing supercooling to decrease nucleation rate, thereby achieving dehumidification, and (2) more directly converting generated moisture back to steam by adding heat. This study focuses on the first approach.

A novel dehumidification structure is proposed, as shown in [Figure 9: see original paper]. High-temperature steam from the blade leading edge is injected into regions with high nucleation rates to achieve dehumidification. The red line represents the blade camber line, with the black line tangent to the camber line at the front and terminating in the region of maximum supercooling. d represents the channel diameter. This structure' s advantage lies in achieving fundamental dehumidification without requiring additional steam.

[Figure 10: see original paper] clearly shows that as the through-flow diameter increases, wetness loss decreases and humidity is reduced. At a 3 mm diameter, humidity decreases by nearly 0.01%. While this may appear modest, increasing the number of flow channels would yield further substantial humidity reduction.

4 Conclusions

This study analyzed the governing equations for wet steam formation and flow in steam turbine final stages, performed numerical calculations on a nozzle with available experimental data, examined the entire wet steam formation process, and validated the model and methodology. Finally, numerical analysis was conducted on a steam turbine stator blade, a novel dehumidification structure was proposed, and its effectiveness was calculated. The results demonstrate that classical nucleation theory and droplet growth models accurately describe the spontaneous condensation phase transition process in steam.

The nucleation rate increases significantly only when steam supercooling reaches approximately 18 K. The maximum nucleation rate occurs downstream of the nozzle throat. The proposed dehumidification structure can fundamentally achieve dehumidification, with effectiveness improving as through-flow diameter increases, resulting in reduced wetness loss.

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