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

Authors: Zhang Guojie, Zhou Zhongning, Wu Jiqing, Li Yimin, Zhang Sen

Date: 2018-01-05T00:00:00+00:00

Abstract

This study addresses the phenomenon of wet steam formation in the last stage of steam turbines, analyzing the governing equations and physical model for non-equilibrium condensation flow of wet steam. Computational analysis is performed on the Moses and Stein nozzle, revealing characteristics of the wet steam formation process; comparison with experimental results verifies the validity of the numerical calculation model and method. Finally, numerical calculation is conducted on a steam turbine stator blade to analyze the moisture and subcooling distributions, and a novel moisture removal structure for stator blades is proposed and evaluated through computational analysis. The results demonstrate that as the channel diameter increases, the average moisture content at the stator blade exit decreases, resulting in reduced wet steam loss.

Full Text

Numerical Study on Dehumidification Methods for Steam Turbine Stator Blades

Zhang Guo-Jie, Zhou Zhong-Ning, Wu Ji-Qing, Li Yi-Min, Zhang Sen
School of Electrical and Power Engineering, China University of Mining & Technology, Xuzhou 221116, China

Abstract

This paper investigates the wet steam formation phenomenon in the final stage of steam turbines by analyzing the governing equations and physical models of non-equilibrium condensation flow in wet steam. Numerical calculations were performed on the Moses and Stein nozzle to characterize the wet steam formation process, and the correctness of the numerical model and methodology was verified through favorable comparison with experimental results. Finally, numerical simulations were conducted on a steam turbine stator blade to analyze the distributions 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 loss is reduced.

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

Received: 2016-04-26; **Revised:** 2017-05-03
Funding: Supported by the Fundamental Research Funds for the Central Universities (No. 2013XK08.2)
Author Introduction: Zhang Guo-Jie (1989-), male, Ph.D. candidate, 主要从事透平机械内部流动的研究
Corresponding Author: Zhou Zhong-Ning, Associate Professor, zznwt@163.com

0 Introduction

Steam turbines are critical energy conversion devices that play a significant role in the national economy. 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 generation of numerous droplets. These droplets are highly detrimental to stage performance, not only reducing turbine stage efficiency but also causing severe erosion of rotor blades as they continue to grow.

The formation of droplets carried by wet steam in turbine flow passages represents an extremely complex problem involving phase change and two-phase flow. The difficulty in studying wet steam lies in the fact that steam 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 and colleagues in 1930. Bakhtar and Zidi experimentally investigated nucleation phenomena in two-dimensional nozzles with three different expansion rates, obtaining data on pressure ratio and droplet size along the nozzle axis, and subsequently conducted theoretical studies on the subject. Gerber et al. developed a numerical model based on the Eulerian-Lagrangian method and applied it to calculations for a Moore low-pressure nozzle, achieving good agreement between numerical and experimental results. Gerber and Kermani later developed an Eulerian-Eulerian numerical method for non-equilibrium condensation under high-pressure, transonic conditions, which also showed excellent agreement with experimental data. Nevertheless, wet steam flow in low-pressure turbine stages remains a highly relevant research topic today, particularly concerning various losses caused by wet steam, as detailed in the work of Moore and Sieverding.

With advances in computer technology and maturation of two-phase flow theory, numerical simulation has become a viable approach 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 steam turbine dehumidification methods are primarily divided into internal and external approaches. External dehumidification improves steam parameters through regeneration to achieve moisture removal, while internal dehumidification consists of three categories: stator dehumidification, rotor dehumidification, and diaphragm dehumidification between stator and rotor rows. This paper focuses on stator dehumidification.

Existing stator dehumidification methods primarily involve manufacturing hollow stator 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. However, these methods feature complex structures and fail to fundamentally eliminate wet steam formation. A well-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 to perform 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 through numerical simulation and result analysis.

---

1 Wet Steam Phase Transition Process and Mathematical Model

The transformation from dry to wet steam flow in turbine stator blades involves three primary stages. In the first stage, saturated steam crosses the saturation line and condensation occurs, generating a small number of extremely fine droplets. In the second stage, as the steam continues to expand, supercooling increases, causing the fine droplets formed in the first stage to suddenly grow larger. However, since their number remains limited, the droplets absorb heat from the steam, their radius decreases, and supercooling increases further. In the third stage, as steam expansion continues, supercooling gradually decreases, the nucleation rate diminishes and eventually stabilizes, the droplet number becomes nearly constant, and droplet radius increases slightly. Throughout this flow process, a non-equilibrium two-phase flow model can be applied for calculations, with droplet diameters typically ranging from 0.1 to 1 m.

1.1 Governing Equations

The small droplets formed in wet steam within steam turbines move with the steam flow. For this study, we assume no relative motion between droplets and steam, meaning both phases share a common velocity. Consequently, the governing equations encompass both the continuous vapor phase and the discrete droplet phase.

The continuous phase mass equation can be expressed as:

$$\partial \rho_c r_c (\rho_c u_i r_c) = -i = 1(S_d + m^* r_c J_d)$$

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

The discrete phase droplet mass equation is given by:

$$\partial \rho_d r_d (\rho_d u_i r_d) = S_d + m^* r_c J_d$$

For each individual droplet, a corresponding mass equation exists, as shown in Equation (3):

$$\partial \rho_d N_d (\rho_d u_i N_d) = \rho_d r_c J_d$$

In the above equations, J_d represents the nucleation rate, defined as the number of droplets formed per unit volume of steam per unit time; r_c is the vapor volume fraction, which relates to the single droplet volume fraction r_d as follows:

$$i = 1$$

$$r_d = 1$$

The continuous phase energy equation is presented in terms of total enthalpy:

$$h_{cl} \rho_l 2\pi RT$$

where T_d denotes droplet temperature.

$$\gamma + 1 C_p (T_d - T)$$

Given the previous assumption of a common velocity for all phases, the momentum equation needs to be solved only for the continuous phase (vapor), with interphase momentum changes represented by the source term S_d. The momentum equation is:

$$\partial (r \rho u_i)_c \partial (r \rho u_j u_i)_c = -r_c \partial (r \tau_{ji})_c i = 1$$

$$\partial (r \rho h)_c \partial (r \rho u_j h)_c \partial x_j)_c + \partial (r \tau_{ji})_c i = 1 \partial x_j (r k_t$$

In the above equation, S_d incorporates the effects of mass and heat transfer. According to Gyarmathy's research, when droplet diameters are small, Equation (14) can be used for calculation:

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

1.2 Nucleation and Droplet Growth Models

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

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

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

Accounting for non-isothermal effects, the nucleation rate expression from Kantrowitz' s correction to classical nucleation theory is employed:

$$-(4\pi r^{3kT})1 + \theta(\rho^2 \rho_l)2\sigma$$

where: α is the condensation coefficient (typically valued at 1), k is the Boltzmann constant, m is the mass of a single water molecule, σ is 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 expressed as:

$$\rho_l \rho_c 2(\gamma - 1)\gamma + 1RT(h_{cl}$$

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

After condensation nuclei formation, the small nuclei droplets continuously exchange heat and mass with surrounding steam. When the droplet radius exceeds the critical radius, droplets grow; when smaller, they evaporate. The critical radius expression is:

$$r^* = \rho_l RT \ln S$$

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

$$P_{sat}(T)$$

where p is steam pressure and $P_{sat}(T)$ is the saturation pressure of water vapor at that temperature.

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

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

Droplet growth involves two mechanisms: first, mass transfer from steam condensation, and second, 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:

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 investigates the Moses and Stein nozzle to validate the correctness of the two-phase flow model, providing a foundation for the dehumidification calculations in the following section.

2.1 Moses and Stein Nozzle Calculation and Result Analysis

The nozzle geometry is shown in Figure 1 [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. From the formula, the nozzle expansion rate is determined to be 8230 sec [1]. The nozzle throat is located at x=0.0822 m with a square cross-section of 10×10 mm.

$$e_x = -(u/p)(dp/dx)$$

represents the mass generation rate in the non-equilibrium condensation process. In classical nucleation theory, the mass generation rate m^* is determined by both nucleation and droplet growth (or evaporation). Their combined mass increase constitutes the total mass generation rate. Therefore, m^* can be written as:

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

Moses conducted multiple sets of experiments, from which seven cases were selected for numerical calculation in this study. The boundary conditions and measurement data are presented in Table 1 .

Table 1 Boundary conditions and Wilson point parameters in different experimental numbers

Grid density significantly impacts numerical calculation reliability. This paper validated grid independence by comparing calculations with different mesh densities against experimental values. The final mesh is shown in Figure 2 [Figure 2: see original paper].

This section first presents numerical calculations for Experiment No. 193 compared with experimental measurements, as shown in Figure 3.

Figure 3 Pressure ratio distribution

As shown in Figure 3, the computational results demonstrate excellent agreement with experimental values, confirming the correctness of the calculation model and methodology.

To explain the droplet formation process 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 [Figure 4: see original paper].

Figure 4 Droplet parameters distribution along nozzle center line

The droplet parameters along the nozzle centerline can be divided into three regions: Region I before the nozzle throat, Region II from the throat to the Wilson point location, and the remaining Region III. To better analyze Figure 4, the Wilson point concept must first be explained. Discovered and proposed by Wilson, the Wilson point represents the location 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 relatively high steam temperature and low supercooling, the nucleation rate remains low, producing only a small number of droplets and resulting in very low steam humidity. Figure 4 also reveals that nucleation does not occur at low supercooling; the nucleation rate only begins to increase significantly when supercooling reaches approximately 18 K. Steam carrying a small number of droplets passes through the throat into Region II, where it continues to expand. As the expansion rate decreases, the saturation temperature changes minimally, while droplet radius reduction due to heat absorption causes steam temperature to decrease, ultimately increasing steam supercooling to its maximum at the Wilson point. Both nucleation rate and droplet number increase with supercooling temperature, but steam humidity remains nearly constant due to decreasing droplet diameter. Upon entering Region III, the steam expansion rate increases, causing the saturation temperature to decrease more rapidly. The large number of droplets generated in Region II grow in radius, 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 methodology, multiple experimental cases were calculated, with the Wilson point locations and corresponding supercooling values compared as shown in Figure 5 [Figure 5: see original paper]. The predicted Wilson point locations show excellent agreement with experimental values, with a maximum error of approximately 2.5%, demonstrating high prediction accuracy. For supercooling at the Wilson point, the difference between calculated and experimental values is less than 4 K, which is acceptable. Through numerical calculations on the Moses and Stein nozzle and comparison with experimental results, the causes and influencing factors of droplet formation were analyzed, validating the mathematical model and computational methodology to provide theoretical assurance for the dehumidification analysis in the following section.

Figure 5 Comparison of supercooling between simulation and experiment in different numbers

3.1 Model Introduction

This section examines the final stage stator blades of a specific steam turbine. The stage contains 60 blades total, with only one flow passage selected for calculation. The hub and shroud diameters are 762 mm and 910 mm respectively, with the computational model shown in Figure 6 [Figure 6: see original paper]. Total pressure and total temperature inlet conditions (26.5 kPa, 340.15 K) and static pressure outlet condition (6.62 kPa) were applied. Grid independence was verified prior to calculation.

Figure 6 Model of computation

3.2 Calculation Result Analysis

Two indicators are generally used to evaluate humidity distribution within flow passages: supercooling, which causes droplet formation, and humidity itself, which represents the result of droplet formation. Therefore, this study investigates dehumidification methods based on these two characteristics.

Figures 7 and 8 present the supercooling distribution and humidity distribution at 50% blade span for the original blade profile.

Figure 7 Supercooling distribution at 50% span
Figure 8 Mass fraction of H2O at 50% span

Comprehensive analysis of Figures 7 and 8 reveals that the location of maximum supercooling does not coincide with the location of maximum humidity. Maximum supercooling occurs 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.

3.3 Dehumidification Structure and Calculation Result Analysis

Based on the analysis in Section 3.2, two dehumidification approaches are possible: first, reducing supercooling to decrease nucleation rate and achieve dehumidification; second, more directly converting generated humidity back to steam by supplying heat. This paper focuses on the first approach.

A novel dehumidification structure is proposed, as shown in Figure 9 [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, the front endpoint of the black line is tangent to the camber line, and the rear segment targets the region of maximum supercooling. The parameter d represents the diameter of the channel. This structure' s advantage lies in fundamentally achieving dehumidification without requiring additional steam flow.

Figure 9 The structure of dehumidification

For convenience, the diameter d of the blade without a channel (original blade) is defined as 0. This section investigates five different stator blade structures with channel diameters of 1 mm, 1.5 mm, 2 mm, 2.5 mm, and 3 mm. Calculations were performed for these five models, with humidity changes and wetness losses shown in Figure 10 [Figure 10: see original paper].

Figure 10 Wetness loss and Mass fraction for different diameters

Figure 10 clearly shows that as the channel diameter increases, wetness loss decreases and humidity is reduced. At a 3 mm channel diameter, humidity decreases by nearly 0.01%. Although this may appear modest, increasing the number of flow channels would yield further substantial humidity reduction.

---

4 Conclusions

Focusing on wet steam formation in the final stage of steam turbines, this paper analyzed the governing equations for wet steam formation and flow, performed numerical calculations on a nozzle with existing experimental data, analyzed the complete wet steam formation process, and validated the model and methodology. Finally, calculations were performed on a steam turbine stator blade, a novel dehumidification structure was proposed, and its effectiveness was evaluated. The results demonstrate that classical nucleation theory and droplet growth models can accurately describe the phase transition process of spontaneous steam condensation. The nucleation rate only increases significantly when steam supercooling reaches approximately 18 K, with the maximum nucleation rate location occurring downstream of the nozzle throat. The proposed dehumidification structure can fundamentally achieve dehumidification, and as the channel diameter increases, dehumidification effectiveness improves and wetness loss decreases.

References

[1] Stodola A. Steam and Gas Turbines [M]. New York: McGraw-Hill, 1927:75-83
[2] Wilson C T R. Investigation of X-Rays and -Ray by the Cloud Method Part II- -Rays [J]. Proceedings of the Royal Society of London,1923,104:192-212
[3] Yellott J I. Supersaturated Steam [J]. Transaction of the ASME, 1934,56:411-430
[4] Rettaliata J T. Undercooling in Steam Nozzles [J]. Transaction of the ASME, 1936,58:599-605
[5] Yellott J I, Holland C K. The Condensation of Flowing Steam Part I - Condensation in Diverging Nozzles [J]. Transaction of the ASME, 1937,59:171-183
[6] Bakhtar F. Nucleation phenomena in flowing high-pressure steam: experimental results[J]. Proceedings of the Institution of Mechanical Engineers,1989, 203(A3):195-200
[7] Bakhtar F, Zidi K. Nucleation phenomena in flowing high-pressure steam

Part 2: theoretical analysis[J]. Proceedings of the Institution of Mechanical Engineers,1990,204(A4):233-242
[8] Gerber A G. Two-Phase Eulerian/Lagrangian Model for Nucleating Steam Flow[J]. Journal of Fluids Engineering, 2002,124(2):465-475
[9] Gerber A G, Kermani M J. A pressure based Eulerian-Eulerian multi-phase model for non-equilibrium condensation in transonic steam flow[J]. International Journal of Heat and Mass Transfer,2004, 47(10-11):2217-2231
[10] Moore M J, Sieverding C H. Two-phase steam flow in turbines and separators [M]. London: Hemisphere Publishing Corporation, 1976:245-268
[11] 丰镇平，李亮，李国君. 汽轮机湿蒸汽两相凝结流动数值研究的现状与进展 [J]. 上海汽轮机,2002,6(2):1-10
FENG Zhenping, LI Liang, LI Guojun. The status and progress of numerical research on flow condensation wet steam turbine [J]. Shanghai Turbine,2002,6 (2):1-10
[12] Ishazaki K, Ikohagi T, Daiguji H. A High-Resolution Numerical Method for Transonic Non-equilibrium Condensation Flows Through a Steam Turbine Cascade [C]. In Proceedings of the 6th International Symposium on Computational Fluid Dynamics, 1995,1:479–484
[13] Young J B. Two-Dimensional Nonequilibrium Wet-Steam Calculations for Nozzles and Turbine Cascades [J]. Journal of Turbomachinery.1992, 114(3):569–579
[14] Grübel M, Starzmann J, Schatz M, et al. Two-Phase Flow Modeling and Measurements in Low-Pressure Turbines-Part I: Numerical Validation of Wet Steam Models and Turbine Modeling [J]. Journal of Engineering for Gas Turbines & Power, 2014, 137(4):042602
[15] Young J B. The Spontaneous Condensation of Steam in Supersonic Nozzle [J]. Physico Chemical Hydrodynamics, 1982,3(2):57–82
[16] Gyarmathy G. Grundlagen einer Theorie der Naß dampfturine [D]. Germany,1962
[17] 张冬阳. 非平衡态湿蒸汽流动快速准确数值模拟方法研究 [D]. 中国科学院工程热物理研究所,2002
ZHANG Dongyang. A fast and accurate numerical simulation method for nonequilibrium wet steam flow[D]. Chinese Academy of Sciences: Institute of engineering thermal physics,2002

Author Introduction: Zhang Guo-Jie (1989-), male, Ph.D. candidate, 主要从事透平机械内部流动的研究
Corresponding Author: Li Yi-Min, Professor, liyimin@cumt.edu.cn
Primary Contact: Zhang Guo-Jie
Phone: 18361220182
Email: zhangguojie2015@cumt.edu.cn
Address: Room A333, Mechanical and Electrical Building, Nanhu Campus, China University of Mining & Technology, Xuzhou, Jiangsu Province, 221116

Note: Figure translations are in progress. See original paper for figures.