Simulation of unsteady flows through three-stage middle pressure steam turbine in operation

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Received: 16 February 2020; Revised: 28 May 2020; Accepted: 8 July 2020

Abstract
The purpose of this study is to simulate unsteady steam flows through three-stage stator and rotor blade passages in a middle-pressure steam turbine operated in a power plant while considering manufactured and secular-changed blade shapes. The shapes of the manufactured and secular-changed blades are measured during overhaul. The numerical method is based on an in-house code developed by Tohoku University. The time-dependent pressure and the Mach number are visualized, and the difference of the results obtained by assuming manufactured and secular-changed blades is explained. The simulated static temperatures assuming the blades shapes are then compared with each other and with the measured data. In addition, a simple method modifying the simulated static temperature considering real gas effect is introduced.

Keywords: Numerical simulation, Steam turbine, Middle pressure, Three stages, Secular change, Heat balance

1. Introduction

The shape change of stator and rotor blades caused by a long-time operation in steam turbines is one of the primary issues considered for the scheduling of maintenance, repair, and overhaul (MRO). Thermal power plants must maintain high performance and safety for long times to reduce the costs of fuels and MRO. The heated steam streaming in boiler pipes may conditionally oxidize the pipe materials, causing a scale layer of oxide on the inner wall of the pipe. The growing scale finally peels off from the wall and moves toward the steam turbine. Solid particles that originated from the scales collide with the stator and rotor blade surfaces and gradually change the shapes over time. The secular-changed blades certainly affect the performance of steam turbines. The MRO of steam turbines deeply depends on the degree of the secular change. However, the degree cannot be accurately predicted only by the measured data obtained from the turbine operation. A numerical approach is desirable to assist the prediction. Some approaches to investigate the effect of erosion on steam turbine blades have been reported (Campos-Amezcua et al., 2007) (Wang et al., 2010) (Edwards et al., 2011). These approaches focused on the prediction of the performance for a single flow passage through a steam turbine nozzle. However, the total performance of a steam turbine unit should be predicted if it assists the MRO. None of such approaches have yet been reported.

The authors at Tohoku University reported numerical studies (Yamamoto et al., 2010) (Miyake et al., 2015) (Miyazawa et al., 2017) simulating unsteady wet-steam flows considering nonequilibrium condensation through three-stage rotor and stator blade passages in a low-pressure steam turbine. The in-house code employs the sliding boundary treatment between stator and rotor boundaries; hence, the numerical method in the code can predict unsteady...
flows through multistage stator and rotor blade passages. Using this code, we have already reported a numerical study simulating unsteady steam flows through single stator and rotor blade passages in a middle-pressure steam turbine while considering manufactured and secular-changed blades (Uemura et al., 2019). We then reported that the secular-changed stator and rotor blades certainly affect the flow field. However, we could not predict the total performance of the steam turbine because of only a single-stage simulation. In this study, we extend the previous study to the investigation of the three-stage stator and rotor blade passages. Each one typical shape of the secular-changed blade for the stator and the rotor in three stages was measured from actual blades during overhaul. The shapes of the manufactured blades for the first-stage stator, first-stage rotor, and third-stage stator that replace the secular-changed blades were also measured. We simulate unsteady steam flows through three-stage stator and rotor blade passages while considering the manufactured and secular-changed blades and show a comparison of the time-dependent results between the two cases. We further predict the performance degradation of the three-stage middle-pressure steam turbine caused by the secular change of blades. The simulated static temperatures assuming the manufactured and secular-changed blade shapes are compared with each other and with the measured data. In addition, a simple method modifying the simulated static temperature considering the real gas effect is introduced.

2. Numerical methods

2.1 Governing equations and numerical schemes

Compressible Navier–Stokes equations are solved as governing equations for steam flows. The equations were transferred to those of relative velocities in curvilinear coordinates as in our previous studies (Yamamoto et al., 2010) (Miyake et al., 2015) (Miyazawa et al., 2017) (Uemura et al., 2019). Flows are supposed to be dry steam because of the high pressure and high temperature through middle-pressure steam turbines. Turbulent flows are modeled by the SST turbulence model (Menter, 1994). We employed a finite-difference method based on the Compact MUSCL (Yamamoto and Daiguji, 1993) and Roe’s approximate Riemann solver (Roe, 1981) for the space difference of convection terms. The second-order central-difference scheme was applied to the viscosity term. The time integration was conducted by the LU-SGS (Yoon and Jameson, 1988). The computational code was parallelized by the MPI (message-passing interface) and executed on the SX-ACE supercomputer at Tohoku University. Unsteady flow simulations based on the URANS approach are conducted in this study. Then, the sliding boundary treatment is applied to each boundary between rotor and stator. Unsteadiness due to larger-scale vortices than the grid size through the boundary can be simulated by the URANS approach.

2.2 Blade shape and computational domain

The 3D blade shapes for the three-stage stators and rotors in an actual middle-pressure steam turbine working at a coal-fired power plant were measured using a 3D scanner (ATOS Core 200) following the method in the study by Yonezawa et al. (2018). The theoretical uncertainty of the 3D scanner was 50 µm, but the measured data may include higher error due to the time limitation even if the error is sufficiently smaller than the secular-changed level. The surface roughness of blades may be necessary to be considered for resolving local flow features over blades. Because the scale of the surface roughness is relatively smaller than that of the secular change of blades, this study focused only on the secular change of blades. The secular-changed blades were removed from the turbine unit during overhaul. One typical blade for each stator and rotor was chosen because of the limited time for measurement. The shapes of the manufactured blades for the first-stage stator, first-stage rotor, and third-stage stator that replace the secular-changed blades were also measured.

Figure 1 represents the schematic of the three-stage stator and rotor blade passages and the computational domains considered herein. The computational domains comprised main six parts for three-stage stator and rotor blade passages with one inlet region, five intermediate regions, and one outlet region: a total of 13 parts. Note that the actual middle-pressure steam turbine has some long blades for assisting the strength of turbine unit, but we did not consider the long stator blades in this study.

Table 1 shows the number of blocks for the six main parts. The grid points for each block were all the same: 61 × 61 × 61 that decided according to the discussion on numerical accuracy in our last studies (Yamamoto et al., 2010) (Miyake et al., 2015) (Miyazawa et al., 2017) (Uemura et al., 2019). The total number of stator and rotor blades was slightly different from that in the actual turbine because of the rounding off the number to the imposed number. The number of blocks for the stator and rotor regions coincided with the least common multiple of the imposed number considered for the computation. The total number of blocks for the rotor passages, stator passages, and additional grids was 75. The total...
number of grid points was 25 million. All blocks were simultaneously computed with the sliding boundary treatment and the periodic boundary treatment is applied to the boundary between neighboring two sets of blocks. We applied the same measured blade shape to each stator and rotor blades.

Table 1 Computational domain and number of blocks

<table>
<thead>
<tr>
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<th>1S</th>
<th>1R</th>
<th>2S</th>
<th>2R</th>
<th>3S</th>
<th>3R</th>
</tr>
</thead>
<tbody>
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<td>Number of blocks</td>
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<td>10</td>
<td>5</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>Imposed number of blades</td>
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<td>100</td>
<td>200</td>
<td>100</td>
<td>100</td>
<td>100</td>
</tr>
<tr>
<td>Real number of blades</td>
<td>196</td>
<td>90</td>
<td>192</td>
<td>90</td>
<td>116</td>
<td>90</td>
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Figures 2(a)–(c) show the 2D blade shapes of the first-stage stator, first-stage rotor, and third-stage stator captured from their 3D shapes compared to that of the manufactured blade shape (blue line) and that of the secular-changed blades (orange line). Fig. 2(a) indicates that the trailing edge of the secular-changed blade is slightly thinned, while the shape of the leading edge may not be changed from that of the manufactured blade. The measured thickness of the secular-changed blade slightly increased on the suction surface. The rough surface may be caused by the deposition of the solid particles flowing from the boiler. Fig. 2(b) indicates that the leading edge of the secular-changed blade is sharpened, while the rear part is thinned. The thickness of the secular-changed blade slightly increased at the intermediate region of pressure surfaces. The reason may be the same as that for the stator. Fig. 2(c) indicates that the leading edge and the trailing edge of the secular-changed blade are slightly eroded. The thickness of the secular-changed blade was slightly increased at the rear region of pressure surface.

Fig. 1 Schematic of the computational grid system.
3. Numerical results

3.1 Flow conditions

The flow conditions were decided according to the actual working conditions. We considered two cases assuming manufactured and secular-changed blades.

Table 2 summarizes the flow conditions. CASE 1 corresponds to a case employing the manufactured blades for the first-stage stator, first-stage rotor, and third-stage stator (second-stage stator, second-stage rotor, and third-stage rotor keep the secular-changed blades). CASE 2 corresponds to a case of secular-changed blades for all three-stage stators and rotors. The total pressure and the total temperature were fixed at the inlet boundary. The static pressure was fixed at the outlet boundary. These values are referred to from the measured data. The sliding boundary conditions were applied to the boundaries between stators and rotors to simulate unsteady steam flows. The periodic boundary conditions were applied to the boundaries between the blade passages at each stator and rotor in the circumferential direction.

<table>
<thead>
<tr>
<th></th>
<th>Inlet total pressure [MPa]</th>
<th>Inlet total temperature [K]</th>
<th>Outlet static pressure [MPa]</th>
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<tbody>
<tr>
<td>CASE 1</td>
<td>4.53</td>
<td>873.45</td>
<td>2.46</td>
</tr>
<tr>
<td>CASE 2</td>
<td>4.56</td>
<td>873.49</td>
<td>2.53</td>
</tr>
</tbody>
</table>

3.2 Time-dependent numerical results

Figure 3 shows the instantaneous pressure distribution in CASE 1 at 50% span visualizing in 3D from the inlet of the first-stage stator to the outlet of the third-stage rotor. The red and blue colors indicate the highest and lowest values. The pressure linearly decreased from the inlet to the outlet through the three-stage stators and rotors. Other typical
features were not observed in Fig.3. The result in CASE 2 was also visualized but was omitted because of no typical difference from that in CASE 1.

Figures 4(a) and 4(b) show the instantaneous absolute Mach number distributions at 50% span in CASE 1 and CASE 2 from the inlet of the first-stage stator to the outlet of the third-stage rotor. Both values at the region between the stator and the rotor in all stages were relatively higher than those at the region between the stages. The wakes from the stator and rotor blades were clearly captured and smoothly connected at the sliding boundaries. The value at the outlet of the third-stage rotor in CASE 1 was slightly higher than that in CASE 2. Viewing overall such in Fig. 4 is too wide to compare the result locally; hence, we next focus on the first- and third-stage stator and rotor passages to show the difference between the two cases.

Figures 5(a) and 5(b) depict the instantaneous absolute Mach number distributions focused on the first-stage stator and rotor passages in CASE 1 and CASE 2 captured from Fig. 4(a) and 4(a). Figs 5(c) and (d) show those in the third-stage stator and rotor passages. The results for the second stage were omitted because of no change of blades. The distributions in Figs. 5(a) and (b) were basically similar with each other. The width of the wakes from the stator in Fig. 5(a) was slightly wider than that in Fig. 5(b), which may be caused by the thinner trailing edge secularly changed by erosion as the comparison of blade shapes shown in Fig. 2(a). This trend on wakes was also found in Figs. 5(c) and (d). The absolute Mach number in Fig. 5(c) was slightly higher than that in Fig. 5(d) at the outlet of the rotor. Those distributions at second-stage stator and rotor were omitted, because the blades were not replaced at the overhaul, setting the same blades in CASE 1 and CASE 2. Even though the flow from the first stage may affect the second stage, no distinguished difference between CASE 1 and CASE 2 is found in the Mach number distributions.

Figures 6(a) and 6(b) show the instantaneous absolute Mach number distributions at the flow cross section after the third-stage rotor. In these figures, periodical low and high Mach number regions were found toward the rotational direction because of the rotor movement. The maximum value of the Mach number in CASE 2 was obviously lower than that in CASE 1. The lower Mach number indicates that the flow speed decreases after the third-stage rotor in CASE 2.
Fig. 5 Instantaneous absolute Mach number distributions at 50% span of the first- and third-stage stator and rotor blade passages.

Fig. 6 Instantaneous absolute Mach number distributions at flow cross section after the third-stage rotor.

Figures 7(a)–(c) show the time-dependent static pressures after the first, second, and third rotors at 50% span in CASE 1 and CASE 2. The static pressures in all cases were periodically varied with the same phase. The averaged pressure values in the three figures for CASE 2 were slightly higher than those in CASE 1. The corresponding phase amplitudes in CASE 2 were slightly less than those in CASE 1, rounding the periodical pressure peak values. The reason may be the weakening of the wakes from the secular-changed rotor blades (Fig. 5). In addition, the shape of the plotted pressure variation for CASE 2 in Fig. 7(c) had an additional peak on the periodic value. The peak is due to the
interaction between the wake from the secular-changed third-stage stator and the following rotor, where the wake may be further influenced by the flow from the first and second stages. This peak indicates that the shape of secular-changed blades certainly affected the unsteady flow field.

Fig. 7 Time-dependent static pressures at 50% span after the first-, second-, and third-stage rotors.
3.3 Prediction of performance

The final purpose of this study is to predict the performance degradation of the middle-pressure steam turbine after a secular change of stator and rotor blades. The actual measured data indicates that the static temperature at the outlet of third-stage rotor was gradually increased by a long-time operation. This trend deeply depended on the performance degradation of the steam turbine.

Figure 8 shows the obtained static pressures at the inlet of the first-stage stator and the outlet of the third-stage rotor compared with the measured data. The values were normalized by the measured data at the inlet in CASE 1. The inlet values in all cases were almost the same with each other. Both measured and simulated values at the outlet in CASE 2 were higher than those in CASE 1. These results indicate that the present simulation could predict the temperature increase as the performance degradation because of the secular change of the stator and rotor blades. The simulated values at the outlet were underestimated from those of the measured data.

Fig. 8 Inlet and outlet static temperatures of the measured data and numerical results

3.4 Modification considering real gas effect

Specific enthalpy can be directly obtained from the simulation, while this value in turbine operation is calculated from the steam table referring to the measured pressure and temperature. The specific enthalpy at the inlet obtained by the simulation was underestimated even though the temperature and the pressure were almost the same with those of the measured data. Our numerical method employed the equation of state for ideal gas; hence, the specific enthalpy at high temperature and high pressure was underestimated. We should employ an equation of state for real gas for the numerical method to consider the real gas effect. However, the modification to our numerical method is currently not easy. It may also spend additional CPU time even if being completed. We introduced herein an alternative simple modification method. The simulated enthalpy drops from the inlet to the outlet in CASE 1 and CASE 2 were 201 [kJ/kg] and 195 [kJ/kg], respectively. We assumed that the real gas effect on these values can be partially canceled because of not the enthalpy itself but the difference between the inlet and outlet enthalpies (i.e., enthalpy drop). The specific enthalpy at the outlet was obtained by subtracting the simulated enthalpy drop from the inlet specific enthalpy referred to from the steam table. The outlet static temperatures were recalculated using the steam table from the modified specific enthalpy at the outlet with the outlet static pressure.

Figure 9 shows the corresponding figure to Fig. 8, in which the simulated static pressures at the outlet were replaced by the modified values. The values were relatively increased both in CASE 1 and CASE 2 as compared with those in Fig. 8. The current errors between the simulation and the measured data were 7[K] and 8[K] at the outlet in CASE 1 and CASE 2, respectively. These errors suggest that additional modifications should be further considered. We note the following results: 1) leakage losses from labyrinth seals are not considered; 2) the shape of each stator and rotor blade is all the same in the circumferential direction even for the secular-changed blade; 3) the long stator blades assisting the strength of the turbine unit are not considered; and 4) the referenced locations estimating the inlet and outlet values in the simulation and the measurement may not be the same with each other.
Fig. 9 Inlet and outlet static temperatures of the measured data and the numerical results after modification

4. Concluding remarks

Unsteady steam flows through three-stage stator and rotor blade rows in a middle-pressure steam turbine while considering manufactured (CASE 1) and secular-changed (CASE 2) blade shapes were simulated by our numerical method. The results obtained are summarized as follows:

1. The obtained instantaneous static pressure and the absolute Mach number were visualized. The static pressure decreased through the three-stage stator and rotor blade passages in both CASE 1 and CASE 2. The absolute Mach number at the outlet in CASE 2 was slightly lower than that in CASE 1. The width of the wakes formed after the stator with secular-changed blades was thinner than that with the manufactured blades both for the first and third stages because of the thinned trailing edge of the stator blades by a long-time operation.

2. The time-dependent static pressures at 50% span after each rotor passage in CASE 1 and CASE 2 were comparatively plotted. The averaged values in CASE 2 were slightly higher than those in CASE 1. The pressure variation phases were basically the same in both cases. The amplitudes of the phase in CASE 2 were slightly smaller than those in CASE 1. The pressure variation after the third-stage rotor passages in CASE 2 had an additional periodical peak. This result indicates that the shape of the secular-changed blades certainly affected the unsteady flow field.

3. The inlet and outlet static temperatures were predicted. The inlet static pressure in CASE 2 were almost the same with those in CASE 1, while the outlet values were higher than those in CASE 1. The numerical results indicate that the performance of the steam turbine was certainly deteriorated by the secular change of blades in a long operation. The outlet static temperature was modified considering the real gas effect. The simulated values approached the measured data.

Acknowledgments

This study was partially supported by ‘Next Generation High-Performance Computing Infrastructures and Applications R&D Program’ promoted by Ministry of Education, Culture, Sports, Science and Technology (MEXT), Japan.

References


