A numerical study of flow pattern in a deteriorated gas turbine under real operating condition

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Abstract
A numerical study of aerodynamics in a deteriorated gas turbine was carried out. In the present study, a hybrid method of CFD simulation and heat balance analysis were used to estimate entire performance of the gas turbine cycle and local flow characteristics in the turbine section. Using this analysis method, flows in a deteriorated gas turbine were simulated under an operating condition, which was designated using the same procedure with that used in a real power plant. In our previous study, the degradation of the gas turbine performance due to blades deterioration had been observed. In the present study, flow patterns were observed and mechanisms underlying the degradation of the gas turbine performance were discussed in detail. Influences of two kinds of deterioration on flow patterns were examined: the first was thinning of nozzle guide vanes in the first stage and the second was increase of rotor tip gap in the first stage. The results showed that the deterioration of nozzle guide vanes leaded to decrease of the Mach number through the nozzle, and this leaded to increase of the machine isentropic efficiency due to decrease of the friction loss. The increase in the rotor tip gap affected a loss generation of the following stage. Changes of flow patterns and its effects were discussed varying operating points of a gas turbine.

Keywords: Gas turbine, Blade deterioration, Aerodynamics, Maintenance, Computational fluid dynamics, Heat balance analysis

1. Introduction
Thermal power generation system using gas turbine combined cycle plays an important role as a base-load power system. In addition, the gas turbine combined cycle has become a regulator of demand and supply balance in power grid due to increase of renewable power. As a base-load power system, the gas turbine is required to be stable and highly efficient. However, the gas turbine deteriorates through long-time operation. Turbine blades deteriorate severely because of high temperature oxidation (Eliaz, G., et al, 2002). Nozzle guide vanes (NGVs) and rotor blades can deform due to corrosion and cracks also occur. Such damages can reduce the reliability and efficiency of the gas turbine. As a regulator of the power balance, the gas turbine is used under hard conditions. To keep the demand and supply balance in the power grid, daily startup and shutdown and partial load operation are often conducted. Such use of the gas turbine may also damage blades due to thermal stress and unsteady fluid force, but the details were not fully understood.

According to these situations, it is beneficial to understand influence of the turbine blade deterioration on the gas turbine performance and on flow characteristics. Measured data in a power plant and thermodynamic theory were used to estimate gas turbine performance such as power output and thermal efficiency in several studies (Dikunchak, S., 1992; Kurz, R. et al., 2009; Morini, M. et al., 2010; Spina, P. R., 2002; Zwebek, A., Pilidis, P., 2003a, 2003b, 2003c). Although these approaches make possible to understand influences of gas turbine deterioration on plant performance quantitatively, mechanisms of the performance degradation cannot be clarified. On the other hand, computational fluid dynamics (CFD) simulations were used to understand mechanisms of loss increase in deteriorated gas turbines (Bouchard, D. et al, 2014; Edwards, R. et al, 2012; Wang, S.-S. et al., 2009). The authors also carried out CFD simulations for deteriorated gas turbine and steam turbine using real blade geometries measured at a power plant during a maintenance work (Yonezawa,
K., et al., 2016, 2017, 2018, 2019). Although mechanisms of the loss increase and local flow patterns in a deteriorated turbine can be discussed by means of CFD simulations, influence on entire performance of a gas turbine cannot be discussed directly. This is because CFD simulation is still so expensive that it is difficult to comprehensively capture fluid flows in the entire system including a compressor, combustor and turbine. In addition, operation control parameters, such as air flow rate and fuel flow rate, are changed flexibly in actual power plants in order to satisfy power demands even if aerodynamic performances of compressor and turbine decrease. Therefore, it is both required to estimate a gas turbine performance and to discuss the flow characteristics under various operation conditions, which occur in actual power plant. According to these requirements, we developed a procedure using a heat balance analysis based on power plant data and CFD simulation of flow in a turbine region (Yonezawa, K., et al., 2019). In our previous report, influences of blade deterioration on gas turbine performance were discussed. Numerical results were also validated by comparing a measured data at an actual power plant with a gas turbine combined cycle. In the present study, mechanisms of the change in aerodynamic performance of the gas turbine are discussed by observing flow characteristics obtained by CFD simulations.

2. Numerical procedure

The numerical procedures were firstly reported and validated in the previous report (Yonezawa, K., et al., 2019) and an outline is reviewed here. Geometries of NGVs and rotor blades were measured using a 3-D scanner. The specification of the examined gas turbine is shown in Table 1. CFD simulation was carried out using ANSYS-CFX as a flow solver. The computational domain was from the turbine inlet until the turbine outlet of the three-staged turbine as shown in Fig. 1. Each stage consisted of a nozzle and a rotor, respectively. Flow paths in the nozzle and the rotor were treated as one-blade-periodic using a periodic boundary condition. At the inlet boundary of the computational domain total pressure and total temperature was specified. At the outlet boundary, the static pressure was specified. The interfaces between the nozzle and rotor were treated as mixing plane and values were averaged circumferentially at each radius in upstream plane and these values were transferred to the downstream plane. The computational mesh was unstructured and consisted of mostly tetrahedral elements and prism elements in the vicinity of walls. Numbers of elements of the computational mesh are shown in Table 2. In the present study, flow patterns in a three-staged turbine region of a gas turbine were examined by means of CFD for various operation conditions, which simulated the actual operation in a power plant. The operation condition of the examined gas turbine is designated with monitoring the compressor outlet pressure and the turbine exhaust temperature because state quantities at turbine inlet are not measured due to high temperature. To determine inlet and outlet boundary conditions of CFD simulations, total temperature and total pressure were required as

| Table 1 Specification of gas turbine. Flow characteristics were obtained from numerical simulation |
|---------------------------------------------------------------|
| 1st stage | 2nd stage | 3rd stage |
| Reynolds number (mid-span of rotor blade) | $1.9 \times 10^6$ | $2.8 \times 10^6$ | $6.5 \times 10^5$ |
| Aspect ratio of rotor blade | 1.6 | 1.1 | 3.8 |
| Zweifel number | 1.1 | 1.2 | 0.9 |
| Rotor blade tip Mach number | 0.55 | 0.56 | 0.86 |
| Mach number at stage outlet | 0.37 | 0.40 | 0.51 |
| Number of NGV | 48 | 48 | 60 |
| Number of rotor blade | 92 | 92 | 92 |
| Rotational frequency | 50 Hz | 50 Hz | 50 Hz |
| Rotor tip radius (m) | 1.4 | 1.6 | 1.7 |
| Rotor blade height, s (m) | 0.20 | 0.38 | 0.57 |
| Rotor blade chord length (mid-span) (m) | 0.12 | 0.11 | 0.13 |

| Table 2 Numbers of elements in each domain of computational mesh |
|---------------------------------------------------------------|
| 1st/NGV | 1st/Rotor | 2nd/NGV | 2nd/Rotor | 3rd/NGV | 3rd/Rotor |
| 6,421 | 5,437 | 12,915 | 13,520 | 9,710 | 9,205 |

($\times 1000$)
well as outlet static pressure. Therefore, the boundary conditions were determined by iterative simulations along a
flowchart presented in Fig. 2. In addition to the CFD results, values in the compressor and combustor were required to
estimate the performance of entire the gas turbine. Then, these values were obtained using a heat balance analysis
(Umezawa, S. 2006). Table 3 shows the values obtained from the measurement and CFD results at the optimum operating
condition without the deterioration. Since the measured value included a certain amount of errors due to uncertainty, the
values were corrected using the heat balance analysis. Through the comparison, the numerical method was validated
quantitatively as can be seen in Table 3.

Fig. 1 Schematic of computational domain. Examined turbine consists of three stages of NGV and rotor, and each stage was
treated as one-blade-periodic. Interfaces between stator and rotor regions are treated as mixing layer in steady state simulation.

Fig. 2 Flowchart of calculation for real operating conditions. Turbine inlet total pressure and total temperature are assumed and
iterative CFD simulations are repeated until the solution satisfies relationships in combustion chamber, at the outlet of turbine
and in compressor (Yonezawa, K., et al., 2019).
Table 3  Results of heat balance and CFD analyses. Measurement results were corrected using the heat balance analysis (Yonezawa, K., et al., 2019).

|                      | Measurement | CFD  |
|----------------------|-------------|------|
| Mass flow rate, kg/s | 597         | 592  |
| Inlet total temp.*, °C| 1200        | 1210 |
| Outlet temp., °C     | 598         | 595  |
| Inlet total press., MPa | 1.49      | 1.47 |
| Turbine isentropic efficiency | 0.871 | 0.872 |
| Turbine axial power, MW | 429       | 428  |

*Inlet temperature was calculated from enthalpy of mixture of combustion gas and cooling gas

Fig. 3 Profile of the 1st stage NGV at mid-span. Serious thinning takes place around trailing edge of the NGV in 1st stage. Scales in horizontal and vertical axes are normalized by chord length of NGV. Dips of the blade contour are film cooling injection holes.

Fig. 4 Thinned NGV model. Thickness around trailing edge and downstream of the cooling gas injection part is varied. Thinner 1.0 is almost identical to actual thinned geometry and Thinner 0.5 and 1.5 were made by interpolating and extrapolating between geometries Original and Thinner 1.0.

The blade geometries were measured, and the blade damage pattern were observed at a working power plant. NGVs in the first stage were damaged seriously and blade profile was clearly changed around trailing edge due to corrosion, as shown in Fig. 3. The NGV became thinner around the trailing edge, which leads to increase of the nozzle throat area. Since the nozzle throat area affects the rotor inlet velocity, influences of the NGV thickness were examined using various thicknesses around of the trailing edge, as shown in Fig. 4. The change of the nozzle throat area is presented in Table 4. On the other hand, original geometries of rotor blades in three stages and NGVs in the 2nd and 3rd stages were used for all computations to clarify the influence of the thinning of the 1st stage NGV.

The rotor tip gap in the 1st stage also affected on the gas turbine performance significantly. In the actual gas turbine, the tip gap changed due to the corrosion and abrasion of the rotor blade tip and the casing wall. In the present study, the tip gap was varied by changing radius of the casing wall for simplicity, as shown in Fig. 5.
Table 4 Throat area in a pitch of 1st stage nozzle

| Area change | Original (new) | Thinner 0.5 | Thinner 1.0 | Thinner 1.5 |
|-------------|---------------|-------------|-------------|-------------|
|             | -             | +2.0 %      | +3.6 %      | +5.3 %      |

Fig. 5 Meridional geometries of 1st stage nozzle and rotor with various tip gaps. Four kinds of tip gaps were examined. Tip gap was varied by shifting casing wall and the identical geometry of the rotor blade was used for simplicity.

Fig. 6 Monitoring parameters to determine the gas turbine operating condition. The left figure shows gas turbine outlet temperature versus compressor outlet pressure. The right figure shows the compressor characteristics as pressure ratio versus mass flow rate. Values are normalized by the values of optimum condition with original blades (Yonezawa, K., et al., 2019).

3. Results and Discussion

3.1 Effect of NGV thinning in 1st stage

It had been found in the previous study that the thinning of NGV caused increase of mass flow rate, turbine axial power, and turbine isentropic efficiency (Yonezawa, K., et al., 2019). The results are reviewed in Figs. 6-8. As mentioned in Section 2, the gas turbine was controlled along the designated relationship between the compressor outlet pressure and gas turbine outlet temperature, which was shown as a reference line in the left of Fig. 6. As the throat area changed, the compressor outlet pressure decreased. The mass flow rate was determined by the compressor characteristics as shown in the right of Fig. 6. Figure 7 shows that changes of the mass flow rate, turbine power output, and isentropic efficiency due to change of the nozzle throat area. The increase of the mass flow rate leaded to axial power output increase in the 2nd and 3rd stages, but not in the 1st stage. Isentropic efficiency increased in three stages due to thinning of the 1st NGV, which was contrary to the expectation. Mach number distributions in mid-span planes of the original and Thinner 1.0
NGVs are shown in Fig. 8. It was found that the Mach number decreased around the nozzle outlet due to thinning of the NGV.

Based on these results, flow characteristics were examined in the present study. Flow angle distributions were compared at the inlet of the 1st stage rotor as shown in Fig. 9. The flow angle was circumferentially averaged and shown as span-wise distribution. The decreases in the flow angle became severer as the NGV got thinner. Decrease of the flow angle means the decrease of the inflow angular momentum of the rotor. Since the blade profile of the rotor was identical for all cases, the fluid work on the 1st stage rotor became smaller as the inflow angular momentum decreased. That is why the stage output decreased in the first stage although increased in the 2nd and 3rd stages.

Fig. 7 Mass flow rate increment, output increment, and isentropic efficiency change versus throat area increment for various length models (Yonezawa, K., et al., 2019). As the throat area increases, the mass flow rate and the turbine power output increase as well as the turbine isentropic efficiency.

Fig. 8 Mach number distribution at mid-span plane around 1st stage (left: Original, right: Thinner 1.0) (Yonezawa, K., et al., 2019). In the nozzle with original geometry, the supersonic region occurs around the nozzle throat, but the flow Mach number decreases with the thinned NGV (Thinner 1.0) due to increase of the throat area.

Fig. 9 Flow angle at 1st stage rotor. The flow angle decreases due to the thinning of the NGV due to decrease of the flow velocity at the outlet of the nozzle.
Wall friction distributions on the suction surfaces of the NGVs in the first stage are shown in Fig. 10. The left panels are with the original NGV of the 1st stage and right panels are with thinned NGV, Thinner 1.0. In the 1st stage, the wall friction on the original NGV is larger than that on the thinned NGV around the mid-chord region which is around the nozzle throat. This difference was due to difference in the flow Mach number as shown in Fig. 8. Around the high Mach number region, the wall friction was also large. In addition, the large wall friction occurred in the vicinity of the side walls. These were caused by shock-boundary layer interactions. In the 2nd and 3rd stages, influence of the increase in the throat area of the 1st stage nozzle was observed. The wall friction was larger with the original NGV in the 1st stage than with the thinned NGV. In addition, there were high wall friction regions near the casing in the 2nd and 3rd stages. As shown in figures, these were due to tip leakage vortices from the upstream rotors. These results confirmed that the influence of the throat area change in the 1st stage nozzle extended entire the turbine.

Velocity coefficients in nozzles and rotor disks are shown in Fig. 11. It is confirmed that the wall friction is a major cause of aerodynamic loss. In the nozzle, the velocity coefficient increased as the nozzle throat area increased. The gradient of the curves of the 1st stage is the largest and of the 3rd stage is the smallest. This tendency is consistent with the difference of the wall friction in Fig. 10.
It may be unexpected results that the thinning of NGV reduces the aerodynamic loss in the turbine. As discussed previously, however, the thinning of the NGVs increases the mass flow rate and it may lead to the decrease of the compressor efficiency and the thermal efficiency.

3.2 Effect of increase of rotor tip gap in 1st stage

Numerical simulations were carried out using various rotor tip gaps of the 1st stage and using original geometries of nozzles and rotors of the 2nd and 3rd stages (Yonezawa, K., et al., 2019). Results are presented again in Fig. 12. The mass flow rate was not affected by the tip gap increase because the flow rate and the compressor outlet pressure were determined by the throat area of the nozzle in the 1st stage. The stage power output of the 1st stage was significantly affected by the tip gap increase as reported by many researchers (Ameri, A. A. and Bunker, R. S., 1999; Bunker, R. S. and Bailey, J. C., 1999). The power output in the 2nd and 3rd stage was not affected so significantly. The isentropic efficiency in the first stage was affected as expected. In the second stage, the isentropic efficiency was decreased as much as in the first stage. The velocity coefficients were also examined as shown in Fig. 13 to discuss this reason. In the first stage, most of loss was due to the increase of the tip gap itself. On the other hand, the loss in the second stage occurred in the nozzle, but not in the rotor. This suggested that the tip clearance vortex in the 1st stage might affect the flow in the 2nd stage. In the present study, the flow pattern is discussed in detail focusing on characteristics of the interaction between the tip vortex and the NGV.

Tip vortex structure is visualized in Fig. 14 by total pressure distributions in axial cross sections for two kinds of tip gaps, \( g = 0.042s \) and \( 0.051s \). Locations of each cross section are shown in top of Fig. 14. The cross section (a) is on the mid-chord and the cross section (b) is at the downstream of the rotor blade. The tip vortex is visualized as a low-pressure region near the casing wall. The tip vortex occurred in the vicinity of the blade tip in cross section (a), which was at mid-chord location. In cross section (a), there was difference of pressure distribution on the blade tip. In cross section (b), the pressure distributions around the vortex show obvious difference between the two cases. More significant pressure drop occurs with larger tip gap, which means increase of the vorticity and increase of the pressure loss in the 1st stage rotor.

![Fig. 12 Mass flow rate increment, output increment, and isentropic efficiency change versus first-stage rotor considering real operating conditions. Mass flow rate was not affected significantly by the increase of the tip clearance. (Yonezawa, K., et al., 2019)](image1)

![Fig. 13 Velocity coefficient of rotor and NGV versus tip gap of first-stage rotor considering real operating conditions. Velocity coefficients in nozzle, \( \phi_s \) and in rotor, \( \phi_R \) are shown in left and right, respectively. The decrease of the velocity coefficient was obvious in the 2nd stage nozzle and 1st stage rotor. (Yonezawa, K., et al., 2019)](image2)
Fig. 14 Distribution of normalized total pressure in relative coordinates in axial cross section around 1st stage rotor blade (left: \(g=0.042s\), right: \(g=0.051s\)). The cross section (a) is on the mid-chord and the cross section (b) is at the downstream of the rotor blade. The total pressure was normalized by the turbine inlet total pressure. The tip vortex core is visualized as a low-pressure region and the low-pressure region around the casing wall becomes larger as the tip gap increases.

The flow pattern around the blade tip were also discussed in many literatures (Ameri, A. A. and Bunker, R. S., 1999; Bunker, R. S. and Bailey, J. C., 1999, for example) and the results obtained in the present study showed similar characteristics.

Fig. 15 shows the pressure distribution on NGV in the 2nd stage at 90% span position. For both cases, the inverse pressure occurred between the suction and pressure surfaces around the leading edge, in which the pressure on the suction surface is larger than on the pressure surface. The amount of pressure-drop on the pressure surface around the leading edge with \(g = 0.051s\) is more than that with \(g = 0.042s\). This meant that the influence of the tip vortex from the upstream was more significant, and consequently, the loss in the nozzle increased.

According to these results, it was confirmed that increase of the tip gap affects the flow in the nozzle in the following stage as well as the rotor tip leakage. Though it has not been examined in the present study, further investigation is required to examine the influence of tip gap increases in the 2nd and 3rd stages.
Fig. 15 Pressure distribution on 2nd stage NGV (top: $g=0.042s$, bottom: $g=0.051s$). The inverse pressure between the suction and pressure surfaces around the leading edge. The pressure-drop on the pressure surface for $g=0.051s$ is more than that $g=0.042s$.

4. Conclusion

Influences of NGV and blade deterioration was discussed. In the present study, the influences of two kinds of corruptions were examined in detail: the thinning of the NGV in the first stage and the influence of the increase of the tip gap.

The thinning of the NGV in the first stage resulted in the increase of the mass flow rate. However, the flow Mach number and velocity around the nozzle throat decreased due to decrease of the area ratio between the nozzle inlet and the throat. In the 1st stage, consequently, the power output decreased due to decrease of the inflow angular momentum. By contrast, the stage isentropic efficiency increased due to decrease of the friction loss.

The increase of the tip gap in the first stage resulted in the decrease of the isentropic efficiency in the 2nd stage as well as in the first stage. This was because the tip vortex affects the flow pattern around the NGV in the 2nd stage. According to the results obtained in the present study, it was found that the deterioration of the NGV and the blade affects the flow in the following stages and operation condition of another component of the gas turbine. Further investigations are required to clarify the influence of such inter-stage interaction on the blade deterioration as well as the gas turbine efficiency, which may lead to enhancement of the reliability and efficiency of the gas turbine.

Nomenclature

- $C_p$: Total pressure in relative coordinates normalized by turbine inlet pressure [-].
- $g$: Tip gas of rotor [m].
- $h$: Static enthalpy [J/kg]
- $h_s$: Isentropic static enthalpy [J/kg]
- $h_0$: Total enthalpy [J/kg]
- $h_{0,rel}$: Total enthalpy in relative coordinates [J/kg]
- $m$: Mass flow rate [kg/s]
- $p_{01}$: Total pressure at turbine inlet [Pa].
- $p_{c2}$: Total pressure at compressor outlet [Pa].
- $p_7$: Static pressure at turbine outlet [Pa].
- $\rho$: Density of working fluid [kg/m$^3$]
- $s$: Blade span length [m].
- $T_{01}$: Total temperature at turbine inlet [K].
ϕ₉ : Velocity coefficient in rotor [-], \( \phi_9 = \sqrt{1 - \zeta_R} \)
ϕₛ : Velocity coefficient in nozzle [-], \( \phi_s = \sqrt{1 - \zeta_s} \)
ζ₉ : Loss coefficient in rotor [-], \( \zeta_9 = 1 - \left( \frac{h_9 - h}{h_0 - h} \right) \) _Rotor_outlet_
ζₛ : Loss coefficient in nozzle [-], \( \zeta_s = 1 - \left( \frac{h_0 - h}{h_0 - h_s} \right) \) _nozzle_outlet_
ζ' : Loss coefficient in combustor [-] \( \zeta' = \frac{2P}{m^2} (P_{c2} - P_{b1}) \)

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