ABSTRACT

In the present work, unsteady RANS simulations were performed to clarify several interesting features of the unsteady three-dimensional flow field in a turbine stage. The unsteady effect was investigated for two cases of axial spacing between stator and rotor, i.e. large and small axial spacing. Simulation results showed that the stator wake was convected from pressure side to suction side in the rotor. As a result, another secondary flow, which counter-rotated against the passage vortices, was periodically generated by the stator wake passing through the rotor passage. It was found that turbine stage efficiency with the small axial spacing was higher than that with the large axial spacing. This was because the stator wake in the small axial spacing case entered the rotor before mixing and induced the stronger counter-rotating vortices to suppress the passage vortices more effectively, while the wake in the large axial spacing case eventually promoted the growth of the secondary flow near the hub due to the migration of the wake towards the hub.

INTRODUCTION

There have been numerous efforts to improve aerodynamic performance and efficiency in turbomachinery. As a result, experimental and numerical works have produced a lot of information and understanding of three-dimensional flow field in turbomachinery. However, they have been limited to the steady flow, not unsteady flow. As the unsteady flow field in turbomachinery has not fully been clarified, it is difficult to estimate an effect of unsteadiness of flow field on performance and to incorporate the unsteady effect in design of turbomachines. The unsteady flow field is induced by rotor/stator interaction, which consists of mainly two factors. One is an interaction of potential fields, which is an inviscid effect and is caused by relative motion between stator and rotor. The other is an interaction of upstream wakes on boundary layer, vortex, separation and secondary flow in downstream passages, which is a viscous effect and leads to entropy generation.

Recently, a lot of work has been done on estimating unsteady effect of rotor/stator interaction, especially wake passing, upon performance of turbomachinery blade rows [1, 2]. The additional losses, which were related to the effect of so-called negative jet due to the wake, was investigated in [3, 4]. Some models were proposed to estimate the unsteady effect of the wake passing through downstream passage on performance [5, 6, 7]. These models were based on the concept of so-called wake recovery proposed by Smith [8]. The wake recovery was explained by a mechanism of the wake attenuation due to inviscid and reversible manners. Assuming two-dimensional, inviscid and incompressible flow, the wake deficit is attenuated as the wake is stretched, because the circulation around the wake is constant by Kelvin’s theorem. The wake recovery was related to performance benefit since the wake attenuates without viscous dissipation. Effects of axial spacing between stator and rotor also has been discussed in association with the unsteady effect [6]. As a whole, it could be expected to achieve the higher efficiency as the axial spacing decreased, because the wake would have little experience of mixing in the axial spacing.
region and more benefit of the wake recovery is expected. However, this anticipation does not include the effect of potential interaction. The effect of potential interaction in compressors and turbines has been investigated by a number of research groups [9, 10, 11]. In transonic compressors additional losses have been brought about in small axial spacing case since a shock wave near the rotor leading edge interacted with the stator [10, 11].

In this way, although the unsteady effects have been revealed progressively, much remains unclear about unsteady flow effects upon the performance of turbomachines. For example, the effect of wake recovery is not confirmed if three-dimensionality dominates the unsteady flow field like in turbine stages with low aspect ratio. The purpose of the present work is to clarify details of unsteady and three-dimensional field in a turbine stage, and to elucidate the unsteady effect of rotor-stator interaction on the performance. The unsteady and three-dimensional flow fields have been investigated in a steam turbine stage with low aspect ratio by unsteady three-dimensional Navier-Stokes flow simulations based on an implicit high-resolution upwind scheme using the TVD formulation. The simulation has been made to clarify the effect of the axial spacing and the results for large and small axial spacing cases were compared to make the unsteady effect more approachable.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>$p$</td>
<td>pressure</td>
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<tr>
<td>$T$</td>
<td>temperature</td>
</tr>
<tr>
<td>$\eta$</td>
<td>turbine stage efficiency</td>
</tr>
<tr>
<td>$\kappa$</td>
<td>specific heat ratio</td>
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<tr>
<td>$T_{rt}$</td>
<td>blade passing period</td>
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<tr>
<td>$T_{st}$</td>
<td>period taken for rotor to pass through one pitch of stator</td>
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<tr>
<td>$C_p$</td>
<td>pressure coefficient</td>
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<tr>
<td>$\rho$</td>
<td>density</td>
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<tr>
<td>$U_t$</td>
<td>rotor tip speed</td>
</tr>
</tbody>
</table>

Subscripts

0 total
1 upstream of stator
2 downstream of rotor
wall rotor blade surface

A STEAM TURBINE STAGE

A single turbine stage was analyzed in the present work. The present turbine stage had 44 stator blades and 98 rotor blades. In order to simulate the stator/rotor interaction problem in the turbine stage, the number of the stator blades was changed from 44 to 49 to avoid excess computational loadings and resources for simulation of the turbine stage. The cross-sectional profile of the stator blade was accordingly scaled down while the height of the stator blade was not changed to keep throat area, that is, solidity was not changed. The calculations were performed between a single passage for the stator and two passages for the rotor by applying the periodical boundary condition in circumferential direction. However, it should be noted that the aspect ratio of the stator has been changed to a little higher than that of the original. If the number of the rotor blades was changed from 98 to 88, the secondary flow in the rotor passage was supposed to become extremely strong because of lower aspect ratio. This is why the number of the stator blades, not the rotor blades, was changed. Two cases of axial spacing between the stator and the rotor, i.e. large and small axial spacing were investigated in the present work in order to make the unsteady effect more clear. It was expected that the unsteady flow field in the turbine could be affected by the axial spacing. One of the axial spacings was set to 46% rotor chord length (large axial spacing case) and the other was set to 12% rotor chord length (small axial spacing case). Although the target was the steam turbine stage, the air was used in the present calculation.

NUMERICAL ANALYSIS METHOD

Numerical scheme

Unsteady three-dimensional flow simulation was performed by solving the compressible Navier-Stokes equations using an unfactored implicit upwind relaxation scheme with inner iterations [12, 13]. The numerical method used in the present flow solver is outlined in the following.

The three-dimensional Reynolds-averaged Navier-Stokes equations were discretized in space using a cell-centered finite volume formulation and in time using the Euler implicit method. The inviscid fluxes were evaluated by a high-resolution upwind scheme based on a TVD formulation [14], where a Roe's approximate Riemann solver of Chakravarthy [15] and a third-order accurate MUSCL approach of Anderson et al. [16] with the Van Albada limiter were implemented. The viscous fluxes were determined in a central differencing manner with Gauss’s theorem. As for turbulence model the k-$\omega$ turbulence model [17] was employed to estimate the eddy viscosity. Simultaneous equations linearized in time were solved by a point Gauss-Seidel relaxation method using no approximate factorization [12, 18]. To obtain a time-accurate solution, inner iterations, so-called Newton iterations, were introduced at each time step according to Chakravarthy [19]. To achieve second-order accurate in time by applying the three-point-backward difference approximation to the temporal derivative [13] and performing inner iterations of three times at each time step. A nondimensional time step size normalized by the rotor tip radius and the inlet sound speed was set to 0.0005. More than 430 time steps were included in the period taken for a rotor blade to pass through one pitch. The blade passing period of the rotor corresponded to the nondimensional time of 0.216. The multi-block parallel algorithm was introduced into the present code and implemented using Message Passing Interface (MPI). More than 300 hours of CPU times were needed for each case of the present unsteady simulations by using a PC-cluster with 8 nodes, which had Intel Xeon 2.4GHz processors and a Gigabit
Computational grid

Figure 1 shows a computational grid used in the present simulations. The computational domain consists of two regions, which are stator and rotor regions, to calculate in different frame of reference. A structured H-type grid system was generated in each region of stator and rotor. As for the stator region, the composite-grid system was employed to avoid grid skewness around the trailing edge, and the stator region was divided into two zones of upstream and downstream. The stator grid consists of 110 cells in the streamwise direction and 69 cells in the pitchwise direction for upstream region, and 44 cells (24 cells for the small axial spacing case) in the streamwise direction and 138 cells in the pitchwise direction for downstream region. Each region had 89 cells in the spanwise direction. The rotor grid consists of \(160 \times 89 \times 69\) cells in the chordwise, spanwise and pitchwise directions, respectively. The whole grid system for stator of one passage and rotor of two passages had 1,363,520 cells in the large axial spacing case. The ratio of the minimum grid spacing on solid walls to the blade tip chord length was less than \(5 \times 10^{-5}\) to evaluate the viscous fluxes at the walls by applying the no-slip and adiabatic conditions with no wall function method. This minimum grid spacing gave \(y^+<1\) at the walls.

Boundary condition

Boundaries of the computational domain were formed by cell interfaces in the present numerical scheme where the cell-centered finite volume approach was applied to the spatial discretization. Fictitious cells were introduced just outside all the boundaries to treat boundary conditions. Values of conserved variables satisfying the boundary conditions were given at the fictitious cells. Using the fictitious cells, numerical fluxes through the boundaries were evaluated in the same way as interior cell interfaces. This treatment of the boundary conditions eased non-physical reflections at the inflow and outflow boundaries, because the inviscid fluxes through the boundaries were evaluated according to the approximate Riemann solver in which the signal propagation properties of the Euler equations were simulated. Details of the boundary conditions can be found in [18].

At the fictitious cells adjacent to the inflow boundary, the conserved variables were given by flow conditions upstream of the stator. Meanwhile, for the outflow boundary, all the variables at the fictitious cells were set equal to those at interior cells adjacent to the fictitious cells. It should be noted that the magnitude of the velocity at the fictitious cells adjacent to the outflow boundary was scaled so as to maintain the imposed total flow rate. At the fictitious cells adjacent to solid wall boundaries, the variables were given so that no-slip and adiabatic conditions could be satisfied. Periodic boundary conditions were imposed in the circumferential direction, because the computational domain was limited to a single stator passage as mentioned above. At the fictitious cells adjacent to the interface boundary connecting the computational domains of the stator and the rotor, the variables were given to interpolate those of the interior cells in the corresponding computational domain, because the fictitious cells overlapped with the interior cells in the corresponding computational domain.

RESULT AND DISCUSSION

Time-averaged Flow Field

The unsteadiness of the flow field in the turbine stage is considered to be closely related to the axial spacing between the stator and the rotor. We investigated the effect of the axial spacing on the unsteady flow field and aerodynamic performance in the turbine stage. The time-averaged flow field, which was calculated by averaging over the time period taken for the rotor blade to pass through one pitch of the stator after a transitional state of one revolution, is discussed in this section. Figure 2 shows the spanwise distribution of circumferentially averaged turbine stage efficiency. The turbine stage efficiency is defined as follows:
where $p$ is pressure, $T$ is temperature, and $\kappa$ is the specific heat ratio. A subscript of 0 denotes ‘total’, and subscripts of 1 and 2 indicate upstream of the stator and downstream of the rotor, respectively. The turbine stage efficiency is normalized by its mass-averaged value for the large axial spacing case. As shown in the figure, the simulation result for the small axial spacing has higher turbine stage efficiency in region from the hub to the midspan. In addition, a drop of the turbine stage efficiency near the casing is less in the small axial spacing case. Therefore, the whole turbine stage efficiency is higher in the small axial spacing case. This is confirmed by the experiment. The drops near the endwall in the turbine stage efficiency distribution can be attributed to the existence of passage vortex. This implies that the passage vortices are less in the small axial spacing case judging from less drop near the endwall.

Figure 3 shows the time-averaged entropy distribution on some cross flow planes in the rotor passage. Plane I through Plane V locates at 0%, 40%, 60%, 80%, and 100% chord, respectively. As seen in the Plane V, high entropy region appears near the suction side. The reason is that the low-energy fluid in the stator wake is transported from the pressure side to the suction side by the circumferential pressure gradient in the rotor passage. In addition, the passage vortices, which are formed in the rotor, also yields high entropy near the casing and the hub. It could be found that in the large axial spacing case another high entropy region emerges around the hub. Compared to the small axial spacing case, the high entropy region around the casing-side passage vortex seems to be larger in the large axial spacing case. In the present turbine stage, an extremely strong swirling flow is formed in the region between the stator and the rotor by the large outlet flow angle of the stator, and at the same time a pressure gradient in radial direction is induced by the swirling flow. The pressure gradient in radial direction transports the low-energy fluid to the hub. The actual path of the fluid from the stator to the rotor was much longer compared to the axial spacing because of the large outlet flow angle of the stator. A little increase of the axial spacing would provoke the accumulation of the low-energy fluid near the hub, which would result in decrease of the turbine efficiency near the hub. This could be easily understood by comparing locations of the stator passage vortices on Plane I between two axial spacing. The accumulation of the low-energy fluid near the hub might enhance the secondary flow, i.e. the passage vortex near the hub.

Figure 4 shows the limiting streamlines on the suction surface of the rotor blade, which were described using LIC (Line Integral Convolution) visualization technique [20]. It is found that the separation lines, which are indicated by the dashed lines, appears near the casing and the hub. The separation on the blade...
suction surface is caused by the secondary flow near the endwall, that is passage vortex. The separation region near the hub extends to 25% span at the trailing edge in the large axial spacing case while it is 10% span in the small axial spacing case. That means in the large axial spacing case the passage vortex near the hub is enhanced by the accumulation of the low-energy fluid near the hub. These characteristic points noted in Figs. 3 and 4 corresponds so well to those in the turbine stage efficiency distribution shown in Fig. 2. In other words, the unsteady losses would be mainly generated downstream of the stator, and they are closely related to the unsteady flow phenomena occurred downstream of the stator.

**Unsteady Flow Phenomena**

**Effect of wake passing** Figure 5 shows the instantaneous entropy distribution at the midspan. As shown in this figure, stator wake is slashed with the rotor leading edge, and then stretched in streamwise direction. While passing through the rotor, the wake is transported to the suction side because of the circumferential pressure gradient in the rotor passage. The wake accumulates around the suction surface till advecting at the aft-passage of the rotor, as mentioned in the previous section. However, it is found that there is a difference between the large and the small axial spacing case, that is, in the large axial spacing case the wake decays earlier by mixing before it enters the rotor compared to that in the small axial spacing case. It should be noted that the wake is transported in radial direction as well as in circumferential direction and passed through the rotor with significantly complicated deformation. Therefore, the effect of the wake recovery is not completely identifiable in the present turbine stage like with low-aspect ratio.

Figure 6 shows the entropy distributions and secondary flow vectors of velocity fluctuation on a cross flow plane at 20% chord in the rotor for the large axial spacing case. Two passages in the rotor are shown in the figure because the present simulations were performed by single passage for the stator and two passages for the rotor. In the figure, \( T_{\lambda} \) indicates the period of time taken for a rotor blade to pass through one pitch. The fluctuation velocity is defined by the difference of velocity vector between instantaneous and time-averaged results, and projected onto the cross flow plane to describe the secondary flow field. In the figure, the stator wake could be identified by high entropy region. The passage vortices from the stator are recognized near the wake. It is found that another secondary flow (hereinafter referred to as wake-induced secondary flow), which is distinguished here from the well-known secondary flow near the endwall, is induced in the wake and the passage vortices as seen in the fluctuation velocity. This is a result that a low energy fluid in the wake and the passage vortices could be convected apart from main flow by pressure gradient. In the axial spacing region between the stator and the rotor, the low energy fluid in the wake and the passage vortices could be convected apart from main flow by pressure gradient. In the axial spacing region between the stator and the rotor, the low energy fluid in the wake and the passage vortices could be convected apart from main flow by pressure gradient. In the axial spacing region between the stator and the rotor, the low energy fluid in the wake and the passage vortices could be convected apart from main flow by pressure gradient. In the axial spacing region between the stator and the rotor, the low energy fluid in the wake and the passage vortices could be convected apart from main flow by pressure gradient. In the axial spacing region between the stator and the rotor, the low energy fluid in the wake and the passage vortices could be convected apart from main flow by pressure gradient.
vortices in the rotor. The passage vortex near the hub is stronger than that near the casing because the low-energy fluid is transported to the hub and accumulates around the hub. In the left passage, the wake-induced secondary flow counter-rotated against the passage vortices near the endwall, and thus the passage vortices seem to disappear as seen in the secondary flow vector in the figure. To the contrary, it is seen at $t/T_{rt}=0.00$ that there are the passage vortices in the left passage and the
counter-rotating vortices in the right-passage. That means the counter-rotating vortices are periodically brought about in the rotor by the incoming wake. Since the low-energy fluid is circumferentially transported to the suction side in the rotor.
have more effect on the suppression of the passage vortices.

Figure 11 shows entropy distribution along axial direction downstream from the leading edge of the rotor. On the abscissa axis in the figure, the leading edge and the trailing edge of the rotor correspond to 0.0 and 1.0, respectively. In the small axial spacing case, less entropy were generated at downstream from the mid-chord, where there must be a difference in the secondary flow as discussed above. Figure 12 shows a comparison of entropy in three regions of the turbine. The value shown in this figure is normalized by the total value of the large axial spacing case. The 30% span from the endwall is defined as the hub and the casing regions and the other is midspan region. The distribution of the mass-averaged value of entropy corresponds well with that of the turbine stage efficiency in Fig. 2. The highest mass-averaged entropy is in the hub region and the lowest one is in the midspan region. The large axial spacing case has higher entropy in the hub and the midspan because of the accumulation of the low energy fluid in the hub. On the other hand, the mass-weighted entropy indicates that in the large axial spacing case most of the loss seems to be generated around the midspan, where there is a difference in the mass-weighted entropy between two axial spacing cases while it is almost same in the hub and the casing regions. This is related to the axial velocity distribution. In the large axial spacing case a blockage effect of the accumulated low energy fluid around the hub increase the axial velocity in the midspan, which result in the increase of mass-weighted entropy in the midspan. In the small axial spacing case, since the secondary flow is weakened compared to the large axial spacing case, the loss remains almost same despite the larger axial velocity in the hub. As a
of the interaction between the pressure fields of the stator and the rotor. Accordingly, the potential interaction may affect growth of the boundary layer on the stator blade surface. Figure 14 shows the time variation of the total pressure loss downstream of the stator. The total pressure loss is normalized by its time-averaged value in the large axial spacing case. As expected, the fluctuation of the total pressure loss has large amplitude in the small axial spacing case while in the large axial spacing case the total pressure coefficient does not almost fluctuate. The strong potential interaction in the small axial spacing case brings about the large fluctuation of the pressure near the trailing edge of the stator as mentioned above. Therefore, the boundary layer thickness on the blade suction surface of the stator varies with time, and the total pressure loss also fluctuates accordingly. Surprisingly, as for the time-averaged value of the total pressure loss coefficient its value in the small axial spacing case is less than that in the large axial spacing case although it is expected to be large. The loss, which is generated in the stator, is less in the small axial spacing case as well as that in the stage. The further investigation should be needed for this result.

result, the total loss is lower in the small axial spacing case as shown in the figure.

Potential interaction Figure 13 shows the instantaneous pressure distributions on the midspan. In the small axial spacing case, the high-pressure region appears between the trailing edge of the stator and the leading edge of the rotor. This is the result

Fig. 13 Instantaneous pressure distributions at midspan

Fig. 14 Time variation of total pressure loss downstream of stator

Fig. 15 Pressure distributions on rotor blade surface at midspan
Figure 15 shows the time variation of the pressure distribution on the rotor blade surface at the midspan. In the figure, the pressure coefficient is defined as follows,

\[ C_p = \frac{p_{01} - p_{\text{wall}}}{\rho U_t^2/2} \]  

where \( p_{01} \) is the total pressure upstream of the stator, \( p_{\text{wall}} \) is the pressure on the rotor blade surface, \( \rho \) is density, and \( U_t \) is rotor tip speed. As shown in the figure, the pressure distribution on the blade surface drastically changes with time in the small axial spacing case. It is believed that this is attributed to the strong potential interaction in the small axial spacing case. It is noted that the strong potential interaction may give rise to the violation of the blade oscillation and finally the fatigue failure of the blade although the small axial spacing has a potential to decrease the loss in a turbine stage.

**CONCLUSIONS**

Unsteady three-dimensional Navier-Stokes flow simulation based on the high-resolution upwind scheme using the TVD formulation was performed in order to investigate the effect of the unsteadiness of the flow field on the performance in an axial steam turbine stage. The mechanism of the unsteady loss was analyzed especially focusing on the effect of the wake passing. The results are summarized as follows:

1. In the large axial spacing case, the high entropy region appears near the hub because the low-energy fluid in the wake is transported in the inward direction by the radial pressure gradient due to the strong swirling flow downstream of the stator.
2. The wake-induced secondary flow, which is distinguished by the well-known secondary flow near the endwall, is periodically induced in the rotor passage since the low-energy fluid in the wake is convected by the pressure gradient.
3. The wake-induced secondary flow counter-rotates against the passage vortices near the endwall. This is because the low-energy fluid circumferentially transported impinges against the suction surface, and thus counter-rotates against the passage vortices near the blade suction corners.
4. The counter-rotating vortices are expected to have an effect on the suppression of the passage vortices.
5. The effect of the counter-rotating vortices on the passage vortex suppression is less in the large axial spacing case because of the wake decay. Conversely, the small axial spacing has a potential to decrease the loss due to the secondary flow in a turbine stage.
6. The strong potential interaction occurs in the small axial spacing case, which may lead to the violation of the blade oscillation and finally the fatigue failure of the blade.

**REFERENCES**


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