Numerical analysis of condensation effects on final-stage rotor-blade rows in low-pressure steam turbine

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Abstract
The causal relationship between unsteady forces of wet-steam flows on rotor-blade rows and the steam condensation in low pressure steam turbines is still one of unresolved issues. In this study, we investigate the effect of condensation on the time-dependent torque of final-stage long-rotor blade rows in a low pressure steam turbine. Then we simulate unsteady wet-steam flows through the three-stage stator-rotor blade rows in the steam turbine while changing the inlet temperature condition. The variation of the inlet temperature results in creating a flow field with a different amount of wetness due to condensation. The temperature and pressure in the flow field obtained from a different inlet temperature are compared with each other, and the torques calculated from the time-dependent pressure on a final-stage long-rotor blade are also relatively compared. The calculated results in this study indicate that the time-dependent torques on the final-stage long-rotor blade significantly depend on the latent heat added by condensation and that the amount of condensation is highly sensitive to variations in the inlet temperature. This study also suggests that an optimal inlet temperature may be exist for optimizing the torque of low pressure steam turbines.

Keywords: Steam turbine, Wet-steam, Nonequilibrium condensation, Long-rotor-blade, Unsteady force, Numerical simulation

1. Introduction
Numerical simulation of the effects of off-design operation is crucial for the development of novel multistage steam turbines. As rotor blades increase in length, the accurate prediction of blade oscillations is especially important. These oscillations, owing to stator–rotor flow interactions or suboptimal operating conditions, may cause otherwise unforeseen accidents. Low-flow operation (Filippenko et al., 2011) (Stanciu et al., 2013) (Tanuma et al., 2015) (Rzadkowski et al., 2016) and rotational instability (Zhang et al., 2011) (Qi et al., 2013) (Megerle et al., 2015) have already been investigated numerically and experimentally. A numerical study concluded that blowing steam jets into the final stage was effective for suppressing the large-scale vortex that can occur under low-load conditions (Haller et al., 2016). Experimental research has shown that spraying water influences the blade vibration in a low pressure steam turbine (Shibukawa et al., 2016). These results have been essential for establishing the reliability of steam turbines in off-design conditions.

In previous work (Miyake et al., 2015) (Miyazawa et al., 2016), we investigated the effects of unsteady pressure forces due to a wet-steam flow on multi-stage long-blade rows by assuming an actual number of blades. Our study indicated that strong shocks are generated by the trailing edges of the final-stage stator blades near the hub, and that those shockwaves come into contact with the adjacent rotor blades. A reflection wave is then generated from each rotor-blade nose, after which the shock from the stator blade is transferred to the rotor blade. The reflection waves then propagate toward the stator blades as pressure waves that merge together to form a long streaking structure in the multi-passage case. These complex interactions combine to induce a pressure fluctuation on each rotor blade that results in fluctuating torque. An interesting unresolved issue is the question of how such torque fluctuations are affected by the condensation that occurs as stream passes through several stages of stator and rotor blades. Of the literature we
reviewed on blade oscillations in low-flow conditions, no studies presented a model for the effect of condensation on torque fluctuations.

Herein, we investigate how condensation influences the time-dependent torque along the final-stage long-rotor-blade rows in the low pressure steam turbine. We focus particularly on the latent heat released when steam condenses because this heat increases the temperature of the gas in the turbine, which in turn increases the pressure. We simulate wet-steam flows through the three-stage stator-rotor blade rows in the steam turbine operating at different inlet temperatures. Varying the inlet temperature induces changes in the nonequilibrium condensation as steam flows through the blade rows. This variation in condensation behavior causes flow fields with a different wetness at the final stage. We compared resultant temperature and pressure values in the flow field influenced by the release of latent heat from condensation. Our computational analysis allows us to predict how inlet temperature and the resultant condensation behavior affect the fluctuations in torque on the final-stage long-rotor-blade rows.

**Nomenclature**

- $c$ Speed of sound
- $e$ Total internal energy per unit volume
- $I$ Nucleation rate
- $J$ Jacobian for transformation
- $k$ Turbulent kinetic energy
- $n$ Number density of water droplets per unit mass
- $p$ Static pressure
- $R$ Gas constant
- $r$ Average radius of droplets
- $r_*$ Critical radius of a droplet
- $S_k$ Source term for $k$ equation
- $S_\omega$ Source term for $\omega$ equation
- $T$ Static temperature
- $t$ Physical time
- $W_i$ Relative contravariant velocities
- $w_i$ Relative physical velocities
- $x_i$ Cartesian coordinates

**Subscripts**

- $v$ Water vapor
- $l$ Water liquid

**2. Numerical methods**

**2.1 Fundamental equations**

The fundamental equations in our in-house solver comprise of the conservation laws for total density, momentum, total energy, liquid water density, and water droplet number density. These conservation laws are coupled with SST turbulence model (Menter, 1994) with relative velocities in general curvilinear coordinates. The equations simulate homogeneous flows without any slippage between gas and water droplets. Water droplets form monodisperse spheres with a constant radius locally. The set of equations is written as

$$\frac{\partial Q}{\partial t} + \frac{\partial F_i}{\partial \xi_i} + S + H = 0$$

(1)

where $Q$, $F_i$ ($i = 1, 2, 3$), $S$, and $H$ are the vector of unknown variables, vector of flux, viscous term, and the source term, respectively. They are written in the following form:
\[ Q = J \\
\begin{bmatrix}
\rho \\
\rho \omega_i \\
\rho \omega_2 \\
\rho \omega_3 \\
\rho \beta \\
\rho n \\
\rho k \\
\rho \omega
\end{bmatrix} = \\
\begin{bmatrix}
\rho W_i \\
\rho W_i + \partial \xi_j / \partial x_i p \\
\rho W_i + \partial \xi_j / \partial x_i p \\
\rho W_i + \partial \xi_j / \partial x_i p \\
\rho \beta W_i \\
\rho n W_i \\
\rho k W_i \\
\rho \omega W_i
\end{bmatrix} \\
\begin{bmatrix}
0 \\
\tau_{ij} \\
\tau_{ij} \\
\tau_{ij} \\
\tau_{ij} \\
0 \\
\sigma_{ij} \\
\sigma_{ij}
\end{bmatrix} \\
\begin{bmatrix}
0 \\
0 \\
0 \\
0 \\
0 \\
0 \\
0 \\
0
\end{bmatrix} \\
\begin{bmatrix}
\rho (Q^x x_j + 2\Omega W_i) \\
\rho (Q^x x_j - 2\Omega W_i)
\end{bmatrix}.
\]

The source term takes account of Coriolis and centrifugal forces due to rotation.

The state equation and the speed of sound in wet steam were derived by Ishizaka et al. (1995). Assuming that the condensate mass fraction \( \beta \) is sufficiently small (\( \beta < 0.1 \)), they obtained the following:

\[ p = \rho RT (1 - \beta) \quad (2) \]

\[ c^2 = \frac{C_{pm} \rho}{C_{pm} - (1 - \beta) R \rho} \quad (3) \]

where \( C_{pm} \) is the isobaric specific heat obtained by a linear combination of the specific heats of the gas and liquid phases using the mass fraction \( \beta \).

### 2.2 Condensation model

The mass generation rate \( \Gamma \) of water droplets is based on the classical condensation theory. It is defined as the sum of the mass generation rate of nuclei at a critical size and the growth rate of a water droplet. Ishizaka et al. (1995) further simplified the equation as follows:

\[ \Gamma = 4 \frac{3\rho \beta \left(\frac{d^3 r}{dt^3} + 3\rho \rho_r^2 \frac{dr}{dt}\right)}{\rho} \quad (4) \]

where the homogeneous nucleation rate \( I \) is as defined by Frenkel (1955), and the growth rate \( dr/dt \) of a water droplet is based on Gyarmathy's model (1963). Based on the discussion by the International Wet Steam Modeling Project (IWSMP) (Starzmann et al., 2016), we modified some empirical parameters for the nucleation model and Gyarmathy's model coupled with Young’s modification (Young, 1982) in this study. The condensation coefficient and the empirical parameter \( \alpha \) in the Young’s modification were specified to 1.0 and 5.0 in this study. The wetness values that we obtained varied slightly from those obtained in our previous study (Miyazawa et al., 2016).

### 2.3 Numerical schemes

A high-order, high-resolution, finite-difference method based on Compact MUSCL (Yamamoto and Daiguji, 1993) and Roe’s approximate Riemann solver (Roe, 1981) was employed for the space difference of the convection terms. The viscosity term was calculated by the second-order central-difference scheme. The lower-upper symmetric Gauss–Seidel (LU-SGS) scheme (Yoon and Jameson, 1988) was employed for time integration. These numerical methods were customized for the SX-ACE supercomputer at Tohoku University, based on the message passing interface (MPI).

### 2.4 Computational domains

The actual three-stage low-pressure steam turbine developed by Mitsubishi Heavy Industries (MHI) (Watanabe et al., 2003) is modeled for this study as in our previous studies (Miyake et al., 2015) (Miyazawa et al., 2016). The final long-rotor blade is 36-in long and rotates at 3,600 rpm.

Figure 1 shows the schematic of the simulated three-stage stator- and rotor-blade rows. We will refer to the stator and rotors as marked in the figure (1S, 1R, 2S, 2R, 3S, and 3R where S is a stator and R is a rotor from first to third
Table 1 gives the number of blades in each simulated stage, which differ slightly from those in the actual turbine. The numbers were rounded off so that the greatest common divisors could be employed for this computation.

Figure 2(a) shows a schematic of the overall computational grid we used. A total of 64 blocks for the blade passages and the additional intermediate grid region between the stator and the rotor are calculated simultaneously by assuming a sliding boundary condition.

Figure 2(b) shows the computational mesh for final-stage stator and rotor blades at 50% span. We generated $91 \times 91 \times 181$ grid points for each blade passage, $61 \times 91 \times 181$ grid points for each inlet and outlet region, and $31 \times 91 \times 181$ grid points for each intermediate region in the axial, tangential, and radial directions. The grid topology was the same as that used in our previous studies (Miyake et al., 2015) (Miyazawa et al., 2016) and the grid points employed in this study was the same with those in the study (Miyazawa et al., 2016).

| Table 1 | Real and imposed blade numbers |
|---------|--------------------------------|
|         | 1S  | 1R  | 2S  | 2R  | 3S  | 3R  |
| Real blade number | 78  | 62  | 48  | 62  | 48  | 58  |
| Imposed blade number | 80  | 60  | 50  | 60  | 50  | 60  |
| Computational blade number | 8   | 6   | 5   | 6   | 5   | 6   |
3. Results

Table 2 shows the computational flow conditions. For our simulation, we set the inlet total pressure to 0.19 MPa, and the pressure ratio between the inlet and outlet static pressures to 13:1 to correspond with the experimental conditions under which the turbine was originally evaluated (Watanabe et al., 2003). The outlet static pressure was calculated from the pressure ratio and supplied the boundary condition. The inlet total temperature under design conditions is 398 K. Then the inlet flow occurs under a dry-steam condition. Homogeneous nucleation and nonequilibrium condensation should occur all along in the three stages under this case. This design condition is defined as CASE 1. We consider an additional three cases with inlet temperatures of +50, +25, and −25 K from the design specification (448, 423, and 373 K, respectively).

| Table 2  | Computational flow conditions |
|----------|-------------------------------|
|          | CASE 1 | CASE 2 | CASE 3 | CASE 4 |
| Inlet total pressure [MPa] |       |       |       | 0.19   |
| Pressure ratio              |       |       |       | 13     |
| Inlet total temperature [K] | 398   | 448   | 423   | 373   |
| Temperature difference [K]  | 0     | +50   | +25   | −25   |

First, we checked the reliability of our numerical solutions by comparing temporally and spatially mass-averaged static and total pressure distributions in the pitch-wise direction at the outlet of 3R rotor blades for CASE 1 with the experimental results, as shown in Figs. 3(a) and 3(b). The static pressure distributions in Fig. 3(a) agree well with the experimental values near the blade tips, whereas the numerical solutions are slightly overestimated from the experimental values toward the hub. However, this difference may be insignificant when compared with the pressure ratio of 13. The total pressure distributions near the mid-span region also agree well with that of the experiments, even if those values differ near the hub and tip. These discrepancies may arise because our model does not consider coarse water droplets. Coarse water droplets may tend to move outward in the radial direction owing to the centrifugal force. This centrifugal distribution may increase the mass-flow rate near the tip and decrease it near the hub, changing the total pressure distribution. Our approach does not consider such coarse water droplets that would depart from water-vapor stream lines. The integrated value of the total pressure along the radial direction in our preliminary simulation is almost identical to the corresponding value for the experiments.

Next, we compare the differences among the simulated results to clarify the effect of condensation on the final-stage long-rotor-blade rows. Figures 4(a), 4(b), 4(c), and 4(d) show the instantaneous wetness distributions at 50% span for all four inlet temperatures. As shown in Fig. 4(a), condensation starts from the outlet of the 1R rotor blades, and the wetness value increases gradually downward. As compared with Fig. 4(a), the onset of condensation is delayed as shown in Figs. 4(b) and 4(c) for high off-design inlet temperatures, whereas condensation soon started after the inlet as shown in Fig. 4(d) for the low off-design inlet temperature. After the onset of condensation, water droplets seem to grow in a similar manner for all inlet temperature. Changing the inlet temperature clearly moved the location at which condensation begins in our simulation.

Figures 5(a), 5(b), and 5(c) present temporally and spatially averaged pressure-temperature curves at the 50%, 10%, and 90% spans, respectively, for all four inlet temperatures. The values of pressure and temperature at the cross section of the turbine axis direction are averaged in time and space. The curve at 50% span through the turbine obtained with CASE 1 indicates that the pressure and temperature decrease gradually and cross the saturation curve before the throat of the 1S stator blade rows. When the pressure and temperature decrease to around 0.09 MPa and 340 K, respectively, the temperature increases rapidly toward the saturation curve; this point is located at the outlet of the 1R rotor blade. Nonequilibrium condensation begins at this pressure and temperature, and the sudden release of latent heat dominates the temperature increase. After the 2S stator blade, mainly the same pressure and temperature tides repeat through the remaining stages along the saturation curve, whereas the balances of the tides at 10% and 90% of the span are different from that at 50% span. The changes in pressure and temperature are dominated by the stator passages at 10% span, whereas the rotor passages dominate those changes at 90% span.

The curves obtained with CASE 2 to CASE 4 start from a different position at the inlet and cross the saturation
curve at a different point than the curves for CASE 1, because the inlet temperature is different. The pressure and temperature curve for CASE 2 crosses the saturation curve near the inlet of the 2S stator blade. The values decrease further toward the outlet of the 2S stator blade and then the temperature increases with the beginning of condensation at 50% span. A similar trend was also found at 10%. However, the temperature increase was relatively small at 90% span. The pressure and temperature curve for CASE 3 crosses the saturation curve near the inlet of the 1R rotor blade. After those values reach around 0.09 MPa and 350 K, the temperature slightly increases once. However, the increase is relatively smaller than the increase that for CASE 1. The temperature increases up to the saturation value after the outlet of the 2S stator blade. A similar trend for temperature and pressure as observed for CASE 1 may also repeat for CASE 3, again with smaller increases than that for CASE 1. Steam for CASE 4 is supersaturated and condensation started soon after the inlet of the 1S stator blade. Then temperature increases once toward the saturation value at the inlet. The tide of temperature and pressure is observed once at the 1S stator blade passage. After the tide, the curve traces the saturation line toward the outlet of the 3R rotor blade. These results show that varying the inlet temperature has a dynamic effect on nonequilibrium condensation occurring in the stator and rotor stages and the resultant increases in temperature and pressure.

Fig. 3 Temporally and spatially averaged static and total pressure distributions after the 3R rotor blades plotted with the experimental data for model confirmation.
Fig. 4 Instantaneous wetness distributions at 50% span.

Fig. 5 Temporally and spatially mass-averaged pressure and temperature values in pitch-wise direction for all four inlet temperatures at the 50%, 10%, and 90% spans.

Next, we focus on the time-dependent torque on the 3R rotor blade. Our previous study (Miyazawa et al., 2016) indicated that strong shocks are generated from the trailing edges of the final-stage stator blades near the hub, and that those shock waves come into contact with the adjacent rotor blades; the interactions induce pressure fluctuations on rotor blades that results in torque fluctuations. We visualized instantaneous density gradients similar to Schlieren photos at the final stage to show how off-design inlet temperatures deform the shock structure.

Figures 6(a) and 6(b) show instantaneous density gradient distributions for CASE 2 and CASE 4 as the cases of the highest and lowest inlet temperatures (448 and 373 K, respectively) at 10% span. The calculated shock structures were
essentially identical, even though the difference of the inlet temperatures is 75K. These results indicate that the variation of the inlet temperature does not influence the shock deformation that leads to the torque fluctuations reported in our previous study (Miyazawa et al., 2016).

Figures 7(a) and 7(b) show instantaneous density gradient distributions for CASE 2 and CASE 4 at 90% span. As compared with those at 10% span, the difference of shock strength and the structure between those in Fig. 7(a) and 7(b) is obvious. Because the wetness value for CASE 4 is to be higher than that for CASE 2 at the final stage, the strength of oblique shocks generated from the trailing edge of the 3R rotor blades may have been weakened by the release of latent heat during condensation.

![Image](image1)

(a) CASE 2 (448 K)  
(b) CASE 4 (373 K)

Fig. 6 Instantaneous density gradient distributions at 10% span.

![Image](image2)

(a) CASE 2 (448 K)  
(b) CASE 4 (373 K)

Fig. 7 Instantaneous density gradient distributions at 90% span.

Figure 8 shows the time-dependent torque on a 3R rotor blade for CASE 1 to CASE 4 normalized by the averaged torque value at each nondimensional time. The unit of the normalized time equals the physical time per pitch movement of the rotor. The maximum peaks shown in Fig. 8 correspond to the shock passing through the 3R rotor nose. This result indicates that the effect of unsteady shocks coming into contact with the adjacent 3R rotor blade is not trivial. The amplitude of the torque variation reaches approximately 10% of the torque. The maximum peaks are mostly in-phase for all four cases, whereas the torque magnitude of the variation is certainly distinct. The torque distributions for CASE 4 are approximately 9% higher than that for CASE 1, while those for CASE 4 and CASE 2 were 6% and 10% lower than that of CASE 1.
Fig. 8  Time-dependent torques on a 3R rotor blade.

Figures 9(a) and 9(b) show the pressure distributions on the suction surface of a 3R rotor blade at the time of maximum torque for CASE 2 and CASE 4. Also Figs. 9(c) and 9(d) show pressure distributions on the pressure surface. The visualized distributions in Fig. 9(a) and 9(b) differ from each other locally. Streak layers with lower pressure values located at the front of the shock are observed in these figures. The pressure downstream from the streak layers near the tip in Fig. 9(a) is slightly higher than that in Fig. 9(b), whereas the pressure upstream from the streak in Fig. 9(a) is relatively lower overall than that in Fig. 9(b). The pressure distributions shown in Figs. 9(c) and 9(d) are even more distinct. The pressure in Fig. 9(c) is lower overall than that in Fig. 9(d). An especially high-pressure region occurred near the tip at the lowest inlet temperature of 373K for CASE 4 (Fig. 9(d)). The pressure differences represented in Figs. 9(a)-9(d) may cause the torque difference shown in Fig. 8. The pressure difference is obviously originated with the release of latent heat which occurs with condensation. Consequently, the numerical results in this study indicate that accurately predicting the amount of latent heat due to condensation is crucial for evaluating the torque incident on the final-stage long-rotor-blade rows.

(a) Suction surface (CASE 2, 448 K)
Fig.9  Pressure distributions on the suction and pressure surfaces of a 3R rotor blade at the maximum torque.

4. Conclusions

We investigated the effect of condensation on the final-stage long-rotor-blade rows in a low pressure steam turbine. Then we simulated unsteady wet-steam flows through the three-stage stator- and rotor-blade rows for the design case and additional three inlet temperatures. On comparing the results, the following conclusions can be drawn.

a) The temporally and spatially averaged static and total pressure distributions at the outlet of the third-stage rotor blades for the original condition were first compared with the experimental data. The static pressures near the tip were in good agreement with the experimental values, and the total pressures near the mid-span region completely coincided with the experimental values, regardless of the discrepancies near the hub and tip, which may be due to the lack of the consideration for coarse water droplets.

b) The location where condensation starts shifted upward or downward according to the decrease or increase in the inlet temperature as compared with that of the design, and the wetness obtained for all four cases increased linearly downward, similar to the design case.
c) The shock structures at 10% and 90% span obtained for the cases of the highest and lowest inlet temperature were compared with each other. Then the structures at 10% span were identical for all inlet temperatures, whereas the shock structures at 90% span were distinct. A higher release of the latent heat in the case of the lowest inlet temperature may relatively weaken the strength of the shock at 90% span.

d) Off-design inlet temperatures changed the torque applied to the last-stage rotor blades. Inlet temperatures higher than the design value decreased the torque on the final rotor, whereas an inlet temperature less than the design value increased torque on the final rotor. The changes in pressure on the final rotors were clearly influenced by the amount of condensation that released latent heat before the final rotor at different inlet temperatures.

Consequently, the calculated results in this study indicate that accurately predicting the amount of latent heat due to condensation at the final stage is a crucial issue to evaluate the time-dependend torque on the final-stage long-rotor-blade rows. Our results also suggest that an optimal inlet temperature may be exist for optimizing the performance of low-pressure steam turbines.

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